



OLIVIER LE ROUX DE BRETAGNE

THERMODYNAMIC AND COMBUSTION CHARACTERISTICS OF
ULTRA-HIGH PERFORMANCE ENGINES FOR MOTORSPORT
APPLICATIONS

SCHOOL OF AEROSPACE, TRANSPORT AND
MANUFACTURING
Advanced Vehicle Engineering Centre

Doctor of Philosophy (PhD)
Academic Year: 2022 - 2023

Supervisor: Professor Matthew F. Harrison
Associate Supervisor: Clive Temple
August 2023



SCHOOL OF AEROSPACE, TRANSPORT AND
MANUFACTURING
Advanced Vehicle Engineering Centre

Doctor of Philosophy (PhD)

Academic Year: 2022 - 2023

OLIVIER LE ROUX DE BRETAGNE

Thermodynamic and Combustion Characteristics of Ultra-High
Performance Engines for Motorsport Applications

Supervisor: Professor Matthew F. Harrison
Associate Supervisor: Clive Temple
August 2023

This thesis is submitted in partial fulfilment of the requirements for
the degree of Doctor of Philosophy (PhD)

© Cranfield University 2023. All rights reserved. No part of this
publication may be reproduced without the written permission of the
copyright owner.

ABSTRACT

Using steady-state and transient one-dimensional gas dynamic engine models developed with AVL Boost™, critical assessment of the performance characteristics of the current 2014+ Formula One™ engine and of the future 2026 Formula One™ Power Unit are investigated.

For the 2014+ regulations, a Digital Twin, aiming at replicating the trends of the real engine despite a lack of component-level-detail data, is created and used to scientifically explain how this engine achieves 50+% brake fuel conversion efficiency and to rank the contribution of each enabling technologies (high compression ratio, lean combustion, passive pre-chamber, direct injection, asymmetric valve profiles, MGU-H and waste gates used as pressure-relief valves).

The impact of the 2026 Formula One™ Power Unit regulations on engine performance is investigated and highlights that the reduction in fuel flow will not only result in the obvious reduction in power output but also in in-cylinder pressure which introduces opportunities for enhanced combustion process and higher air/fuel ratios. Nevertheless, with the high MGU-K power, both the 2014+ and 2026 Power Units are predicted to have similar peak output power despite an advantage at low speed for the 2026 regulations thanks to the capacity of electric motor to produce torque at low speed. Using transient simulations, the impact of the removal of the MGU-H in the 2026 regulations is assessed and an anti-lag solution using the MGU-K called torque consuming is investigated. It is demonstrated that always operating the engine at full load during acceleration phases and using the MGU-K to absorb the excess power compared to the power demand / to control the amount of power delivered to the wheels helps to reduce turbo lag, improve engine efficiency, and reduce the need for the MGU-K to torque fill, but at the expense of a higher fuel consumption.

Keywords:

Engine simulation, Digital Twin, AVL Boost™, Formula One™ Power Unit, Energy, Exergy, Turbo lag mitigation

ACKNOWLEDGEMENTS

I would like to express my gratitude to Professor Matthew Harrison, my supervisor, for his help along the whole project, for the time he dedicated to our meetings during these three year and for his unwavering support and motivation when I needed it. I am truly convinced that without the discussions we had, including those about “exotic” fuels, this thesis would have been very much different.

I would like to thank Pat Symonds and Nick Hayes for trusting me for this project, supporting me with the elements I needed to complete my work and for attending our fortnightly meetings. I am very happy to have been able to work with Formula One™.

I would like to also thank Clive Temple for his help during the project and I am sure that he was an important contributor in securing this project at Cranfield University.

I would like to thank all the other persons that I have contacted and that helped me at some stages of this project including Dr Kim Blackburn, Professor James Brighton, Dr John Williams, and Professor Mani Sarathy.

Et finalement, en français, pour remercier mes parents pour leur soutien inconditionnel quel que soit le sujet tout au long de mes études. Sans votre soutien, je n'aurais pas pu poursuivre les études que je souhaitais faire et mon rêve de travailler en F1, une partie de la réalisation de cette thèse vous revient.

TABLE OF CONTENTS

ABSTRACT	i
ACKNOWLEDGEMENTS.....	iii
LIST OF FIGURES.....	ix
LIST OF TABLES	xviii
LIST OF ABBREVIATIONS.....	xx
NOMENCLATURE	xxiii
1 Introduction.....	1
1.1 Context	1
1.2 The role of simulation in high-performance engine development.....	10
1.3 Position of the Formula One™ industry regarding engine development and the arising knowledge	12
1.4 Methodology adopted for this project.....	12
1.5 Objectives of this research.....	13
1.6 Statement on the significant and personal contribution to the body of knowledge.....	15
2 Literature review.....	19
2.1 The future of internal combustion engine.....	19
2.1.1 Place of the ICE in the future energy mix	19
2.1.2 Place of the ICE in motorsport	21
2.1.3 Steps to reduce environmental impact of motorsport	22
2.1.4 Opportunities linked to internal combustion engine technology improvement	23
2.2 Identification of engine improvement areas using thermodynamic engine analysis	26
2.3 High engine efficiency enabling technologies	29
2.3.1 Wide view of the enabling technologies	29
2.3.2 Specific enabling technologies	31
2.4 Evolution of Formula One™ Power Unit regulations.....	43
2.5 Engine modelling in powertrain development processes	50
2.5.1 The different types of engine simulations	50
2.5.2 Cranfield University and 1D engine simulations	56
2.6 Scientific knowledge about the performance characteristics of Formula One™ engines.....	60
2.6.1 Formula One™ engines in the literature.....	60
2.6.2 Cranfield F1™ Design Sprint.....	61
3 Assessment and management of uncertainties for the creation of the 2014+ Digital Twin.....	63
3.1 Categories of parameters defining an engine and associated uncertainties	63

3.2 Design methodologies adequacy to cope with uncertainties and analysis of the 2014+ regulations using engine governing equations	64
3.3 Uncertainty management strategy	68
3.4 Review of the data necessary for the creation of the 2014+ Digital Twin and of the selected values	70
3.5 Assessment of uncertainty significance on the 2014+ Digital Twin performance prediction	77
3.5.1 Analysis of the predictions made for compound parameters.....	77
3.5.2 Sensitivity analysis on the 2014+ Digital Twin parameters.....	83
3.6 Conclusions regarding the uncertainty significance and management ...	91
4 Tools developed for this research project.....	95
4.1 Introduction	95
4.2 Engine governing equations	98
4.3 Introduction to AVL Boost™.....	99
4.4 Steady-state AVL Boost™ models.....	104
4.4.1 Digital Twin of the 2014+ Formula One™ engine.....	104
4.4.2 2026 Formula One™ engine model	107
4.5 Transient AVL Boost™ model of the 2026 Formula One™ Power Unit	112
4.6 Energy and exergy analysis MATLAB tool.....	119
4.7 MATLAB results data extraction tool.....	122
4.8 Engine efficiency chain	123
5 In-depth investigation of the 2014+ Formula One™ engine operation using a Digital Twin.....	125
5.1 Introduction	125
5.2 Path to 50+% brake efficiency: how does the 2014+ Formula One™ engine reach very-high fuel conversion efficiency?	127
5.2.1 High compression ratio.....	131
5.2.2 High dilution	134
5.2.3 Pre-chamber, improved ignition reliability for lean mixtures and shorter combustion duration.....	139
5.2.4 Direct injection.....	145
5.2.5 Asymmetric valve lift profiles (not Millerisation).....	149
5.2.6 MGU-H and unconventional waste gate operation.....	151
5.2.7 Summary and conclusions	159
5.3 Energy and exergy analyses of the 2014+ Formula One™ engine.....	165
5.3.1 Energy and exergy analyses of the CWG and OWG modes.....	167
5.3.2 Energy and exergy distribution of the 2014+ Formula One™ engine compared to that of an engine from the first era of Formula One™ turbocharged era.....	173
5.4 Evolution of turbocharger operating characteristics with varying engine load and influence of the fuel flow limit on the engine behaviour	176

5.5 NO _x production rate of the 2014+ Formula One™ engine compared to conventional road car engines	183
5.6 Conclusions on the insights brought by the 2014+ Digital Twin	185
6 Assessment of the impact of the 2026 Formula One™ Power Unit regulations on engine performance	187
6.1 Evaluation of the performance differences between the 2014+ and 2026 Formula One™ Power Unit Regulations	187
6.2 Effect of combustion characteristic modifications on engine performance and assessment of the most effective parameter to increase fuel conversion efficiency	198
6.3 Requirements for the 2026 Formula One™ engine to run at $\lambda = 1.6$...	204
6.4 Conclusions on the impact of the 2026 Formula One™ Power Unit regulations	211
7 Investigations of different ICE/MGU-K interaction modes to minimise turbo lag in the 2026 Formula One™ Power Unit transient operation	215
7.1 Comparison of the transient performance of a motorsport ICE first fitted with an e-turbocharger and then with a conventional turbocharger with added P1-hybrid electric machine.....	216
7.1.1 Motivation for work	216
7.1.2 Presentation of the models used for this study.....	217
7.1.3 Simulation results – How does an eTC improve engine efficiency?.....	220
7.1.4 Simulation results – Impact of the MGU-K on transient behaviour of the engine and turbocharger	223
7.1.5 Assessment of the difference in engine performance depending on the position of the electric motor on the engine	225
7.1.6 Conclusions on the impact of the electric motor on the ICE	229
7.2 Assessment of the MGU-K as a torque consuming device for the 2026 Formula One™ Power Unit to minimise turbo lag	231
7.2.1 Presentation of the “torque consuming” concept.....	231
7.2.2 Presentation of the torque consuming modes investigated in this case study	232
7.2.3 Modelling of the torque consuming modes using AVL Boost™	234
7.2.4 Analysis of the 2026 Formula One™ Power Unit behaviour depending on the ICE/MGU-K interaction modes for the 70-320 km/h speed event.....	241
7.2.5 Evolution of the impact of the different ICE/MGU-K interaction modes on the 2026 Formula One™ Power Unit behaviour with different speed events.....	259
7.2.6 Conclusions on the usefulness of torque consuming modes.....	261
7.3 Conclusions on the impact and opportunities related to coupling an internal combustion engine with an electric motor	262
8 Conclusions and future work	265

8.1 Conclusions	265
8.1.1 Contribution to the body of knowledge	265
8.1.2 Results achieved relating to the objectives of the research.....	266
8.2 Future work.....	268
REFERENCES.....	271
APPENDICES	287

LIST OF FIGURES

Figure 1-1 - Power evolution of Honda's engines from 1983 to 2008 (top figure) [1] and for BMW between 2000 and 2009 (bottom figure) [2]	2
Figure 1-2 - The different eras of Formula One™ engines.....	2
Figure 1-3 - Power and BMEP predictions for four different Formula One™ engine generations [1]; [6]; [7].....	4
Figure 1-4 - 2014+ Formula One™ Power Unit energy flow diagram [12].....	6
Figure 1-5 - Summary of the main properties of each Formula One™ engine era	8
Figure 1-6 - 2026+ Formula One™ Power Unit energy flow diagram [15].....	9
Figure 2-1 - Automotive Council UK Thermal Propulsion Systems Technology Roadmap [36]	25
Figure 2-2 - Evolution of gasoline and gasoline flame laminar flame speed with equivalence ratio [49]	32
Figure 2-3 - Evolution of the number of pages dedicated to the engine / Power Unit in Formula One™ technical regulations	45
Figure 2-4 - Formula One™ Power Unit regulations' evolution summary	46
Figure 2-5 - Example of Ansys Chemkin's simulation results for propane flame speed calculations (Blue curve: 300 K – 1 bar / Green curve: 700 K – 1 bar)	54
Figure 2-6 - Comparison of Ansys Chemkin propane flame speed prediction (green dots) against literature data (background graphics from [132])	55
Figure 2-7 - 2022 Cranfield F1™ Design Sprint - Step Forward group geometry	57
Figure 2-8 - 2022 Cranfield F1™ Design Sprint – Brute Force group geometry	58
Figure 2-9 - 2022 Cranfield F1™ Design Sprint – Step Forward and Brute Force results comparison	59
Figure 3-1 - Otto's cycle PV diagram.....	66
Figure 3-2 - Comparison between governing equations and 2014+ Digital Twin power predictions	78
Figure 3-3 – 2014+ Digital Twin internal volumetric efficiency predictions	79
Figure 3-4 – 2014+ Digital Twin peak in-cylinder pressure predictions	79
Figure 3-5 – 2014+ Digital Twin heat transfer predictions in CWG mode.....	80

Figure 3-6 - Average over all engine speeds of heat transfer coefficients during a cycle	81
Figure 3-7 – 2014+ Digital Twin mean heat transfer coefficient predictions	82
Figure 3-8 – 2014+ Digital Twin MGU-H power predictions	82
Figure 3-9 - Spider diagram example	84
Figure 3-10 - Sensitivity results regarding effective efficiency predictions.....	90
Figure 3-11 - AVL Boost™ IMEP prediction vs measurement [151].....	92
Figure 3-12 - Relative error between AVL Boost™ prediction and measurement	92
Figure 3-13 - AVL Boost™ NOx production prediction vs measurement [151].	93
Figure 4-1 - Summary of the different levels of engine simulations	95
Figure 4-2 - Effect of initial conditions on transient simulation results	101
Figure 4-3 - Port pressures and temperatures comparisons between steady-state and transient simulations (from top to bottom: 10,000 RPM / 10,500 RPM / 11,000 RPM / 11,500 RPM).....	103
Figure 4-4 - AVL Boost™ layout of the 2014+ engine steady-state model (2014+ Digital Twin).....	104
Figure 4-5 - Geometry of the 2014+ engine steady-state model	105
Figure 4-6 - 2014+ Digital Twin turbine swallowing curve	106
Figure 4-7 – AVL Boost™ layout of the 2026 engine steady-state model	107
Figure 4-8 - Geometry of the 2026 engine steady-state model	109
Figure 4-9 - AVL Boost™ layout of the 2026 engine transient model.....	112
Figure 4-10 - Compressor map (plus steady-state simulation compressor operating points) and turbine swallowing curve for the transient 2026 engine model.....	116
Figure 4-11 - Schematic of the turbocharger used in the transient 2026 engine model.....	116
Figure 4-12 - Example of simulation control leading to compressor surge	117
Figure 4-13 - Example of vehicle speed evolution with the transient 2026 engine model.....	119
Figure 4-14 - Example of exergy analysis bar diagram	122
Figure 4-15 - Example of AVL Boost data extraction using the dedicated MATLAB code.....	122

Figure 4-16 - Engine efficiency chain	123
Figure 5-1 - 2014+ Formula One™ regulations fuel mass flow limit.....	130
Figure 5-2 - Expected window for the engine power curve.....	130
Figure 5-3 - PV and TV diagrams at 10,500 RPM - Baseline engine (Red curve: CR = 10:1) and increased compression ratio (Green curve: CR = 16:1) models	131
Figure 5-4 - Performance improvements brought by a higher compression ratio (Red curve: 10:1 CR / Green curve: 16:1 CR)	132
Figure 5-5 - Lambda evolution implemented in the 2014+ Digital Twin	135
Figure 5-6 - PV and TV diagrams at 10,500 RPM – Increased compression ratio (Green curve: $\lambda = 1$ for all engine speeds) and high dilution (Blue curve: Lambda evolution of the 2014+ Digital Twin) models	135
Figure 5-7 - Performance improvements brought by high dilution (Green curve: $\lambda = 1$ for all engine speeds / Blue curve: Lambda evolution of the 2014+ Digital Twin).....	136
Figure 5-8 - Evolution of gasoline and gasoline flame laminar flame speed with equivalence ratio [49]	138
Figure 5-9 - Increase of combustion cyclic variability with dilution [34].....	139
Figure 5-10 - PV and TV diagrams at 10,500 RPM - High dilution (Blue curve: 50 °CA combustion duration) and short combustion duration (Black curve: 2014+ Digital Twin combustion duration) models	141
Figure 5-11 - Performance improvements brought by short combustion duration (Blue curve: 50 °CA combustion duration / Black curve: 2014+ Digital Twin combustion duration (41 °CA))	141
Figure 5-12 - Sensitivity analysis results on knock onset models.....	142
Figure 5-13 - Impact of the modification of the pressure exponent in Douaud's knock onset model on the octane number requirement predictions.....	144
Figure 5-14 - PV and TV diagrams at 10,500 RPM - Short combustion duration (Black curve: Port injection) and GDI (Pink curve: Direct injection) models	146
Figure 5-15 - Trapped fuel mass flow for port-injection (Black curve: Port injection) and direct injection (Pink curve: Direct injection)	147
Figure 5-16 - Performance improvements brought by direct injection (Black curve: Port injection / Pink curve: Direct injection).....	147
Figure 5-17 - Valve lift profile comparison	149

Figure 5-18 - Improvements brought by asymmetric valve profiles (Pink curve: Conventional intake valve lift profile / Brown curve: Modified intake valve lift profile, early IVC)	150
Figure 5-19 - Modifications brought by the addition of the MGU-H (Brown curve: No MGU-H, normal waste gate operation / Green curve: MGU-H and waste gate closed)	152
Figure 5-20 - Pressure ratio across the engine with and without the MGU-H (Brown curve: No MGU-H, normal waste gate operation / Green curve: MGU-H and waste gate closed)	153
Figure 5-21 - PV diagrams (zoomed on the right graph) at 10,500 RPM – Asymmetric valve profiles (Brown curve: No MGU-H, normal waste gate operation) and MGU-H (Green curve: MGU-H and waste gate closed) models	153
Figure 5-22 - Schematic of the 2014+ Digital Twin behaviour at 10,500 RPM in CWG-recharge mode.....	154
Figure 5-23 - Schematic of the 2014+ Digital Twin behaviour at 10,500 RPM in CWG power mode	155
Figure 5-24 - Internal volumetric efficiency (left) and pressure ratio across the engine (right) evolutions with engine speeds for CWG and OWG modes of the 2014+ Digital Twin	156
Figure 5-25 - PV diagrams (zoomed on the right graph) at 10,500 RPM - MGU-H (Light green curve: Waste gates closed, excess turbine power recovered by the MGU-H) and Open Waste Gate (Dark green curve: Waste gates open, the MGU-H supplies most of the power required by the compressor) models	156
Figure 5-26 - Improvements brought by fully opening the waste gates (Brown curve: No MGU-H, conventional waste gate operation / Light green curve: MGU-H, closed WG / Dark green curve: MGU-H, open WG)	157
Figure 5-27 - Schematic of the 2014+ Digital Twin behaviour at 10,500 RPM in OWG mode.....	158
Figure 5-28 - MGU-H power evolution with engine speed for CWG and OWG modes.....	158
Figure 5-29 - PV and TV diagrams at 10,500 RPM - Conventional (Red curve: Base engine model) and the 2014+ Digital Twin in its highest performance mode (Green curve: 2014+ Digital Twin in OWG mode)	162
Figure 5-30 – 2014+ Digital Twin power and effective efficiency curves for the CWG and OWG modes compared to the expected power window from Figure 5-2	162
Figure 5-31 - PV diagram - Idealised pressure-limited cycles	166

Figure 5-32 - TS diagram - Idealised pressure-limited cycles.....	166
Figure 5-33 – 2014+ Digital Twin energy distribution - OWG mode	167
Figure 5-34 - Evolution of the cylinder surface area with bore for a given volume	169
Figure 5-35 – 2014+ Digital Twin exergy distribution - OWG mode	171
Figure 5-36 – 2014+ Digital Twin energy distribution - OWG and CWG comparison	172
Figure 5-37 – 2014+ Digital Twin exergy distribution - OWG and CWG comparison	173
Figure 5-38 – Energy distribution – OWG mode 2014+ Digital Twin and Honda RA168E comparison.....	174
Figure 5-39 - Exergy distribution - OWG mode 2014+ Digital Twin and Honda RA168E comparison.....	175
Figure 5-40 – Load demand distribution over a lap (x = Load demand)	177
Figure 5-41 - Normalised average engine speed distribution over a lap	177
Figure 5-42 – 2014+ Digital Twin compressor operating points vs engine load and speed.....	178
Figure 5-43 – 2014+ Digital Twin compressor operating points vs engine load and speed (Focus on the engine operating points).....	180
Figure 5-44 – 2014+ Digital Twin compressor operating points vs engine load and speed (Focus on the most used compressor points)	181
Figure 5-45 - MGU-H power evolution with engine load.....	182
Figure 5-46 – 2014+ Digital Twin compressor operating points at 100% load – CWG (red) vs OWG (light blue) modes	182
Figure 5-47 – NO _x production – 2014+ Digital Twin and Honda F20C comparison	184
Figure 5-48 - Specific NO _x production – 2014+ Digital Twin and Honda F20C comparison	184
Figure 6-1 - Fuel mass and energy flow - 2014+ vs 2026 comparison.....	188
Figure 6-2 - Lambda evolution with engine speed - 2014+ vs 2026 comparison	188
Figure 6-3 – BMEP (top left), torque (top right) and power (bottom) curves – 2014+ vs 2026 comparison	189
Figure 6-4 – Peak in-cylinder pressure and temperature curves – 2014+ vs 2026 comparison	189

Figure 6-5 - Effective efficiency and BSFC curves - 2014+ vs 2026 comparison	190
Figure 6-6 - Internal volumetric efficiency and pressure ratio across the engine curves - 2014+ vs 2026 comparison	190
Figure 6-7 - Air mass flow (top left), boost pressure (top right) and compressor power (bottom) curves - 2014+ vs 2026 comparison	191
Figure 6-8 - Cooling losses (top left), intercooler load (top right) and combustion chamber wall heat transfer coefficient (bottom) curves - 2014+ vs 2026 comparison	193
Figure 6-9 - Absolute and specific NO _x production - 2014+ vs 2026 comparison	193
Figure 6-10 - Total power unit power and torque curves (ICE + MGU-K) - 2014+ vs 2026 comparison	194
Figure 6-11 - Average energy distribution - 2014+ vs 2026 comparison	195
Figure 6-12 - Energy distribution at 10,500 RPM - 2014+ vs 2026 comparison	195
Figure 6-13 - Average exergy distribution - 2014+ vs 2026 comparison	196
Figure 6-14 - Average impact of combustion advance, combustion duration and lambda evolution on the 2026 Formula One™ engine performances.....	199
Figure 6-15 - Average exergy repartition of reference combustion advance, combustion duration and lambda evolution modifications compared to the reference 2026 engine model	201
Figure 6-16 - Exchange rates to improve effective efficiency by 1 percentage point using combustion advance, combustion duration and lambda evolution .	202
Figure 6-17 - Lambda evolution with engine speed – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models	205
Figure 6-18 - Air mass flow (top left), boost pressure (top right), compressor power (bottom left) and intercooler load (bottom right) curves – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models	205
Figure 6-19 - Internal volumetric efficiency and pressure ratio across the engine curves – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models.....	206
Figure 6-20 - BMEP (top left), power (top right) and effective efficiency (bottom) curves – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models.....	207
Figure 6-21 - Otto cycle indicated efficiency evolution with lambda	207
Figure 6-22 - Peak in-cylinder pressure and temperature curves – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models	208

Figure 6-23 - Cooling losses and exhaust enthalpy flow curves – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models	208
Figure 6-24 – Absolute and specific NOx production – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models	209
Figure 6-25 – Average energy distribution – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models	209
Figure 6-26 - Average exergy distribution – $\lambda_{\max} = 1.4$ vs $\lambda_{\max} = 1.6$ 2026 engine models	210
Figure 7-1 - AVL Boost layout of the engine model used for investigating the coupling of an electric motor with the ICE (here with an eTC)	217
Figure 7-2 - Engine load evolution.....	220
Figure 7-3 - Engine speed evolution for the reference and eTC cases	220
Figure 7-4 - Turbocharger rotor speed (top), absolute boost pressure (middle) and compressor air mass flow (bottom) evolutions for the reference and eTC cases	221
Figure 7-5 - Lambda and effective efficiency evolutions for the reference and eTC cases	222
Figure 7-6 - Engine speed evolution for the reference, eTC and MGU-K cases	223
Figure 7-7 - Turbocharger rotor speed (top), absolute boost pressure (middle) and compressor air mass flow (bottom) evolutions for the reference, eTC and MGU-K cases	224
Figure 7-8 - Lambda and effective efficiency evolutions for the reference, eTC and MGU-K cases	224
Figure 7-9 – Vehicle speed (solid lines) and electric motor electric power (dashed lines) evolutions for the reference, eTC and MGU-K cases.....	226
Figure 7-10 - Performance indicators for the reference, eTC and MGU-K cases	228
Figure 7-11 - Lambda (top left), effective efficiency (top right), engine speed (bottom left) and turbocharger rotor speed (bottom right) evolutions for the reference case and the reference case with a fuel enrichment limiting strategy.....	230
Figure 7-12 - Different possible MGU-K usages for the 2026 Formula One™ Power Unit	234
Figure 7-13 - AVL Boost™ transient engine model used to investigate the ICE/MGU-K interaction modes	235
Figure 7-14 - Target lambda for torque consuming transient simulations.....	235

Figure 7-15 - Fuelling strategy for torque consuming transient simulations ...	236
Figure 7-16 - Throttle position evolution depending on the speed event	240
Figure 7-17 - Example of equivalent vehicle speed evolution for the 70-320 km/h speed event	241
Figure 7-18 – 70-320 km/h speed event – TFO power and vehicle speed evolutions (1/3)	242
Figure 7-19 - 70-320 km/h speed event – TFO power and vehicle speed evolutions (2/3)	243
Figure 7-20 - 70-320 km/h speed event – TFO power and vehicle speed evolutions (3/3)	244
Figure 7-21 - 70-320 km/h speed event – TCB power and vehicle speed evolutions	245
Figure 7-22 - 70-320 km/h speed event – TCA power and vehicle speed evolutions	246
Figure 7-23 - 70-320 km/h speed event – TCAB power and vehicle speed evolutions	247
Figure 7-24 - 70-320 km/h speed event - Turbocharger rotor speed evolutions of the different ICE/MGU-K interaction modes.....	248
Figure 7-25 - 70-320 km/h speed event – Effective efficiency evolutions of the different ICE/MGU-K interaction modes	249
Figure 7-26 - 70-320 km/h speed event – Turbocharger and effective efficiency characteristics of the different ICE/MGU-K interaction modes.....	250
Figure 7-27 - 70-320 km/h speed event – Fuel and electric energy use of the different ICE/MGU-K interaction modes	253
Figure 7-28 - 70-320 km/h speed event - Fuel penalty vs turbo lag duration of torque consuming modes	256
Figure 7-29 - 70-320 km/h speed event - Fuel penalty vs electric energy deployment for torque filling of torque consuming modes	257
Figure 7-30 - 70-320 km/h speed event - Fuel penalty vs estimated ES energy level reduction of torque consuming modes	257
Figure 7-31 - 70-320 km/h speed event - Acceleration effectiveness of the different ICE/MGU-K interaction modes	258
Figure 7-32 - Evolution of the fraction of fuel consumption dedicated to torque consuming with different speed events.....	259
Figure 7-33 - Evolution of turbo lag time with different speed events	260

LIST OF TABLES

Table 1-1 – 2014 Formula One™ engine efficiency improvement breakdown .	15
Table 2-1 - Notable high engine efficiency enabling technologies.....	31
Table 3-1 - List of the information used to guide the development of the 2014+ Digital Twin	70
Table 3-2 - List of parameters necessary for the 2014+ Digital Twin engine model	76
Table 3-3 - Non-exhaustive list of the 2014+ Digital Twin compound parameters	76
Table 3-4 - Beginning of hybrid era engine cooling loads [5].....	80
Table 3-5 - Summary of the sensitivity analyses conducted to assess input dataset uncertainty significances on performance prediction for the 2014+ Digital Twin	90
Table 4-1 - 2014+ Digital Twin speed-dependent parameters.....	105
Table 4-2 - Properties of gasoline, ethanol, and the 2026 engine model's fuel	110
Table 4-3 - Parameters controlled by the ECU and the Formula Interpreter in transient simulations.....	113
Table 4-4 - Vehicle settings for transient simulations	114
Table 4-5 - MGU-K parameters for the 2026 transient engine model.....	115
Table 4-6 - Measuring points considered on each engine model for energy and exergy analyses.....	121
Table 5-1 - Key technologies of the first generation of turbocharged Formula One™ engines (1977 - 1988)	125
Table 5-2 - Key technologies of the second generation of turbocharged Formula One™ engines.....	127
Table 5-3 - Summary of the properties of each engine model investigated for the 2014+ Formula One™ engine breakdown.....	129
Table 5-4 - Summary of the key solutions enabling the 2014+ Formula One™ engine high efficiency	161
Table 5-5 – 2014+ Digital Twin performance predictions	163
Table 5-6 - Average energy distribution for conventional gasoline and Diesel engines.....	168

Table 6-1 - Average relative evolution between the 2014+ and 2026 Formula One™ regulations.....	198
Table 6-2 - Average relative evolution between the 2026 engine model running at $\lambda_{\max} = 1.6$ compared to the same model running at $\lambda_{\max} = 1.4$ and the 2014+ Digital Twin in CWG mode.....	211
Table 7-1 - eTC electric motor parameters.....	218
Table 7-2 - MGU-K parameters	218
Table 7-3 - Energy consumption during the studied manoeuvre for the reference, eTC and MGU-K cases.....	226
Table 7-4 - Presentation of the ideal energy usage distribution.....	252

LIST OF ABBREVIATIONS

0D	0 Dimension
1D	1 Dimension
3D	3 Dimension
AFR	Air Fuel Ratio
AFR _{st}	Stoichiometric Air Fuel Ratio
AI	Artificial Intelligence
ATDC	After Top Dead Centre
BMEP	Brake Mean Effective Pressure
CA	Crank Angle
CAD	Computer Assisted Design
CFD	Computational Fluid Dynamics
CWG	Closed Waste Gate
DOC	Duration of combustion
ECU	Engine Control Unit
ERS	Energy Recovery System
ES	Energy Store
eTC	Electrically assisted turbocharger
F1™	Formula One™
FEA	Finite Element Analysis
FIA	Fédération Internationale de l'Automobile
FMEP	Friction Mean Effective Pressure
GDI	Gasoline Direct Injection
HCCI	Homogeneous Charge Compression Ignition
HP	Horse Power
HT	Heat Transfer
ICE	Internal Combustion Engine
IMechE	Institution of Mechanical Engineers
IMEP	Indicated Mean Effective Pressure

IVC	Intake Valve Closing
KERS	Kinetic Energy Recovery System
LLNL	Lawrence Livermore National Laboratory
MFB	Mass Fraction Burned
MGU-H	Motor Generator Unit – Heat
MGU-K	Motor Generator Unit – Kinetic
ML	Machine Learning
MSc	Master of Science
MBT	Maximum Brake Torque
NA	Naturally Aspirated
NO _x	Nitrogen Oxides
ON	Octane Number
OWG	Open Waste Gate
PU	Power Unit
ROHR	Rate of Heat Release
RON	Research Octane Number
SoC	Start of Combustion
SOC	State Of Charge
TC	Turbocharger
TCA	Torque Consuming during Acceleration
TCB	Torque Consuming during Braking
TCAB	Torque Consuming during Acceleration and Braking
TFO	Torque Filling Only
TDC	Top Dead Centre
WG	Waste Gate

NOMENCLATURE

$A_{arrhenius}$	Arrhenius law pre-exponential factor	Depends on reaction order
A_{car}	Vehicle frontal area	m ²
A_{HT}	Heat transfer surface area	m ²
A_p	Piston area	m ²
B	Engine bore	m
C_D	Drag coefficient	-
C_L	Downforce coefficient	-
C_p	Constant pressure heat capacity	J/kg/K
E	Energy	J
$\dot{E}$	Energy flow	kW
E_a	Activation energy	J/mol
E_x	Exergy	J
$\dot{E}_x$	Exergy flow	kW
(F/A)	Fuel/Air ratio	-
g	Earth's gravitational acceleration (9.81)	m/s ²
h	Specific enthalpy	J/kg
$\dot{H}$	Enthalpy flow	kW
h_c	Heat transfer coefficient	W/m ² /K
$I_{performance}$	Vehicle acceleration effectiveness	g/MJ
k	Chemical reaction rate constant	Depends on reaction order
m_a	Mass of air trapped in cylinders	kg
$\dot{m}_a$	Air mass flow	kg/s
m_{car}	Vehicle mass	kg
$\dot{m}_f$	Fuel mass flow	kg/s
n	Arrhenius law temperature exponent	-
N	Engine speed	RPM
p	Pressure	Pa

P_b	Brake power	kW
P_f	Friction losses power	kW
P_I	Indicated power	kW
Q_{lvh}	Fuel lower heating value	MJ/kg
r	Specific gas constant	J/kg/K
R	Universal gas constant (8.314)	J/mol/K
r_c	Parameter of order of turbulence integral scale	m
R_f	Flame radius	m
s	Specific entropy	J/K/kg
S	Entropy	J/K
S_L	Laminar flame speed	m/s
$\bar{S}_p$	Piston mean speed	m/s
S_T	Turbulent flame speed	m/s
t	time	s
T	Temperature	K
T_I	Indicated torque	N.m
u'	Turbulence intensity	m/s
V	Vehicle speed	m/s
V_{swept}	Engine swept volume	m ³
w	Local average gas velocity in the cylinder	m/s
γ	Specific heat ratio	-
ε^{chem}	Chemical specific exergy	J/kg
ε^{therm}	Thermomechanical specific exergy	J/kg
η_c	Combustion efficiency	-
η_e	Effective efficiency	-
η_f	Fuel conversion efficiency	-
η_m	Mechanical efficiency	-
η_{th}	Thermal efficiency	-

η_v	Engine volumetric efficiency	-
λ	Relative air/fuel ratio	-
μ	Tyre friction coefficient	-
π	Pi (3.1416)	-
ρ	Density	kg/m ³
$\tau_{ignition}$	Ignition delay	s
φ	Fuel exergy factor	-

1 Introduction

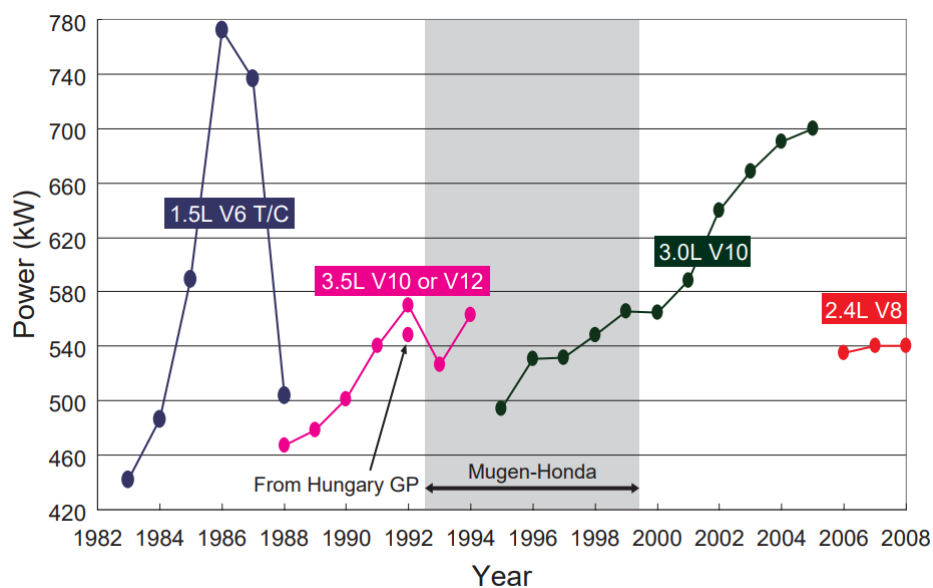
1.1 Context

Since its creation in 1950, and thanks to its teams pushing to the limits to extract as much as possible within regulations, the FIA Formula One Championship™ has been an innovation laboratory, promoting a range of technical solutions now used in the whole automotive industry.

One of the distinctive features of Formula One™, is that small margin improvements in performances are pursued despite cost or inconvenience. The motivation for truly exceptional performance has always been there.

That motivation remains, but every year, sporting, technical and now financial regulations are published by the Fédération Internationale de l'Automobile (FIA), to which teams and engine manufacturers must comply. The more recent editions have aimed at continuing to ensure a fair and secure competition playground for all competitors but also at limiting research costs by excluding research and development in specific technologies or materials for example and, significantly for the topic of this Doctoral thesis, by limiting testing and simulation time and costs.

Despite engine regulations being updated every year, an engine generation lasts for several years with the core concept staying the same and limited modification or precision being brought to specific items. Performance from the end-of-generation engines has always been significantly higher than start-of-generation ones due to constant improvement motivated by the desire of winning races as shown in Figure 1-1 which is extracted from the papers published respectively by Sakurahara reporting the power evolution of Honda's Formula One™ from 1983 to 2008 [1] and by Theissen et al. regarding BMW's engine between 2000 and 2009 [2].



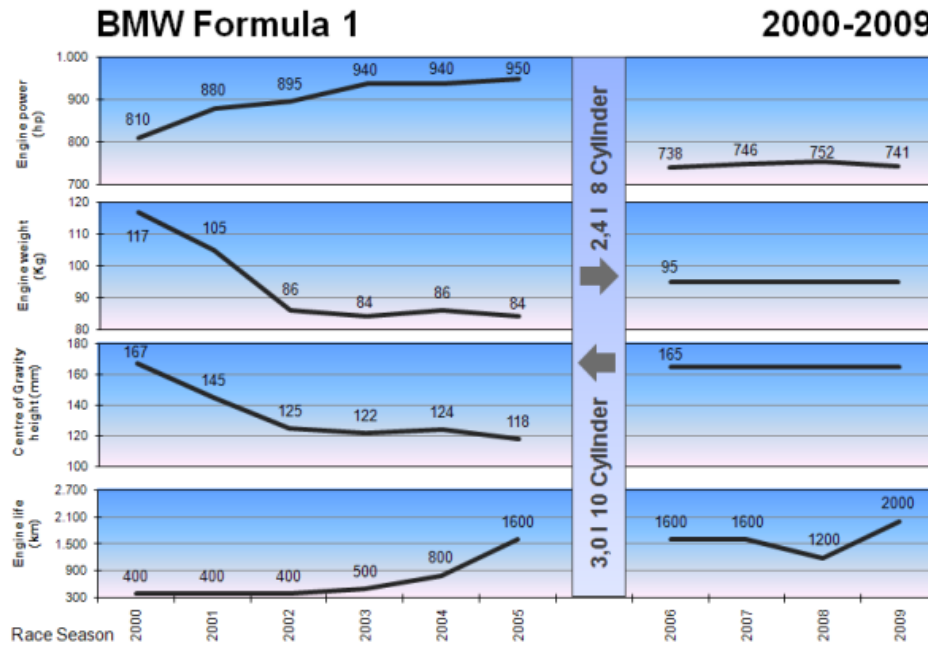


Figure 1-1 - Power evolution of Honda's engines from 1983 to 2008 (top figure) [1] and for BMW between 2000 and 2009 (bottom figure) [2]

Prior to 2014, both forced induction and naturally aspirated (NA) engines had been used in the Championship, under different forms and at different periods. Both the turbocharged and naturally aspirated engines were high-performance, but they were also air-limited by one form of regulation or another, meaning that their performance was limited by the amount of air that could be trapped in the cylinders (from 1987 limited by regulations on air inlet conditions or engine speed or both) and their thermo-chemical fundamentals were predictable and widely understood by anyone familiar with high-speed engines, particularly motorcycles. The proposition made here is that the secrets kept by engine manufacturers mostly concerned the details at component-level product engineering: the materials, fuel and lubrication science, and the practical know-how of engine calibration rather than relying heavily on the underlying thermo-fluid science of the engine.

The evolution of Formula One™ engines can roughly be described by four main eras which are presented in Figure 1-2:

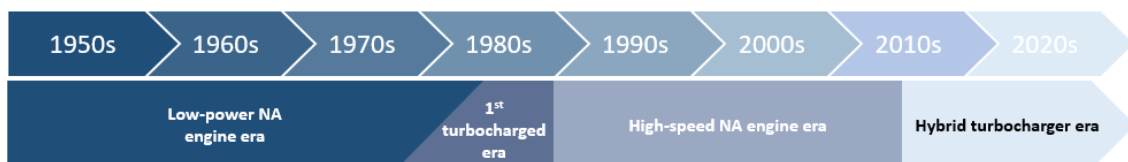


Figure 1-2 - The different eras of Formula One™ engines

At the beginning of Formula One™, both naturally aspirated and supercharged engines were allowed. These engines existed under various forms (architecture, swept volume) and right after World War II, Formula One™ engines were

reasonably powerful, the BRM P14 V16 being reported at 600 HP (1947 - 1955). But over the years, the swept volume had been progressively reduced which also progressively reduced output power. In 1965, the Lotus 33 which won the Championship with Jim Clark was powered by a 1.5L V8 FWMV Coventry Climax engine producing a very modest 158 kW [3].

In 1966, faced with the competition of sports cars, the regulations changed to allow again both turbocharged (1.5 L) and naturally aspirated (3.0 L) engines, which will lead to the creation of the famous Cosworth DFV engine. It is only in 1977 that an engine manufacturer, Renault, tackled the challenge of a turbocharged engine. After a rough start [4], the turbocharged engines matched the performance of their naturally aspirated rivals and proved their superiority by 1983. Despite no regulations amending engine architecture, most of these turbocharged engines were V6 and 4-inline with BMW's 4-inline being the first turbocharger engine to be crowned champion in 1983 (claimed both the Drivers' and Constructors' Championships).

With the change in regulations allowing naturally aspirated engines to vary from 1.5 L to 3.0 L, the power figures went up but this first generation of turbocharged Formula One™ engine went much higher.

In qualification mode, the turbocharged engines were run at over 5 bar of boost pressure to achieve published power figures as high as 780 kW (see Figure 1-1 [1]), but rumoured to have exceeded 1,000 kW [5], requiring very large tyres and wings to be able to transfer the power to the ground. Nevertheless, with these operating conditions, the engines were extremely unreliable with only a few laps of lifespan, meaning that the boost pressure had to be significantly reduced for the engine to last a whole race.

This first generation of turbocharged Formula One™ engines featured conventional turbochargers but relied on highly engineered fuels with high octane ratings (with high proportions of toluene and other aromatic components) burnt at or below stoichiometric air/fuel ratio [6]. By running $\lambda \leq 1$, the thermal efficiency is sacrificed but the loss in released fuel energy was made up by injecting large quantity of fuel and reliable combustion was ensured in addition to other advantages like fuel cooling effects.

In 1987, a two-year period of transition began which eventually saw the ban of turbocharged engines and 3.5 L naturally aspirated engines becoming mandatory. During those two years, a boost pressure limit of 4.0 bar and then 2.5 bar was introduced, inevitably leading to a significant reduction in engine power as demonstrated in Figure 1-1.

As shown in Figure 1-3, the turbocharged engine developed by Honda for the 1988 Formula One™ season (Honda RA168E) produced between 20 and 25 bar BMEP and a maximum of 504 kW [6] with a fuel mass flow around 140 kg/h. These data are obtained using a 1D gas-dynamic engine performance model of this engine developed for this thesis using AVL Boost™ and which is validated against the power curve given by Otobe et al. [6]. Obviously, without boost pressure limit, the engines from the 1st turbocharged era achieved much higher

experience, the realistic maximum of 130% volumetric efficiency is achievable with separate intake pipes for each cylinder that are optimised for length, leading to modest BMEP values compared to that of turbocharged engines. To produce high power values, naturally aspirated engines relied on very high engine speeds which are mostly limited by the mechanical limits of specific components like the piston and the connecting rod. A maximum of over 20,000 RPM was achieved before successive regulations limited the engine speed to 19,000 RPM in 2007 and 18,000 RPM in 2009 with the aim to reduced engine development costs [1].

Despite their lower BMEP, at the end of the V10 era, power output was approaching the values that were reached by turbocharger engines. As shown in Figure 1-1, the 2005 Honda engine produced 700 kW. In addition, between 1990 and 2005, the specific power output of Honda's V10s nearly doubled, from 143 to 240 kW/L, with increased reliability, from 400 to 1400 km, and significant weight reduction, from 160 to 89 kg, demonstrating the possible extent of improvement over an engine generation [1].

In 2014, the regulations changed once again and this time very significantly [8]. 1.6 L V6 turbocharged engines became mandatory alongside an Energy Recovery System (ERS) to form a wider propulsion system referred to as a Power Unit (PU). Extending the experience gained with the KERS (Kinetic Energy Recovery System) used in the early 2010s, Formula One™ cars were now powered by electric-hybrid powertrains.

In addition to this change in Power Unit concept, the engine philosophy was also completely modified with the introduction of a fuel flow limit and of the MGU-H (an electric motor linked to the turbocharger rotor shaft). For the time, the engine is limited in the fuel flow it can consume, therefore the engine power output is not limited by the amount of air it can swallow (air-limited engines). Previous regulations achieved effective air limitation by restricting engine speed or by imposing a limit on inlet manifold pressure. In addition to introducing a fuel-flow limit, the quantity of fuel that can be consumed during a race has also been limited (100 kg in 2014 [9] / 110 kg in 2021 [10]). The fuel mass limit has since been removed (2022). Indeed, there can be a lap time gain in having less fuel mass onboard even if it means limiting engine power at specific moments. Therefore, it could be argued that limits on the quantity of fuel allowed on board a vehicle and a restriction on refuelling are also forms of effective fuel limitation, particularly when re-fuelling was also banned by regulations. This was a radical change.

While of secondary importance for previous engine generations, fuel conversion efficiency became the engine designers' primary focus to make the most out of the limited amount of energy available. In this process of increasing fuel conversion efficiency, the regulations also crucially made mandatory, for the first time, the use of direct injection which enables more precise fuelling control. Despite using specifically developed fuels, their composition is not as different to that of road cars as it was in the first era of turbocharged engines. Shell, Scuderia Ferrari's fuel supplier, claims that its Formula One™ fuel contains "99% of the same types of compounds as Shell V-Power road fuels available to customers worldwide" [11].

In the 2014+ PU (refers to the 2014 – 2025 Formula One™ Power Unit generation), both the MGU-K (electric motor linked to the engine crankshaft) and MGU-H can act in torque delivery and torque absorption modes (recovering kinetic and thermal energy respectively). Unlimited amount of energy can be transfer between both MGUs but the energy flow between the MGU-K and the Energy Storage (ES – the “battery”) is limited, both in and out of the ES, on the contrary to that of the MGU-H (see Figure 1-4). Moreover, the difference between the minimum and maximum State of Charge (SOC) of the ES must never exceed 4 MJ when the car is on track. All these features made the overall energy management of the 2014+ PU very complex and key for on-track performance.

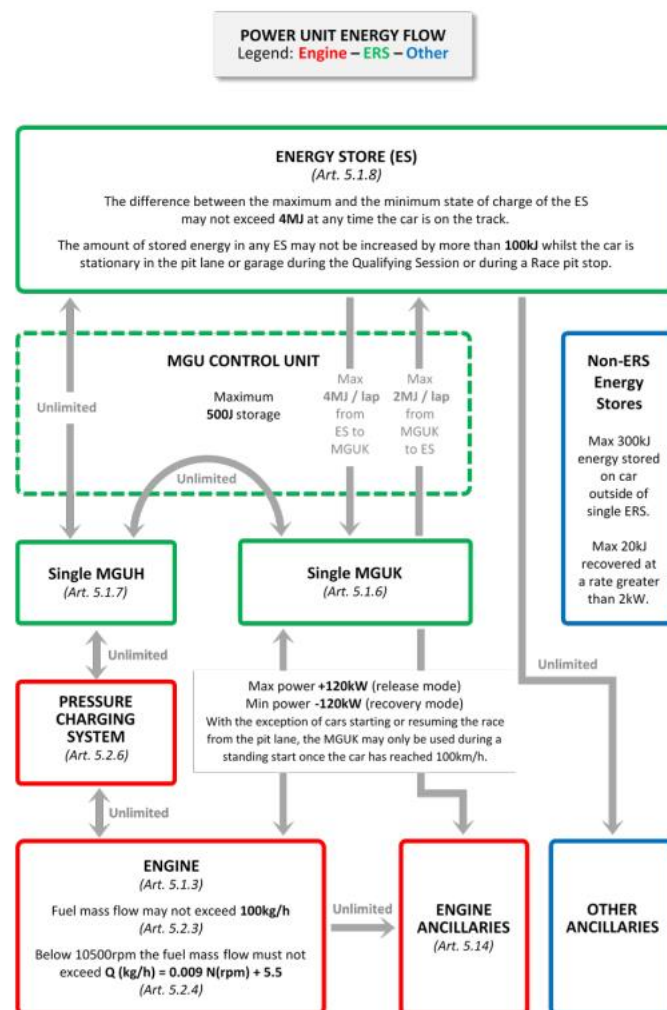


Figure 1-4 - 2014+ Formula One™ Power Unit energy flow diagram [12]

Furthermore, in consequence of the limited available fuel flow, the rapid move in the direction of hyper-dilution (very high air/fuel ratio), to maximise fuel energy release (combustion efficiency) and conversion of heat into work (thermal efficiency), means a totally different combustion system had to be imagined to promote rapid combustion thanks to high turbulence intensity; without direct-injection, the use of a pre-chamber (spark-plug surrounded by a perforated gap) would not have been possible. Indeed, to operate reliably, the mixture in the pre-chamber needs to be rich with the rest of the combustion chamber running at

higher air/fuel ratio than stoichiometry, which requires a precise position and operation from the sole injector allowed per cylinder.

With hyper-dilution combined to high compression ratio and short combustion duration enabled by the pre-chamber, in-cylinder pressures can reach up to 300 bar (without considering connecting rod deformation) where current manufacturing processes meet their mechanical limits. Thus, the 2014+ engine is also pressure-limited, meaning that the efficiency cannot be further increased because of mechanical reasons.

When first introduced, the 2014+ regulations featured a 15,000 RPM engine speed limit which has since been removed as the engine is known to be barely operated above 12,500 RPM [5]. Because of the 100 kg/h fuel flow limit, the output power cannot be increased by injecting more fuel, and maintaining reliable lean combustion at higher engine speeds is very difficult. Compared to the high-speed V10 and V8 eras, the 2014+ engine is limited by pressure forces rather than by inertia forces.

As the philosophy and complexity of the Power Unit completely changed compared to previous Formula One™ and known commercial engines, it became difficult, using logic and experience alone, for an experienced engine designer to imagine or predict how this new Power Unit worked in all of its details.

According to the model of the 2014+ engine developed for this research project, this new generation of turbocharged engines produces more power out of a lower fuel flow than the last engine of the first turbocharged era (Figure 1-3) demonstrating its higher efficiency. In addition, the 2014+ engine higher BMEP despite the higher swept volume highlights that by running lean mixture, high compression ratio and a pre-chamber ignition system, the combustion applies a higher stress on the engine.

With the limited fuel mass flow and despite its higher efficiency, the 2014+ engine, cannot compete with the 780 kW produced by the most powerful engines of the first turbocharged era (Figure 1-1). Nevertheless, by considering the MGU-K, the Power Unit can deliver higher power values, closing the gap between both generations. With the introduction of a hybrid Power Unit, the total output power is now a balance between electric and thermal power.

However, another significant difference compared to the first turbocharged era is reliability. The 780 KW announced for Honda's 1986 engine is a peak value only achievable during qualifications for a very limited number of laps before engine re-build. The engine power output was significantly reduced to survive race distance and the engine could be changed after each session. Even if the 1,000 hp reported by engine manufactures for the 2014+ generation [13]; [14] is also considered to be a peak value, only achieved when the compressor is powered by the MGU-H and the waste gates are fully open, the 2014+ Power Unit can operate at high stress levels and deliver high levels of power for much longer period with only four internal combustion engines allowed for the 2023 23-race Formula One™ season.

A summary of the main properties of each Formula One™ engine era is presented in Figure 1-5:

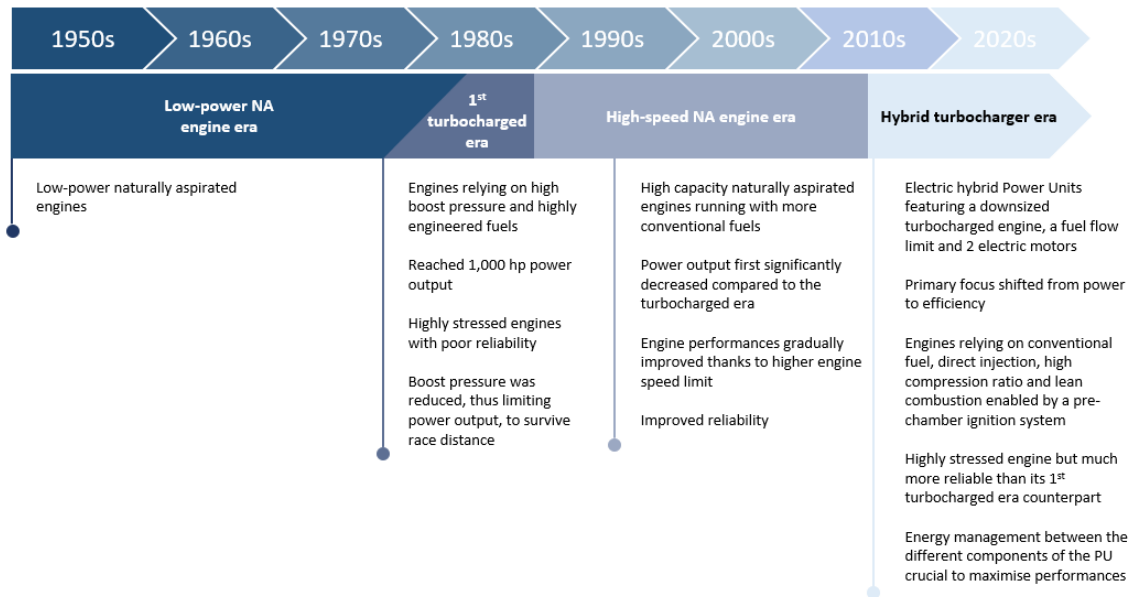


Figure 1-5 - Summary of the main properties of each Formula One™ engine era

In 2026, a new set of regulations of Formula One™ PU will be introduced [15]. The engine will keep the same architecture as in 2014+ but is only allowed a conventional turbocharger with the MGU-H ban, and the MGU-K power is increased from 120 to 350 kW. The regulations will still impose a fuel flow limit (no fuel mass limit announced) but based on energy flow rather than on mass flow for an overall lower fuel limit than in 2014+, see Figure 1-6. On the contrary to the 2014+ Formula One™ Power Unit, the 2026 engine and electric system will be distinct parts of the Power Unit.

By featuring a conventional turbocharger without electric assistance, the 2026 engine will be exposed to turbo lag unlike the 2014+ engine. For the last decades, Formula One™ engines have relied on rather conventional fuels but by setting an energy flow limit and only allowing advanced sustainable fuels, defined as “derived from a carbon capture scheme, a renewable fuel of non-biological origin (RFNBO), municipal waste, or non-food biomass” [15], the regulations reopen the competition for highly engineered fuel development.

Like for the 2014+ regulations, achieving lean combustion will be key to maximise fuel conversion efficiency and extract the maximum out of the limited amount of fuel available. Understanding the new advanced sustainable fuels to achieve reliable, efficient, and stable combustion will be of primary importance.

In consequence of the fuel flow reduction, the engine model built for this project and based on the 2026 regulations inevitably predicts a reduction in power output despite fuel conversion kept high (Figure 1-3). To compensate the loss of thermal power, the MGU-K power has been more than doubled to achieve similar Power Unit power to that of 2014+ as shown in Figure 1-3 but with a different thermal / electric balance.

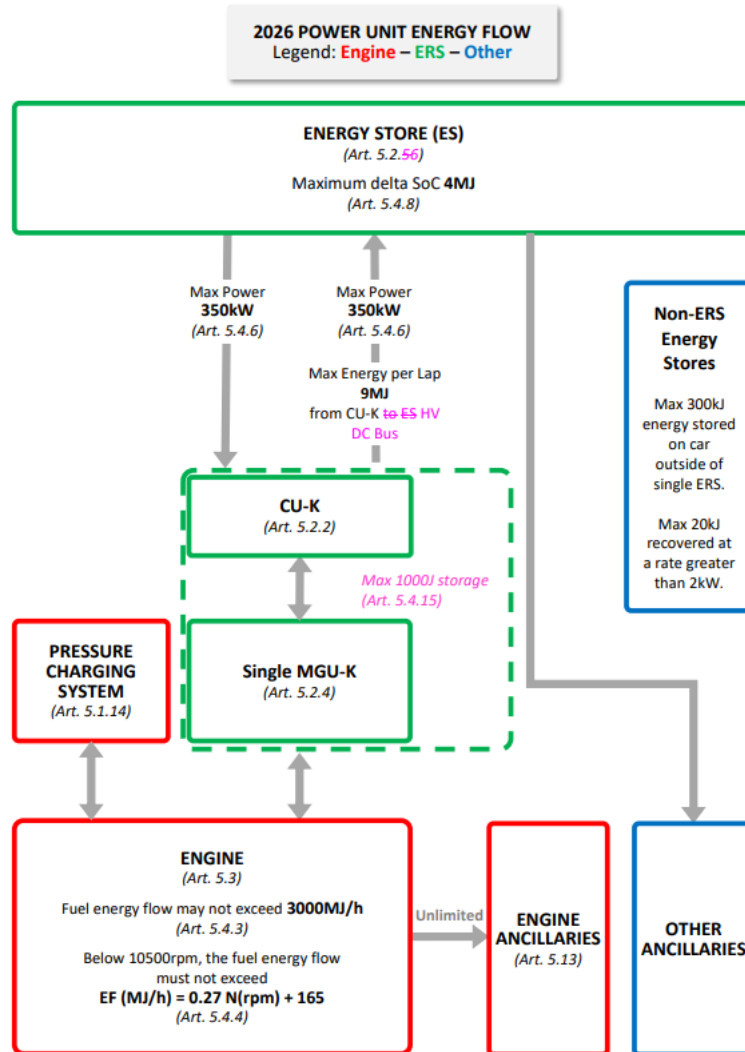


Figure 1-6 - 2026+ Formula One™ Power Unit energy flow diagram [15]

As the architecture does not change but power is lower, the BMEP of the 2026 engine is reduced to the level of a conventional high-performance engine. Very different to its predecessor, the 2026 engine will be less challenging to design from a mechanical point of view as engine speed is lower than that of the high-speed naturally aspirated engine and it will not be pressure-limited because of the lower in-cylinder pressures due to the fuel flow reduction. Whilst there are always reliability issues to remember, this leaves room for the engine designers to push combustion characteristics (dilution and combustion duration for example) further.

The 2026 engine will face three main challenges: minimising turbo lag, developing advanced sustainable fuel allowing engine designers to push combustion further and energy management (including overall cooling loads) to ensure that the 350 kW MGU-K can be fed with enough electric energy considering that all the electric energy comes indirectly from fuel energy.

Due to the 2014+ Formula One™ Power Unit complexity, the Cranfield F1™ Design Sprint was created in 2018 at Cranfield University to try to understand the secrets of this engine year on year using advanced engine simulations to complete / understand what was being published or spoken about with these engines.

The Cranfield F1™ Design Sprint is an extra-curricular student-led activity aiming at establishing an independent scientific appraisal of the development of Formula One™ powertrains. Every year, Cranfield students of the Advanced Motorsport Engineering and Mechatronics MSc are invited to participate voluntarily in this project run for two months. To keep its independent view, the project does not receive any data input from the Formula One™ industry but each year, the results are shared with members of that industry for feedback and advice regarding improvements. In addition to the work done on the understanding of the Formula One™ engine, this activity trains future engineers of the industry and has made a significant contribution to the development of pedagogy at Cranfield University.

Due to the intrinsic competitive aspect of the sport, engine manufacturers do not share details regarding their engine and the main knowledge transfer within the industry happens when engineers move from one manufacturer to another. As done with the F1™ Design Sprint, the only way to look deep into the black box which is the 2014+ Formula One™ engine and to expose its key features is to run the best available engine simulations and build the knowledge step by step.

1.2 The role of simulation in high-performance engine development

In 2000, very few people in Formula One™ and in the motorsport industry was using engine performance simulations, including 1D simulations, for engine development. Engine dynamometer testing, track testing and know-how underpinned the development effort.

Since, the sport has massively changed with the introduction of test restrictions, the cost cap, and the increase in computation power. Engine simulations, 1D and CFD (Computational Fluid Dynamics), are now used by everyone and are crucial for improved and faster engine development.

Another argument for this massive adoption is the improvement of engine simulation software which have been tested and validated for more than two decades. There is now good confidence across the automotive and motorsport industry in engine simulation results to help and increase engine design capabilities.

This adoption has also been allowed by the increased capabilities developed by external institutions like Cranfield University. Cranfield University has strong expertise and capacities in simulation across different fields (engine, CFD, FEA for instance) and has a pedagogy built around heavy use of simulation for

teaching but also group and individual projects. As students are trained to use simulation tools, the industry benefits from this knowledge.

When it comes to 1D engine simulations, Cranfield University distinguishes the learning potential from different approaches:

- The Step Forward philosophy: taught for generations at Cranfield University and other universities, this approach is the classic method and is very similar to what is done when only physical testing is available for development. Based on simulation results, rapid design decisions are made and tested. Those that seem promising are continued while the others are rejected and so the design 'steps forward' one step at a time. Along the route, a mental map of the engine behaviour is built and orientates the designer's decisions. The results are not necessarily optimum as all variables are not optimised simultaneously. But improvements on the baseline are found and there is confidence in the designer that the reasons for the improvements are understood as he / she had to direct himself / herself the development. This philosophy is the one deployed consistently (more latterly amongst others) during the Motorsport Powertrain Design module at Cranfield University and used during the F1™ Design Sprint.
- The Brute Force philosophy: Enabled by recent advances in high computation power and investment in user licences for simulation software, an attempt at true multi-variable optimisation is made by testing a very large number of combinations of variables values. Nevertheless, the human designer is still in charge of deciding which tests are to be undertaken and then of analysing the generated results to extract the preferred design. Improvements on the baseline engine performance are made but to understand where they came from, the designer needs to step back by perturbing key variable values and assessing the performance impact. This approach has been tested during the 2022 Cranfield F1™ Design Sprint and has been used for several MSc thesis at Cranfield University.
- The Computer Guided Design Optimisation philosophy: This approach relies on algorithms to direct an optimal search against a composite objective function within user-defined limits. The objective is a set of weighted performance indicators forming a single composite objective. Like brute force in that a global optimum is sought, it relies less on human intervention. Tracking the rate of change in the objective function against changes in variables selected by the algorithm allows the designer to follow how the code is improving the design over a large period (72 hours for example). Despite being less intuitive and less suitable as a learning tool compared to the Step Forward approach, this philosophy tolerates having a very sparse input data as it starts with a 'blank sheet' of parameters and can lead to design that a designer would not have thought of. This approach has been the subject of the Frankie Chapman's MSc thesis [16].

As witnessed during the 2022 Cranfield F1™ Design Sprint and shown in Chapter 2, these different approaches are not incompatible but rather complementary. The group of students following the Step Forward philosophy would have like the computation power of the Brute Force group whereas the latter would have benefitted from the deeper engine behaviour understanding of the Step Forward students.

The engine design process can benefit from the different philosophies which can be used at different steps of a project.

1.3 Position of the Formula One™ industry regarding engine development and the arising knowledge

When developing their engine, the manufacturer's goal is very simple, this is to make the car go faster. Engineers look for high-performing and reliable technical solutions which improve the overall vehicle / Power Unit package with the stopwatch being the only judge. As Nick Hayes would say "In every decision, we will choose the option that makes the car go faster on the track" [5].

Nevertheless, despite pushing engine concepts to their limit, scientific knowledge is extracted only as a biproduct of this approach as this is not the objective. The industry does not have time to delve into scientific details if in-depth analysis is not required to improve the technical solution.

In addition, if ever detailed scientific investigations are conducted, their results are kept internally and only rarely shared with the scientific community once the PU regulations change, and the solutions become obsolete.

Information about the 2014+ Formula One™ Power Unit can be found in the specialised press with articles about different components. However, these articles are written based on what the teams and engine manufacturers are willing to share, i.e. no intellectual property or information that could be detrimental if in their competitors' hands. Their aim is to share the information to a wider audience but do not provide any scientifically based explanations and can sometimes be erroneous.

Teams and the sport's institutions (FIA and Formula One™ Management) communicate on the engine's performance to promote the sport. The two most famous totemic claims regarding the 2014+ Formula One™ PU are its 50% overall efficiency and 1,000 hp power [13]; [14] but no hard evidence, like engine dynamometer results, have ever been shared.

1.4 Methodology adopted for this project

Because of the secrecy in Formula One™, the lack of validation data available in the literature and no knowledge of component-level details, this project does not

aim at creating an engine model perfectly replicating the performance of the 2014+ engine in all aspects.

Rather based on years of feedback from the industry for the Cranfield F1™ Design Sprint and input from the sponsor of this Doctoral Thesis, the idea is to create a Digital Twin¹ of the 2014+ Formula One™ engine to expose its interesting features.

The objective of the 2014+ Digital Twin is thus not to perfectly replicate the performances of the 2014+ engine but to be a model exhibiting the same trends, allowing to understand the behaviour of an engine following the 2014+ regulations in a systematic way.

Furthermore, the 2014+ Digital Twin allows the engine developer to access data which is normally unavailable or not measured on a real engine, enabling to push investigations further and to get more insights on the engine's behaviour.

For instance, the 2014+ Digital Twin can give insight into detailed turbocharger operational characteristics which are normally only measured if placed on a separate turbocharger dynamometer like the equipment available at the Imperial College London [17]. The 2014+ Digital Twin can be equipped with temperature measurement points in large number and/or in places unreachable on a real engine, extending the range of possible analyses, including exergetic analysis.

The 0D/1D engine simulation software AVL Boost has been used to create this 2014+ Digital Twin with the previously introduced Step Forward approach being used to develop the engine when required (for instance, for maximising the volumetric efficiency). This method provides a layered approach to building up understanding on how the engine behaves. In addition, 1D simulations has also been used backwards. Rather than using the simulation results to influence the engine design, results have been used for investigation and understanding of the engine, to gather insights into the engine behaviour.

The 2026 Formula One™ engine model investigated in this thesis has been created following the same Step Forward methodology approach and AVL Boost™.

1.5 Objectives of this research

The operation of the current Formula One™ engine differs greatly from that of conventional road car and even high-performance engines because of unusual features like the fuel flow limit and the MGU-H. Due to the intrinsic secretive aspect of the sport, very few data regarding these engines is available and the

¹ The term 'Digital Twin' refers to the best equivalent engine model of an existing engine which has benefited from input and feedback from the industry and for which the impact of the input dataset on performance predictions has been critically assessed.

public must satisfy with the performance claims made by engine manufacturers. Furthermore, the internet is full of partial, misleading and incorrect data.

Specific independent research actions, like the Cranfield F1™ Design Sprint, manage to step by step uncover the mystery surrounding the 2014+ Formula One™ Power Unit and the engineering solutions allowing it to reach high-performance and efficiency. But, to date, a precise scientific thermodynamic explanation / understanding of the 2014+ engine uncommon operation does not exist. More generally, the same observation can be made if considering an engine combining all the following characteristics: lean mixture, pre-chamber ignition system, direct injection, high compression ratio, electrically assisted turbocharger and waste gates used as pressure-relief valves.

In 2026, Formula One™ will introduce new Power Unit regulations. The engine will still be fuel-limited but will run with advanced sustainable fuels and the turbocharger will not be allowed to receive electrical assistance. In addition, the electric / thermal balance will greatly evolve compared to the 2014+ regulations. Extensive literature is available on electric hybrid powertrains, however such a technology combination has never been investigated for high-performance motorsport applications, specifically the potential interaction modes between the electric motor and the internal combustion engine to enhance the performance of the whole Power Unit.

Using steady state and transient 0D/1D engine simulations, including the 2014+ Digital Twin, the objectives of this thesis are the following:

- To create a refined Cranfield 2014+ Formula One™ engine Digital Twin using the Step Forward methodology,
- To get insights on the uncommon behaviour / operation of the 2014+ Formula One™ engine using simulation results of the 2014+ Digital Twin,
- To provide a scientific explanation on how the technical choices made for the 2014+ Formula One™ engine enable it to reach very high level of efficiency,
- To predict how the 2014+ technical solutions can be transferred to the future generation of Formula One™ Power Unit which will be introduced in 2026 and investigate how the 2026 Formula One™ Power Unit could evolve in its own regulation framework,
- To investigate how the 350 kW MGU-K featured in the 2026 Formula One™ Power Unit can be used to improve the performance of the internal combustion engine and specifically, how it can interact with it to mitigate turbo lag.

1.6 Statement on the significant and personal contribution to the body of knowledge

The work exposed in this thesis presents the following personal and original contribution to the scientific understanding of the 2014+ and 2026 Formula One™ engines:

- Compared to a conventional engine, i.e. stripped from all mentioned enabling technologies and similar to Formula One™ first turbocharged era engines (see Section 5.2), the 2014+ Formula One™ engine can achieve high level of efficiency thanks to high compression ratio, high dilution, a passive pre-chamber ignition system, direct injection, asymmetric valve profiles, an electrically assisted turbocharger combined with pressure relief waste gates which respectively increase effective fuel conversion efficiency by the values shown in Table 1-1 (Chapter 5),

Enabling technology	Percentage point increase in effective fuel conversion efficiency in average over all simulated engine speeds brought by each solution (Baseline engine effective efficiency: 34.9%)
High compression ratio	3.62
High dilution	4.07
Passive pre-chamber ignition system	0.57
Direct injection	1.38
Asymmetric valve profiles	0.12
Electrically assisted turbocharger with pressure relief waste gates	1.96 (if WG closed and the MGU-H sends energy to the MGU-K) 1.75 (if WG open and the MGU-H supplies energy to the compressor)

Table 1-1 – 2014 Formula One™ engine efficiency improvement breakdown

- By exploiting the Power Unit Energy Flow diagram from the 2014+ Formula One™ regulations and the unlimited link between the MGU-H and the MGU-K, the energy of the exhaust gas can be recovered and subsequently transferred to the crankshaft to increase the engine effective efficiency by 3.1 percentage points across engine speeds, This is a form of turbo-compounding where it is adopted because the energy gain outweighs the losses in converting from mechanical energy to electrical and back again. This is a sustainable mode of operation (at least in steady state conditions), without requiring any energy input from the ES, making it the most efficient engine mode (Chapter 5),

- Influenced by the fuel flow limit and the MGU-H introduction, running the 2014+ engine at high dilution leads in average over all simulated engine speeds to only 7.5% of the fuel energy (3.4% exergy) to be transferred to the cooling system, increasing the exhaust enthalpy to 40.9% (21.7% exergy) in its highest engine performance mode, resulting in higher exhaust energy recovery potential and reduced cooling requirements (Chapter 5),
- In addition to reliably igniting lean mixture, the 2014+ Digital Twin shows that the pre-chamber ignition system lowers the engine's knock sensitivity which enables the use of engineering solutions resulting in higher achievable peak cylinder pressure and higher efficiency as reported in the literature for conventional road car engines (Chapter 5),
- The 2014+ and 2026 Formula One™ engines will have similar thermodynamic behaviour (energy and exergy) but the latter will not be pressure-limited, opening the way for higher dilution levels and shorter combustion duration (Chapter 6),
- Introducing speed targets to 1D simulations, i.e. switching from steady state to transient simulations, massively extends the usefulness of the engine models by exposing additional characteristic of the engine behaviour (Chapter 7),
- Coupling an electric motor to a turbocharged, fuel flow limited internal combustion engine, either at the turbocharger shaft or at the crankshaft, helps to improve acceleration performances and fuel conversion efficiency but in different ways (Chapter 7),
- In the 2026 Formula One™ Power Unit, by using the MGU-K to torque consume, i.e. deliberately absorb energy produced by the engine, turbo lag and the dependency on the MGU-K to accelerate the car can be reduced, but with a higher fuel consumption penalty. The impact is different depending on the ICE/MGU-K interaction mode chosen but torque consuming during acceleration (TCA) is the most promising mode (Chapter 7),
- According to the engine models developed for this thesis, the 2014+ and 2026 Formula One™ engines produce similar amount of specific NO_x (NO_x production rate / output power) than a conventional road car engine before after-treatment system (Chapters 5 and 6),

- This Doctoral Thesis demonstrated the robustness of the currently available 0D/1D engine simulation software which managed to successfully model the behaviour of high-performance Formula One™ engines despite the sparse input dataset (Chapter 3).

2 Literature review

2.1 The future of internal combustion engine

2.1.1 Place of the ICE in the future energy mix

Greenhouse gases emissions overview

With the issues regarding climate change being more and pressing in society, all sectors of the economy having a significant impact on greenhouse gas emissions are examined to try to reduce their emissions and to decarbonise them.

The transport sector is the biggest source of emissions in Europe with 27% of the total emissions and it is the only sector whose emissions have increased since 1990 [18]. The observation is very similar in the United States with the transportation sector representing 29% of the US greenhouse gases emissions in 2021 and recording the highest increase than any other sector between 1990 and 2021 [19].

The emissions of the transport sector come mainly from the combustion of fossil (and more recently other organic) fuels which contains carbon atoms and thus leads to the emissions of carbon dioxide (CO₂). It is important to note that the increase in CO₂ emissions is for the transport sector as a segment of the economy. Individual vehicles in Europe have been subject to ever tightening pollutant emissions since the arrival of Euro emissions standards in 1991. These drove technological development such as standardisation on closed-loop-control fuel injection which has been an important building block for subsequent improvements in specific fuel consumption and hence CO₂ emissions [20].

Decarbonising the transport sector is a big issue and is critical not to exceed the scientists' global temperature increase target. In response to the problem, the European Union has proposed that Member States ban the sale of new petrol and diesel cars from 2035 [21]. The solution which has been the most adopted for now is the promotion of zero-emissions vehicles, mainly electric vehicles, whose operation does not emit any greenhouse gases at the vehicle source.

The potential of the internal combustion engine

However, for internal combustion engine experts, this technology has not reached its thermal efficiency limit yet. There is room for improvements if new engineers/researchers are educated on this field [22]; [23]. especially when considering that internal combustion engines are used for light duty vehicles but also for heavy duty ones and energy production for which zero emissions technologies are not yet ready.

As exposed by Reitz et al. [19], "a sustainable mobility future will require a diverse portfolio to ensure the right technologies for the right applications". Internal combustion engines will be part of the solutions but will not have the monopoly

they have today. On the contrary to the other technologies currently developed, the internal combustion engine has a maturity that makes it usable for a wide range of applications and at low cost. By further developing it and increasing its efficiency, the internal combustion engine can accompany the development of new solutions until these are ready and participate in the first steps towards decarbonisation.

The modifications of the transport sector will come from the development of new technologies but will also be based on the development of new infrastructures that will make these technologies usable [23].

Global warming and the social dimension

The social dimension on the transport sector must also be considered. As explained by Thøgersen et al. [24], the issues linked to climate change requires individuals, companies, and institutions to change their behaviour. Fossil fuels are currently fundamental fuels for our society so actions across all levels of society are required to reduce emissions. The study shows that despite climate change being a concern, it does not yet have a big influence on people's choice regarding their way of commuting or travelling for leisure. Travel modes are influenced by physical reasons (distance to cover, availability or not of an ICE car, infrastructure) but also by personal/social norms. To change people commuting and travel habits, more climate-friendly transport means must be made more attractive (availability, cost) and the people perception of the different transport options must be changed.

In addition, as highlighted by Reitz et al. [22], even if internal combustion engine emissions are a problem, the main issue is the increase in energy demand which makes fuel demand always higher. Alongside the improvement of technologies' efficiencies, there is a need for consumer behaviour change to reduce energy demand.

The downsides of battery electric vehicles

As mentioned previously, electric vehicles operation does not emit any local greenhouse gases making them attractive for the zero emissions future. Nevertheless, to get the full picture regarding the impact of a technology on the environment, life cycle analyses must be conducted, i.e. considering the production of the car and of its "fuel" in addition to its operation.

The production of batteries for electric vehicles consumes large amounts of energy and materials and emits significant quantities of greenhouse gases [25]. In addition, the emissions of an electric vehicle highly depends on the country it is used and of the electricity production system. If the electricity is mainly produced by low-carbon emissions technologies (renewable energy, nuclear energy, ...), the vehicle life cycle analysis appears much "cleaner" than if the electricity is produced from coal or other fossil fuels.

With current technologies, the battery production offsets the benefits of electric vehicle, and it takes around 150,000 km, depending on the scenario selected for assessment, for it to be competitive in terms of environmental impact compared to an internal combustion engine vehicle [25].

It seems certain that the internal combustion engine will be part of the short to mid-term future mix for the transport sector, but its share will decrease, and its form will evolve. However, its efficiency must be improved, and its emissions reduced to be relevant while developing new technologies that will progressively replace it, at least for some applications.

2.1.2 Place of the ICE in motorsport

Despite the introduction of electric series in the past years, the most famous one being the ABB FIA Formula E World Championship launched in 2014, the internal combustion engine still powers most motorsport series.

However, a significant number of elite series have introduced electric hybridisation to their powertrain but at different levels. The Formula One™ powertrain currently features two electric motor (MGUs), one on the turbocharger and one on the crankshaft, which are integral parts of the Power Unit. The Hypercar class of the World Endurance Championship uses electric motors for regenerative braking and propulsion on the front axle. With the introduction of the Rally1 regulations in 2022, the World Rally Championship switched to a hybrid powertrain for the cars to travel in electric mode close to the service park and to have a boost during acceleration and regenerative braking. The Indycar Championship has also implemented an electric motor on the engine for a “push-to-pass” use.

Nevertheless, for smaller series than elite ones, hybrid and electric powertrains have only been implemented on rare occasions, BTCC being an example with a hybrid system designed to provide a 40-hp 15-second boost per lap.

Regarding the future of motorsport, the internal combustion engine is likely to stay a key component at least for the next decade, especially in amateur series, even though more and more series with alternative powertrains should appear. The FIA Sustainable Energies Roadmap shows that even if more and more elite series will switch to electric powertrains, internal combustion engines will still be used [26].

In 2026, Formula One™ will introduce new Power Unit regulations [15] in which the fraction of electric power relative to the total Power Unit power will increase but a high efficiency turbocharged internal combustion engine will still be used. Considering the cost of an engine development, these regulations are likely to last at least until 2030 meaning that internal combustion engines will be used in the pinnacle of motorsport at least for around one more decade.

2.1.3 Steps to reduce environmental impact of motorsport

Motorsport emissions

According to Formula One™ [27], for the 2019 season, the race car's Power Unit emissions only represent 0.7% of Formula One™ total footprint whereas logistics accounts for 45%, event operations for 7.3%, business travel for 27.7% and facilities and factories for 19.3%. Similar observations are made for the 2022 FIA Motorsport Games [26].

What this report shows is that the on-track action only represents a very small fraction of the impact of the Formula One™ on the environment. By orienting regulations so that engine manufacturers produce always more efficient Power Units, the environmental impact can be reduced but only marginally. Nevertheless, it can be imagined that technologies developed in Formula One™ can be transferred to road car manufacturers which will have a greater impact on worldwide footprint.

However, to survive, motorsport needs to take actions on-track but more importantly off-track as this represents the larger fraction of the sport's impact on the environment.

Motorsport sustainability strategies

Motorsport governing bodies, like the FIA and Motorsport UK, have established sustainability / environmental strategies to guide the sport towards a more viable future [28]; [29]. Formula One™ has done the same with its sustainability strategy [27].

Through these sustainability strategies, roadmaps are defined to achieve net zero carbon, by 2030 for all three mentioned documents. As explained previously, it is important to reduce the on-track emissions and the development of sustainable fuels and innovative solutions is of great importance for governing bodies and championships. In addition, by pushing for alternative fuel development and making them usable on a large number of applications, the impact of logistics and transport can be significantly reduced.

As climate change is a global issue, the governing bodies, here the FIA and Motorsport UK, offer their members the tools necessary for them to also accelerate their transition towards a low-carbon future and thus reduce the emissions due to off-track operation but also set deadlines in their roadmap so that objectives can be achieved in time. The objective is to reduce the impact of the organisation itself but also of the events organised under the authority of the FIA by influencing the behaviours of members, the public attending the events, the event promoters, etc. so that the whole industry can secure a sustainable future. The aim is to help motorsport but also indirectly global mobility by promoting innovative technologies that can address the issue globally.

For example, the FIA has created the FIA Environmental Accreditation for stakeholders to measure and enhance their environmental performance. The highest accreditation (3 stars) will become mandatory for FIA events, championships, and competitors. On the 20th of June 2023, the FIA announced that all Formula One™ teams have achieved this accreditation [30].

2.1.4 Opportunities linked to internal combustion engine technology improvement

As mentioned in Section 2.1.1, the internal combustion engine will be part of the future energy mix, but improvements are required to increase its efficiency to reduce the amount of energy lost and reduce its impact on the environment.

As investigated by Caton [31], the internal combustion engine efficiency can be further improved but is physically limited by the different energy conversion processes involved in the work generation. An internal combustion engine will never be able to reach the level of efficiency observed for electric motors for example (>90% [32]) primarily because of the combustion process.

Despite the combustion being at the origin of the work generation, it is also the most limiting factor when it comes to improving the internal combustion engine efficiency [31]; [33] because of the irreversibility created during the different chemical reaction and energy transfer inherent to combustion (further described in Section 2.2). Depending on the engine, around 20% of the fuel exergy is lost during combustion [34]; [35] which inevitably limit its efficiency to ~80% without even considering additional losses.

Nevertheless, the internal combustion engine has not yet reached its physical efficiency limit with current peak brake efficiency around 42% [36] and several processes can be improved to limit the different energy losses and reduce the gap to this limit.

Based on the work done by Reitz et al. [22], Zhao et al. [23], Caton [31] and Wang et al. [33], the potential improvement opportunities on the internal combustion engine concern the following items:

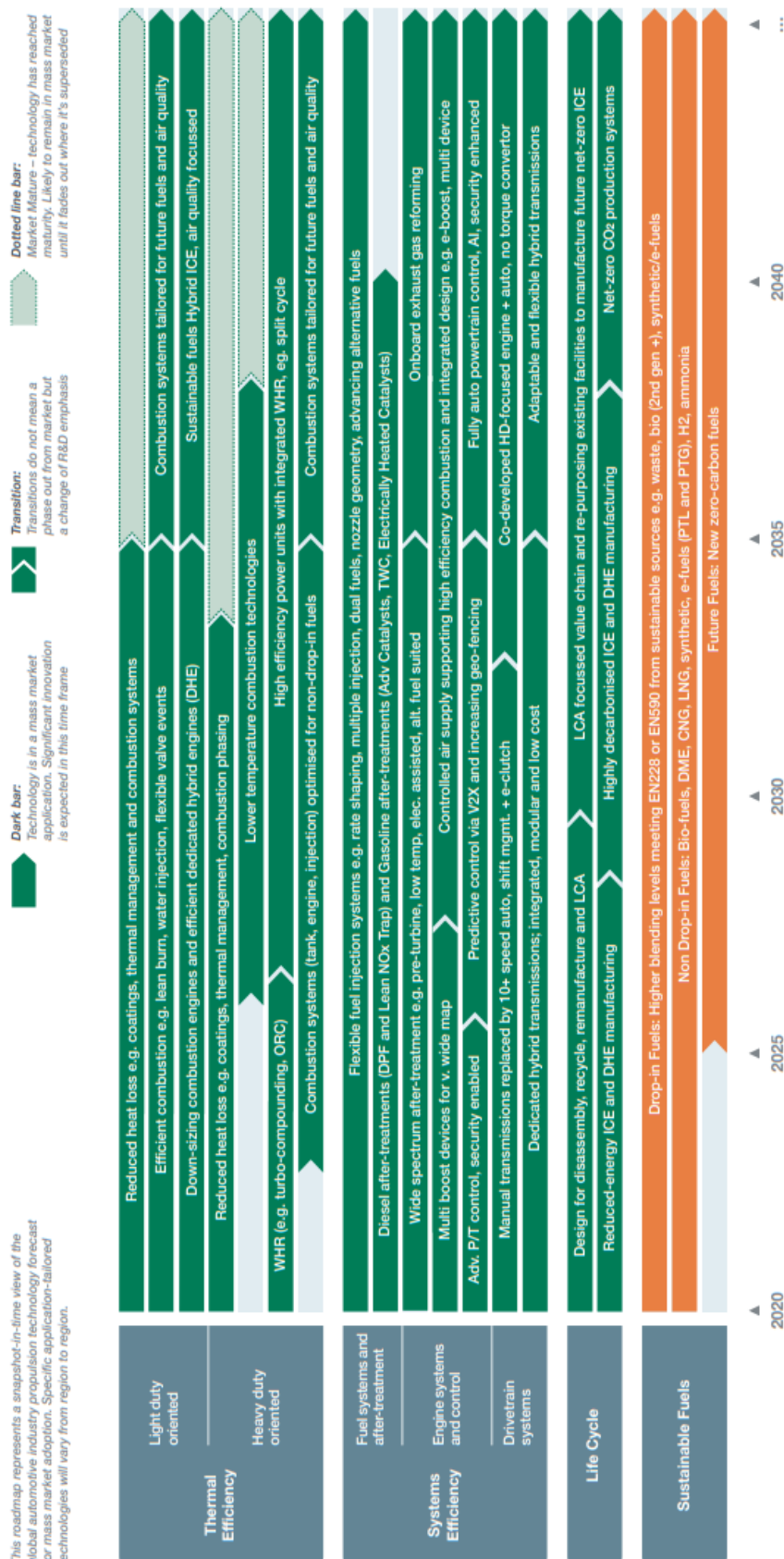
- Combustion systems: Improve the combustion event by enhancing the mixture preparation and implementing technologies enabling combustion strategies which result in higher fuel energy conversion into work,
- Engine breathing: Increase the breathing capabilities of the engine to reduce pumping losses via variable systems for example and implement solutions to increase the engine reactivity to fast changes in load/speed demand to limit low-efficiency operation,
- Electrification: Combine the internal combustion engine with electric motors and battery to modify its operation. The goal is to operate the engine over a limited speed-load range where its efficiency can be

maximised and thus reducing the need for complex systems to achieve high efficiency over a wide operating range. By doing so, the engine cost can be reduced and more investment in the electric part is possible,

- Friction and lubrication: Reduction in friction losses directly results in higher output performances with barely any drawbacks,
- Engine advanced controls: Develop advanced controls to limit cycle-to-cycle variation and ensure the engine can operate in its maximum efficiency conditions using real-time data control and improve the interaction between the engine and its environment (with the electric part of the powertrain but also globally, with other vehicles to optimise engine behaviour with traffic),
- Fuels: Develop flexible and optimised combustion systems capable of accommodating different fuels at separate times or together, develop fuels suitable for newly developed combustion modes and increase the production of bio or e-fuels to reduce the overall impact of internal combustion engines on the environment when considering well-to-wheels emissions,
- Engine simulations: Improve simulation capabilities, including the use of artificial intelligence tools, for design optimisation and develop vehicle simulations including the powertrain to design optimal control strategies without need of physical testing.

In addition to being discussed in the scientific community, the industry is also aware of these improvement opportunities as shown by the Thermal Propulsion Systems Roadmap proposed by the Automotive Council UK [36] (see Figure 2-1) which aims at increasing internal combustion engine brake efficiency up to 53% by 2035 for light duty vehicles and to 60% for heavy duty ones.

Regarding motorsport, the FIA Environmental Strategy [28] also aims at “promoting the automobile technology of tomorrow with a special focus on environmental considerations”. The regulations of the different championships will push the development of different technologies and sustainable fuels are one of their objectives as in 2025, 100% sustainable fuels must start being introduced in World Championships.



2.2 Identification of engine improvement areas using thermodynamic engine analysis

Energy and exergy definitions

To increase the internal combustion engine efficiency, the behaviour of the engine and losses must be understood so that the improvement areas previously mentioned can be tackled.

When analysing the behaviour of an engine, it is useful to understand how the fuel energy is distributed and to assess the fraction of the fuel energy converted into useful work to define the efficiency of the conversion process.

This is what the energy, i.e. the first principle of thermodynamic, applied to an engine allows to do, to look at how the input energy is transformed as it changes from one form to another through a control volume.

However, as stated by Caton [35], Rakopoulos et al. [37] and Farrell et al. [38], the first principles of thermodynamic only result in an incomplete treatment of the engine operation.

The second principles of thermodynamics, by considering irreversibility, allows for an in-depth understanding of the thermodynamic operation of the engine.

The exergy, or availability, is based on the second principles of thermodynamics and is a complement to energy-flow assessment. The exergy defines the maximum useful work that can be extracted from a system in a given environment, i.e. the capacity of the system in a specific state to produce useful work. The maximum work will be extracted if the system is brought to equilibrium with its surrounding through a series of isentropic processes. The rest of the energy which cannot be converted into useful work is called anergy. The exergy analysis shows that a system cannot convert all its energy into useful work.

On the contrary to energy, exergy can be destroyed when the system goes through irreversible processes. When entropy is created, exergy is destroyed. In addition, from the exergy point of view, heat is of lower energy quality than work as it cannot directly produce useful work.

Energy and exergy analyses on internal combustion engines

By conducting an exergy analysis on an internal combustion engine, it is possible to identify the processes creating entropy and thus limiting power production. To achieve high performance and efficiency, this analysis is necessary to get insights on when the engine capacity to produce power is minimised and understand how the entropy creation can be reduced. As stated by Rakopoulos et al., “first-law efficiencies do not explicitly penalize the system for internal irreversibilities” [37].

Classically, when conducting first law of thermodynamics (energy) analysis on an internal combustion engine, the fuel energy is transformed and distributed to four distinct categories [34]; [37]:

- Useful work: The fraction of fuel energy converted into usable output power,
- Heat losses to the coolant: Heat transfer through the combustion chamber walls to the cooling systems,
- Exhaust enthalpy: The exhaust gases exiting the cylinder are hot and contain high level of enthalpy,
- Friction: Fraction of the fuel energy dedicated to overcoming resistive forces within the engine.

A first law of thermodynamics analysis thus describes how the energy has been distributed within the engine but does not determine how much work potential has been lost via undesirable energy transfer (colling losses, exhaust enthalpy and friction) but also how much work potential has been destroyed in the necessary energy conversion processes for the engine operation on the contrary to a second law of thermodynamics (exergy) analysis.

As shown by Caton [35], and Razmara et al. [39] when conducting both energy and exergy analyses on an internal combustion engine, the exergy distribution also considers the useful work produced, the heat transfer to the coolant, friction and exhaust enthalpy but as exergy only considers the work potential and not only energy, the repartition of fuel exergy into these categories is different to that of the fuel energy.

In addition, exergy analysis calculates the amount of exergy that has been destroyed during the engine operation, i.e. the proportion of fuel energy that is definitively impossible to convert into work anymore.

In an internal combustion engine, the irreversibility sources are multiple [35]; [37]; [38]; [40]; [41]; [42]; [43]:

- Combustion irreversibility: This is the highest source of exergy destruction,
- Heat transfer irreversibility: Work potential is lost during heat transfer because of the finite temperature difference,
- Mixing irreversibility: Occurs when the different species in the combustion chamber mix to tend towards a homogeneous mixture,
- Aerodynamic irreversibility: Occurs via throttling but also in the cylinder's inlet and outlet ports,
- Friction losses: Also account for work potential destruction but is easily measurable as it is directly calculated as a work.

Combustion irreversibility

Combustion irreversibility is the highest source of work potential loss and usually represents around 20% of the fuel exergy [34]; [35]. Rakopoulos et al. [37] claim that 90% of all in-cylinder irreversibility is due to the combustion process. Despite being necessary for the operation of the internal combustion engine, combustion is also its biggest limitation.

Dunbar et al. [43] determined that combustion irreversibility could be break down into four different contributions:

- Reactant diffusion: Fuel and oxidant drawn together,
- Fuel oxidation: Chemical reactions during which the fuel molecules are transformed into products,
- Internal thermal energy exchange: Heat transfer between the different species in the combustion chamber, the hot products transfer heat to the cold reactants,
- Mixing: Mixing of the different system constituents to form a homogeneous mixture.

As reported by Dunbar et al. [43] and by Rakopoulos et al. [37], the heat transfer between the combustion products and the reactants are the biggest source of exergy destruction during the combustion process.

Considering this observation and the fact that heat transfer irreversibility increases as the temperature difference decreases, a solution to reduce combustion irreversibility is to increase combustion temperature.

Nevertheless, one of the current solutions to increase internal combustion engine efficiency is to run with highly diluted mixture which results in lower combustion temperature and thus higher combustion irreversibility. As observed in different publications [35]; [38]; [40]; [44], running at high air/fuel ratios effectively increases fuel conversion efficiency despite the higher combustion irreversibility (higher mixing losses and lower temperature heat transfer) as it also reduces the cooling losses and the energy lost to the exhaust which overall help the engine to be more efficient.

It is interesting to note that, as reported by Rakopoulos et al. [37], and Farrell et al. [38] reducing cooling losses does not necessarily result in higher efficiency. Indeed, the work potential saved by limiting heat transfer to the coolant is usually transferred to the exhaust gas. This observation also stresses the importance of exhaust energy recovery system to improve the overall efficiency of the internal combustion engine.

Impact of fuel selection on exergy destruction

The fuel selection can have an impact on the exergy destruction as investigated by Rakopoulos et al. [37], Farrell et al. [38], Liu et al. [44] and Caton [45]. Small fuel molecules, such as hydrogen, usually result in lower entropy creation during combustion as the change in molecule number between the reactants and the products is reduced compared to bigger molecules. However, mixing irreversibility for small fuel molecules is higher as a small fuel will be equivalent to a larger molar proportion of the unburned mixture than a large one.

However, as shown by Farrell et al. [38] and Caton [45], the bigger the molecules become (i.e. more and more atoms), the lower the variation in entropy creation between molecules. This means that going from a molecule with one carbon atom to another with two will cause a greater change in irreversibility than going from a molecule with ten carbon atoms to one with eleven. This led them to conclude that for fuels like gasoline or Diesel, a small molecular structure change does not result in significant entropy creation difference.

The influence of fuel selection on combustion irreversibility can also be considered from the combustion temperature point of view as done by Caton [45]. Different fuels at the same air/fuel ratio will lead to different combustion temperature and as described previously, the higher the temperature, the lower the combustion entropy creation.

Nevertheless, as previously discussed and reinforced by Liu et al. [44], reducing combustion irreversibility does not necessarily result in higher efficiency which led them to conclude that “it is more important and reasonable to design or reform fuel for increasing the thermal efficiency, rather than reducing the exergy destruction of IC engines”.

Considering the difference between energy and exergy, both are necessary to fully understand the behaviour of an engine and how its efficiency is limited. In addition, when investigating energy recovery system, conducting an exergy analysis can help to make the best choice by identify the source with the most work potential, which is not possible with an energy analysis.

2.3 High engine efficiency enabling technologies

2.3.1 Wide view of the enabling technologies

Many technologies are currently deployed to improve the characteristics of internal combustion engine by focusing on the areas describing in Section 2.1.4 to increase its efficiency but also to reduce its overall impact on the environment.

Table 2-1 proposes a summary of the most notable technologies while in Section 2.3.2, the specific technologies that are of interest for the work presented in this thesis are described in detail.

Area	Classification	Notable technologies
Combustion systems	Lean combustion	Lean Spark Ignition Combustion Homogeneous Charge Compression Ignition (HCCI) Gasoline Compression Ignition (GCI) Direct Injection Stratified Charge (DISC)
	Fuel injection	High pressure direct injection Dual fuelling Octane on-demand
	Ignition system	Active pre-chamber ignition Passive pre-chamber ignition High energy spark plug
	Combustion chamber geometry	High compression ratio High millerisation Offset crankshaft
Engine breathing	Variable geometry	Variable camshaft Variable length trumpets
	Forced induction	Electrically assisted turbocharger Variable geometry turbocharger Two-stage compression with intercooling
Electrification	Hybrid propulsion	Electric motor on one axle and thermal propulsion on the other Electric motor and ICE on the same axle Parallel or series hybrid architecture Regenerative braking
	Ancillaries electrification	Electrically assisted turbocharger Electric pumps
Friction and lubrication	Materials	Optimised material selection for friction but also heat transfer properties
	Surface treatment	Optimised surface treatment for each engine component

Area	Classification	Notable technologies
Friction and lubrication (Follow-up)	Oil technologies	Improved oil characteristics at low and high temperatures
Thermal and energy management	Exhaust energy recovery	Turbo compounding Electrically assisted turbocharger
	Waste heat recovery	Organic Rankine Cycle
	Cooling losses reduction	Combustion chamber insulation
Engine advanced controls	Transient efficiency control	Electrically assisted turbocharger
Fuels	Sustainable fuels	E-fuels Biofuels
	High octane fuel blends	Fuel blend composition modified to achieve high knock resistance
Engine simulations	Engine simulations	Improved CFD models dedicated to specific engine components (pre-chamber, injector for instance) Vehicle / Powertrain simulations Fuel blend combustion properties prediction models

Table 2-1 - Notable high engine efficiency enabling technologies

2.3.2 Specific enabling technologies

2.3.2.1 Pre-ignition system and direct injection for lean combustion and high compression ratio engines

Challenges associated with higher compression ratio and lean combustion

By considering the efficiency equation for Otto's cycle (Equation 2-1) [46] it can be noticed that the efficiency of an internal combustion engine can be enhanced by increasing compression ratio (CR), the engine geometric compression ratio, and/or by increasing γ , the specific heat ratio:

$$\eta_{th} = 1 - \frac{1}{CR^{\gamma-1}} \quad (2-1)$$

Operating at higher γ is possible by increasing the mixture's air/fuel ratio, i.e. operating at higher air/fuel ratio than stoichiometry [47]; [48]. As already mentioned in Section 2.2, increasing air/fuel ratio results in higher combustion irreversibility but is counter-balanced by reduced heat transfer to the cooling

systems and the engine can be operated at higher throttle angle, thus limiting pumping losses and increasing the overall engine efficiency. In addition, for road car engines, running at high dilution level leads to reduced combustion temperature which minimise NO_x production [47].

Nevertheless, higher compression ratio is limited by higher knock sensitivity and lean combustion also comes with drawbacks. As shown in Figure 2-2, laminar flame speed varies with air/fuel ratio. The higher the dilution, the lower the laminar flame speed which affects the combustion.

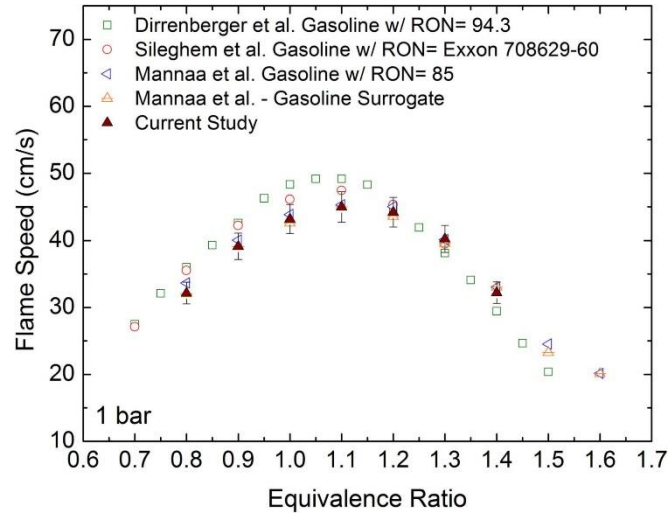


Figure 2-2 - Evolution of gasoline and gasoline flame laminar flame speed with equivalence ratio [49]

Because of the reduced laminar flame speed with air/fuel ratio, the development of the flame kernel and the flame propagation are negatively impacted [47]; [48] which can lead to compromised combustion stability (high cycle-to-cycle variation), longer burn duration, and incomplete combustion. These phenomena decrease engine efficiency which is the opposite of the lean combustion objective.

The automotive industry is not the only one to look at lean combustion as a way to make its activity more sustainable and less polluting. With duty cycles much different from the engines used in road vehicles, lean combustion is also a field of research for maritime engines [50] and stationary engines [51]; [52]. Even research on lean combustion for smaller engines (>30 kW) is carried out to try to reduce the impact of all applications of internal combustion [53].

Lean combustion is also of interest for other means of propulsion which rely on different thermodynamic cycles than that of internal combustion engines. For examples, lean combustion for turbines, where the combustion happens constantly on the contrary to an ICE, is investigated using different combustion techniques [54]; [55]. For instance, lean-rich combustion is one tested solution where the combustion is split into 2 zones, physically separated or not, with one zone being a rich combustion and the second being lean.

Pre-chamber ignition system to enable lean combustion

To overcome these challenges, the ignition energy and the in-cylinder turbulence must be increased [47]. Ignition systems can be classified into two categories, point and space ignitions [48]. For the first category, which includes conventional spark ignition, the flame starts from a unique point from which it propagates. For space ignition, the ignition is characterised by a higher ignition energy and reduced flame traveling distance. Pre-chamber ignition systems are part of this latter category and are regarded as one of the most promising solutions to overcome the challenges of lean combustion for gasoline engines [47]; [48]; [56].

A pre-chamber ignition system refers to a perforated cap placed on top of the spark plug in the combustion chamber. By doing so, the cylinder is divided into two combustion chambers, the main and pre ones, connected via one or more orifices.

The objective of a pre-chamber ignition system is to ignite the pre-chamber rich (thus easily ignitable) mixture so that the combustion leads to an increase in pressure in the pre-chamber resulting in jets of hot chemical species being ejected into the main combustion chamber via the orifices. The jets create multiple, distributed ignition sites that ignite the lean mixture of the main combustion chamber.

Thanks to these multiple ignition sites, the flame traveling distance is reduced, the combustion duration is reduced, and combustion variability is significantly reduced, thus making lean combustion possible and increasing the engine's fuel conversion efficiency [47]; [48]; [56]; [57].

In addition, as the combustion duration is reduced, the unburned mixture is exposed to high temperature conditions for a reduced period which results in lower knock sensitivity of the engine and thus the possibility to run higher compression ratio to push efficiency even further as reported in [47]; [56]; [57]; [58].

With a pre-chamber, the overall combustion chamber surface area is increased, leading to higher cooling losses but its impact on the overall energy balance was judged negligible by Benajes et al. [59] and outweighed by the benefits of running a pre-chamber [47].

Furthermore, Toulson et al. [47] state that implementing a pre-chamber ignition system is easier than other low-temperature combustion technologies, like HCCI, as it does not require as complex hardware and control systems and it can operate over a wider range of conditions with similar dilution levels.

Active and passive pre-chambers

Different types of pre-chambers exist. Depending on the size of the orifices connecting both the pre-chamber and the cylinder, the flow expelled from the pre-chamber following its combustion can be sub or supersonic. In the first case, the technology is referred to as Torch Ignition, whereas in the second case, the flame

is quenched, and it is hot radicals contained in the jets that ignite the main combustion lean mixture at a distance from the pre-chamber rather than a flame and the technology is referred to as Jet Ignition [56].

Then, going further, the pre-chamber can possess its own dedicated fuelling system, in which case the pre-chamber is qualified of active and in the opposite case, the term passive pre-chamber is used.

As fuel but also air can be directly injected into an active pre-chamber, the maximum dilution level at which it can operate is higher than that of a passive one [48]; [58] with some studies reporting that passive pre-chamber does not yield higher lean limit than a conventional spark-ignition engine despite more stable combustion and lower knock sensitivity [47]; [48]; [56]; [60]. For passive pre-chamber, scavenging is an issue and is directly controlled by the movement of the piston and the force it applies on the gas during the compression stroke [48]; [59]. According to Benajes et al. [59] there is a given amount of residual gas, directly related to the engine compression ratio, that cannot be scavenged in passive pre-chambers, and this affects the effectiveness of the system.

For passive pre-chamber, fuelling control is also very important to ensure that the air/fuel ratio within the small combustion chamber can be precisely controlled to ensure good combustion efficiency and stability. As explained by Palakunnummal et al. [57], the filling of a passive pre-chamber is highly dependent on the conditions at its vicinity, with the interaction between the fuel spray and main combustion chamber mixture motion having an impact. Nevertheless, the pre-chamber ignition system is less sensitive to large scale variation than conventional spark-plug systems.

As demonstrated by Almatrafi et al. [61], the operation of a passive pre-chamber can be improved, and its lean limit increased by switching from port-fuel to direct injection and using two-phase injection strategy. The first phase is used to inject most of the fuel and create the lean mixture of the main combustion chamber. The second phase, which happens later, during the compression stroke, is used in collaboration with the upwards motion of the piston to create the rich mixture required for the pre-chamber operation. A similar fuelling strategy is report by Toulson et al. [47] with the Bowl pre-chamber ignition concept.

Impact of the pre-chamber design

The geometry of the pre-chamber (volume, position, number of orifices, orifice's diameter, orifice's orientation) is of primary importance for the pre-chamber to work efficiently as it affects the scavenging, the jet characteristics, and the charge motion within the chamber [47]; [48]; [56]; [57]; [60]; [62]; [63]; [64]. However, as highlighted by Novella et al. [63], a universal perfect design does not exist, and it is critical for the pre-chamber characteristics to be chosen for a clearly defined target operation.

For Zhu et al. [48], the most important parameter is the diameter of the orifices as it affects the scavenging characteristics of the pre-chamber, the flame

quenching, and the jet velocity. Depending on the diameter of the holes, the pressure build-up in the pre-chamber will change and so will the jet velocity and penetration in the main combustion chamber. The diameter must be small enough so that the jet velocity can overcome the pressure in the main combustion chamber and achieve a jet penetration that limit the flame traveling distance, thus leading to reduced burn duration [56]; [64]. On the other hand, if the diameter is too small, the jet velocity will be too high which can compromise the ignition of the main chamber charge and lead to higher heat losses if the jets make contact with the cylinder walls [47].

The active pre-chamber concept opens the door for dual fuel solutions as two different injectors can be used for the main and pre chambers [47]; [56]. As the characteristics of the jets coming out of the pre-chamber also depend on the fuel combustion properties, two different fuels could be used to optimise the overall combustion process. Dual fuel solutions are not further investigated in this thesis due to the current and future Formula One™ regulations only allowing one injector per cylinder which prevents the injection of two different fuels at two different points of the engine. Nevertheless, considering the engineered nature of the future 2026 Formula One™ fuels, the impact of two different fuels could be assessed if combined in a blend, but not separately.

Benefits of Gasoline Direct Injection

Gasoline Direct Injection (GDI) effects the operation of a passive pre-chamber ignition system but is also useful for a conventional spark-ignition engine. As presented by Heywood [34] and by Zhao et al. [65], direct injection possesses several advantages over port-fuel injection precisely because the fuel is directly injected in the cylinder.

Firstly, with direct injection, the control of the mixture preparation is more rapid and more precise as there is no delay for the fuel to travel from the port to the cylinder. This precise control leads to better engine drivability and better fuel economy [66]. Secondly, the fuel distribution within the cylinder can be more precisely controlled and the injection event characteristics can influence the air/fuel ratio distribution in the combustion to modify the combustion properties by, for example, either creating a homogeneous or a stratified mixture to allow overall lean operation without compromising the ignitability of the mixture in the vicinity of the spark plug [34].

A third advantage of direct injection is that as the fuel is injected under the form of small liquid droplets, these need to evaporate before the combustion event. By extracting the necessary energy for evaporation from the air trapped in the cylinder, the charge temperature can be reduced which, in turn, reduces the knock sensitivity of the engine. In consequence, high compression ratio and/or higher boost pressure can be used and thus achieve higher efficiency [34]; [65]; [66]. Direct injection, in addition to improving fuel preparation, is an enabling technology for higher fuel conversion efficiency solutions.

The interaction of the spray with the combustion chamber geometry, especially the piston, can have a great influence as it impacts the droplet distribution and evaporation and the air/fuel ratio distribution [34]; [65]. As seen with the Bowl pre-chamber ignition concept [47], the interaction of the spray with the piston and the stratification of the charge can be key for the operation of a passive pre-chamber.

The design of the injector (number of holes, hole geometry for examples) and its position in the cylinder will play a role in the successful implementation or not of direct injection [34]; [67].

Another very important characteristic is injection pressure. A change in injection pressure will affect the properties of the spray and the fuel preparation [34]; [65]; [67]; [68]. As the injection pressure is increased, a better atomisation is achieved (smaller droplet size) and the spray angle is increased which results in easier air/fuel mixing and more controlled air/fuel ratio distribution. In addition, as presented by Kaminaga et al. [69], by increasing injection pressure, intensity level and small-scale turbulence can be increased resulting in fastest flame propagation and thus reduced combustion duration.

The development of the fuel spray is also heavily affected by the conditions in the cylinder [34]; [67]; [68], i.e. the cylinder pressure (the higher, the lower the spray penetration) and the large-scale movement of the trapped air which influence spray development.

2.3.2.2 Engine breathing

Downsizing and engine breathing

Engine breathing is of primary importance for an internal combustion engine as the quantity of air trapped in the cylinders determined the quantity of fuel that can be burned.

The current trend regarding engine development is the “downsizing” concept. The aim is to reduce the engine swept volume while producing similar performance as a larger engine with potential benefits on fuel consumption and fuel conversion efficiency [70]; [71]; [72]. By reducing the engine volume, friction losses are reduced, pumping losses can be decreased and as the engine is operated at higher BMEP, the efficiency can be raised.

Nevertheless, as shown by Equation 2-2 which is a governing equation for internal combustion engines [34], if only the engine swept volume V_{swept} is reduced, the output power is also reduced. To achieve similar performance level as a larger engine, the engine operation must be modified.

$$P_I = \frac{\eta_{th}\eta_c\eta_v NV_{swept} Q_{lvh}\rho_a (F/A)}{2} \quad (2-2)$$

One solution is to implement forced induction solutions and more specifically, turbocharging followed by a cooling stage. By doing so, the inlet air density ρ_a is

increased, more air is introduced, more fuel can be burned, and the power output of the downsized engine can be increased [71]; [72].

Turbocharging

Different forced induction solutions exist but turbocharging is interesting as it relies on the turbine extracting energy from the exhaust enthalpy to power the compressor, thus making use of the otherwise wasted exhaust gas enthalpy. Nevertheless, this solution comes at the price of an increased exhaust pressure which can limit the engine breathing if the turbocharger is not selected appropriately [71]. However, if correctly implemented, pumping losses can be significantly reduced as the cylinder is provided with high-pressure air which can produce positive work on the piston.

Turbocharging is also interesting when the engine is aimed at running a lean mixture. With a naturally aspirated engine, to run high air/fuel ratios, the volumetric efficiency can be improved to trap more but only limited gains are possible with optimising wave tuning; otherwise the fuel supply can be reduced but this results in lower output performance. However, with a turbocharged engine, there is the possibility to raise the boost pressure, leading to higher air mass flow and air/fuel ratios without modifying the fuel flow and compromising on the performance.

Turbocharging also enables the use of a Miller cycle which brings several advantages to the internal combustion operation.

Miller cycle

The Miller cycle relies on a lower effective compression ratio than the expansion ratio thanks to an early or late intake valve closing (IVC). As the compression work is reduced compared to the expansion, efficiency is theoretically increased.

With an early IVC, the intake valve lift and duration are reduced leaving less time for air to enter the cylinder while with a late IVC (after BTC), as the piston is moving upwards, a fraction of the charge could be pushed back in the intake port; in both cases, the engine volumetric efficiency is reduced. However, by using a turbocharger, boost pressure can be raised to compensate for the lower volumetric efficiency and achieve similar performance [70]; [73]; [74]; [75]. In addition, with an early IVC, as the valve lift is reduced, so is the flow coefficient meaning that higher throttle angles are necessary to reach the target air mass flow which reduces throttling losses and thus the overall pumping losses despite increased losses at the intake valve [75].

As the effective compression ratio is reduced, the temperature and pressure at the end of the compression stroke are reduced compared to a conventional cycle [73]; [74]; [75]; [76]. By consequence, the knock sensitivity of the engine is reduced opening the door for higher compression ratio and/or boost pressure, i.e. higher fuel conversion efficiency and performance.

The impact of the Miller cycle depends on how it is implemented. According to Huang et al. [73], with an early IVC, there is a significant reduction in tumble ratio and turbulent kinetic energy which slows down flame propagation and thus increases combustion duration. However, as reported by Tavakoli et al. [74], an early IVC results in lower peak pressure and temperature, i.e. lower knock sensitivity. A Miller cycle with late IVC is reported to achieve the best output performance without significant change in pressure and temperature [74]; [75].

Wave tuning for turbocharged engines

Despite not relying as much on wave tuning for air intake as naturally aspirated (NA) engine, Rosetti et al. [77] showed how intake trumpets length can be used to optimise the operation of a turbocharged engine, specifically to reduce engine knock sensitivity.

Trumpets can be designed to tune the intake wave to improve volumetric efficiency as for a NA engine wave action tuning (tuning), but it can also be designed to produce nearly constant pressure at the inlet (un-tuning) or even negative pressure at IVC (anti-tuning). Despite having a negative impact on volumetric efficiency, these solutions allow to reduce in-cylinder temperature at IVC and thus to reduce knock sensitivity. The loss in volumetric efficiency is compensated by slightly higher boost pressure but this comes at the expense of higher intercooler load.

For fixed length trumpets, the authors report that the un-tuning concept (constant intake pressure) is the most promising solution as it requires very short trumpets which reduces packaging issues and intermediate boost pressure and intercooler load increases.

However, if a variable trumpet length system is available, the engine can switch between the different tuning concepts depending on its needs.

2.3.2.3 Electrification of the internal combustion engine

Benefits of a hybrid powertrain

One limitation of internal combustion engines is that compromises must be made regarding the engine speed – load range optimisation. Specific operating conditions can be optimised at the expense of others resulting in the engine being particularly inefficient in given conditions.

One solution to improve the transient operation of the internal combustion engine is to couple it with an electric motor in a parallel configuration, i.e. both the engine and the electric motor are connected to the vehicle's wheels on the same axle. By doing so, the powertrain has an additional degree of freedom with the possibility to split the power demand between the ICE and the electric motor [78].

The electric motor, which efficiency is far superior to that of an internal combustion engine, can be used in collaboration with the engine so that this latter only operates in a limited range of high efficiency operating points, i.e. the dependence of the engine operation on the power demand is reduced [78]; [79]; [80]. By consequence, the design of the engine can be optimised only for a very specific narrower operation window resulting in higher peak efficiency. The efficiency of a hybrid powertrain is highly dependent on the internal combustion efficiency. By doing so, the fuel consumption and the emissions of the powertrain can be reduced.

To fully maximise the benefits of a hybrid powertrain, the electric motor can also be used for regenerative braking to recover kinetic energy during braking events that would otherwise be wasted under the form of heat in the braking system [80]; [81]. This recovered energy is stored in the battery to be later used when the electric motor is required to provide tractive power.

Nevertheless, optimal control of the powertrain is required so that the internal combustion engine is only used in high efficiency conditions and that the battery state of charge (SOC) is correctly managed [78]; [79]; [80]. A good energy control allows both the internal combustion engine and the electric motor to operate in high-efficiency zones and can result in significant efficiency improvement.

Electrified turbocharger

As previously discussed, turbocharging is an important feature for the downsized engine to trap enough air in the reduced swept volume and match the performance of bigger engines. In addition, the turbocharger makes use of the exhaust enthalpy which would otherwise be wasted to the environment, thus increasing the overall efficiency of the engine.

Nevertheless, when selecting a turbocharger, there is a compromise to be made in terms of design between transient responsiveness at low exhaust enthalpy (small turbine) and peak power targets (big turbine). Not to compromise the peak performance of the engine, the turbine cannot be too small, resulting in the turbine not to be very efficient in low exhaust enthalpy situation meaning that the turbocharger is not capable of immediately matching boost pressure demand during sudden change in load [82]; [83]; [84]. The delay in boost responsiveness is called turbo lag.

The turbo lag results in poor drivability and low engine efficiency, because of sub-optimal air/fuel ratios if the injected fuel mass flow is not reduced to match the trapped air mass flow, meaning that improving the turbocharger responsiveness is key to achieve better engine performance.

According to Park et al. [82], turbo lag is caused by several factors. The primary factors are related to the physical properties of the turbocharger, i.e. the rotor moment of inertia and the intake duct volume, the fluid in the intake system has inertia and so does the turbocharger. These inertias must be overcome when operating conditions are changed. The secondary factor is a decrease in turbine

efficiency as a result of a change in operating point, i.e. the turbine does not operate at its peak efficiency / design point because of the reduced exhaust enthalpy. These factors mutually affect each other and are responsible for turbo lag.

Turbo lag can be mitigated by minimising the turbocharged moment of inertia [85]; [86]; [87], selecting a turbine with a lower aspect ratio or utilizing specific turbine technologies to match a wider range of engine operating conditions with only one design (variable geometry turbine, multi-stage turbocharger or twin-scroll turbine) [85]; [88]; [89], increasing exhaust enthalpy [90] and externally assisting the turbocharger [85]; [87].

This is in this last category that electrification can prove useful with an electrically assisted turbocharger, i.e. connecting an electric motor to the turbocharger shaft [83]; [84]; [88]; [91]. By doing so, as for hybrid powertrains, a new degree of freedom is added and there is the possibility to decouple the work required by the compressor and the work recovered by the turbine.

The electric motor can provide the turbocharger with power when the turbine is lacking exhaust enthalpy, thus nearly eliminating turbo lag, improving the responsiveness of the turbocharger and the performance of the engine, including its efficiency.

As presented by Lee et al. [83], different configurations are possible each with their advantages and drawbacks. According to Katrasnik et al. [88], the ratio of electric motor torque over moment of inertia must be maximised to achieve the best performance of an electrically assisted turbocharger.

With the electric motor connected to the turbocharger rotor, it is possible to recover with the turbine more energy than required by the compressor to do electric energy recuperation. By doing so, more exhaust enthalpy is recovered, and the battery connected to the electric motor can be recharged [91]. To yield significant recovery levels, the matching of the turbocharger must be modified, i.e. selecting a bigger turbine able to swallow a higher flow, to recover more energy but this comes at the expense of higher exhaust pressure and moment of inertia.

As suggested by Lee et al. [83], because the turbocharger speed can be controlled using the electric motor, implementing waste gates to prevent the turbocharger from overspeeding is not necessary anymore.

2.3.2.4 Sustainable fuels and potential for fuel composition optimisation

Review of available sustainable fuels

As mentioned previously, decarbonising the transport industry is a critical objective to ensure a sustainable future, which requires moving away from fossil fuels [92]; [93]; [94]. One solution is the adoption of theoretically carbon neutral / renewable fuels whose production requires removing carbon dioxide from the atmosphere.

This can be achieved either by using biofuels of second generation which are derived from biomass produced from waste (rather than from food compared to the first generation [95]) or by using e-fuels which are synthesized using hydrogen produced via electrolysis using renewable energy and CO₂ from carbon capture. For both, CO₂ is taken from the atmosphere for their production, either by plant growing process or carbon capture, which theoretically compensate for the CO₂ that will be emitted during their combustion in an internal combustion engine.

In addition to being a source of power, e-fuels can be seen as an effective energy carrier, i.e. e-fuels can be produced to balance the intermittent production of renewable electricity. In addition, compared to electricity, e-fuels can be easily stored, transported, and used in different industries in need for power using the current infrastructures [92]; [93]; [94].

Currently, e-methane, e-methanol and e-ammonia are the fuels the most produced using this process [92]; [93].

Both biofuel and e-fuel productions can be combined to produce bio-e-fuels using products of biomass and hydrogen obtained via renewable electricity electrolysis to make e-fuels more competitive [93].

Challenges associated with large-scale sustainable fuel production

Despite their potential for decarbonisation, e-fuels are not yet produced and used at large-scale because of several challenges. As highlighted by Brynolf et al. [93], their exact impact on the environment is not yet known and it is too early to make definitive decisions. In addition, to drive down the cost of e-fuels and make them attractive, the electrolysis (hydrogen production) and carbon capture processes must be improved. Currently, the production of e-fuels in large quantities is impossible because of the limited production of green hydrogen and carbon capture technologies [94].

For e-fuels to play a significant role in decarbonisation, their attractiveness needs to be increased which will require the development of competing fuel options, renewable electricity production, the implementation of carbon capture systems, the availability of biomass and adapted policies [93].

However, it must be highlighted that the different processes involved in the production of e-fuels are highly energy-consuming thus requiring large amounts of energy which must be renewable for e-fuels to make sense, and that for each process, the efficiency is different from 100%. The more steps in the production, the more losses there are, so the sustainability of e-fuels must be assessed with that in mind when comparing it to other solutions involving less steps to obtain the moving energy.

Optimisation of fuel composition to improve the combustion process

As highlighted by Splitter et al. [96], the increase of engine efficiency has been closely linked to the development in engine technologies and on fuels; there is a co-evolution of engine and fuel. With internal combustion engines pushed towards always higher fuel conversion efficiency, the needs in terms of fuels will evolve [97]. For example, as the efficiency of spark ignition engines is improved, the issue of abnormal combustion (knock and pre-ignition) becomes more likely, which increases the need for fuels with higher knock resistivity.

However, with the development of synthetic fuels where specific molecules are produced separately through sustainable processes, the possibility of engineering / designing fuel opens. The composition of future fuels could be optimised through the blending of synthetic fuels to obtain the desired fuel properties [98].

As investigated by Corrubia et al. [99] and Rockstroh et al. [100], the fuel's octane number can be increased by optimising the fuel composition of the fuel blend, in particular by adding alcohol molecules, which results in higher efficiency as higher compression ratio can be used. In addition, as highlighted by Heywood [34] and Rossi et al. [101], if the fuel composition is adapted to yield higher heat of evaporation, higher compression ratio can also be used as the evaporation of the fuel will result in an increased cooling of the charge.

The fuel composition can also influence the combustion characteristic, specifically laminar flame speed, making possible the optimisation of fuel blends to improve flame propagation [98]; [100]. As reported by Naiki et al. [102], the lean limit of a fuel can be increased if the laminar flame speed of the blend is optimised for that, making lean combustion more easily achievable with the benefits previously described. Rockstorh et al. [98] showed that a synthetic fuel aimed at replacing gasoline could have the same RON but improved laminar flame speed and thus have more interesting combustion characteristics.

Flame speed is related to the molecular structure. The use of olefins results in faster flame with a higher lean limit whereas aromatics (excluding benzene) will slow down the flame [98]; [102].

Nevertheless, the development of fuel blends must be accompanied by improvements in terms of simulation. For instance, the RON of a blend is not obtained as straightforwardly as linearly combining the values of the individual species; there is an octane blending effect [99]. Models to help with the optimisation of fuel blend [100] and to capture combustion characteristics [103] to support engine development must be researched to create less computationally expensive simulations and to push internal combustion engine efficiency further [101], with potentially the help of machine learning and artificial intelligence tools [104].

2.3.2.5 Successful implemented high engine efficiency solutions

A number of solutions presented in the literature review have been implemented in commercially available systems and are discussed here.

Regarding pre-chamber ignition systems, MAHLE Powertrain has developed both passive and active solutions [105]; [106] which have been validated on demonstrators. The passive pre-chamber allows for up to 40% combustion duration reduction at stoichiometric air/fuel ratio, the use of higher compression ratio for higher efficiency and the system fits in a conventional spark plug package space. The active pre-chamber achieves fast and stable combustion up to $\lambda = 2$ for gasoline while also reducing knock sensitivity and achieving higher efficiency thanks to higher dilution and compression ratio.

Maserati with the Nettuno engine that will be placed in its MC20 supercar also implemented a passive pre-chamber ignition system [107]. The engine features both high-pressure direct (350 bar) and port fuel (6 bar) injections and two spark plugs, one in the main chamber and one in the pre-chamber. The pre-chamber ignition system is used to achieve faster combustion and improve performance at high load, most probably combining it with the high-pressure direct injection to create a fuel distribution adequate for the pre-chamber operation. At low load, when the control of the pre-chamber is more difficult, the engine is operating as a conventional spark ignition gasoline engine and ensure good drivability over a wide range of operating conditions.

To achieve higher engine efficiency and low emissions, Mazda developed a new combustion process called Spark Controlled Compression Ignition combining the advantages of spark ignition and compression ignition engines [108]. The engine can ensure reliable and fast combustion of lean mixtures thanks to high compression ratio and controlled combustion start using the spark plug.

Regarding engine breathing, Garrett Motion has developed an electrified turbocharger which has been tested on engine dyno but also on demonstration vehicles [109]. This device allows for transient control of air/fuel ratio, can recover additional energy from the exhaust gas to recharge the battery and results in improved manoeuvre time.

Variable Valve Actuation systems were not discussed earlier in this section as they are banned from the Formula One™ regulations (see the following section) and not further investigated in this thesis, but Fiat developed a system called MultiAir allowing the engine to operate without throttle body by actively modifying the valve actuation to control pumping losses and combustion efficiency [110]; [111].

2.4 Evolution of Formula One™ Power Unit regulations

Every year, the Fédération Internationale de l'Automobile publishes an update set of regulations for the coming Formula One™ season. The regulations include a sporting component which regulates how a Grand Prix weekend is managed, a

technical component which dictates the design of the car and of the Power Unit and recently a financial component has been added.

The sustainability of the sport is steered from an environmental point of view as discussed previously but there is a financial aspect too with the aim to limit teams' spending so that the future of the smallest teams can be secured and the gap with the leading teams reduced.

This section focuses on the technical regulations and more specifically on the sections that define the engine / Power Unit. The evolution of the regulations will be assessed using the four latest generations of Power Unit regulations starting from the V10 era (2000-2005), then the V8 era (2006-2013), the first hybrid era (2014-2025) and the future regulations that will be introduced in 2026 and that will constitute the second hybrid era.

To observe the evolution of the regulations between generations but also within a given generation, the regulations documents of the first and last years of each generation are considered when possible. The 2000 regulations document was not found online so the 2001 one will be used, the 2025 regulations have not yet been published when writing this thesis and for the upcoming 2026 generation, obviously it is not known when this generation will end and only the document for the first year has been published. This means that the 2001 [112], 2005 [113], 2006 [114], 2013 [115], 2014 [8], 2024 [12] and 2026 [15] Formula One™ Technical regulations are investigated.

The first notable insight regarding the evolution of the engine / Power Unit definition in the Formula One™ regulations concerns the number of pages dedicated to this subject.

As shown in Figure 2-3, for the V10 era, only one page described the requirement for the engine. This number changed to 5 at the beginning of the V8 era to reach 8 by the end of this generation. Between 2014 and 2024, the page number increased from 8 to 18 and for the 2026 regulations, 21 pages are dedicated to describing the requirements of the Power Unit.

This increase has two reasons. Firstly, in the V10 era and at the beginning of the V8 one, the engine was the only element of the Power Unit whereas with the current and future regulations, as the Power Unit is composed of the ICE and of electric motor(s), there are more components to define thus resulting in an increased number of pages. Secondly, as discussed below with Figure 2-4, an increased number of parameters are regulated which requires more articles in the regulations. This is done by the FIA to control the diversity between the different engines and maybe ensure a greater equity between them but also to control the cost and only allowing engine manufacturers to research a specific number of parameters.

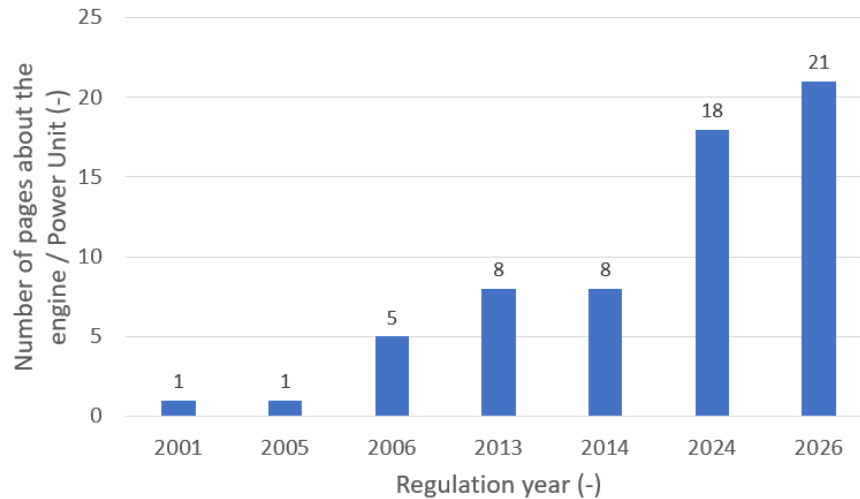


Figure 2-3 - Evolution of the number of pages dedicated to the engine / Power Unit in Formula One™ technical regulations

Inspired by the work done by Montgomery [116], a summary of the regulations' evolution is shown in Figure 2-4. The data used to create this graphic are shown in Appendix A with the associated article numbers from the regulations. Only a selection of the parameters described in the regulations are inspected here.

The colour of each box in Figure 2-4 represents the number of regulations documents dictating a value / behaviour / limit for the given parameter. If the box is red, it seems that only one document / set of regulations out of the seven investigated here has an article about this parameter. Whereas if it is dark green, all of them regulate it. The legend of Figure 2-4 shows the colours associated with each number. The graphic will be discussed in a sequence from top to bottom.

It must be highlighted that for the parameters shown in Figure 2-4 with a small number of regulations having a paragraph on it (colours close to red), it is most of the time a paragraph added in the latest regulations to regulate a parameter that was not in the previous years, not the opposite.

Means of propulsion

As shown in Figure 2-4, all seven investigated regulations documents dictate the allowed means of propulsion, but the requirements are different. For the first three regulations (2001, 2005 and 2006), solely the internal combustion engine was allowed to power the car whereas in the 2013 regulations, the KERS system (introduced in 2009 [117]) was also allowed to transmit power to the wheels. From 2014, with the introduction of mandatory electric hybrid powertrains, the means of propulsion includes the ICE but also the MGU-K.

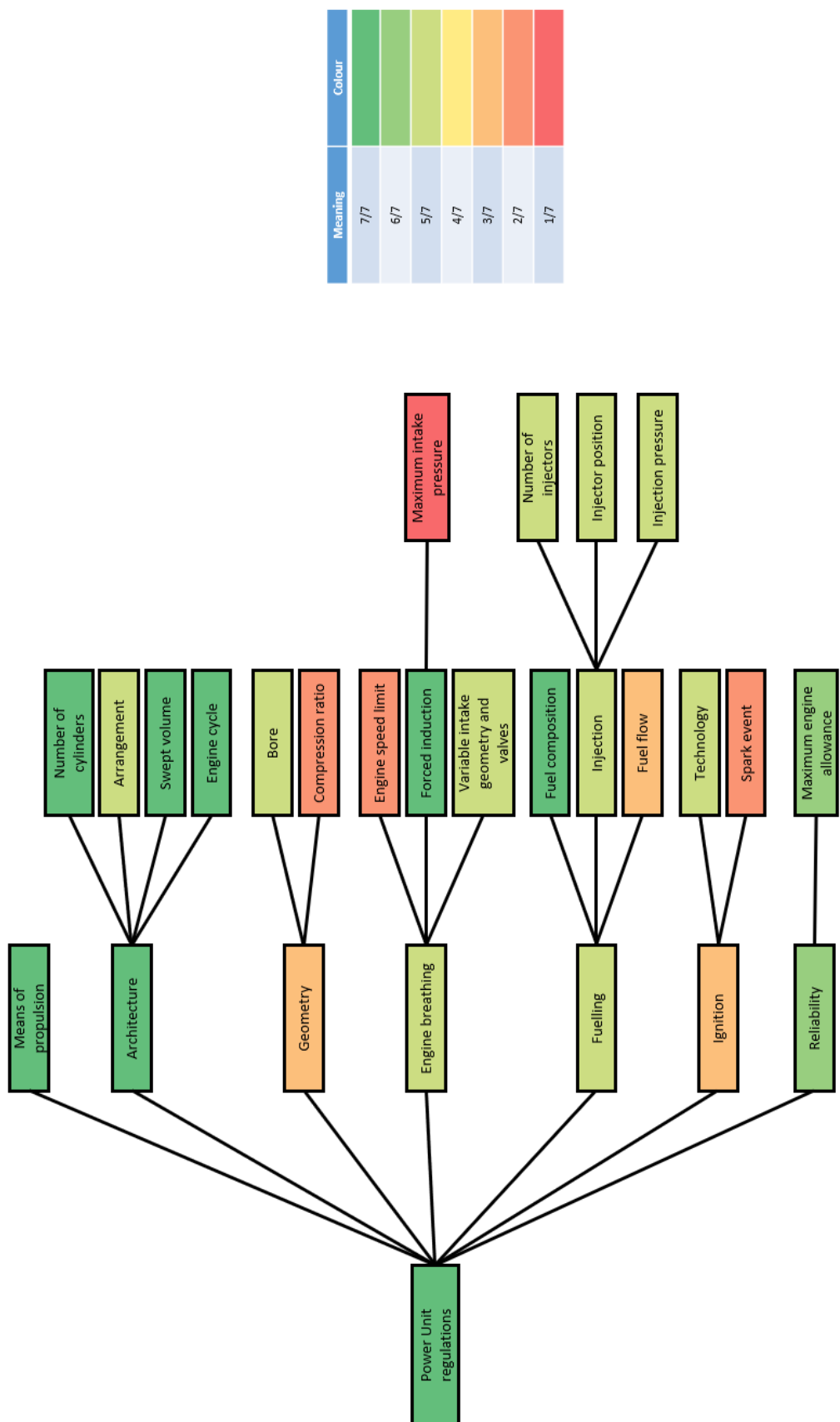


Figure 2-4 - Formula One™ Power Unit regulations' evolution summary

Architecture

Regarding the architecture of the engine, all investigated regulations dictate the use of 4-stroke engines, a specific swept volume and a given number of cylinders. The first two were already regulated in pre-2000 regulations but it is only in 2000 that the regulations made mandatory a number of cylinders, in this case 10. However, for the generation previously referred to as the V10 era, the regulations did not stipulate that the engine must have a V-arrangement even though all engine manufacturers chose this configuration. Since 2006, the regulations made mandatory the use of a V-arrangement but also the 90° angle between the two banks.

With the reduction in cylinder number and swept volume with the years and the introduction of an electrical system to create a hybrid Power Unit, the Formula One™ regulations followed the downsizing and electrification trends also observed in the road car industry.

Geometry

The geometry of the combustion chamber has been regulated via two parameters, the piston bore and the geometric compression ratio. As for engine architecture, it is only since 2006 that a specific piston bore is defined by the regulations, thus resulting in an indirectly regulated stroke as the engine swept volume is fixed. This addition is likely to have been made to prevent engine manufacturers investigating different values and thus control cost.

However, it is only since 2017 [118] that a maximum geometric compression ratio is stated in the regulations (visible from the 2024 regulations in Figure 2-4). Previously, engine manufacturers were only limited by their fuel's knock properties. However, as it will be described in Chapter 5, a specific pre-chamber ignition system is used in the 2014+ Formula One™ engine alongside direct injection which help reducing the engine knock sensitivity and thus allow to push compression ratio to higher values to reach higher efficiency. Once again, the compression ratio limit is likely to have been implemented to limit engine development and manufacturing costs.

Engine breathing

Regarding engine breathing and specifically engine speed limit, the trend is different compared to the other parameters. A limit was set in the regulations in the second part of the V8 era from 2007 and was progressively reduced to keep the cost and/or power down. A 15,000 RPM limit was kept in the first years of the V6 hybrid era but was then removed from 2022 [119] as it was unnecessary for reasons explained in Chapter 5. As the 2026 engine will be very similar to the 2014+ one, the engine speed limit will not be reintroduced for the next Power Unit generation.

Regarding forced induction, all regulations comment on this topic, either to ban it when the engine is naturally aspirated (2000-2013) or to authorise it and regulate the type of turbocharger allowed in case of turbocharged engines (2014-2026). However, only the 2026 regulations state a maximum intake pressure that engine manufacturers cannot exceed (comments on this limit are made in Chapter 6).

All the investigated regulations documents ban the use of variable exhaust geometry systems, but the situation is different for the inlet. In the V10 era, nothing was mentioned regarding variable intake valve and geometry systems indirectly meaning that they were de-facto authorised. However, from 2006, variable cam systems have been banned presumably as they are very expensive systems to develop and make reliable. Variable intake geometry systems were banned in the V8 regulations and in the first year of the V6 hybrid era before not being mentioned any more, i.e. being de-facto authorised, but will again be banned in 2026. Such systems are time-consuming to develop but allow to increase engine efficiency even for turbocharged engine as demonstrated by Rosetti et al. [77].

Fuelling

Being a very important element of the engine performance, fuel composition has been regulated for the whole period under review here (2000-2026). It is understood that this is to ensure that fuel suppliers do not engineer very specific fuels only compatible with Formula One™ and that the development made for the sport is also applicable to road car fuels. In addition, this maintains a level of parity between the different teams as everyone uses a fuel with common characteristics.

From 2026, the regulations will impose the use of 100% advanced sustainable fuels, i.e. with no fossil origin, with the aim to push forward the development of sustainable fuels and accompany the decarbonisation of the transport industry.

Fuel preparation with regards to the injection system has been regulated from 2006. Previously, nothing was mentioned in the V10 era regulations, but all manufacturers were using port injection. From 2006, the number of injectors, their position and the injection pressure have been fixed by the regulations.

By imposing only 1 injector per cylinder, the regulators prevented researching complex fuel preparation methods and dual fuel solutions and the limited injection pressure limits the development cost. In the V8 era, the engines were port-injected as imposed by the regulations but from 2014, the regulations made direct injection mandatory in the attempt to improve the engine efficiency and follow the development trend of road car engines. In addition, direct injection makes possible the precise control of pre-chamber fuelling as previously discussed.

Importantly, from 2014, the maximum fuel flow usable by the engine became limited by the regulations. Previously, engine manufacturers had the possibility to burn as much fuel as desired to produce more power, but with the introduction of a fuel flow limit, the regulators turn the attention of engine manufacturers towards

efficiency. This also is in line with the development of road engines and, as discussed in Chapters 5 and 6, has a significant impact on the engine behaviour and performance.

Ignition

From the introduction of the V8 engines, the combustion event has also been controlled via the articles on ignition systems. Engine manufacturers are only allowed to use a single conventional spark plug and a single ignition coil per cylinder, with more advanced techniques being forbidden.

From 2015, a limit of five spark events was introduced which is interesting regarding the fuel-limited characteristic of the 2014+ engine. As discussed previously, running at high air/fuel ratios helps to improve efficiency but makes flame propagation more difficult hence the use of a pre-chamber to speed up the combustion and benefit from the higher level of dilution. Considering the dynamics of the combustion event, the passive pre-chamber can only be filled once per cycle with fresh charge, so the five spark events are not made to fire it multiple times per cycle. Multiple spark events might be necessary to ensure that the pre-chamber will fire at some point, to decrease the probability of misfire if the air/fuel mixture of the pre-chamber is suboptimal and regulations imposed a maximum number of spark events to again prevent manufacturers to research more complex control and technologies. Even without the use of a pre-chamber, the observation can be the same with multiple spark events authorised to maximise the chances to ignite a lean mixture only using the conventional spark plug.

Interestingly, even though the 2026 engine philosophy will have similarities with the 2014+ one and a pre-chamber ignition system will still be used, only one spark event per cycle will be allowed. This means that a specific attention to fuel preparation and pre-chamber filling will be necessary to ensure optimal behaviour and minimise risks of misfire as there will only be one chance to ignite the charge.

Reliability

Over the years, Formula One™ regulations imposed ever more constraining limits on Power Unit component allowances with the objective to increase the reliability and enhance the sustainability of the sport. In 2004 [120], a driver was allowed to use a new engine for each Grand Prix weekend whereas in 2023 [121], one driver is only allowed four internal combustion engines despite the number of Grand Prix increasing from 18 to 23 in the same period. Reliability has significantly increased.

Overall, the evolution of Formula One™ technical regulations pushed the sport towards higher reliability, higher efficiency, and reduced development cost to ensure the sustainable future of the sport.

As in the wider automotive industry, Formula One™ adopted the trends of downsizing, electrification, and turbocharging to improve engine efficiency. The most important change for efficiency improvement has been the introduction of a fuel flow limit which shifts the focus to maximising efficiency to enhance output performance.

Compared with the technology roadmap proposed by the Automotive Council UK for thermal propulsion systems [36] (Figure 2-1), it can be highlighted that Formula One™ is a leader. Formula One™ engines already reached 50% peak brake thermal efficiency, achieves reliable and efficient lean combustion, uses high-performance downsized engine coupled with electric motors and relies on an electrically assisted turbocharger to optimised energy flows whereas these objectives are only set for 2025 or later for the automotive sector. Nevertheless, it must be highlighted that Formula One™ engines have very specific operating conditions, with most of the period of use spent at full load.

2.5 Engine modelling in powertrain development processes

2.5.1 The different types of engine simulations

A range of simulation techniques are used to model engine behaviour and/or predict engine performance, with different levels of complexity and accuracy. Those that strive to be most physically realistic tend to be the most complex but also require more computing effort and time. Those that replace the most complex physical processes (such as combustion rate-kinetics chemistry) with empirical data are faster and can be very accurate (or at least they can be relied on) although these need appropriate amounts of often detailed input data.

Depending on the amount of data available on the engine and the desired output predictions, different tools can be suitable / appropriate.

Governing equations

A set of ‘governing equations’ [34], presented in Chapter 4, can be used to describe the engine behaviour from an analytical perspective, the physical phenomena are hidden from view by functions relating input to output and do not require component-level details to be used. They constitute the tools with the lowest complexity, require low computing power and can be applied using a spreadsheet. Experience with the Cranfield F1™ Design Sprint suggests that familiarity with the inter-relationships between the governing equations helps engineers understand the origins of excellent performance more readily.

With minimal data on the engines and sound assumptions, the overall engine behaviour can be modelled, either to estimate the range of power output possible for a given design or to predict the necessary requirements to reach a desired level of performance. Governing equations exist for the engine, but others are applicable for additional parts of the Power Unit, like the turbocharger.

0D simulations

The next level of complexity is embodied by zero dimensional (0D) simulations for which calculations are time dependent, but no spatial dimension is considered. The models rely on semi-empirical equations and thermodynamic laws [122]. On the contrary to the governing equations, these simulations can model gas exchange processes, combustion, the behaviour of charging devices (turbocharger, supercharger) but with some limitations has no spatial effect is modelled. For instance, the effect of pressure waves on volumetric efficiency cannot be represented as the intake and exhaust geometries are not provided, also the impact of the combustion chamber shape on the rate-of-heat-release cannot be investigated.

Compared to the simple application of the governing equations, the use of 0D simulations requires more detailed information about the engine to provide accurate results as more phenomena are being investigated using the semi-empirical relationships. Nevertheless, as some performance predictions of these models highly dependent on the user-input data regarding the engine under study fed to the semi-empirical models, if the model's operating conditions are changed compared to that of the real engine, the levels of uncertainty regarding the simulation increase as the input data may not remain suitable for the operating conditions and the model is not made to adapt by itself with these modifications. The limit in the modelling complexity also limits range of conditions that can be investigated for a given input dataset.

It must also be kept in mind when using (semi-) empirical models that these models have often been developed for specific conditions. If the engine operating conditions differ too much from that of the model, the probability of producing insignificant results increases.

1D simulations

Expanding from 0D to 1D simulations, the spatial dependency of a length parameter is added making possible to model gas flow in pipes [123]. The effect of pipe sizes on pressure waves can be captured, fuel spray can be modelled to a first level of complexity using the dimensions of the combustion chamber. More in-depth investigation of the engine behaviour is possible with these models, the level of reliability of the performance prediction and of the model adaptability to different operating conditions are increased. Nevertheless, for this to be achieved, it is necessary to possess more detailed data about the engine design and as the modelling complexity is increased, the required computation power increases too.

3D CFD simulations

The last level of complexity is 3D CFD engine simulations for which a CAD (digital) model of the engine flow and combustion chamber geometry is required.

With these models, complex flow structures, energy transfer and the impact of engine geometry can be investigated in great details. Because of the 3D nature of these simulations, the constantly varying geometry (movement of the piston) and the varying in-cylinder conditions (combustion, compression, expansion, ...), they require a lot of computation power and can take up to several days on high-performance computers to run. In addition, with this technique, generally only one part of the engine is modelled [57]; [101]. For instance, the combustion chamber and the combustion are modelled but the cooling system is not and inversely, using boundaries conditions to model the effect of one on the other. Another solution to mitigate the required computing power is to simplify the simulation to only focus on a specific aspect of the engine as done by Badiger who investigated the use of two-phase cooling using a simplified geometry instead of the exact engine design [124].

Even though very time consuming, 3D CFD simulations are necessary for in-depth understanding and development of high efficiency engine components and the continuous development of improved models allowing the accurate representation of innovative solutions is necessary to drive engine efficiency up.

Regarding the different simulation approaches mentioned in Chapter 1 and considered at Cranfield University, namely Step Forward, Brute Force and Computer Guided Design Optimisation, they are not applicable to 3D simulation tools. These three methods rely either on fast decision-making or fast computation of a large number of cases. This is thus not applicable in reasonable time for 3D CFD combustion simulations which take hours / days to complete. These simulation approaches require simulations capable of running in minutes, which is the case for steady state 0D/1D simulations.

Rate-kinetics simulations

In addition to complete engine simulations, combustion can be simulated alone using 2-zone zero dimensional rate-kinetics simulation. Based on very detailed fuel combustion mechanism file [125], rate-kinetics simulations can predict combustion progress and track the time evolution of the concentration of the different reactant, product and intermediate species for different temperature, pressures, and air/fuel ratios. In addition to the combustion process, rate-kinetics simulations can also predict the levels of emissions, such as NO_x, if included in the mechanism. This is by using this type of simulation that combustion can be incorporated into 3D CFD simulations [101].

Rate-kinetics simulations present an attractive alternative to the use of engine testing for the derivation of mass-fraction burned curve and their associated heat release curves. Those curves are likely to remain at the heart of 0D engine simulations as they have proven to be both rapid to calculate and reliable for a long time. But the engine testing required to provide the empirical basis of 0D models, commonly accomplished with single cylinder engines for Formula One™, has always been costly and with limits on testing imposed by regulations, alternatives must be kept under appraisal.

To set up a rate-kinetics simulation, the fuel/air combustion mechanism must be defined using a so-called ‘chemistry set’. A chemistry set is made of three data files describing the reaction mechanism, the thermodynamic properties (entropy, heat capacities) and the transport properties (density, viscosity) of the species contributing to the chemical reaction. The mechanism and thermodynamic properties files are necessary for all simulations, whereas the transport properties one is only required for simulations in which diffusion is investigated.

The mechanism file has three sections. The first is a list of the elements composing the species of the mechanism (carbon, oxygen, hydrogen for example), the second one announces all the molecules which will appear in the mechanism (reactants, final products and intermediate products) and the third one is the mechanism, i.e. a list of reactions, each associated with coefficients describing the modified Arrhenius law [125] of the specific reaction:

$$k = A_{arrhenius} \times T^n \times e^{(-\frac{E_a}{RT})} \quad (2-3)$$

where k is the rate constant, T the temperature, n the temperature exponent, $A_{arrhenius}$ the pre-exponential factor, E_a the activation energy, R the universal gas constant.

The thermodynamic file contains the NASA / Janaf coefficients [126] of every species featured in the mechanism. These coefficients describe the evolution of the species constant pressure heat capacity, enthalpy, and entropy with temperature.

The transport file contains coefficients used to calculate the viscosity and diffusion properties of every species intervening in the mechanism.

Even if fuels in the same families have reactions that are common across their mechanisms, the path to reach the final products through key intermediate species of the reaction is highly sensitive to the fuel molecular structure and to the combustion environment (pressure, temperature, air/fuel ratio for instance). For each fuel, a specific mechanism file is necessary, and it can vary depending on the conditions under which the fuel has been studied.

As described on the Lawrence Livermore National Laboratory (LLNL) website [127], which is an important reference for fuel combustion mechanism research with more than 90 publications made on this subject between 2016 and 2022 and mechanisms for more than 30 species publicly available on their website, several steps are necessary to obtain a species combustion mechanism. Once the reaction paths are identified, reactions are classified into different categories which have specific reaction-rate rules. These reaction rules are based on calculations at the atomic level (ab initio calculations) and fundamental measurements of species thermodynamic properties and reaction rates. Once the reaction rates are attributed and merged into a complete chemical kinetic model (mechanism), the model's predictions are compared against experimental

results from fundamental combustion devices such as shock tubes or rapid compression machines.

Mechanism files are highly detailed and only those with access to complex knowledge of the reaction sequence can configure them for a new or unusual blend of fuels. This restricts the use of rate-kinetics calculations as a tool for innovation. A lot needs to be known about a fuel before rate-kinetics can be used to accurately represent a fuel combustion behaviour.

As further explained by the Lawrence Livermore National Laboratory, common fuels like Diesel or gasoline are not pure chemical species, but are compounds and blends of hundreds of components, and it is very impractical to simulate the oxidation of each. Most of the combustion reaction mechanisms available in the literature concern single species fuel [128]. For gasoline and Diesel, mechanisms can be found under the form of 'surrogates' in which their composition has been narrowed down to a few components [129]; [130]. There appears to be opportunities to use surrogates as the basis for fuels innovation and that topic could be explored by future researchers but is beyond the scope of this present thesis.

Rate-kinetics simulation software like Ansys Chemkin [131] and fuel chemical kinetic models can be used to predict the combustion behaviour of the fuel (flame speed, rate-of-heat-release) under different conditions (pressure, temperature, air/fuel ratio) using models of well-known combustion devices (shock tubes, constant-volume 'bomb', etc.). This can be used in place of experimental testing with a degree of uncertainty depending on the mechanism, the combustion conditions, and the rate-kinetics code. The results of the rate-kinetics simulation can then be used in the engine simulation for a more detailed description of the combustion with varying combustion conditions.

An example of simulation results is shown in Figure 2-5 for a simple, single species fuel, in this case propane. The flame speed of propane is calculated at 1 bar at 300 and 700K.

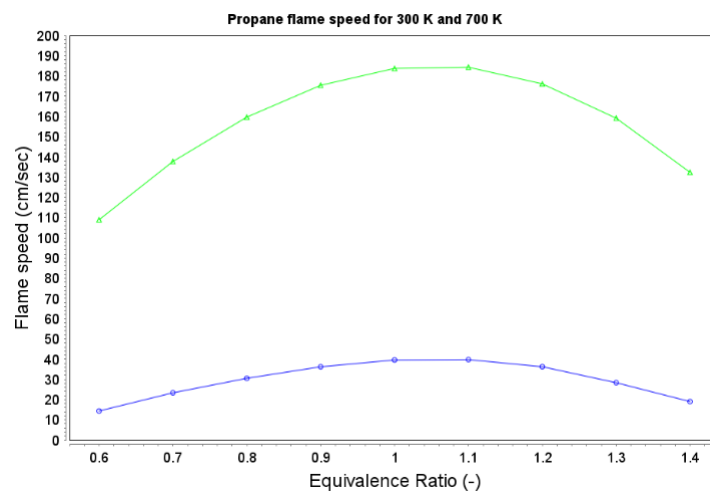


Figure 2-5 - Example of Ansys Chemkin's simulation results for propane flame speed calculations (Blue curve: 300 K – 1 bar / Green curve: 700 K – 1 bar)

The simulation results shown in Figure 2-5 are compared for equivalence ratios of 0.75, 1 and 1.2 against experimental data from the literature which are summarised in a graphic made by Zheng et al. [132] and shown in Figure 2-6. The green dots highlighted by the green arrows in Figure 2-6 represent the values predicted by Ansys Chemkin (refer to Figure 2-5) and are to be compared against the other points of the graphics.

It must be highlighted that the six simulation results placed within Figure 2-6 match the experimental measurements. This demonstrates that the combustion mechanism developed for propane and used in this simulation correctly models the combustion of propane under these specific conditions. The results produced using Ansys Chemkin and this mechanism can thus be considered reliable and used for engine simulation for example.

Nevertheless, Chemkin which could have been an interesting tool for this thesis has not been used for reasons described in Chapter 4.

The scope of Ansys Chemkin is not limited to combustion. This software targets at solving complex chemical kinetics problems, this includes combustion but also any chemical reaction for which a reaction mechanism exists.

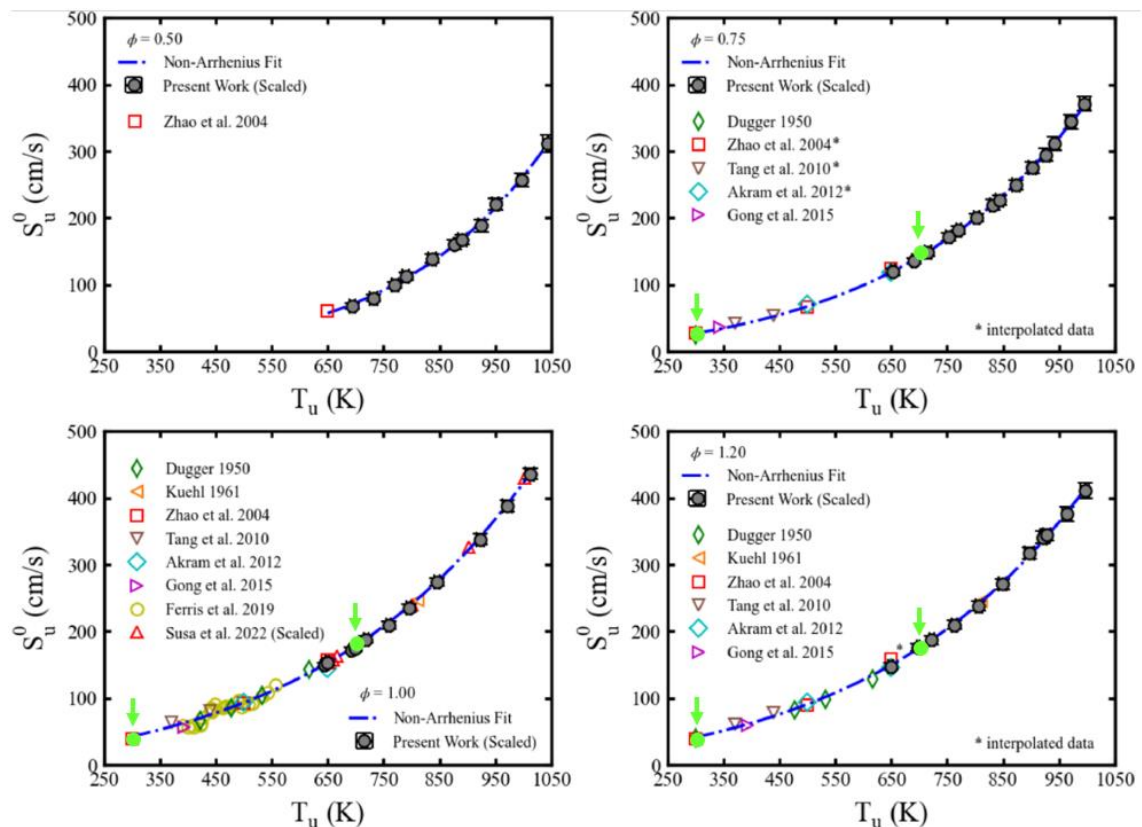


Figure 2-6 - Comparison of Ansys Chemkin propane flame speed prediction (green dots) against literature data (background graphics from [132])

Using open OD reactors and flow reactors, Ansys Chemkin can be used to simulate chemical reactions occurring in large-scale chemical-producing plants. Using CVD and LPCVD reactors, the software can investigate chemical vapor

deposition (CVD) which is used to deposit layers of a substance onto a surface. Using shock tubes and flame simulators, Ansys Chemkin can calculate flame speeds for different conditions and validate reaction mechanisms against experimental data. Using 0D engine models, it can be used to model the mixture within the cylinder during the closed-valve phase of an engine cycle and predict the performance of this part of the engine operation but does not simulate gas exchange processes. If the selected mechanism incorporates NO_x formation, those emissions levels can be predicted.

In addition to the simulation models, Ansys Chemkin also features tools to modify reaction mechanisms (merging, reduction, optimisation), optimise the composition of the fuel surrogate to match user-defined targets and estimate thermodynamic and liquid properties of a given chemical structure.

Kintekus is another software very similar to Chemkin. There is also KinTek, which is based on the same principles, but this is specialised in enzyme chemical kinetics. Rate kinetics software is often specialised in specific fields but if a chemical reaction occurs and if a detailed mechanism of this latter exists in the correct format, it can be studied using any rate kinetics tool.

To design effective combustion systems in an environment such as Formula One™ where engine testing is restricted, the motorsport industry needs to simulate combustion events in 3D environments, i.e. conduct simulations where combustion is part of a CFD simulation. By integrating fuel combustion mechanisms, it becomes possible for the CFD simulations to predict the propagation of the flame and its interaction with the flow within the combustion chamber. Commercial CFD codes such as AVL Fire and Converge CFD apply rate kinetics approaches although these are used differently to tools such as Chemkin which are used for more fundamental studies.

Nevertheless, reaction mechanisms will still need to be located, adapted or specifically developed to provide accurate results under engine operating conditions as mentioned by Yang [133]. As in-cylinder CFD simulations are already extremely heavy in their use of computing power, adding a detailed combustion mechanism to describe the flame propagation in addition to the flow in the combustion chamber significantly increases the simulation duration. This is why, as described by Lin et al. [134] and Ismail et al. [135], that it is often necessary to rely on reduced mechanism which only model the main reactions of the mechanism while not compromising too greatly the predictions.

2.5.2 Cranfield University and 1D engine simulations

As discussed in Chapter 1, Cranfield University considers the learning potential from three different approaches regarding 1D engine simulations:

- Step Forward philosophy: Based on simulation results, rapid design decisions are made and tested. Those that seem promising are continued while the others are rejected and so the design 'steps forward' one step at

a time. This is the classic method and is very similar to what is done when only physical testing is available for development,

- Brute Force philosophy: Based on high computation power, an attempt at true multi-variable optimisation is made by testing a very large number of combinations of variables values with the engineer analysing and extracting the preferred design from the pool of results,
- Computer Guided Design Optimisation philosophy: Based on optimisation algorithm finding the optimal design against a composite objective function within user-defined limits.

The first two philosophies were compared during the 2022 Cranfield F1™ Design Sprint with two groups of students, one for each philosophy, asked to develop an engine model based on the 2026 Formula One™ Power Unit regulations. The idea was to assess how each philosophy will impact the development of the engine model, what the students would learn from it and if the performance of the final models would be different.

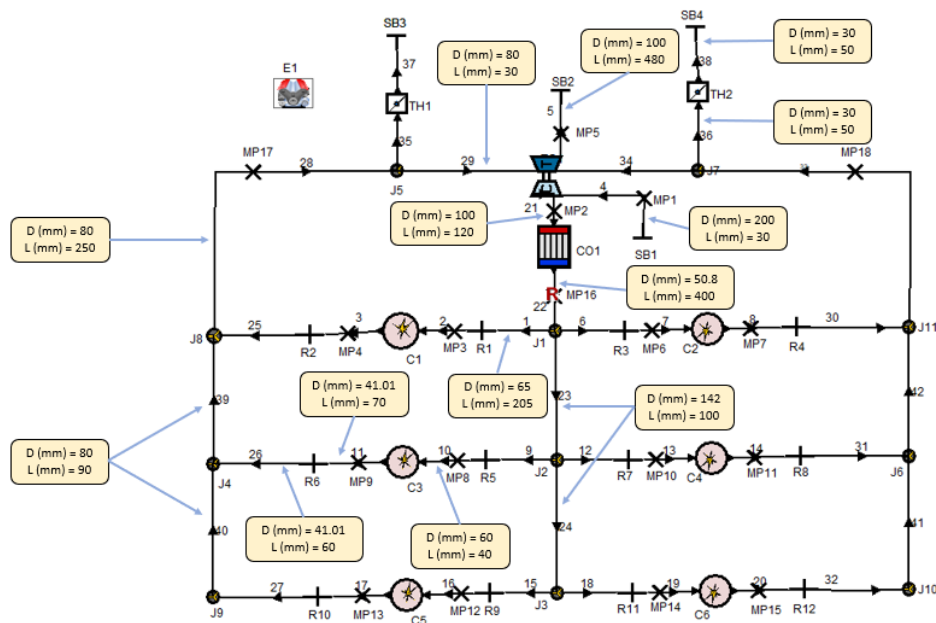


Figure 2-7 - 2022 Cranfield F1™ Design Sprint - Step Forward group geometry

The geometries of each group are shown in Figure 2-7 and Figure 2-8. For the Brute Force group, the optimal geometry was selected by extracting the most interesting results from the large number of simulations they ran. On the other hand, the Step Forward group designed its engine by “trial and error”. Both philosophies result in different geometries which have an impact on the engine performance.

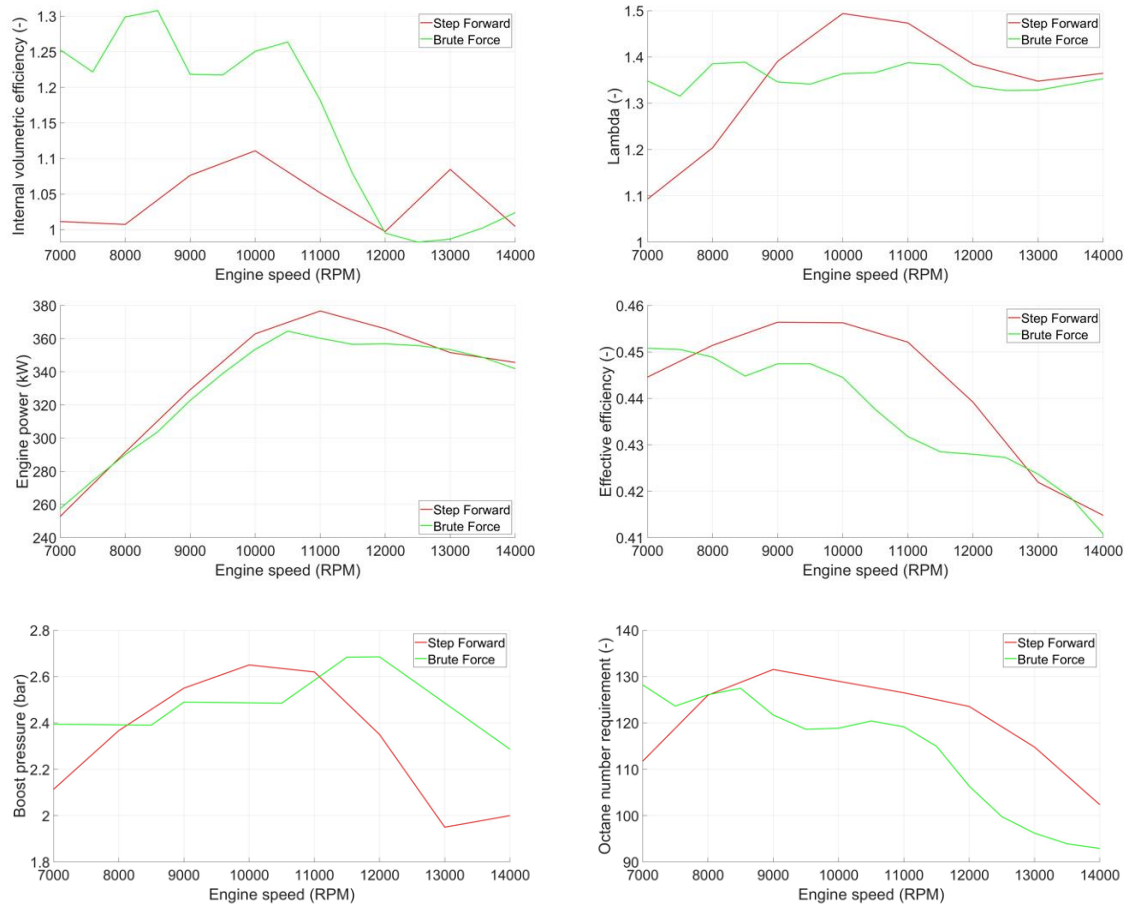


Figure 2-9 - 2022 Cranfield F1™ Design Sprint – Step Forward and Brute Force results comparison

Beyond the simulation results, the feedback received from the students was that each group would have liked having the skills developed by the other group. The Step Forward would have benefited from more computational power to test more of their ideas following the analysis of previous simulation results whereas the Brute Force group would have enjoyed developing a more in-depth understanding of the engine behaviour to better understand the results produced by the simulations and how the development should be oriented. A hybrid philosophy with intermediate computing power and higher control of the engineer on the tested design choice seems to be an interesting solution to test.

Regarding the Computer Guided Design Optimisation philosophy, this approach was developed by Frankie Chapman during his MSc thesis at Cranfield University for the development of a road car engine [16]. AVL Boost™ engine simulations were coupled with a MATLAB script running optimisation algorithms aimed at finding the optimal results within user-defined limits to constrain engine design and using a scalarisation function to weigh the different performance indicators.

The interesting finding of this thesis is that beyond improving performances compared to real results, the designs proposed by the optimisation algorithms were uncommon, with solutions that an experienced engine designer would not

have necessarily thought about because very different from conventional practices.

This work demonstrates the benefits that could be brought by introducing Machine Learning and Artificial Intelligence tools at specific steps of the engine development process with a physical person validating the final result to ensure its feasibility.

2.6 Scientific knowledge about the performance characteristics of Formula One™ engines

2.6.1 Formula One™ engines in the literature

Due to the competitive nature of the sport, compared to the efforts put into the development of these engines, only a small number of papers on the actual design or operation of Formula One™ engines is published by the Power Unit manufacturers.

When it is done, the papers are often published once the engine generation of interest has become obsolete to ensure that no competitive advantage is given away to the competitors. This was done by Coventry Climax in the 1960s [3] to describe the development of their engine between 1961 and 1965, by Honda at the end of the first turbocharged era to present the behaviour and the sensitivity analyses conducted on their RA168E engine [6] but also in 2009 when Honda retired from the sport at the end of the 2008 season to show the development made on their V10 but also V8 engines during their third period of presence in Formula One™ [136]. In the same period, BMW which was present in Formula One™ from 2000 and withdrew in 2009 presented details on their engine performances in 2010 [2]. Prior to that, in 1996, Chrysler published papers on the performance and the control systems of their 1993 engine, i.e. the engine that raced in their final year in the sport [137]; [138].

As the current generation of Power Unit, introduced in 2014, is still in place and because the future regulations that will be introduced in 2026 will not constitute a complete change, it is likely that in-depth insights about these high-efficiency Power Units, which is currently unavailable for the public as highlighted by Palakunnummal et al. [57], will stay this way for again several years.

Nevertheless, it is possible to find articles written by engineers working in the industry or researchers related to specific Formula One™ teams about concepts applicable to the current Power Unit but that do not reveal any in-depth knowledge about the engine [58]; [77]; [139]; [140].

Researchers are also looking at the technologies potentially enabling the high efficiency of the 2014+ Formula One™ Power Unit without evident input from the industry [57]; [141]; [142].

But to the author knowledge, there is no in-depth analysis of the thermodynamic behaviour of the 2014+ Formula One™ or of an electrically assisted turbocharged lean-mixture high-compression engine featuring a pre-chamber ignition system, which would share similarities with the current Formula One™ engine, available in the literature.

2.6.2 Cranfield F1™ Design Sprint

As presented in Chapter 1, the Cranfield F1™ Design Sprint is an extra-curricular student-led activity aiming at establishing an independent scientific appraisal of the development of Formula One™ powertrains. Every year, Cranfield students of the Advanced Motorsport Engineering and Mechatronics MSc are invited to participate voluntarily in this project run for two months. To keep its independent view, the project does not receive any data input from the Formula One™ industry but each year, the results are shared with members of that industry for feedback and advice regarding improvements. In addition to the work done on the understanding of the Formula One™ engine, this activity trains future engineers of the industry and has made a significant contribution to the development of pedagogy at Cranfield University.

This project has been run every year since 2018 (in 2022, the project turned its attention to the 2026 Formula One™ Power Unit regulations) and with the feedback of the industry on the results produced by the students, several facts about the 2014+ Formula One™ engine have been established:

- Lean mixture: The engine runs at higher air/fuel ratio than stoichiometry ($\lambda_{peak} \approx 1.45$) to ensure complete combustion and increase fuel conversion efficiency,
- In-cylinder pressure limit: The engine is operating at its mechanical limit in terms of peak in-cylinder pressure. The pressure limit is of 300 bar which corresponds to 280 bar if the connecting rod deformation is accounted for,
- Waste gates operation: The turbine can swallow the entirety of the engine mass flow and the waste gate role is to be a pressure-relief valve rather than a flow diverting device,
- MGU-H power: The MGU-H peak power is around 65 kW.

Nevertheless, the Cranfield F1™ Design Sprint focused on the analysis of the operation of the 2014+ Formula One™ engine and never on the detailed investigation of its thermodynamic behaviour, the same observation is done when looking at the literature. There is thus a gap in the body of knowledge regarding the scientific understanding of the 2014+ Formula One™ Power Unit.

In addition, the engine models created for the Cranfield F1™ Design Sprint did not benefit from industry inputs or from sensitivity analyses regarding the impact

of the input dataset on the model's predictions unlike the 2014+ Digital Twin proposed in this thesis. This means that the confidence inspired by these models regarding their capacity to exhibit similar performance trends as the real engine and to be used as a reliable source of data is lower than that of the proposed 2014+ Digital Twin.

3 Assessment and management of uncertainties for the creation of the 2014+ Digital Twin

3.1 Categories of parameters defining an engine and associated uncertainties

When modelling the performances of an internal combustion engine, creating a Digital Twin, a number of parameters are required to describe its geometry and behaviour. These parameters can be differentiated into three categories:

- **Design parameters:** This refers to the geometry of the engine (intake, exhaust, combustion chamber for example). The best way to estimate these parameters is to directly measure them on the engine or better from a 3D CAD model. These are linear functions. They are in fact lines or planes in space,
- **Operational / calibration parameters:** These represent the parameters set by the engineers to control the engine and extract the maximum out of a fixed design. Ignition timing, injected fuel flow and waste gate angle belong to this category. The best way to know about the operation / calibration parameters is to ask practitioners as they are part of the control strategies and not visible from the outside. The data would be delivered as look-up tables of calibration parameters against engine speed and load. So, these are in fact 2D surfaces in space,
- **Compound parameters:** These parameters are a consequence of prior selection of both design and operation parameters; engine designers cannot directly modify them but only influence them with other parameter selections. Combustion duration is one of these as it is influenced, among others, by turbulence intensity which is a consequence of intake geometry (design parameter) and air-fuel mixture (calibration parameter). It varies with both engine speed and load. It is very difficult to estimate these compound parameters even when having access to the engine. If ever measured, this is done while the engine is running with intrusive instrumentation and even sometimes indirectly (Ex: rate-of-heat-release (ROHR) curve derived from pressure trace). At their simplest, because they inevitably vary with both engine speed and load, there are a set of 2D surfaces in space but in fact are often more multi-dimensional than that.

When modelling an engine using a simulation software, the precision with which these parameters are included in the model will influence the realism and accuracy of the performance prediction and/or the behaviour of the Digital Twin.

As previously explained, for this research project, only a limited number of these parameters are available to set up the engine models. The rest of the parameters need to be estimated, leading to inherent uncertainties on the chosen values and therefore on the predicted performances. The uncertainties are more limited for

those parameters that are simple linear functions. There is likely to be a single optimum value that can be uncovered through sensitivity testing of the model. But the 2D surfaces will have many local optima and only one global optimum which is notoriously hard to find with confidence.

The model of the 2014+ Formula One™ engine presented in this thesis is referred to as a 'Digital Twin' as it is the best equivalent to the physical V6 engine that can be obtained using the available data. This model benefited from the input and feedback of the industry through the Cranfield F1™ Design Sprint and the sponsor of this thesis and from an assessment of uncertainty linked to its input dataset which discussed in this chapter.

To obtain the qualification of Digital Twin, the model must validate the three aforementioned criteria. The complete 2014+ Formula One™ engine model is the only model discussed in this thesis which is a Digital Twin.

The 2014+ Digital Twin created for this project does not claim nor aim to perfectly replicate the performance behaviour of the 2014+ engine but aims at inspiring confidence in its ability to faithfully exhibit similar performance trends against varying engine speed and load. To fulfil this objective, the Step Forward method has been implemented and sensitivity analyses have been conducted on the different parameters influencing the model's performance. The Step Forward method is commonly used to choose parameters which are not known in detail but expected to be linear functions. Experience in the Cranfield F1™ Design Sprint shows that it is less well suited to selecting values for parameters which are 2D surfaces.

3.2 Design methodologies adequacy to cope with uncertainties and analysis of the 2014+ regulations using engine governing equations

As explained in Chapters 1 and 2, the learning potential of three design methodologies are considered at Cranfield University when developing 1D engine simulations (Step Forward, Brute Force, Computer Guided Design Optimisation). All three can help to cope with the lack of information on Formula One™ engines but in different manners which don't necessarily result in the same performance prediction and the way the results are obtained influences the manner in which the user builds up his / her understanding of the engine and how it operates.

The Brute Force and Computer Guided Design Optimisation approaches, by testing a large number of combinations, are potentially more likely to find an optimal design which maximises performance, making them attractive for design tasks. Nevertheless, when creating an engine's Digital Twin, maximising performance is not the sole or even main target; on the contrary replicating the behaviour trends of a specific engine is as important.

As one of the objectives of this thesis is to bring a scientific understanding of the 2014+ Formula One™ engine, the Step Forward methodology is deemed the most suitable option. This approach relies significantly more on the engine designer and the development of his / her understanding of the engine by referring to the governing equations to orientate the model improvement.

The engine model developed using the Step Forward methodology is more likely to only be a local optimal within the field of variables tested (rather than the global optimum of all possible variables) but by basing the development on the designer's understanding of the engine rather than entirely on computer science, it is easier to surface the scientific reasons than might have influenced F1™ engineers design decisions and to minimise uncertainties by setting parameter values in line with the observation brought by the governing equations for engine performance.

By looking at the 2014+ regulations through the governing equations, a number of conclusions can be drawn regarding how to deal with a fuel-limited engine and make the most out of the limited energy.

From a broad point of view, for a given indicated power P_I , the brake output power P_b can be increased by minimising the friction losses P_f (Equation 3-1). As the 2014+ regulations limit the amount of energy usable, the indicated power that can be produced is also inevitably limited. Thus, reducing the friction losses (i.e. increasing η_m , the mechanical efficiency) helps, as for all engines, to maximise the power output of the engine but becomes even more significant when it cannot be overcome by burning more fuel.

$$P_b = P_I - P_f = \eta_m \times P_I \quad (3-1)$$

The indicated power P_I can be further described by the following equation:

$$P_I = \eta_f \dot{m}_f Q_{lvh} \quad (3-2)$$

As a fuel-limited engine, the fuel mass flow $\dot{m}_f$ of the 2014+ engine is fixed by the regulations. In addition, Q_{lvh} , the fuel lower heating value, is a characteristic of the supplied fuel and does not change with engine design which leaves the fuel conversion efficiency η_f as the only lever.

Equation 3-3 gives a more detailed description of fuel conversion efficiency:

$$\eta_f = \eta_c \times \eta_{th} \quad (3-3)$$

where η_c represents the combustion efficiency, i.e. the ability to release fuel energy, and η_{th} the thermal efficiency, i.e. the efficiency to convert heat into work.

Combustion efficiency is influenced by the air/fuel ratio at which the engine operates. By running at stoichiometric air/fuel ratio or higher, enough oxygen is introduced in the combustion chamber to oxidise all injected fuel molecules. The more fuel molecules are oxidised, the more fuel energy can be released, thus maximising the combustion efficiency. For fuel-limited engines, running at $\lambda \geq 1$ is key to maximise the amount of fuel energy released during combustion and

make a step towards high fuel conversion efficiency. Nevertheless, increasing lambda changes the burn rate significantly as past stoichiometry, the leaner the mixture, the lower the laminar flame speed (see Chapter 2). On a real engine, to make the most out of dilution, the loss in laminar flame speed must be compensated, by increasing in-cylinder turbulence, with a pre-chamber ignition system for example.

Despite not being practically achievable, being based on ideal assumptions including instantaneous combustion and isentropic compression and expansion strokes, no heat cooling losses and idealised intake and exhaust processes for the Otto cycle [46], analysing the equations describing ideal thermodynamic cycles alongside engine governing equations highlight potential ways of maximising the power output of a fuel-limited engine.

By considering the Otto cycle (Figure 3-1), despite being an ideal cycle, means to increase thermal efficiency can be highlighted. Its thermal efficiency is calculated using the following equation [46] (already shown as Equation 2-1):

$$\eta_{th} = 1 - \frac{T_4}{T_3} = 1 - \frac{1}{CR^{\gamma-1}} \quad (3-4)$$

where T_3 represents the mixture temperature at the end of the combustion, T_4 the mixture temperature at the end of the expansion stroke, CR the engine geometric compression ratio and γ the mixture heat capacity ratio.

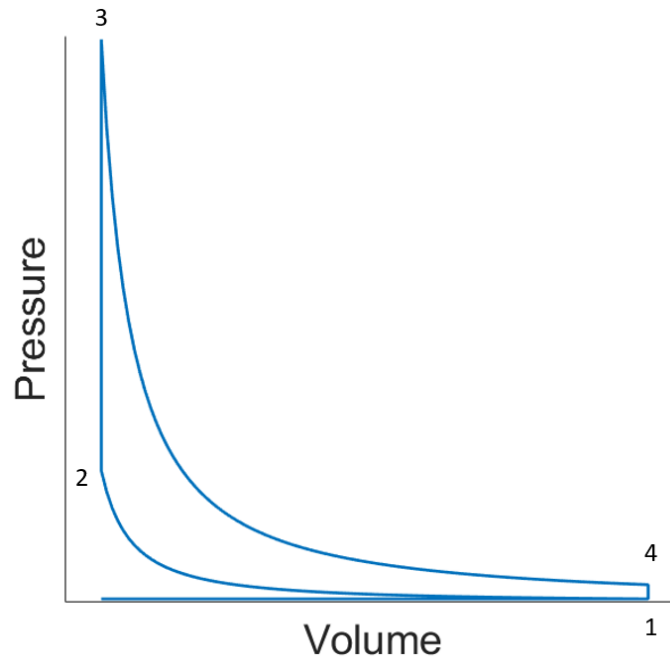


Figure 3-1 - Otto's cycle PV diagram

Equation 3-4 shows that thermal efficiency, i.e. the conversion of heat released during the combustion into useful work, increases as the ratio of the temperature at the end of the expansion stroke over the temperature at the beginning of this same stroke increases. The higher the temperature difference, the more efficient the expansion stroke and the more heat is converted into work.

Increasing the geometric compression ratio of the engine, $CR = \frac{V_4}{V_3}$ will lead to a higher temperature/pressure at combustion TDC and a higher expansion ratio, thus maximising the temperature difference and the thermal efficiency as shown by the second expression of η_{th} . Equation 3-4 also shows that increasing the heat capacity ratio of the mixture γ will drive the efficiency up.

The heat capacity ratio of air being higher than CO₂ (at 25°C and 1 bar, 1.400 against 1.289 [46]), increasing the proportion of air, i.e. increasing air/fuel ratio, will increase the heat capacity ratio of the whole mixture for the expansion stroke and lead to higher thermal efficiency. In addition to maximising fuel energy release, increasing λ also helps with the conversion of heat into work by reducing the temperature of combustion, and thus heat transfer to the cooling system, and enables to reach higher fuel conversion efficiency. Running high level of dilution is thus a natural decision when looking at maximising efficiency.

Even though, its value is fixed depending on engine speed, the fuel mass flow $\dot{m}_f$ can be expressed by the following equation which is not limited by simplifying assumptions on the contrary to the previous Otto cycle equations:

$$\dot{m}_f = \frac{\eta_v N V_{swept} \rho_a}{2 \times AFR_{st} \lambda} \quad (3-5)$$

where η_v is the engine volumetric efficiency, N the engine speed, V_{swept} the engine swept volume, ρ_a the inlet air density, AFR_{st} the fuel stoichiometric air/fuel ratio and λ the relative air/fuel ratio (AFR/AFR_{st}).

For a given air/fuel ratio target, the higher the volumetric efficiency η_v , the lower the air density ρ_a needs to be. By carefully designing the engine intake and exhaust systems with associated valves, ports and camshafts, volumetric efficiency can be increased, reducing in return the requirement in terms of inlet air density, i.e. of compressor power. If the compressor power is reduced, either more power can be recovered by the MGU-H at equivalent turbine, or the turbine design can be modified, and the backpressure reduced.

As demonstrated by Heywood [34], compared to ideal pressure-limited and constant pressure cycles, the Otto cycle is the most efficient. This demonstrates the necessity to achieve a combustion as close as possible to constant volume, that is minimising the combustion duration.

Nevertheless, when using these governing equations, the designer needs to keep in mind that they do not consider practical limitations like mixture ignitability, spontaneous ignition or mechanical limits which will condition the design of the real engine.

As stated in the introduction (Chapter 1), two totemic claims are made by the Formula One™ industry: the 2014+ PU produces more than 1,000 hp and the engine reaches over 50% brake efficiency [13]; [14].

By combining Equations 3-1, 3-2 and 3-3, the engine brake power can be expressed as:

$$P_b = \eta_m \eta_{th} \eta_c \dot{m}_f Q_{lvh} \quad (3-6)$$

The lower heating value of the fuel used in Formula One™ engines is estimated at 44 MJ/kg [5]. By using this value, the maximum fuel flow allowed by the regulations, 50% as brake efficiency, the maximum power at the crankshaft of the engine is predicted at 819.5 hp (611.1 kW). With the MGU-K providing 160.9 hp (120 kW), the total PU power is estimated at 980.4 hp (731.1 kW) which is 20 hp shy of the claimed 1,000 hp.

According to the same equation, a brake efficiency of 51.2% is required to reach 1,000 hp. By considering 5% mechanical losses, typically 10% for automotive engines [34], an engine fuel conversion efficiency (indicated efficiency) of 53.9% is needed.

As observed in the literature, internal combustion engines with specific combustion technology, namely HCCI, are promising combustion solutions which can reach thermal efficiency approaching 50% [143]; [144]; [145]. The engines used in these publications are Cooperative Fuels Research (CFR) engines or commercially available Diesel engines converted to HCCI, i.e. engine which have not been specifically developed for HCCI. With specifically developed engines, it can be imagined that the 50% thermal efficiency barrier can be overcome. Nevertheless, HCCI engines are mostly operated at low BMEP values (< 5 bar).

From the point of view of the governing equations, the claims made by Formula One™ engine manufacturers are pertinent. Nevertheless, to understand in detail how these figures can be achieved, building a Digital Twin of the engine is necessary.

3.3 Uncertainty management strategy

Under more conventional circumstances, engine performance models are created and validated by measurements made on the engine of interest, either by directly conducting experiments as part of the research project or by using data provided by a third party. The model is considered validated when its predictions match the measured data.

For this research project, it is not possible to test a current Formula One™ engine to make measurements and access to third party data is extremely limited, thus there is no opportunity to validate the Digital Twin of the 2014+ engine against a full set of measurements.

When selecting the input parameters necessary to construct and run the 2014+ Digital Twin, assumptions must be made because of this limited access to geometrical data about the engine and calibration data on how the engine operation is managed. This lack of hard data leads to inevitable uncertainties. It is unlikely that the values selected for even simple geometrical dimensions, for

example exhaust pipes diameters and lengths, are exact matches for the ones on the real engines, plus, specific calibration parameters vary with engine operating conditions, such as air/fuel ratio, which make uncertainties in the input dataset for the 2014+ Digital Twin greater.

This uncertainty in the input dataset results in uncertainty over the realism of the model's performance prediction output dataset. Under different circumstances, the inputs would be validated by measurement or inspection and the outputs by performance measurements of the engine on a dynamometer or of the vehicle on a track. However, in this unusual case of sparse data, to make sure the predictions made by the engine model can be trusted and used for additional analyses, the level of uncertainty on output performances caused by the uncertainties in the input dataset must be assessed and managed.

The methodology adopted in this case is to verify the input dataset making sure that the values chosen for the design and operational input dataset parameters are within expected ranges individually and are coherent with one another as a set. For instance, there are common lengths and diameters used for exhaust systems. Expected dimensions can be set and although Formula One™ engines are very high-performance, not all their characteristics differ too significantly from a more conventional engine for which complete input datasets are available.

After establishing the plausibility of the input dataset, the next step is to check that the compound parameters, which are influenced by the select input dataset, also lie in expected / common ranges according to the literature, private communication, or feedback from the industry. The term 'compound parameter' is used here to describe measurable engine metrics that result from more than one input parameter. For example, internal volumetric efficiency (which is here referenced to the intake plenum conditions as the 2014+ Digital Twin is turbocharged) depends on many parameters including the intake and exhaust geometries, valve diameter and lift, flow loss coefficients, camshaft timing and others. If simulation results, with plausible input dataset where all parameters are within expected ranges, predict compound parameters to be also in expected ranges, then confidence in the modelling is raised further.

A third element to managing uncertainty is the use of sensitivity analysis on individual parameters. By sweeping across a range of values for the parameter under consideration, it can be reasonably assumed that the real engine actual value is covered. It is also possible to assess the importance of any particular input parameter on the output performance and therefore how great the performance gap could be if there is a difference between the value selected for the 2014+ Digital Twin and that of the real engine.

The lower the importance of single or multiple uncertainties over input data, the more the model's performance prediction can be trusted. Inversely, the higher the significance, the more precautions must be taken for selecting the value of the parameter(s).

The sensitivity analyses are also used to confirm some of the choices made for the 2014+ Digital Twin. It can show that when a specific parameter is modified,

the engine performance prediction does not respect anymore the facts listed in Table 3-1 (see below) which are considered as references for the creation of the 2014+ Digital Twin. This makes sure that the overall behaviour of the engine is coherent with that of the real engine. This is complementary to the verification made on the compound parameters and their expected ranges discussed previously.

3.4 Review of the data necessary for the creation of the 2014+ Digital Twin and of the selected values

The AVL Boost™ simulations used for this research project are based on parameters of all three categories presented in Section 3.1. Even though some of the parameters are easily measurable when having access to the engine, the secretive aspect of Formula One™ means that even this information is not made available to the public.

For the setting up of the 2014+ Digital Twin, the selection of the input parameters has been guided by data provided by the sponsor of this Doctoral Thesis and insights gathered during the Cranfield F1™ Design Sprint which are summarised in Table 3-1 [5].

Information	Source
The 2014+ engine relies on shorter combustion duration than conventional spark ignition engine	Feedback from the industry for the Cranfield F1™ Design Sprint
The 2014+ engine is pressure limited with a maximum pressure of 300 bar (around 280 bar if connecting rod compression is considered)	Feedback from the industry for the Cranfield F1™ Design Sprint
The compression ratio used by the 2014+ engine is lower than the regulation limit	Sponsor
The turbine of the 2014+ engine can swallow the whole engine mass flow at any engine speed	Feedback from the industry for the Cranfield F1™ Design Sprint
The engine operation window is mainly between 9,000 and 12,000 RPM	Sponsor
The 2014+ engine dilution limit is $\lambda = 1.45$	Feedback from the industry for the Cranfield F1™ Design Sprint
Specific range for the fuel RON	Sponsor
The MGU-H maximum power is 65 kW	Feedback from the industry for the Cranfield F1™ Design Sprint

Table 3-1 - List of the information used to guide the development of the 2014+ Digital Twin

Table 3-2 gives the list of the parameters necessary to run engine models with AVL Boost™, and explains how the values have been selected for the 2014+

Digital Twin in addition to presenting how the potential uncertainty linked to each can affect the simulation predictions:

Parameter	Adopted value(s) and explanation	Potential uncertainty impact
Design parameters		
Geometries		
Intake geometry	<p>The intake pipe geometries have been selected following the Step Forward methodology and targeting at improving the engine volumetric efficiency using common lengths and diameters</p> <p>Dimensions shown in Chapter 4, Figure 4-5</p>	<p>Affect the engine volumetric efficiency and thus the boost pressures required from the compressor to achieve the target air/fuel ratios</p> <p><i>Internal volumetric efficiency assessed in Section 3.5.1</i></p>
Exhaust geometry	<p>The exhaust pipe geometries have been selected following the Step Forward methodology and targeting at improving the engine volumetric efficiency using common lengths and diameters</p> <p>Dimensions shown in Chapter 4, Figure 4-5</p>	
Combustion chamber geometry	<p>Bore and stroke defined using the regulations [8]</p> <p>Compression ratio chosen as 16:1 because known to be lower than 18:1 (regulations limit) and [57] suggests that it is higher than 13:1. Plus, the 2026 regulations set the compression ratio limit at 16:1 which can suggest that this is the currently used value</p> <p>Con-rod length and piston offset provided by the sponsor of this Doctoral Thesis</p>	<p><i>Can affect the engine performance prediction but considered accurate</i></p>
Heat transfer		
Heat transfer surface area – Combustion chamber	Defined using the recommendations of AVL Boost User Manual [146].	<p>Affect the heat transfer and performance predictions and thus the engine energy balance</p> <p><i>Sensitivity analysis in Section 3.5.2</i></p>
Heat transfer surface area – Ports	Defined using the pipe dimensions attributed to the ports (considered as cylinder's surface area)	<p>Affect the heat transfer and performance predictions and thus the engine energy balance</p> <p><i>Sensitivity analysis in Section 3.5.2</i></p>

Parameter	Adopted value(s) and explanation	Potential uncertainty impact
Turbocharger		
Turbine swallowing curve	The turbine swallowing curve has not been taken from an existing commercial turbocharger but is a user-defined curve aiming at swallowing the whole engine mass flow and recovering adequate amount of power from the exhaust gas	Affect the exhaust back pressure, thus the volumetric efficiency, the amount of power produced by the turbine and the power of the MGU-H <i>Internal volumetric efficiency and MGU-H power assessed in Section 3.5.1</i>
Air delivery		
Valve lift profiles	Provided by the sponsor of this Doctoral thesis	Affect the volumetric efficiency of the engine and thus the boost pressure required from the compressor <i>Internal volumetric efficiency assessed in Section 3.5.1</i>
Intercooler		
Intercooler settings	Set for a target outlet temperature of 40 °C with a 0.2 bar pressure drop	Affect the boost pressures required to achieve the target air/fuel ratios, volumetric efficiency and octane requirements <i>The exact behaviour of the intercooler is not of interests for this Thesis and internal volumetric efficiency assessed in Section 3.5.1</i>
Friction		
Valve train geometry	Set as Double Direct Acting Overhead Cams, 10 camshaft bearings, Flat Follower	Influence the friction losses predictions and thus the engine energy distribution <i>Camshaft bearing number sensitivity analysis in Section 3.5.2</i>
Operation / calibration parameters		
Combustion		
Start of combustion	Based on the ROHR curve provided by the sponsor of this Doctoral Thesis Kept constant for all engine speeds	Affect the engine performance prediction and is likely to change with engine operating conditions <i>Sensitivity analysis in Section 3.5.2</i>

Parameter	Adopted value(s) and explanation	Potential uncertainty impact
Air delivery		
Lambda evolution with engine speed	<p>For all engine speeds, lambda is kept between 1 and 1.45</p> <p>For the highest engine speeds, lambda is decreased to consider the difficulty to maintain stable and reliable lean combustion</p>	<p>Affect the amount of air necessary in the combustion chamber and thus the boost pressures required from the compressor. Will also affect the engine performance prediction</p> <p>Sensitivity analysis in Section 3.5.2</p>
Compressor operating points	<p>Compressor pressure ratios have been defined manually to match the air/fuel ratio targets and are influenced by the engine volumetric efficiency</p> <p>Isentropic efficiencies have been set to values ranging from 0.75 to 0.8, corresponding to efficiencies of a commercially-available high-efficiency compressor</p>	<p>Affect the power recoverable by the MGU-H</p> <p>Values congruent with the feedback from the Cranfield F1™ Design Sprint and MGU-H power assessed in Section 3.5.1</p>
Fuel		
Fuel stoichiometric air/fuel ratio	<p>Set at 14.7 as one of the possible values for gasoline</p> <p>Shell's F1™ fuel contains 99% of the same types of compounds as their high-performance road fuel [11]</p>	<p>Affect the amount of air required in the combustion chamber to achieve a specific air/fuel ratio and thus the boost pressures required from the compressor</p> <p>Sensitivity analysis in Section 3.5.2</p>
Fuel heat of evaporation	Set at 419 kJ/kg, the recommended value by AVL Boost User Manual for Premium Gasoline [146]	<p>Affect the amount of heat required to evaporate the fuel droplets and thus the in-cylinder conditions</p> <p>Sensitivity analysis in Section 3.5.2</p>
Fuel temperature	Set at 25 °C	<p>Affect the amount of heat required to evaporate the fuel droplets and thus the in-cylinder conditions</p> <p>Sensitivity analysis in Section 3.5.2</p>
Friction		
Oil type	<p>Set as SAE 10W60</p> <p>The oil with the highest viscosity is selected considering the loads faced by the moving parts of the engine and the necessity to maintain a sufficient oil film thickness</p>	<p>Depending on the oil chosen, the viscosity will vary, influence the friction calculations and the engine energy distribution</p> <p>Sensitivity analysis in Section 3.5.2</p>

Parameter	Adopted value(s) and explanation	Potential uncertainty impact
Oil temperature	Oil temperature set at 100 °C as a common value	The higher the oil temperature, the lower the viscosity, the lower the friction losses Influence the engine energy distribution Sensitivity analysis in Section 3.5.2
Compound parameters		
Combustion		
ROHR curve	Provided by the sponsor of this Doctoral thesis for a specific engine speed Represented as a Weibe curve Used for all engine speeds	Influenced by the conditions within the combustion chamber during the combustion event so likely to vary with engine speed and affect engine performance prediction Sensitivity analysis in Section 3.5.2
Combustion duration	Set at 41°CA based on industry advice for the Cranfield F1™ Design Sprint Kept constant for all engine speeds	Influenced by the conditions within the combustion chamber during the combustion event so likely to vary with engine speed and affect the engine performance prediction Sensitivity analysis in Section 3.5.2
Heat transfer		
Combustion chamber wall temperatures	Temperatures set following the recommendations provided by Harrison [147]	Affect the heat transfer prediction and thus the engine energy repartition Sensitivity analysis in Section 3.5.2
Port wall temperatures	Temperatures set at 150 and 240 °C for the intake and exhaust ports respectively	Affect the heat transfer prediction and thus the engine energy repartition Sensitivity analysis in Section 3.5.2
MGU-H		
MGU-H power	The MGU-H power depends on the turbocharger It is set as the difference between the compressor requirement and the power recovered by the turbine and varies with engine speed	This parameter does not impact the performance prediction of the model MGU-H power assessed in Section 3.5.1

Parameter	Adopted value(s) and explanation	Potential uncertainty impact
Fuel evaporation		
Fraction of the heat necessary to evaporate the fuel coming from the combustion chamber walls (Heat from wall)	Set at 0.3 by default	Affect the heat transfer prediction and thus the engine energy repartition <i>Sensitivity analysis in Section 3.5.2</i>
Air delivery		
Valve flow coefficients	Flow coefficients set using recommendations provided by Harrison [148]. These coefficients depend on the geometry of the ports and of the valves and are unknown. It was thus decided to use conventional values The valve flow coefficients used for the 2014+ Digital Twin are shown in Appendix C	Influence the volumetric efficiency, affecting the boost pressure required from the compressor <i>Internal volumetric efficiency assessed in Section 3.5.1</i>
Parameters related to engine modelling		
Heat transfer		
Heat transfer coefficient model	The Woschini 1978 heat transfer coefficient model has been selected as a well-established model	Different models can lead to different predictions and thus heat transfer values, affecting the engine energy distribution <i>Sensitivity analysis in Section 3.5.2</i>
Friction		
Friction model	A friction model is used rather than user-defined values due to the lack of validation data The Patton, Nitschke, Heywood (PNH) model was chosen as a well-established model	The friction model does not impact the indicated performance of the engine but can affect friction prediction, and thus the engine brake energy distribution <i>Sensitivity analysis in Section 3.5.2</i>
Knock onset		
Knock onset model	The parameters of the knock onset model have been left as default except the pressure exponent which has been lowered for reasons detailed in Section 5.2	Estimate the fuel octane number required to prevent knock to occur in the simulated conditions <i>This result is only informative and does not impact the performance prediction</i>

Parameter	Adopted value(s) and explanation	Potential uncertainty impact
Pollutants		
Pollutant model	The parameters of the pollutant model (NO _x , CO, HC) have been left as default	<i>The values predicted by this model will only be used for relative comparisons and will not be considered as representative</i>

Table 3-2 - List of parameters necessary for the 2014+ Digital Twin engine model

As indicated in Table 3-2, some parameters of the input dataset influence compound parameters of which a non-exhaustive list is given in Table 3-3. The parameters in italic are parameters which are compounded when considering a real engine, i.e. influenced by engine operating conditions, but that must be set to specific values for the 2014+ Digital Twin to run. In Section 3.5.1, compound parameters are compared against expected ranges, when possible, to assess if the decisions made for the input dataset yield acceptable performances.

Compound parameter	
Power / BMEP	Wall heat transfer
Effective efficiency / BSFC	FMEP
Intercooler cooling load	Residual gas concentration
Volumetric efficiency	Blow-by
In-cylinder pressures / temperatures	Emissions
In-cylinder turbulence	Exhaust back-pressure
<i>Rate-of-heat-release (ROHR - shape and duration)</i>	Turbine operating point
Octane number requirements	MGU-H power
<i>Wall temperatures</i>	

Table 3-3 - Non-exhaustive list of the 2014+ Digital Twin compound parameters

For parameter whose value contains a degree of uncertainty, a sensitivity analysis has been conducted to assess the impact on performance prediction and estimate the potential gap with a real 2014+ engine. These parameters are listed below, and the corresponding sensitivity analyses' results are presented in the Section 3.5.2:

- Weibe function shape,
- Combustion duration,
- Start of combustion,

- Air/fuel ratios,
- Stoichiometric air/fuel ratio,
- Fuel heat of evaporation,
- Fuel temperature,
- Fraction of heat coming from combustion chamber walls for fuel evaporation,
- Combustion chamber wall temperatures,
- Combustion chamber surface areas,
- Heat transfer coefficient model,
- Wall temperatures for port heat transfer,
- Surface areas for port heat transfer,
- Oil type,
- Oil temperature,
- Friction model,
- Number of crankshaft bearings.

3.5 Assessment of uncertainty significance on the 2014+ Digital Twin performance prediction

3.5.1 Analysis of the predictions made for compound parameters

3.5.1.1 Engine power

Figure 3-2 shows in grey line, the power prediction of the 2014+ Digital Twin in this most powerful mode (Open Waste Gate (OWG) mode – see Chapter 5) compared to curves created using Equation 3-6.

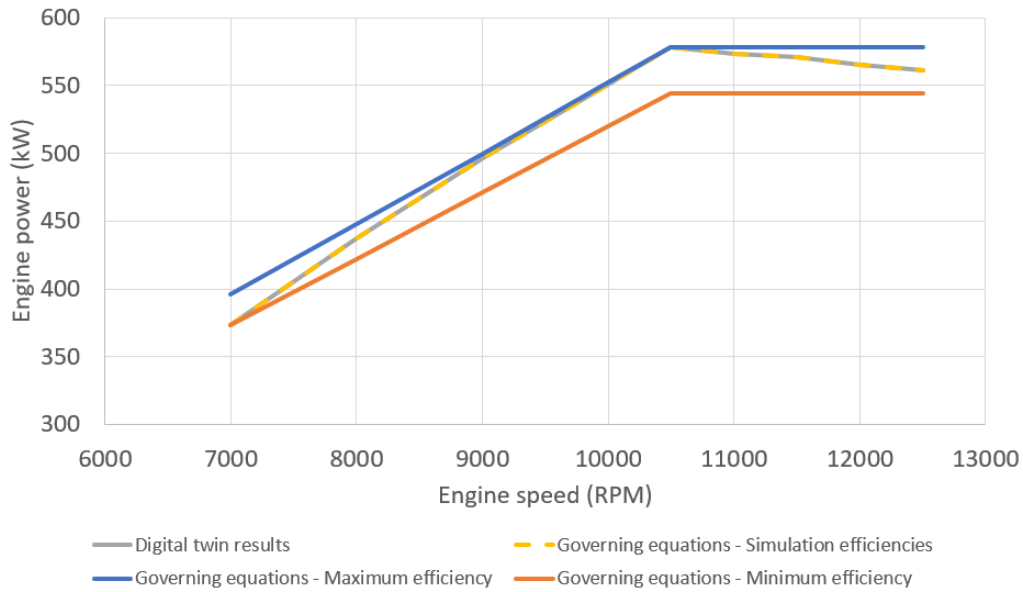


Figure 3-2 - Comparison between governing equations and 2014+ Digital Twin power predictions

The yellow dashed curve has been calculated using the effective efficiency (or brake efficiency) predicted by AVL Boost™ while the two others used the maximum and minimum predicted efficiency. It can be noted that the grey and yellow curves are perfectly superimposed showing that the Digital Twin predictions are in line with the simple calculations that can be done using the governing equations (Equations 3-1 and 3-2). This is a useful gross-error check.

There is confidence that the shape of the power curve predicted by the 2014+ Digital Twin is sensible considering that it follows relatively well the shape of the fuel flow limit which is the most constraining limit of this engine in terms of maximum power output.

3.5.1.2 Internal volumetric efficiency

Figure 3-3 shows the internal volumetric efficiency (referenced to the intake plenum conditions) of the 2014+ Digital Twin for its two different modes that will be further described in Chapter 5.

The volumetric efficiency is, for most points contained within the [0.95, 1.05] window which can be considered, by common practice, reasonably expected. For reasons detailed in Chapter 5, the volumetric efficiency difference increases between both modes as engine speed increases. For the Open Waste Gate mode, the volumetric efficiency keeps rising due to reduced back-pressure, which is an advantage, whereas for the Closed Waste Gate (CWG) mode, it drops down to 0.9. This will require more work from the compressor to reach the air/fuel target, but the reasons and the advantages of each mode will be discussed in Chapter 5.

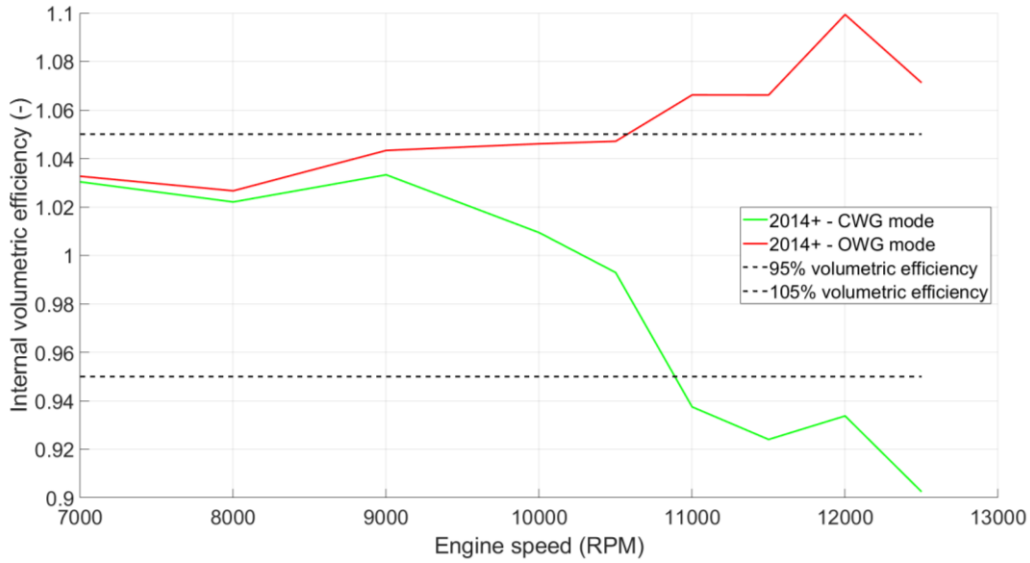


Figure 3-3 – 2014+ Digital Twin internal volumetric efficiency predictions

As the internal volumetric efficiency of the 2014+ Digital Twin is within an expected range, it can be considered that the decisions made on the input dataset, and which influence this compound parameter are adequate.

3.5.1.3 Peak in-cylinder pressure

As stated in Table 3-1, 300 bar is considered as the maximum reachable in-cylinder pressure [5]. As shown in Figure 3-4, for both engine modes of the 2014+ Digital Twin, the selected input dataset leads the engine model to respect this criterion, except for one engine speed in the closed waste gate (CWG) mode.

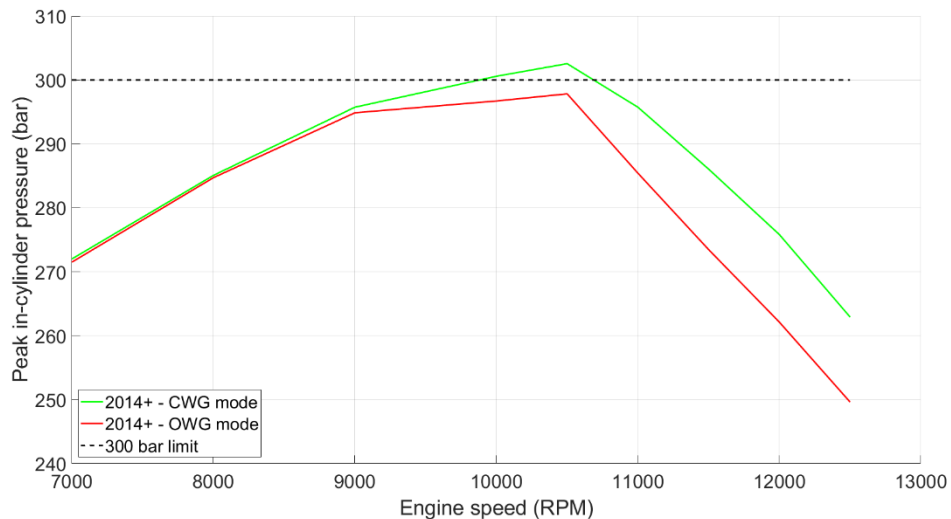


Figure 3-4 – 2014+ Digital Twin peak in-cylinder pressure predictions

At 10,500 RPM for the CWG mode, the pressure reaches 302.5 bar which exceeds the limit by less than 1%. Thus, peak in-cylinder pressure being

influenced by a large number of parameters, Figure 3-4 confirms that the values chosen for the 2014+ Digital Twin's input dataset are appropriate.

3.5.1.4 Heat transfer coefficient

From private communication [5], at 10,500 RPM, the following cooling loads were expected for a specific engine of the beginning of the 2014+ hybrid era:

System	Cooling load (kW)
Water cooling system	85
Oil cooling system	35
Intercooler	60

Table 3-4 - Beginning of hybrid era engine cooling loads [5]

It can be assumed that the water-cooling system dominates cooling of the cylinder head and the liner whereas the oil system is used to cool the piston and cam-box. Figure 3-5 shows the predictions made by the 2014+ Digital Twin in CWG mode for the different cooling loads.

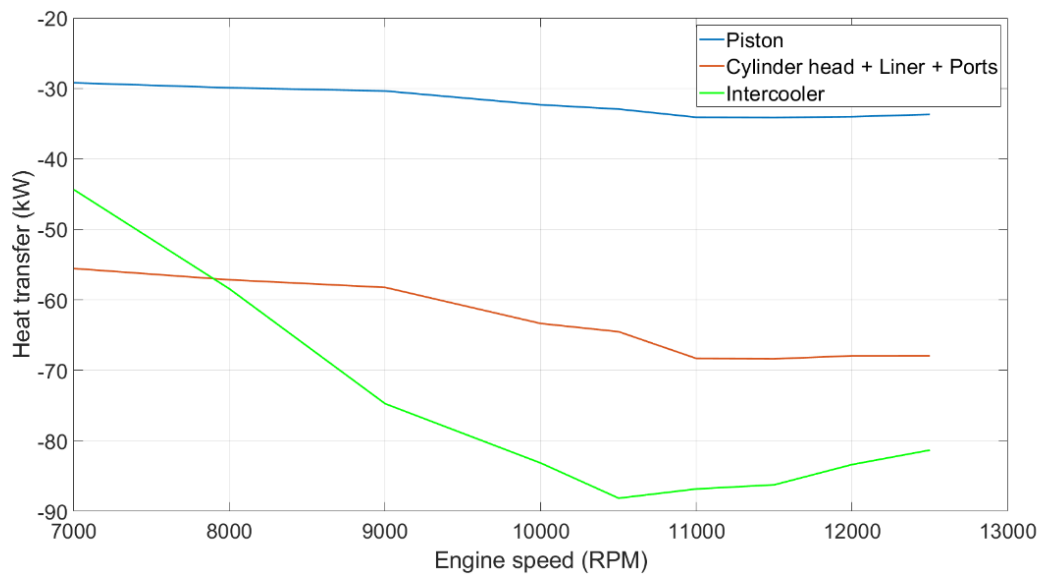


Figure 3-5 – 2014+ Digital Twin heat transfer predictions in CWG mode

The intercooler load is much higher than the value shown in Table 3-4. Nevertheless, it is believed that the engine from which the data comes was not running at the high air/fuel ratios seen in later engines of this type and used in the 2014+ Digital Twin. With a fixed fuel flow and lower air/fuel ratios, the required air mass flow is lower leading to less load on the intercooler by considering equivalent compressor and intercooler outlet temperatures.

The 2014+ Digital Twin predicts lower water-cooling system loads (Cylinder head + Liner + Ports) than the value reported for the real engine. However, as

the engine relating to the test data in Table 3-4 is believed to have run lower level of dilution than the 2014+ Digital Twin, in-cylinder temperatures are expected to be higher and thus to results in higher cooling loads which would explain the 20 kW difference between both values at 10,500 RPM.

Regarding the piston oil cooling system, the values shown in Figure 3-5 are similar to the value given in Table 3-4.

Also from private communication with a Formula One™ Power Unit manufacturer [5], it is known from detailed CFD that heat transfer coefficients on the real engine are between 6,000 and 13,000 W/(m².K) depending on the location in the combustion chamber. The highest values are restricted to the small diameter cooling channels that orbit around each valve seat. The lowest values are found in the flow recirculation zones between individual cylinder liners. As an aside, the CFD work used to uncover these unmeasurable quantities is a good example of using a Digital Twin.

Figure 3-6 shows the average over all simulated engine speeds of the global heat transfer coefficient in the combustion chamber during a cycle in closed waste gate (CWG) mode. For most of the combustion phase, the heat transfer coefficients are within the expected range mentioned in the previous paragraph.

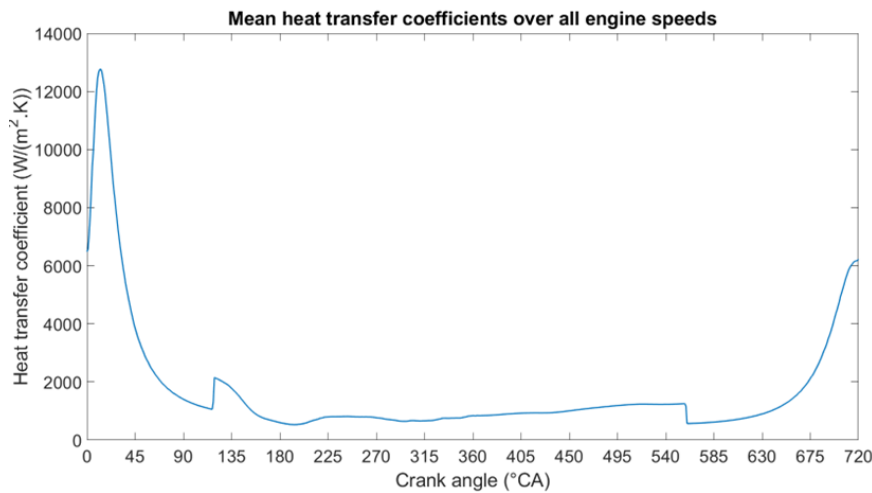


Figure 3-6 - Average over all engine speeds of heat transfer coefficients during a cycle

Figure 3-7 shows the evolution with engine speed of cycle mean heat transfer coefficient values. Heat transfer coefficients are highly influenced by the in-cylinder temperature and pressure as shown by Equation 3-7 which is Woschini 1978 model [34]:

$$h_c = 3.2 \times B^{-0.2} \times p^{0.8} \times T^{-0.55} \times w^{0.8} \quad (3-7)$$

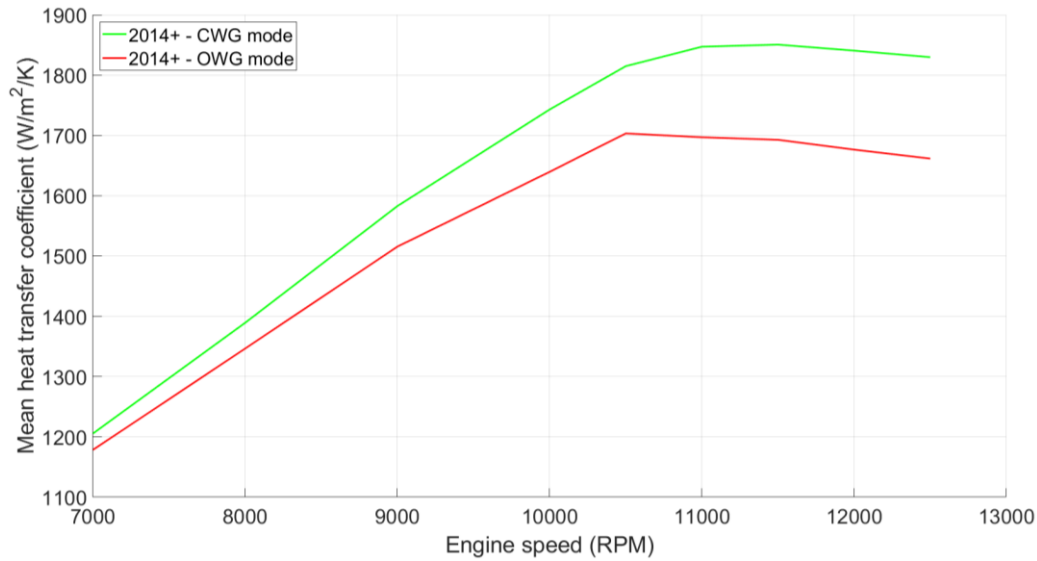


Figure 3-7 – 2014+ Digital Twin mean heat transfer coefficient predictions

During most of the cycle, in-cylinder pressures are low leading to low heat transfer coefficient values as shown in Figure 3-7, hence the lower mean heat transfer coefficient values over the whole cycle than the values previously mentioned which are instantaneous coefficients.

3.5.1.5 MGU-H power

From the feedback of the Cranfield F1™ Design Sprint, the MGU-H power is known to be around 65 kW. The 2014+ Digital Twin can be operated in two modes, either with the waste gates closed for the whole engine speed range (CWG mode) or with the waste gates fully open (OWG mode), which requires respectively the maximum from the MGU-H in terms of energy recovery (negative values) or deployment (positive values).

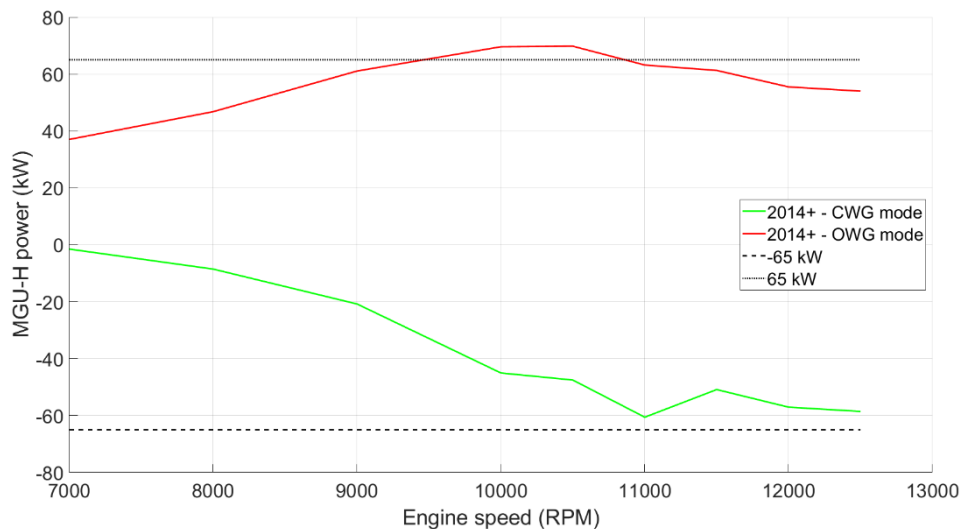


Figure 3-8 – 2014+ Digital Twin MGU-H power predictions

Figure 3-8 shows the MGU-H power predicted by the 2014+ Digital Twin for both modes. Except for two engine speeds in the open waste gate (OWG) mode where 70 kW is reached, all other values are contained in the [-65, 65 kW] window.

This confirms that all the assumptions made previously which impact the power required by the compressor and/or the power recovered by the turbine are congruent with the feedback from the industry when it comes to the MGU-H power.

3.5.2 Sensitivity analysis on the 2014+ Digital Twin parameters

The 2014+ Digital Twin in CWG mode, which is presented in detail in Chapters 4 and 5 and whose parameters are listed in Table 3-2, is the reference against which the performance of the models with modified parameters for the sensitivity analyses are assessed.

To be able to fully evaluate the impact of each parameter, only one parameter is changed at a time compared to the 2014+ Digital Twin and it is ensured that the engine is running under the same conditions. For instance, in the stoichiometric air/fuel ratio sensitivity analysis, the value of this parameter is modified but the operating points of the compressor too so that the engine is operating at the same relative air/fuel ratios in all cases.

The difference between the modified model and the reference is assessed using percentage **relative difference** which is calculated using the following equation:

$$\% \Delta x = 100 \times \left(\frac{x_{modified} - x_{reference}}{x_{reference}} \right) \quad (3-8)$$

Where $x_{reference}$ is the result obtained for the reference model (2014+ Digital Twin in CWG mode) and $x_{modified}$ the modified model result.

Results are then presented under spider diagrams [149] like the one shown in Figure 3-9. For each axis, if the point is in the red zone, it represents a performance reduction compared to the reference and inversely if in the green zone. Please note that the axis extremums are not necessarily all the same as demonstrated in the example, for clarity of presentation. In addition, some axes are reversed, with positive values in the red zone. An increase in fuel octane number requirement is not an improvement for instance, this is why this specific axis is reversed.

If the values obtained for one of the modified cases were to be the exact same ones as that of the model taken as reference, its curve on the spider diagram would lie on the frontier between the red and green curve, at the exact 0 of each axis, as there would be no relative difference between both models.

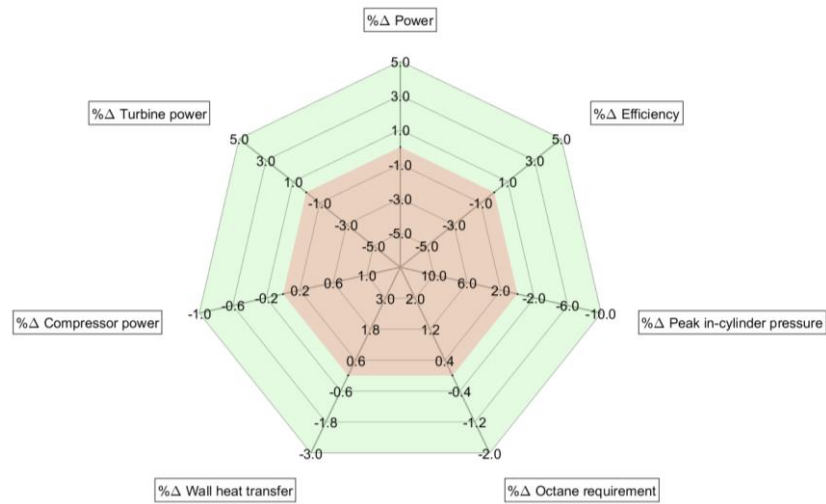
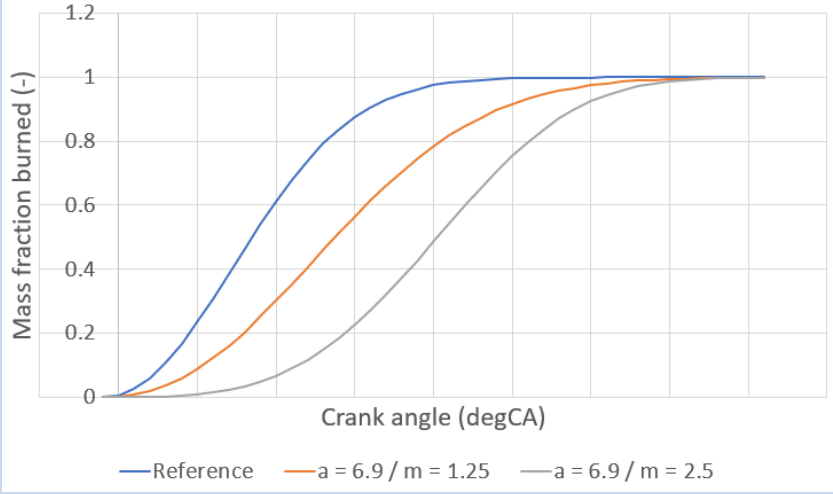


Figure 3-9 - Spider diagram example

For clarity reasons, the spider diagrams are not presented in this section but are available in Appendix B. Here will only be discussed the conclusions brought by the sensitivity analyses.

The results of the sensitivity analyses are presented in Table 3-5.

Parameter	Variation investigated during the sensitivity analysis	Conclusions
Weibe function shape	<p>The reference curve is compared against Weibe functions of same combustion duration but of different evolutions, slower at the beginning of the combustion event (see on the right).</p>	<p>By modifying the Wiebe function, all the parameters assessed are significantly impacted. The biggest change concerns the peak in-cylinder pressure which is reduced by over 20% in both cases.</p> <p>Nevertheless, as the 2014+ engine is known to be pressure-limited, it is important to note that this would no longer be the case if one of these two other Weibe functions was adopted. In addition, with a slower rise at the beginning of the combustion, the efficiency is also significantly reduced, showing that engine designers would have pushed for rapid combustion start to drive the efficiency of this fuel-limited engine up.</p> <p>If the shape of the ROHR is modified compared to the one used in the 2014+ Digital Twin, it is likely to still feature a rapid rise and to lead to much lower performance difference than what is suggested with this sensitivity analysis.</p> 
Combustion duration	<p>The reference combustion duration (41 °CA) is compared to three other values, 45, 50 and 55 °CA and the average impact of a 1 °CA increase is calculated</p>	<p>Combustion duration must be increased by 9 °CA to observe a significant change in efficiency (1%) which is accompanied by a larger reduction in in-cylinder pressure. As the engine is pressure-limited, especially in the high-power zone, it is unlikely that combustion duration will change significantly. As it also drives the efficiency up (Section 3.2), reducing combustion duration can be considered as a development target for engine designers as demonstrated by the use of a pre-chamber.</p>

Parameter	Variation investigated during the sensitivity analysis	Conclusions
Combustion duration (Follow-up)	The reference combustion duration (41 °CA) is compared to three other values, 45, 50 and 55 °CA and the average impact of a 1 °CA increase is calculated	The combustion will slightly vary with engine speed, but the impact is limited with an average relative efficiency increase lower than 0.2% for each 1°CA reduction.
Start of combustion	The reference combustion timing is compared to four other values, namely -3, -5, -7.5 and -10 °CA ATDC and the average impact of a 1 °CA increase in combustion advance is calculated	<p>Like for combustion duration, combustion advance is unlikely to vary significantly in the high-power range of the engine because of the pressure limit. All tested combustion advances lead to an increased peak in-cylinder pressure. With -10 °CA ATDC, the efficiency improvement is reduced compared to -5 and -7.5 °CA ATDC, suggesting that MBT timing has been exceeded.</p> <p>Because the engine is not pressure-limited at all engine speeds, it is possible to increase the combustion advance, however, the maximum relative improvement in efficiency is estimated below 2% which corresponds to around 1 percentage point in overall performance. In average, increasing combustion advance by 1 °CA yields about 0.25% relative increase in effective efficiency.</p>
Air/fuel ratios	A 0.05 increase and a 0.15 decrease in lambda compared to the evolution used in the 2014+ Digital Twin are investigated	<p>Especially in the non-pressure-limited zone, it is possible that lambda is higher than what has been chosen for the 2014+ Digital Twin.</p> <p>By increasing lambda by 0.05 point, a relative efficiency increase of 1% can be achieved at the expense of 6% more compressor power requirement and intercooler load in addition to the pressure increase.</p> <p>It can be highlighted that the amount of power recovered by the turbine is also increased but not as much as the compressor power resulting in less energy recoverable by the MGU-H or more energy to deploy.</p>
Stoichiometric air/fuel ratio	<p>A +/- 10% variation in stoichiometric air/fuel ratio is assessed</p> <p>Reference value: 14.7</p>	<p>The fuel used in Formula One™ being specifically developed for these engines, it is possible that the value differs from that of gasoline (used in the 2014+ Digital Twin) despite still being mainly composed of fossil components.</p> <p>With a 10% variation in AFR_{st}, engine efficiency varies by approximately 1% because of a 5% variation in peak in-cylinder pressure caused with the difference in air quantity that must be introduced in the cylinder to run at equivalent air/fuel ratios.</p> <p>However, the power required by the compressor and the intercooler load change by nearly 20% which is significant. On the other hand, because of the engine mass flow engine change, the turbine power varies by around 8%, It must be noted that the turbine swallowing curve is the same for all cases, so not necessarily optimal for all.</p>

Parameter	Variation investigated during the sensitivity analysis	Conclusions
Stoichiometric air/fuel ratio (Follow-up)	A +/- 10% variation in stoichiometric air/fuel ratio is assessed Reference value: 14.7	With the current AFR_{st} , the 2014+ Digital Twin peak in-cylinder pressures and boost pressures are of the correct magnitude according to the feedback from the Cranfield F1™ Design Sprint which leads to believe that the 14.7 value is satisfactory (see Section 3.5.1).
Fuel heat of evaporation	A +/- 20% variation in fuel heat of evaporation is assessed Reference value: 419 kJ/kg	The impact of this parameter is very limited as the maximum variation observed is below 1% and the efficiency barely even changes. Like for AFR_{st} , despite the F1™ fuel not being gasoline, it can be assumed that its properties are similar. The fuel heat of evaporation is thus not expected to vary by as much as 20% and its impact on performance can be considered negligible.
Fuel temperature	The reference value (25 °C) is compared to 15 and 50°C and the average impact of a 5°C increase is calculated	As for fuel heat of evaporation, the impact of fuel temperature is limited. The average impact of a 5°C increase in fuel temperature and most variations are reported below 0.05%. The highest variation is observed for turbine power, but this difference can be attributed to convergence difference as the different case curves are only different for a small number of engine speeds as shown in Appendix B. Otobe et al. [6] reported that for Honda's RA168E engine, fuel pre-heating, up to a point, helped with efficiency because of the distillation characteristics of their fuel (composed at 84% in volume of toluene). It might also be the case for the 2014+ engine and fuel temperature is believed not to be lower than 25°C but its impact on performance prediction is considered negligible.
Heat from wall	The reference value (0.3) is compared to two extreme values, 0 and 0.7	As the walls of the combustion chamber are hot, it is impossible that 0% of the heat necessary for fuel evaporation comes from the wall and inversely as a direct injection engine, the fuel is injected during the compression stroke at which point the temperature of the inlet air has already increased, it is impossible that 100% of the heat comes from the walls. 0.3 is thus considered as a sensible estimation and a modification of 0.1 lead to very small changes in performance prediction. The higher the amount of heat coming from the wall, the less the mixture will be cooled leading to slightly higher in-cylinder pressure, octane requirement and heat transfer. Nevertheless, the changes are predicted below 1%.

Parameter	Variation investigated during the sensitivity analysis	Conclusions
Combustion chamber wall temperatures	A +/- 10% variation in wall temperatures is assessed	<p>Wall temperatures have very limited impact on performance prediction. The turbine power variation presents the larger impact which can also be attributed to model convergence, like for fuel temperature.</p> <p>Obviously, the biggest impact concerns wall heat transfer which is modified by around 6%. This would affect the design of the engine cooling system however, the amount of power recovered by the turbine does not change significantly.</p> <p>This demonstrates that despite the higher cooling losses, the exhaust enthalpy does not vary significantly, so does the engine power, meaning that the engine energy distribution and behaviour are not considerably modified. As it will be seen in Chapter 5, Section 5.3, exhaust enthalpy is much higher than cooling losses for the 2014+ engine so the impact of x kW on exhaust enthalpy will represent a lower percentage than for cooling losses.</p>
Combustion chamber surface areas	A +/- 10% variation in surface areas is assessed	<p>The conclusions are very similar to that of the combustion chamber wall temperatures. Most performance indicators are barely changed (below 1%) by the modifications in surface areas, except wall heat transfer which varies between 6 and 10%. However, this change does not significantly impact the engine energy balance.</p>
Heat transfer coefficient model	The reference model (Woschini 1978) is compared to the other models available in AVL Boost™	<p>Except the AVL 2000 and Bargende models, the predictions are very similar with impacts lower than 1% everywhere except for heat transfer.</p> <p>AVL 2000 model, which is a modified version of Woschini to better model heat transfer during the gas exchange part of the cycle [150], predicts lower heat transfer by 4%, 3% higher turbine power and 0.45% higher peak in-cylinder pressure, but its impact on effective efficiency is negligible.</p> <p>As described in AVL Boost™ theory manual [150], the Bargende model considers additional factors compared to the other correlations including a term considering flame propagation and another considering in-cylinder turbulence. If the turbulence term chosen in AVL Boost™ is not adapted to the engine operating conditions, the heat transfer prediction can be mis-estimated.</p> <p>Depending on the model chosen, different heat transfer predictions are made, the model used for the 2014+ Digital Twin, Woschini 1978, predicts similar heat transfer as most models except for the Bargende model. A maximum of 1.6% reduction in efficiency can be observed by changing the heat transfer coefficient model.</p>

Parameter	Variation investigated during the sensitivity analysis	Conclusions
Port heat transfer – Wall temperatures	A +/- 10% variation in wall temperatures is assessed	<p>As expected, and like for the combustion chamber wall temperatures sensitivity analysis, the port heat transfer is affected by 6% by this modification.</p> <p>The efficiency is modified by less than 0.1%, so it can be extrapolated that even if the temperatures are modified by +/-100%, the efficiency would be modified by less than 1%.</p> <p>The turbine power variation can again be attributed to simulation convergence and the octane requirement is the most impacted parameter by the modification of the port wall temperature with a 0.2% change. Where again, the impact can be judged negligible as a 100% change in temperature would only lead to a 2% difference.</p>
Port heat transfer – Surface areas	A +/- 10% variation in surface areas is assessed	The conclusions for this sensitivity analysis are very similar to that of the port wall temperatures. Except for port wall heat transfer, the impact on performance is very limited as performance predictions are affected by less than 0.1%.
Friction – Oil type	The reference oil type (SAE 10/W60) is compared to SAE 40	<p>The type of oil plays a more significant role in the friction prediction. As for the fuel, the oil used in Formula One™ engines are specifically developed between the engine manufacturer and its oil supplier.</p> <p>Because of its high viscosity, SAE 10/W60 leads to higher FMEP prediction compared to SAE 40, for which efficiency is increased by 1%.</p>
Friction – Oil temperature	<p>A +/-20% variation in oil temperature is assessed</p> <p>Reference: 100°C</p>	At 80 °C oil temperature, FMEP significantly increases, and the efficiency is reduced by more than 1%, however this temperature is considered by the author too low for a high-performance engine. Inversely, if the oil temperature is set at 120 °C, efficiency can be improved by 0.89%. However, the higher the oil temperature and the lower its viscosity and the lower the oil film thickness. 100 °C is believed to be a good compromise.
Friction – Friction model	The reference friction model (PNH) is compared to the other model available in AVL Boost™ (SLM)	<p>Two friction models are available in AVL Boost™, the Patton, Nitschke, Heywood (PNH) model and the Shayler, Leong, Murphy model (SLM) model. The SLM model predicts nearly 2% higher efficiency than that of PNH but considering the lack of validation data, it is not possible to know which friction mode predicts the closest FMEP values to that of the real engine.</p> <p>However, as indicative performances are not impacted by friction calculations, it will be possible to compare them to the claims of the engine manufactures which are stated in Section 3.2.</p>

Parameter	Variation investigated during the sensitivity analysis	Conclusions
Friction – Number of camshaft bearings	The reference value (10) is compared to 8 and 12	Modifying this number does not significantly affect the performance predictions, with only a 0.07% change in effective efficiency for +/-2 modifications.

Table 3-5 - Summary of the sensitivity analyses conducted to assess input dataset uncertainty significances on performance prediction for the 2014+ Digital Twin

Figure 3-10 shows a summary of the impact of each parameter on effective efficiency (brake fuel conversion efficiency). This does not represent the total impact of a given parameter (the full analysis is given in the spider diagrams in Appendix B), but as effective efficiency is the key feature of the 2014+ engine, it is interesting to understand how much the parameters investigated in the sensitivity analyses influence it.

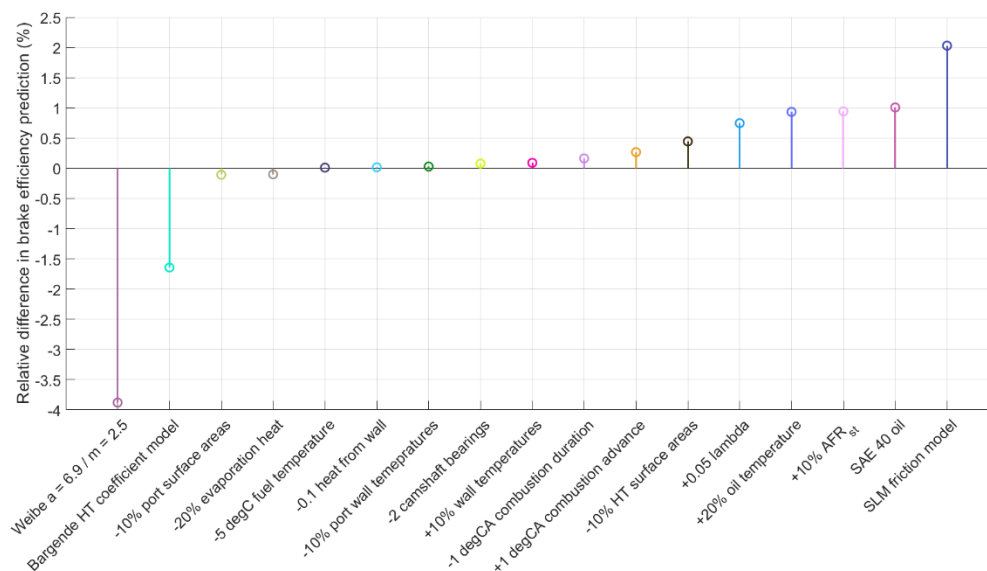


Figure 3-10 - Sensitivity results regarding effective efficiency predictions

As explained in Table 3-5, modifying the Weibull curve shape has a great impact on effective efficiency, the greatest, which confirms that a rapid rise in MFB at the beginning of the combustion is key to achieve high efficiency. Changing the friction model from PNH to SLM yields the best improvement on efficiency with nearly 2% in relative difference (Equation 3-8), which in this case is nearly 1 percentage point in actual performance but due to the lack of validation data, it is not possible to decide which model is the most suitable for the 2014+ engine. The same conclusion can be drawn for the impact of oil type and temperature. Nevertheless, the friction model only affects the effective performance prediction, indicative performances can still be assessed without considering this parameter.

Bargende heat transfer coefficient model predicts heat transfer values significantly different to all the other models (+12%) resulting in an average 1.64% difference in brake efficiency prediction. This model considers additional factors which might cause the difference with the other models.

The two other parameters which have a relative impact close to 1% are stoichiometric air/fuel ratio which is believed to be close to that of gasoline and to vary by less than

10% of the chosen value (14.7) and relative air/fuel ratio which is an operational parameter chosen by the engine designers. However, increased air/fuel ratios lead to higher in-cylinder pressure for an already pressure-limited engine so, higher values than what used for the 2014+ Digital Twin are only possible in the zones the engine is not operating close to 300 bar. However, it is interesting to note that air/fuel ratio is the operational parameter with the most impact on engine efficiency according to Figure 3-10.

The impact of the other parameters is below 1% in relative difference (average from Figure 3-10 is 0.74%), i.e. below 0.44 percentage point in actual effective efficiency points and around 0.36 bar in terms of BMEP. The conclusion is the same regarding IMEP for which a 1% relative difference results in 0.48% percentage points and 0.4 bar IMEP (average relative difference using the cases presented in Figure 3-10 is 0.61%).

3.6 Conclusions regarding the uncertainty significance and management

In the previous sections, the values used to set up the 2014+ Digital Twin have been justified by reviewing the prediction of compounded parameters and sensitivity analyses have been conducted for the parameters containing a degree of uncertainty.

For most parameters, even though differences in performances were observed when values were modified, their significance, especially for effective efficiency, proved to be under control (under 1% compared to 2014+ Digital Twin) and not to significantly impact the behaviour and energy distribution of the engine.

As previously stated, the objective of the 2014+ Digital Twin is not to claim having reproduced the exact behaviour of the 2014+ Formula One™ engine but to have a model experiencing the same trends.

Based on the data provided by the sponsor of this Doctoral Thesis and the feedback received from the industry for the Cranfield F1™ Design Sprint, this chapter showed that despite the uncertainties associated to the values used in the 2014+ Digital Twin, the behaviour of the engine is not significantly affected by these uncertainties, with most parameters affecting the performance prediction by less than 1% in relative difference.

As discussed in the previous section, 1% in relative difference represent approximately 0.4% in effective efficiency for the 2014+ Digital Twin. In Formula One™, with input energy being limited by regulation now, every difference in efficiency provides the equivalent difference in performance, and any positive change is significant, and worth investigating to make the car faster.

Figure 3-11 which comes from the AVL Boost™ validation manual [151], compares the software IMEP predictions against measurements for different operation conditions (engine load, boost pressure, air/fuel ratio, spark timing and compression ratio) for an engine not specified.

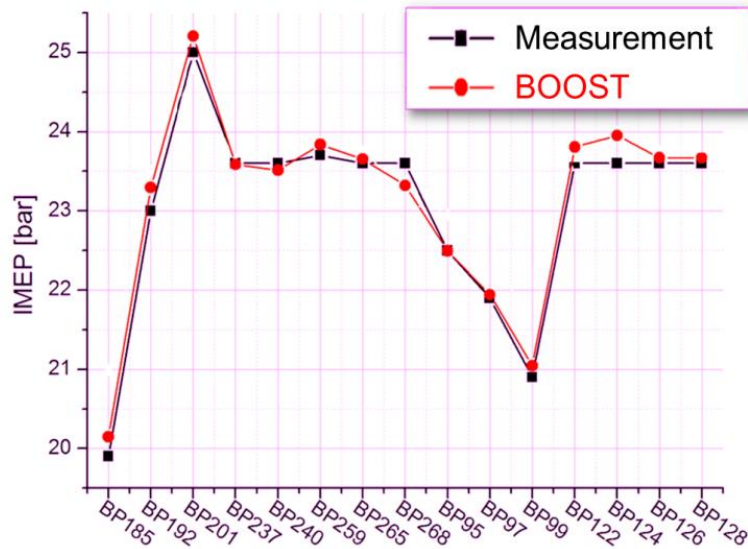


Figure 3-11 - AVL Boost™ IMEP prediction vs measurement [151]

Despite the predictions being close to the measurement values, there is a difference.

The average relative error in AVL Boost™ prediction is of 0.43% with peak values approaching 1.5% as shown in Figure 3-12 (cases are presented in the same order as in Figure 3-11):

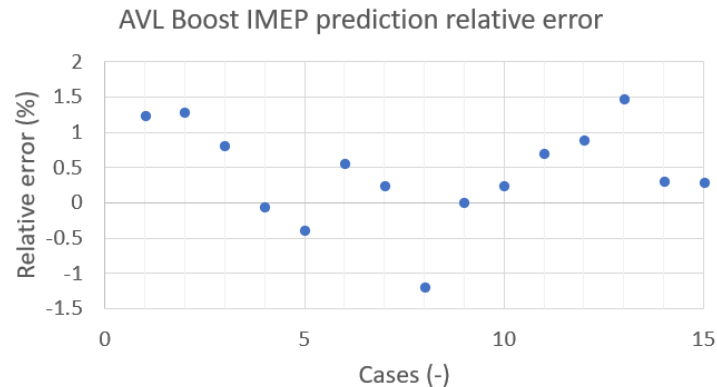


Figure 3-12 - Relative error between AVL Boost™ prediction and measurement

The average absolute difference in IMEP between AVL Boost™ and the 15 measurement cases presented in Figure 3-11 is of 0.15 bar.

The engine under study produces an IMEP of around 23 bar which is approximately half what the 2014+ engine is capable of (see Chapter 5). If the relative error between the simulation and measurement is considered independent of the engine model, this would result in 0.3 bar inherent modelling error in IMEP predictions between the 2014+ Digital Twin and the 2014+ engine if a full input data set was available for the Formula One™ engines. This value is of the same order of magnitude as the 0.4 bar value reported in Section 3.5.2 for the mean impact of input dataset uncertainties on the 2014+ Digital Twin IMEP predictions.

The impact of input dataset uncertainty is thus similar in scale as the margin of uncertainty of the inherent modelling uncertainty of AVL Boost™. Whilst working on

Formula One™ Digital Twins seems to double the margin for modelling error when compared with commonplace engine for which complete input datasets are available, the magnitude of the uncertainty is not too great to raise concerns over the validity of the exercise. The 2014+ Digital Twin can be confidently used to investigate the behaviour of the 2014+ engine and get insights on its most uncommon characteristics. However, fine changes in absolute predicted performance levels need to be assessed in light of the circa **0.7 bar IMEP total uncertainty** which represents less than 2% of the average 2014+ Digital Twin IMEP prediction despite the sparse input data available from the industry. Relative changes between one configuration and the next will be less prone to uncertainty.

In addition to IMEP, the AVL Boost™ validation manual [151] proposes a comparison between the software NO_x production predictions and experimental data as shown in Figure 3-13.

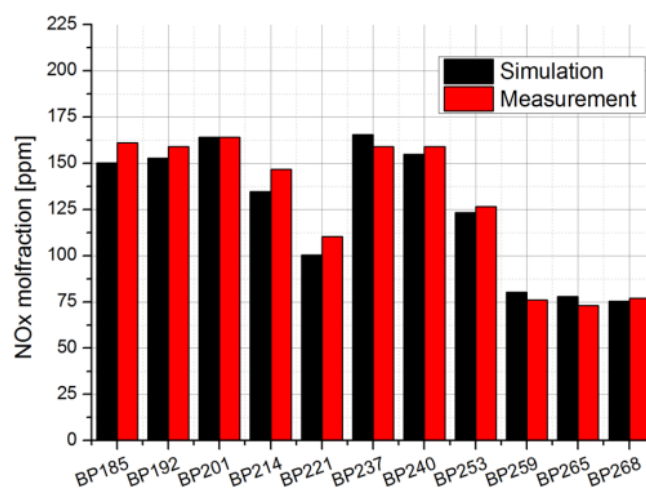


Figure 3-13 - AVL Boost™ NO_x production prediction vs measurement [151]

4 Tools developed for this research project

4.1 Introduction

As set out in Chapter 2 (Section 2.5.1), and summarised in Figure 4-1, a range of simulation techniques are used to model engine behaviour and/or predict engine performance, with different levels of complexity and accuracy.

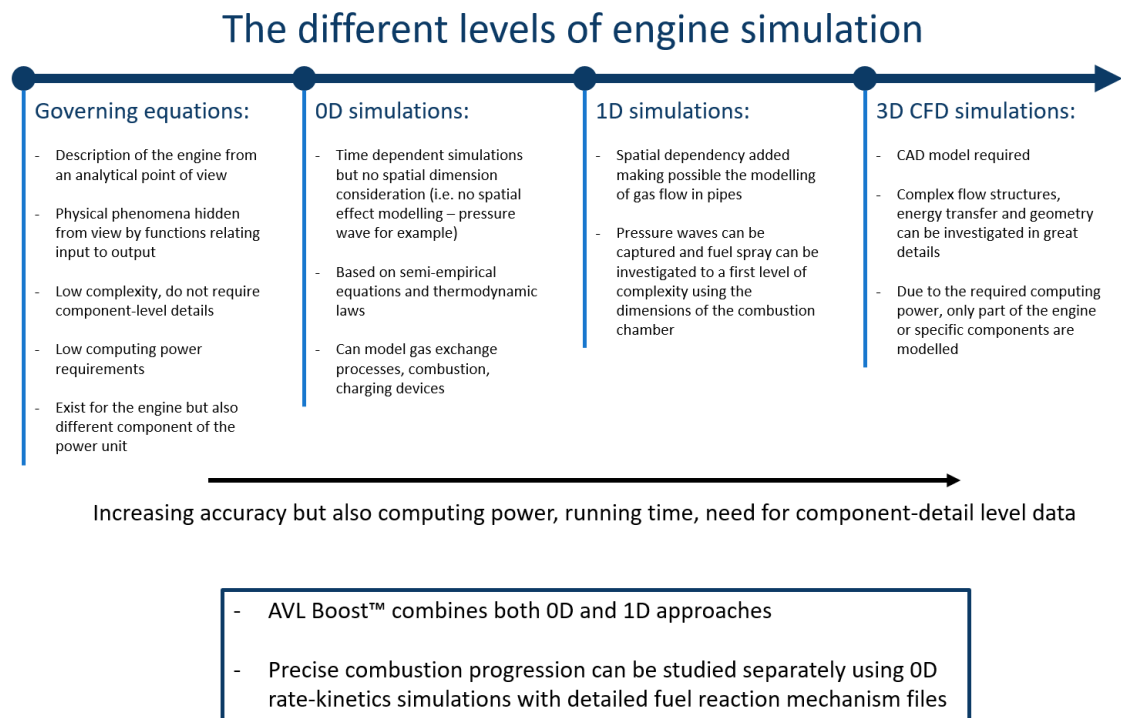


Figure 4-1 - Summary of the different levels of engine simulations

With the introduction of fully advanced sustainable fuels in the 2026 Formula One™ regulations, fuel development will be of primary importance for this next engine generation. The 2026 Formula One™ engineered fuels will be blends of individual species whose compositions will be exactly known as opposed to conventional gasoline or Diesel which are variable blends of different fuel components. Having the possibility to quickly assess the potential of different fuel compositions will be important to find the optimal blend.

Because of the time-consuming and costly nature of engine testing, even with single cylinder engines, rate-kinetics simulation has the potential to improve the fuel development process by allowing fast assessment of fuel combustion properties and selection of the most promising blends (here high flame speed at high lambda and high resistance to auto-ignition for example).

The usefulness of industry-standard rate-kinetics simulation tool Ansys Chemkin to assess combustion properties of advanced sustainable fuels for the 2026 Formula One™ Power Unit regulations has been assessed during this thesis, but several elements made it impractical. This was a very disappointing result and after several months of trials that the decision to discontinue the further use of

Anslys Chemkin for 2026 Formula One™ fuels was made. This is not intended to detract from the usefulness of the software for other purposes, but it has proved unreliable for the conditions reported here.

As stated previously, fuel combustion mechanisms are necessary to describe the combustion event in rate-kinetics simulations and their results are highly dependent on this reaction file.

In first approximation, because the 2026 Formula One™ fuel will be engineered and its composition can be exactly determined and thanks to the literature providing with combustion mechanisms for a significant number of single species fuel which model fuel behaviour accurately, the mechanisms of each component of the fuel blend could be merged to form that of the blend.

Then using a model of the 2014+ fuel as a reference point, the simulation results could be used to modify the rate-of-heat-release curve used for the 2014+ Digital Twin for it to be representative of the 2026 engine operating conditions. Furthermore, NO_x emissions between fuel blends and between combustion rates could be compared if their formation is included in the reaction mechanism.

However, obtaining blend combustion mechanisms is not as simple as linearly merging the mechanism of each component contained in the engineered fuel. The intermediate species of each fuel could interact with that of the others and modified the reaction path in a manner that may not be appreciated by a simple merged mechanism. If so, the rate-kinetics simulation predictions could be significantly different to reality and lead to mistakes. Plus, in consequence of this uncertainty regarding the validity of the merged mechanism, physical testing would still be necessary to draw the ultimate conclusions.

In addition, as the 2026 fuels are currently under development, no information is available regarding their possible composition, so the composition that could be determined using rate-kinetics simulation could have very different properties compared to the fuels that will be used post 2026, making the engine simulation results irrelevant. In addition, as this Doctoral Thesis does not aim at conducting experimental work on fuel reaction mechanism, which could be a research project on its own, the uncertainty associated with the use of a reaction mechanism simply obtained by combining individual fuel mechanism would be too significant.

Combined with important software reliability issues where the models for certain fuel combinations at certain starting conditions would run on the computers at a collaborating laboratory but not at Cranfield University, with no explanation found for this lack of reproducibility, these limitations made rate-kinetics simulations conducted with the leading commercial software too cumbersome, too unreliable and too reliant on sophisticated input data (they rely on a lot of prior-knowledge of intermediate reactions) for use in this Doctoral Thesis.

For the development of fuels fulfilling the requirements of the 2026 Formula One™ Power Unit regulations, the conclusions reached at this point is that fuel suppliers will have to rely on their engineering capacity and physical engine testing to compose a fuel blend with particular performance characteristics by choosing within the range of available “Advanced Sustainable components”

which are defined by the 2026 Formula One™ Power Unit regulations as “derived from a carbon capture scheme, a renewable fuel of non-biological origin (RFNBO), municipal waste, or non-food biomass” [15]. The composition of the fuel will be extremely important as it will define the combustion properties and possible operating conditions for the engine.

Nevertheless, if a fuel supplier were to develop a rate-kinetics simulation tool based on machine learning, having all the necessary mechanisms for single species fuels and for blends, and empirical “rules” for fuel blends, it could be possible to identify patterns regarding how combustion mechanisms of individual species are impacted when combined with that of other to form a blend to obtain more accurate merged mechanisms.

By having such a tool and considering that the composition of the future Formula One™ fuels will be engineered and thus perfectly known, there is the possibility to significantly impact the development of these fuels.

Based on the machine learning tool, fuel suppliers could investigate a large number of combinations fulfilling the regulation requirements, create the associated reaction mechanisms to run appropriate rate-kinetics simulations to estimate key combustion parameters such as laminar flame speed. Based on these observations, the most promising blends could be physically created and tested to confirm or not the expectations. The feedback obtained from the creation of one blend would be used to refine the model of the machine learning tool to improve its prediction accuracy.

As explained previously, creating a blend combustion mechanism is not straight forward but considering the low number of species the 2026 Formula One™ will be composed of, this technique could prove valuable. Obviously, there would be degree of uncertainty linked to the results provided by the rate-kinetics simulation as the mechanism would not be extensively validated and physically testing will always draw the ultimate conclusions.

But combining this technique with high computation power and the fuel supplier practical knowledge could give them the edge on its competitors by exploring unusual blend composition that a fuel expert would not necessarily have thought about as it was demonstrated with optimisation algorithms and engine design by Chapman [16].

Despite not being applicable for fuel development, as accurate blend reaction mechanisms are required, rate-kinetics simulations are very useful for the analysis of a reaction whose mechanism is accurately modelled (single species combustion, large scale chemical reactors).

Considering the issues faced with rate-kinetics simulations, the uncertainty regarding the composition of the 2026 fuels, it was decided to model the 2026 engine using ROHR based simulation and to use simpler tools that allow engineers to blend combinations of fuels to obtain basic thermodynamic quantities like heating value. One example tool is the AVL Boost™ Gas Properties Tool (see Section 4.3).

With knowledge of fuel composition, estimates of laminar flame speed for a blended fuel can be made from published data for the main components of the blend [152] and this result can be used to re-scale ROHR curves. Then engine testing can be used to gain more complete knowledge of rate-of-heat-release relationships with changing engine speed, load, lambda and both GDI and ignition timing. There is likely to be a role for Artificial Intelligence in guiding the blending of future fuels once the database for fuel components and blending properties is more widely available. Machine Learning wrapped around ROHR-based simulation could be used to provide the optimum ROHR performance specification of the fuel for a given engine and its speed and load combinations.

4.2 Engine governing equations

Section 4.1 explains why rate-kinetics simulation are not being taken as the starting point of a scientific study of the operations of current and future generation Formula One™ engines. To provide an alternative, the equations that govern the thermodynamic behaviour of engines of this type are inspected as the first step.

Engine governing equations are set out in most internal combustion engine textbooks and those described by Heywood [34] are shown below for 4-stroke engines:

$$P_I = \frac{\eta_f m_a N Q_{lvh} (F/A)}{2} \quad (4-1)$$

where P_I is the engine indicated power (before accounting for mechanical losses), η_f the fuel conversion efficiency, i.e. the efficiency to convert fuel energy into useful work, m_a the mass of air trapped per cycle, N the engine speed, Q_{lvh} the fuel lower heating value and (F/A) the fuel/air ratio.

It can be observed that the power equation is a function of engine speed. This demonstrates from a mathematical point of view why the V10 and V8 Formula One™ engines generations relied on very high engine speeds to produce power.

Based on the relations between power and torque, and between power and IMEP, these quantities can be expressed as:

$$T_I = \frac{P}{2\pi N} = \frac{\eta_f m_a Q_{lvh} (F/A)}{4\pi} \quad (4-2)$$

$$IMEP = \frac{2 \times P}{N V_{swept}} = \frac{\eta_f m_a Q_{lvh} (F/A)}{V_{swept}} \quad (4-3)$$

where V_{swept} is the engine swept volume.

In addition, fuel conversion efficiency and volumetric efficiency can be calculated using the following equations:

$$\eta_f = \eta_{th} \times \eta_c = \frac{P_I}{\dot{m}_f Q_{lvh}} \quad (4-4)$$

$$\eta_v = \frac{m_a}{V_{swept} \rho_a} = \frac{2 \times \dot{m}_a}{V_{swept} \rho_a N} \quad (4-5)$$

where η_{th} is the thermal efficiency (capacity to transform heat into useful work), η_c the combustion efficiency (capacity to release fuel energy), $\dot{m}_f$ the fuel mass flow, $\dot{m}_{air}$ the air mass flow and ρ_a the inlet air density.

Using Equations 4-4 and 4-5, Equation 4-1 can be transformed into:

$$P_I = \frac{\eta_{th} \eta_c \eta_v N V_{swept} Q_{lvh} \rho_a (F/A)}{2} \quad (4-6)$$

The ability of engine designers to extract the maximum performance out of a given piston area independently of the cylinder size can be assessed by considering specific power which is power divided by piston surface area:

$$\frac{P_I}{A_p} = \frac{\eta_f \eta_v \bar{S}_p Q_{lvh} \rho_a (F/A)}{4} \quad (4-7)$$

where $\bar{S}_p$ is the piston mean speed.

To achieve high engine performance, Equations 4-6 and 4-7 demonstrate the importance of high fuel conversion efficiency, high volumetric efficiency (good engine breathing), high inlet air density (thus the interest in air compression solutions), high fuel/air ratio and high mean piston speed.

4.3 Introduction to AVL Boost™

AVL Boost™ is a thermodynamic engine cycle and gas exchange simulation developed by AVL, an Austrian mobility technology company specialised in development, simulation, and testing for the automobile industry [153]. AVL Boost™ has been used extensively at Cranfield University over the 20 past years for research and teaching purposes. One example being the Cranfield F1™ Design Sprint previously mentioned but in addition there have been over a hundred MSc thesis projects based on this simulation technology.

This software is part of the wider AVL simulation software package designed to support the development of new vehicles and powertrains. AVL Boost™ can interact with the other software, with AVL Fire™ notably for 3D flow simulations, and can also be run in collaboration with MATLAB for optimisation and/or automation tasks.

AVL Boost™ is a 0D/1D engine simulation software like GT-Power (Gamma Technologies) or Wave (Ricardo), designed to provide engine performance, exhaust emissions and acoustics predictions based on engine modelling [123].

The combustion event is described with 0D models using semi-empirical models and the energy conservation law; the spatial distribution within the combustion chamber and the flame propagation are not considered for example. On the contrary, the inlet and exhaust pipes are treated as 1D elements, their dimensions (diameter and length) influencing gas exchange processes.

As explained in the Literature Review (Chapter 2), because no volumes are considered, 0D/1D simulation cannot replicate every physical phenomenon, in-cylinder turbulence for example. This is why careful input dataset and results control is necessary with this type of tool to ensure that predictions are relevant and usable. However, what is lost on precision is gained in computation time. 0D/1D engine simulation software requires less computation power than their 3D counterparts, enabling to run more iterations of a given design in a given time.

AVL Boost™ can run both steady-state (or quasi-steady-state (QSS) when steady-state simulation results are subsequently used as input to a second time-marching performance simulation) and transient simulations. In the first case, the engine is operated at constant engine speed and load with specific initial conditions. The case is run over a user-defined number of cycles to allow acoustics phenomenon to stabilise and the simulation to converge. This type of simulation is very useful for design tasks as it allows engine designers to sweep through a range of values for given parameters and to select the optimum design for their performance targets. By running steady-state simulations, the results can be used to analyse the behaviour of a given engine, to conduct analyses not practically realisable on a real engine, like it is done with the 2014+ Digital Twin.

Regarding transient simulations, the engine is given a set of starting conditions and then a control-command which will make its operating conditions evolve with time. It allows the engine designer to assess transient effects that do not appear with steady-state simulations like turbo lag and to test more detailed engine control strategies, but it also requires a different input parameter dataset as it will be highlighted by Section 4.5 and higher computation power. This type of simulation brings more insights regarding how the engine will behave on the road / track, rather than on the dynamometer.

On the contrary to the steady-state simulations and except for specific cases, the operating conditions are constantly changing in the transient simulation, like on a real engine, with each cycle run only once, meaning that the traditional notion of convergence (where simulations run until the IMEP settles on a converged value) is not applicable.

However, as demonstrated in Figure 4-2, transient simulations run with AVL Boost™ are not heavily dependent on initial conditions. Two transient simulations have been run with two different initial conditions. For the first one, the initial pressures and temperatures are derived from the steady-state simulation results for the exact initial engine speed. For the second simulation, the conditions are kept the same, but the initial engine speed is reduced, i.e. the initial conditions are not perfectly adapted for this new speed. The green curve x-axis is offset so that both simulations reach 225 km/h at the same time.

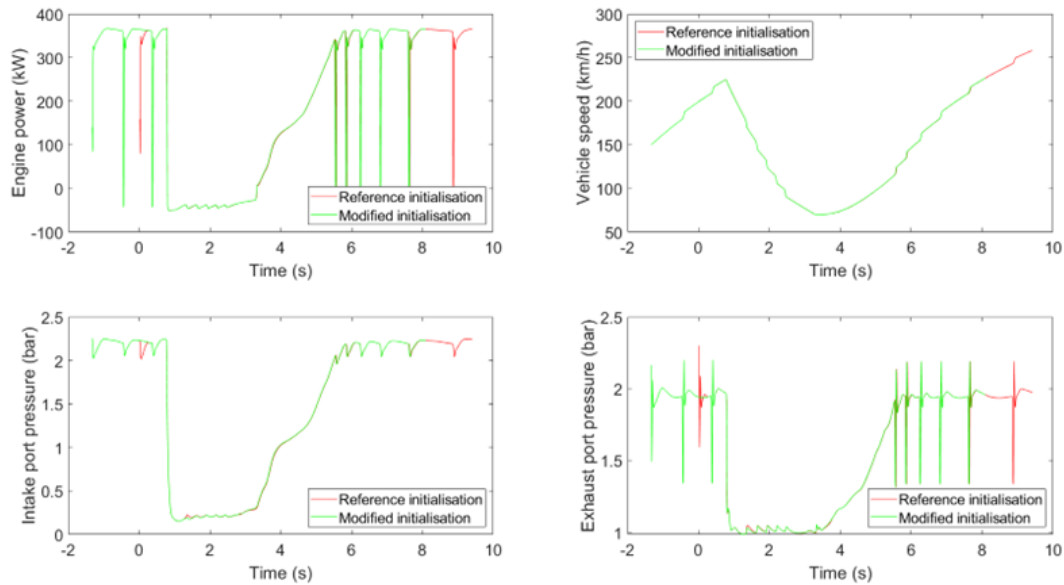


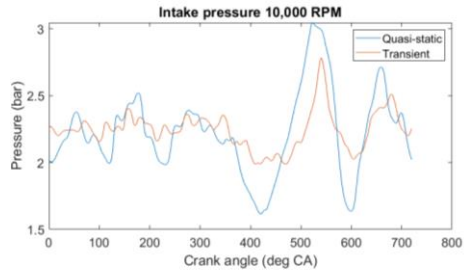
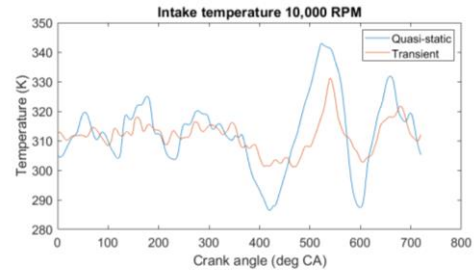
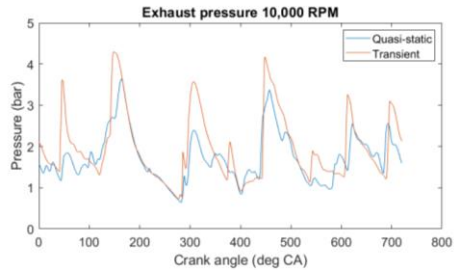
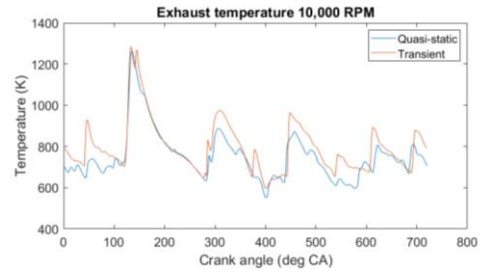
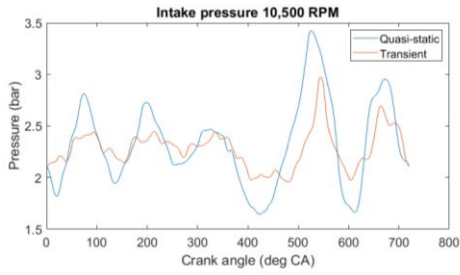
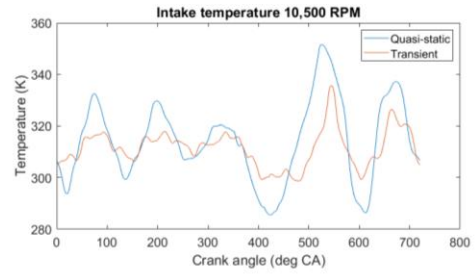
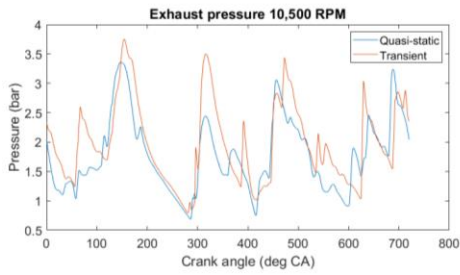
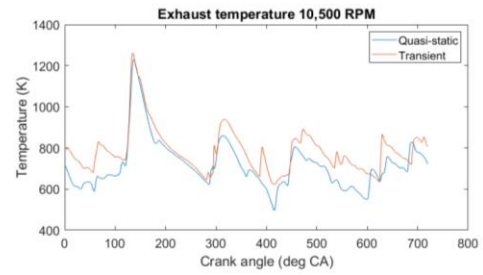
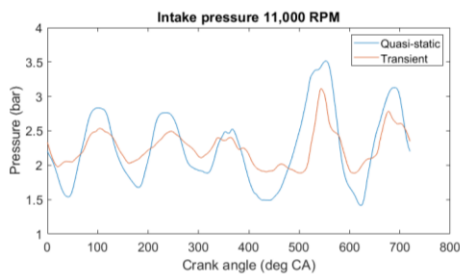
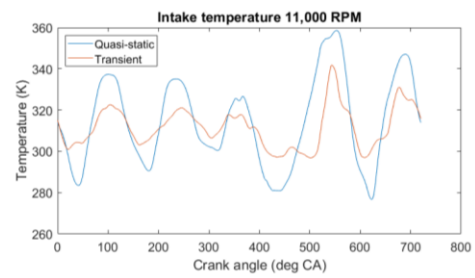
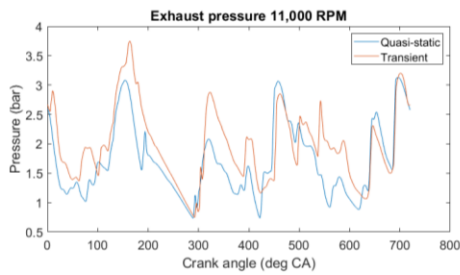
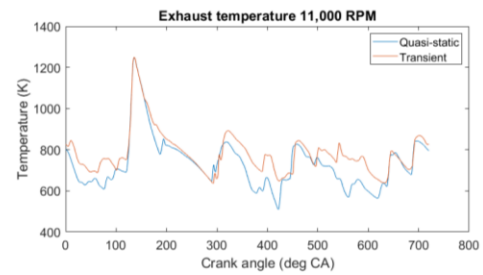
Figure 4-2 - Effect of initial conditions on transient simulation results

Except for the first few cycles, the transient predictions made by both models are very similar. This is why for the later studies (Chapter 7) which are based on transient simulations, the simulation is always started several cycles before the zone of interest begins, to make sure that the initial conditions are overwritten by the time-marching values from the simulation and the results are not dependent anymore on initial conditions.

Regarding this notion of non-convergence for the transient simulations, the pressure waves in the intake and exhaust ports observed for steady-state and transient simulations at specific engine speeds have been compared in Figure 4-3. Inspecting the convergence of pressure waves in the intake and exhaust systems is a much more rigorous test of overall simulation convergence than relying on the IMEP settling to a final value. The first is a test of the convergence of the 1D wave action whereas the second is only a test of convergence of the 0D thermodynamics which can be acceptably converged even though the wave action is ‘wrong’.

It can be observed that steady-state and transient simulations predict similar pressure waves behaviour in the engine ports. At the intake, both simulations follow the same trends, i.e. peaks are in phase, but the amplitudes are lower for the transient cases (which would be expected from the resonant response of a fluid-pipe-network: the resonant response takes time to build up). In addition, the steady-state curves are smoother, with less perturbations. Regarding the exhaust, the curves are even more similar than that at the intake with peaks in phase and lower amplitude difference.

Obviously, the pressure and temperature curves are different as time is given to the acoustic effects to settle in the steady-state simulations which can explain the higher amplitude at the intake port. But it is reassuring to observe that despite being very different type of simulations, the models experience similar trends.



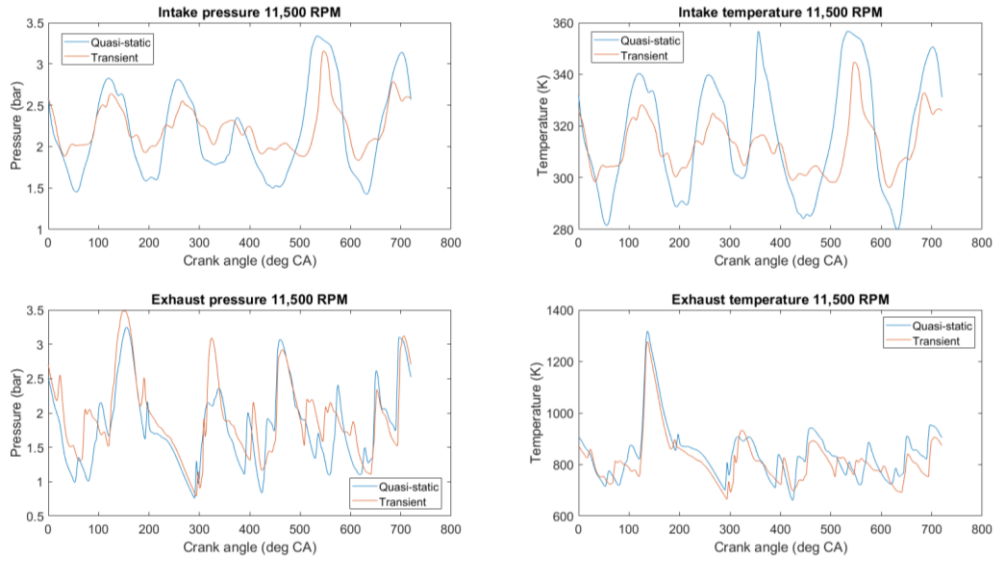


Figure 4-3 - Port pressures and temperatures comparisons between steady-state and transient simulations (from top to bottom: 10,000 RPM / 10,500 RPM / 11,000 RPM / 11,500 RPM)

In addition to its engine performance prediction capabilities, AVL Boost™ provides two tools which have been useful for this Doctoral Thesis.

The first one is the AVL Boost™ Gas Properties Tool which enables the user to compose fuel blend to be used in simulations in addition to the common fuels proposed by the software. Within a list, the user can choose several fuels in specific proportions. Then, based on JANAF / NASA coefficients [126], the tool calculates the properties of the created blend which are summarised in a .bgp file that can be used as simulation input. This file describes the thermodynamic properties of the blend (specific heat capacities, specific internal energy, specific entropy), the physical properties of the blend (molar mass, density, lower heating value, heat of vaporisation, stoichiometric air/fuel ratio, carbon, oxygen, nitrogen to total mass ratio), and the composition (H, O, N, H₂, OH, CO, NO, O₂, H₂O, CO₂, N₂ and Ar) and properties of the exhaust gases (molar mass, specific gas constant, specific heat capacities, specific enthalpy, specific internal energy, specific entropy, heat capacity ratio) depending on air/fuel ratio, pressure, and temperature. This tool does not contain any element of rate-kinetics prediction and does not rely on fuel reaction mechanisms.

The second tool to highlight here is the AVL Boost™ Turbocharger Tool which can be used to create a complete model of the turbocharger. AVL Boost™ gives the user two options to model the turbocharger, either a simplified or a full model. The Turbocharger Tool is useful for the full model for which a compressor map and turbine swallowing curve are required. The tool enables the user to create .btc files containing these two characteristics and to input them into the engine model. Using this file, the simulated turbocharger behaviour is more realistic than with the simplified model and can calculate turbocharger rotor speed and indicate when the compressor operates at the surge line.

4.4 Steady-state AVL Boost™ models

4.4.1 Digital Twin of the 2014+ Formula One™ engine

Figure 4-4 shows the AVL Boost™ layout of the 2014+ engine steady-state model (2014+ Digital Twin) and Figure 4-5 details its geometry. The model is symmetric, so the dimensions in Figure 4-5 are only given for one side of the engine.

The input dataset of this model has already been discussed and analysed in detail in Chapter 3 and is summarised in Appendix C where it can be compared to that of the 2026 engine steady-state model.

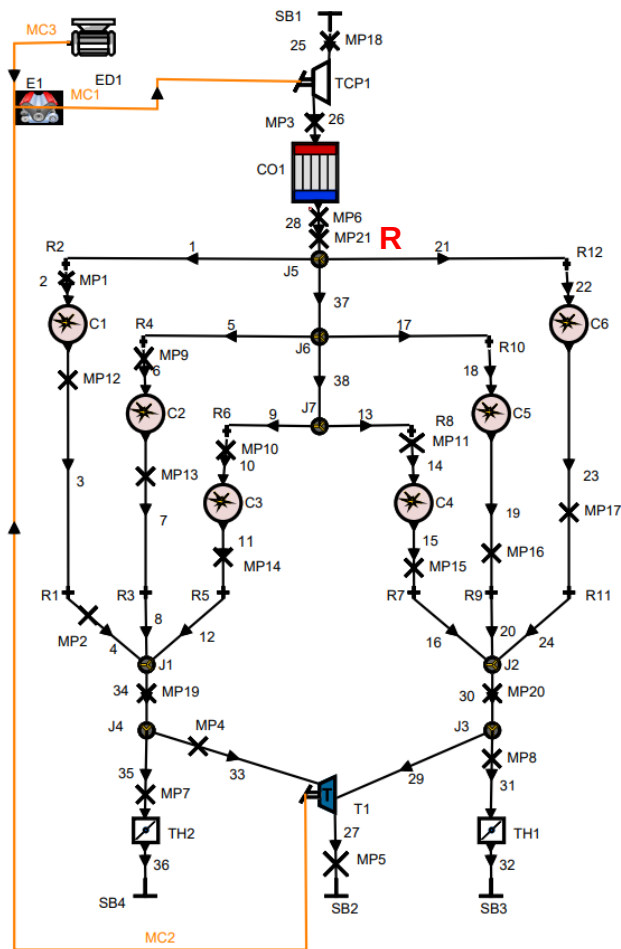


Figure 4-4 - AVL Boost™ layout of the 2014+ engine steady-state model (2014+ Digital Twin)

In addition to the parameters mentioned in Appendix C which are true for every simulated engine speed, the model is provided with a speed-dependent dataset whose parameters are listed in Table 4-1:

Parameters
Engine speed
Mass of fuel injected per cycle per cylinder

The engine does not feature a throttle body as it was not necessary for the simulations run with the 2014+ Digital Twin but this component is included in the 2026 Formula One™ transient model as it is necessary for its operation (see Section 4.5).

In the 2014+ Digital Twin, the turbocharger is split to accommodate the action of the MGU-H (referred to as ED1 in Figure 4-4). The compressor and turbine operate separately, the compressor power consumption can be different to the power produced by the turbine, and are mechanically linked to the engine icon with the compressor consuming work from the engine, as a supercharger, and the turbine provides work to the engine, see the direction of the arrows on the orange mechanical links in Figure 4-4. The mechanical efficiency of the turbocharger is only considered once, on the turbine side. The MGU-H power is set as the difference between the turbine power and compressor requirement so that the engine energy balance is not impacted by this unusual modelling. Considering the direction of the mechanical link between the MGU-H and the engine icon, positive values mean that the MGU-H provides energy to the compressor, whereas if negative, the MGU-H recovers energy from the turbine.

The detailed physical behaviour of the MGU-H is not investigated in this model. It is defined as an 100% efficient electric motor and so are the mechanical connections.

The operating points of the compressor are user-defined to achieve the target air/fuel ratios (no complete compressor map inputted in the 2014+ Digital Twin) and the turbine operating points are calculated by AVL Boost™ using the turbine swallowing curve shown in Figure 4-6 which has been user-defined to allow the turbine to swallow the whole engine mass flow at any engine speed and achieve satisfying power regeneration from the exhaust gases and MGU-H power.

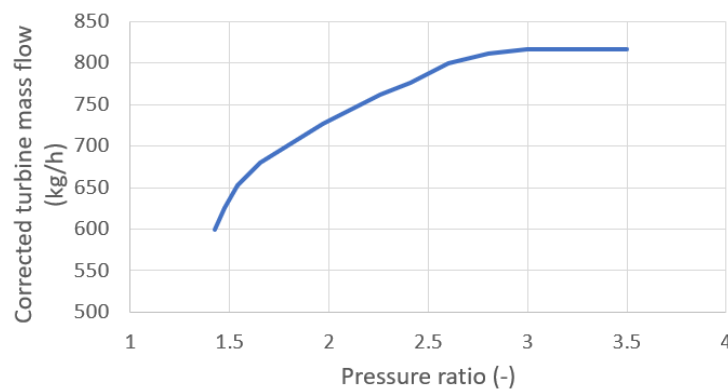


Figure 4-6 - 2014+ Digital Twin turbine swallowing curve

The engine internal volumetric efficiency is calculated based on the conditions measured at measuring point MP21, which is marked by a red “R” in Figure 4-4, at the plenum inlet.

Despite AVL Boost™ offering a plenum model, it was decided to model the 2014+ engine plenum as a series of pipes to have better control of the dimensions and to be able to place, if necessary, measuring points inside the plenum. This

approach also includes for wave action within the ‘plenum’ which is absent from the 0D plenum model included in AVL Boost™.

The two throttle valves located at the bottom of Figure 4-4 on each side of the turbine are the waste gates.

The two junctions located on each side of the engine between the exhaust secondary pipes and the turbine inlets (J3 and J4) are set with specific flow coefficients. To avoid unrealistic back-flow, fluid travelling from the turbine upwards the turbine inlet pipe, when the waste gates are open, only the inlet pipes, pipes 30 and 34, have junction inlet flow coefficients set at 1. For the pipes exiting these junctions, the inlet flow coefficient is set at 0 but the outlet one is set at 1. This specific setting is not done on the other junctions of the model.

For the firing order detailed in Appendix C, the reader should refer to the cylinder numbers given in Figure 4-4 (“CX” mark next to each cylinder) with cylinder 1 being at the top left and cylinder 6 at the top right.

4.4.2 2026 Formula One™ engine model

Figure 4-7 presents the layout of the AVL Boost™ model of the 2026 engine which is deliberately laid out differently compared to that of 2014 to make it easier to differentiate them and because of the different disposition of the turbocharger.

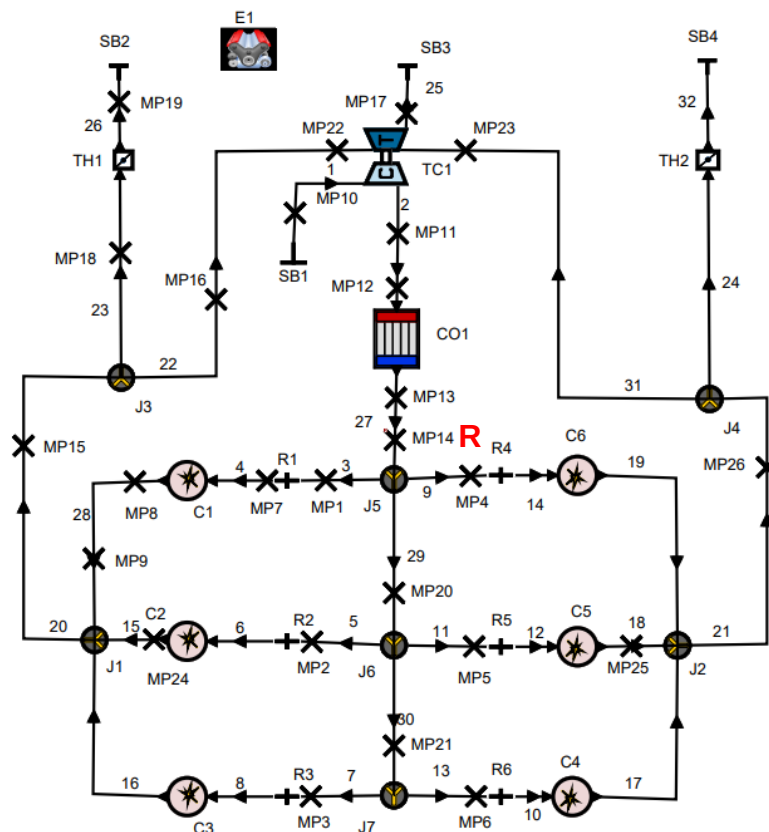


Figure 4-7 – AVL Boost™ layout of the 2026 engine steady-state model

The creation of this model is different to that of the 2014+ Digital Twin as, at the moment of writing this Doctoral Thesis, the 2026 engine is still under development and has not yet been on track. Engine manufacturers are obviously already working on their version of the 2026 engine, but their technical choices are not yet definitive, and no information is available. The choices made for the 2026 engine model thus did not benefit from feedback or input from the industry, discarding the 2026 Formula One™ engine model presented here from the 'Digital Twin' qualification.

This model will thus not be referred to as a Digital Twin as it does not aim at replicating the behaviour of a given engine, its objective is to predict, based on the knowledge developed on the 2014+ Digital Twin and the 2026 regulations [15], what could be the performance of this new generation of Formula One™ engines and investigate how some of the challenges linked to the new regulation can be overcome.

As for the 2014+ Digital Twin, the 2026 engine model requires an input dataset composed of geometric, calibration and compound parameters. As no data on this new engine is available, certain parameters have been transferred from the 2014+ to the 2026 model. This section presents the 2026 model input dataset and discussed how the parameter values have been selected in addition to introducing the model operation.

In addition to Figure 4-7, Figure 4-8 presents the geometry of the 2026 engine model and its detailed input dataset can be found in Appendix C alongside the description of the 2014+ Digital Twin input parameters.

As the MGU-H is removed under the 2026 regulations [15], there is no need to separate the compressor and the turbine as done for the 2014+ Digital Twin. Furthermore, this model does not contain any electric motor as it is aimed at studying only the behaviour of the internal combustion engine.

The 2026 model is run for the same engine speeds as the 2014+ Digital Twin and the same engine speed dependent parameters (see Table 4-1) plus a parameter defining the turbine calculation mode (see below) are provided to the software for each case.

The structure of the 2026 engine model is very similar to that of the 2014+ Digital Twin, the main difference being that the pipe diameters have globally been reduced in consequence of the engine reduced mass flow (Figure 4-8).

As the turbocharger is not split, the work consumed by the compressor and the work produced by the turbine (multiplied by the mechanical efficiency of the assembly) must be equal. AVL Boost™ proposes different calculation modes so that it can be achieved. For this model, it was chosen to set the compressor the same way as in the 2014+ Digital Twin, i.e. the simulation is given a specific user-defined compressor boost pressure which allow to reach the target air/fuel ratio.

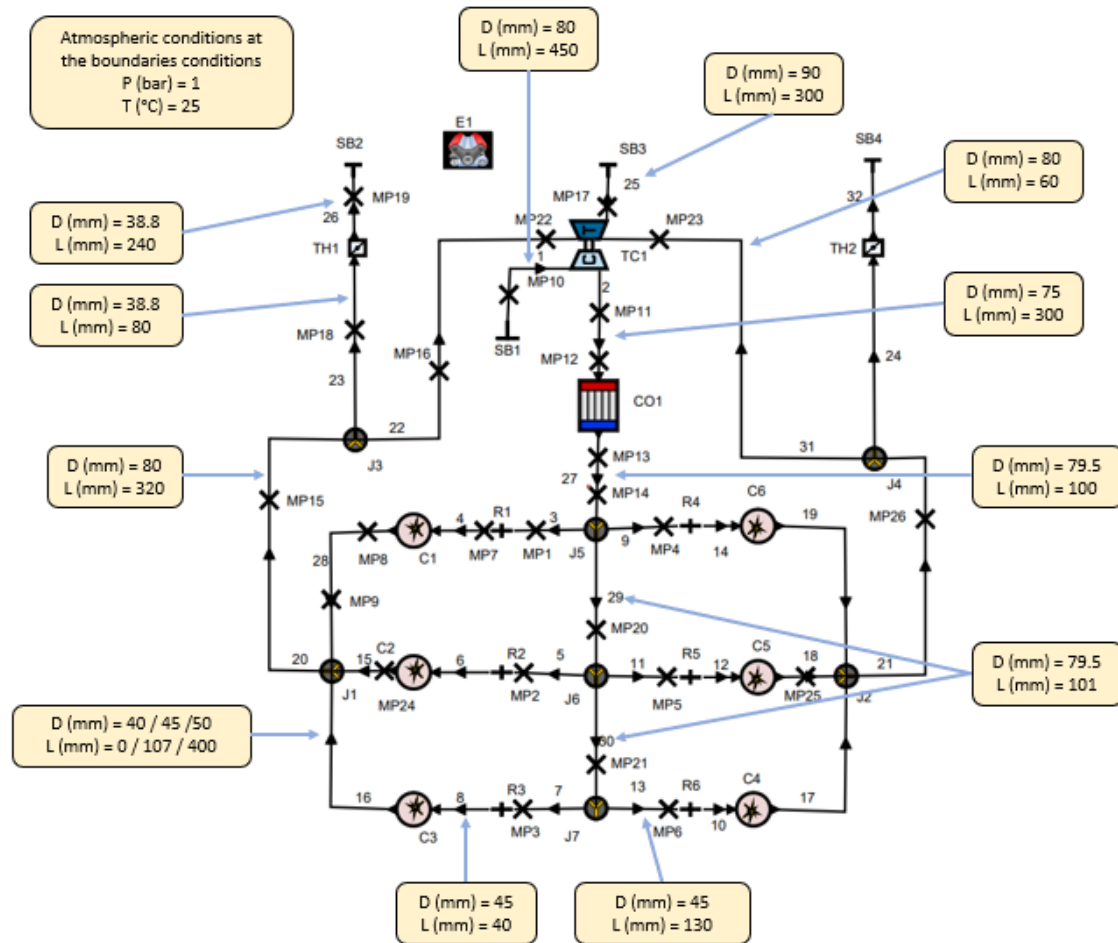


Figure 4-8 - Geometry of the 2026 engine steady-state model

However, the turbine is modelled differently. Instead of using a swallowing curve, AVL Boost™ models the turbine as a restriction with a specific reference cross sectional area. Until 11,000 RPM included (engine speed close to peak power), the Turbine Layout Calculation mode is used, i.e. based on the reference area, AVL Boost™ calculates the required discharge flow coefficient so that compressor and turbine powers match. Above 11,000 RPM, the mode is switched to Waste Gate Calculation where AVL Boost™ uses the turbine reference area (defined as the turbine outlet pipe cross section area) and a user-defined turbine discharge coefficient to determine the amount of mass flow that must be diverted from the turbine, once again to match compressor and turbine powers. This last mode simulates the use of waste gates.

As shown in Figure 4-7, the 2026 engine model also features valves on each side of the turbocharger which are waste gates. Despite being modelled, it was chosen in this model not to use them, the waste gates were left on the model's layout for visual reasons and remind that the effect of this device is modelled. Rather, the use the waste gate is modelled via the turbine module in Waste Gate Calculation mode as explained previously, which proved much easier to control and achieved the same results. In the model, all the flow is directed towards the turbine and based on the user-defined turbine properties and when the turbine model is set into Waste Gate Calculation mode, AVL Boost™ calculates the fraction of the

flow that must grow through the turbine to recover enough energy to power the compressor (turbine to total ratio $R_{turbine/total} = \frac{\dot{m}_{turbine}}{\dot{m}_{total\ exhaust\ flow}}$). If $R_{turbine/total} = 1$, it represents the waste gate fully closed, all the exhaust flow goes through the turbine, if $R_{turbine/total} = 0$, all the flow is diverted, $\dot{m}_{turbine} = 0$. The rest of the flow $(1 - R_{turbine/total})$ represents the fraction of fluid that would go through the waste gate on a real engine. When looking at the simulation results, the total flow at the turbine inlet is equal to the turbine outlet flow, it is just for the calculations that AVL Boost™ determines a coefficient used to calculate the energy recovered by the turbine.

As for the 2014+ Digital Twin, pipes have been used to create the plenum and the reference measuring point for internal volumetric efficiency is also located at the plenum inlet with MP14.

One of the important innovations brought by the 2026 regulations is the switch from fossil fuels to fully advanced sustainable fuels. Freedom is given to fuel suppliers to select the advanced sustainable components of their choice to create a fuel that meets the regulation requirements.

For the 2026 engine model, given the issues faced with rate-kinetics simulation, it was decided to model the fuel using AVL Boost™ Gas Properties Tool, as a mix of 80% in liquid volume of high-performance gasoline (used in the 2014+ Digital Twin) and 20% of ethanol (E20). By using the properties listed in Table 4-2, the properties of this blend have been estimated:

Parameters	Gasoline	Ethanol [154]	2026 fuel
Density (kg/m ³)	748 [154]	789	765.2
Q_{lvh} (MJ/kg)	44 [5]	27.753	40.61
AFR _{st} (-)	14.7	8.9986	13.51
Heat of evaporation (kJ/kg)	419 [146]	963	532.5

Table 4-2 - Properties of gasoline, ethanol, and the 2026 engine model's fuel

As shown in Table 4-2, despite not being directly applicable as it contains fossil gasoline, this composition yields a 765.2 kg/m³ density which is within the regulation limits and a lower heating value Q_{lvh} of 40.61 MJ/kg also within the [38; 41 MJ/kg] window allowed by the regulations [15]. However, as the 2026 fuel flow limit is set in terms of energy, the lower Q_{lvh} , the greater the quantity of fuel required to match the fuel energy limit.

The fuel supplier will thus aim at designing a fuel with the highest possible energy density, this is why it was decided to adopt 41 rather than 40.61 MJ/kg for the 2026 engine model fuel. For stoichiometric air/fuel ratio and heat of evaporation, the values shown in Table 4-2 are used.

In addition, the regulations state that the oxygen content of the fuel must be between 6.7 and 7.1 wt%. Ethanol is made of 34.8% of oxygen in mass whereas gasoline / F1 fuel considered here has 0% oxygen content. Thus, for the E20 chosen here, the oxygen content of the fuel is of 7.3 wt%, i.e. close to the regulations limits thus further justifying the choice of this fuel composition for the 2026 engine model.

Because the fuel composition will change between 2014+ and 2026, its combustion properties, laminar flame speed for instance, will evolve leading to different combustion behaviour and the design of the pre-chamber is also prone to evolve, thus influencing the combustion process. The rate-of-heat-release is very likely to be impacted by the introduction of this new regulations.

However, as discussed in Chapter 3, rate-of-heat-release is a compound parameter which is influenced by a large number of variables including fuel, combustion timing, in-cylinder geometry and fluid flow. Even when knowing the exact fuel composition, it is difficult to estimate it without measuring it on a running engine, so as the 2026 fuel composition is unknown, it proves impossible to make a sound assumption on its combustion characteristics. One way of doing so would consist in using rate-kinetics simulations but as explained above, the efforts put in this solution are not successful.

As the 2026 engine will still be fuel-limited, it will again be run at high-dilution levels to make the most of this limited energy resource, use a pre-chamber to maximise mixture ignition reliability and the regulations do not make combustion chamber geometry evolve significantly compared to 2014+. The combustion is thus likely to be more similar to that of the 2014+ engine than to a conventional engine. This is why it was decided to use the 2014+ Weibe curve in the 2026 engine. In addition, as a fuel-limited engine, high efficiency will still be the target, and this is made possible by a fast rise in mass fraction burned at the beginning of the combustion as shown in Section 3.5.2, further justifying the unchanged ROHR curve compared to the 2014+ Digital Twin.

In the same way, due to the similarities in design, the valve characteristics (lift and flow coefficients) used in the 2014+ Digital Twin have been transferred to the 2026 engine model. The same has been done for wall temperatures as it is a compound parameter difficult to accurately estimate without experimental measurement or detailed thermal simulations.

As it was done for the 2014+ Digital Twin and is detailed in Chapter 5, the pressure exponent of the knock onset model has been modified to be more suitable to the level of in-cylinder pressures experienced by the engine. Because of lower in-cylinder pressure (see Chapter 6), the value of the 2026 pressure exponent has not been reduced as much as for the 2014+ Digital Twin but was changed from 1.7 to 1.62.

4.5 Transient AVL Boost™ model of the 2026 Formula One™ Power Unit

The layout of the transient AVL Boost™ 2026 engine model is shown in Figure 4-9. The geometry of this model is the same than its steady-state counterpart, but some elements have been modified or added to allow the transient simulation to run properly.

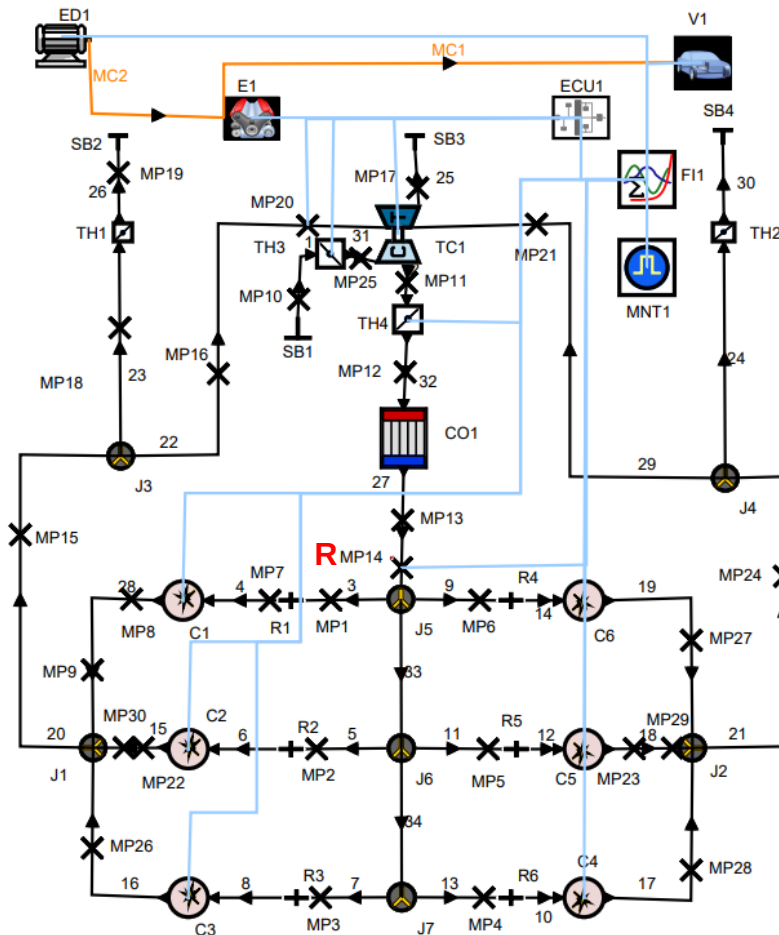


Figure 4-9 - AVL Boost™ layout of the 2026 engine transient model

The aim of this model is to simulate a transient engine dynamometer test with varying load on the engine. The simulation does not include a driver or vehicle model and the electric storage (ERS) is not modelled.

As explained in Section 4.3, for transient simulations, the model is given a set of initial conditions in addition to a control-command, in this case an engine load. Based on the control maps defined by the user, the model defines itself its operating points for the different parameters.

To be able to define these operating points, an ECU icon (ECU1 at the top right of Figure 4-9) has been added. This simulation module can control based on 1 or 2D maps the behaviour of “actuator” engine parameters based on the input of

“sensor” parameters. The data link between the ECU and the elements it controls are possible thanks to the blue data connections visible in Figure 4-9.

As the ECU icon is limited to 1 and 2D maps, more complex control strategies, like the ones described in Chapter 7, cannot be implemented using this element alone, this is why the “Formula Interpreter” module (FI1) has been added. The Formula Interpreter allows the user to write code in C language to control engine parameters, “actuators”, based on the inputs of the same “sensors”.

In the model shown in Figure 4-9, the ECU and the Formula Interpreter control the following parameters:

Parameters
Engine load signal
Load applied to the engine crankshaft
Simulated clutch position
Simulated gear ratio
Fuelling
MGU-K power
Throttle angle
Waste gate opening (within the turbocharger model)

Table 4-3 - Parameters controlled by the ECU and the Formula Interpreter in transient simulations

To run transient simulation, the engine must be coupled with an “energy consumer” which in the case of Figure 4-9 is the vehicle module (V1). This icon aims at modelling in a simplified manner the load that a specific vehicle would apply on the crankshaft of the engine. The vehicle main parameters are shown in Table 4-4:

Parameter	Unit	Value
Total mass	kg	936
Frontal area	m ²	1.2
Drag coefficient C _D	-	1
Downforce coefficient C _L	-	5
Gear ratios	-	2.65 / 2.29 / 1.975 / 1.7 / 1.46 / 1.255 / 1.08 / 0.93:1
Final gear ratio	-	5.35

Parameter	Unit	Value
Gear upshift / downshift engine speeds	RPM	9000 / 12000
Gear shifting time	ms	12
Brake bias F/R	-	60% / 40%
Tyre rolling radius	mm	360
Tyre friction coefficient	-	1.2

Table 4-4 - Vehicle settings for transient simulations

The total mass of the vehicle is assumed to be 796 kg for the car [155], 80kg for the driver plus equipment, seat, and ballast [155] and 60 kg of fuel.

The tyre rolling radius is set at 360 mm, i.e. the value given on Pirelli's website for their Formula One™ tyre [156]. It is assumed that the tyre does not deform significantly under rolling so that this geometric value is also the effective tyre rolling radius.

The simplified vehicle load is assumed to only be due to constant rolling resistance and drag. Rolling resistance is approximated to be 2% of the vertical load with an average downforce estimated at 2.5g [157]. Considering these, the vehicle load is calculated using the following equation:

$$\begin{aligned}
 \text{Vehicle load (N)} &= 0.02 \times m_{car} \times g \times (2.15 + 1) + \frac{1}{2} \times \rho \times C_D \times A_{car} \times V^2 \quad (4-8) \\
 \text{Vehicle load (N)} &= 578.5 + 0.72 \times V^2
 \end{aligned}$$

where m_{car} is the vehicle mass, g the earth's gravitational acceleration (taken as 9.81 m/s²), ρ the air density (1.2 kg/m³), C_D the vehicle drag coefficient, A_{car} the vehicle frontal area and V the vehicle speed.

When deceleration is considered, the maximum available braking force is calculated using the following equation:

$$\text{Braking force (N)} = \mu \times \left(\frac{1}{2} \times \rho \times C_L \times A_{car} \times V^2 + m_{car} \times g \right) \quad (4-9)$$

where μ is the tyre friction coefficient and C_L the vehicle downforce coefficient.

Based on this equation and the vehicle's brake bias and the MGU-K power, during deceleration event, the vehicle load is modified to reflect the braking due to the physical braking system.

In addition, for transient simulations, the moment of inertia of rotating parts must be inputted to calculate variation in angular velocity. The moments of inertia of the given engine and drivetrain have been calculated using the equations presented by Harrison [158] and are equal respectively to 0.034 and 0.1 kg.m².

Compared to the steady-state model, as the transient model will be used to assess interaction modes between the internal combustion engine and the MGU-K, an electric motor has been added to the layout to represent the MGU-K (ED1). The detailed behaviour of an electric motor is not modelled here. The MGU-K is set with a 98% electrical efficiency and is mechanically linked to the engine through a 97% efficient mechanical connection with a 4:1 constant gear ratio. Its power output is controlled by the Formula Interpreter considering the power and torque limits imposed by the regulations [15] but the Energy Store (ES) which is normally linked to the MGU-K is not considered here.

In addition, for transient simulations, the moment of inertia of rotating parts must be inputted to calculate variation in angular velocity. For the MGU-K, the value of the CTSM242 electric motor developed by Helix has been used [32] and is shown in Table 4-5 alongside the other MGU-K parameters:

Parameter	Unit	Value
Maximum power	kW	350
Maximum torque	Nm	500
Peak electrical efficiency	-	0.98
Moment of inertia	kg.m ²	0.042
Mechanical connection efficiency	-	0.97
Mechanical connection gear ratio	-	4:1

Table 4-5 - MGU-K parameters for the 2026 transient engine model

For transient simulation, the turbocharger must be fully modelled so that the engine model can itself calculate what are the compressor and turbine operating points according to the instantaneous engine conditions. To do so, the compressor map of Garrett G30-770-58mm turbocharger [159] and a user-defined turbine swallowing curve are implemented in AVL Boost™ using its Turbocharger Tool and shown in Figure 4-10.

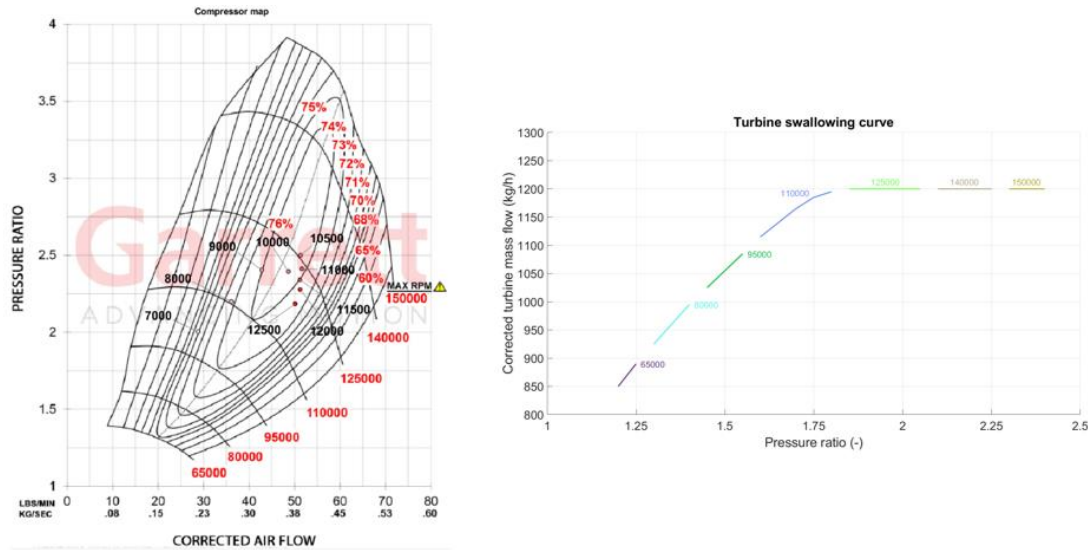


Figure 4-10 - Compressor map (plus steady-state simulation compressor operating points) and turbine swallowing curve for the transient 2026 engine model

The turbocharger moment of inertia has been estimated at $6.55e-5 \text{ kg.m}^2$ via a self-created CAD model of the rotor based on the dimensions of the Garrett G30-770-58mm turbocharger [159] using nickel for the rotor and turbine wheel and titanium for the compressor wheel. A schematic of the turbocharger is shown in Figure 4-11:

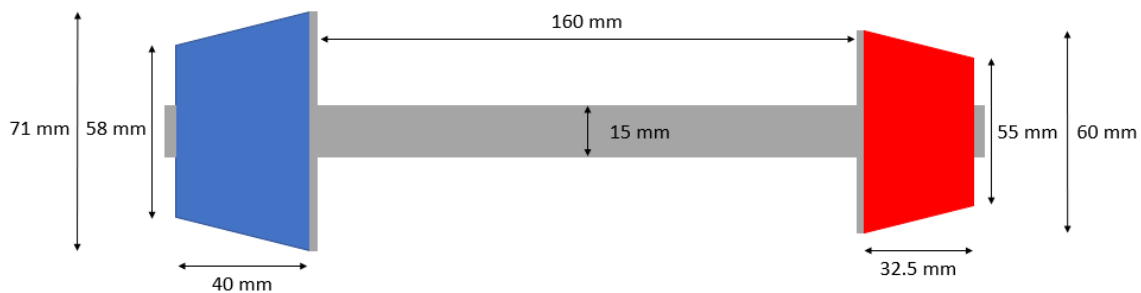


Figure 4-11 - Schematic of the turbocharger used in the transient 2026 engine model

The opening of the waste gate, modelled by the coefficient defining the mass flow going through the turbine relatively to the total exhaust flow $R_{turbine/total}$, is controlled using a 2D map implemented in the ECU which is a function of turbine inlet mass flow and turbocharger rotor speed. The target is to modify $R_{turbine/total}$ to keep the rotor speed around 125,000 RPM when the engine operates at full load.

As the transient model is also used for part-load operation, a throttle body has been added between the compressor and the intercooler at 200 mm from the compressor outlet. Another throttle body has been added in front of the compressor to try to model the effect of a pre-compressor valve (see Chapter 7).

For specific values like wall temperature, only one value can be input on the contrary to steady-state simulations where these parameters are speed dependent. The values used for 10,500 RPM in the steady-state simulations have

been used for the transient model as it is the middle engine speed of the studied window.

The objective with this transient model, and with the others too, is to be the minimum viable model for the simulation objectives. The model must be able to express the phenomenon of interests but without complexifying it too much so that unnecessary computing power is required.

To simplify the fuel control strategy because of the non-practical AVL Boost™ coding interface, it was chosen to cut fuel injection during deceleration event. By doing so, the exhaust enthalpy drops very significantly and thus the power produced by the turbine also decreased. If the throttle valve is kept open to conventional levels during braking events, the air mass flow going through the compressor leads to still significant power requirement from the compressor. In consequence of the power imbalance between the compressor requirement and turbine production, the turbocharger rotor speed drops unrealistic values, nearly zero if the deceleration event is long enough.

To avoid that, during deceleration events, the throttle valve is closed to the minimum angle that does not make fail the simulation, namely 5°. However, at such low angle, the flow coefficients are significantly reduced which leads the compressor to operate beyond the surge line. By doing so, the compressor load is significantly reduced, and the turbocharger rotor speed drop is limited to more expected levels. Figure 4-12 shows the compressor operating points for a simulation running under the described conditions.

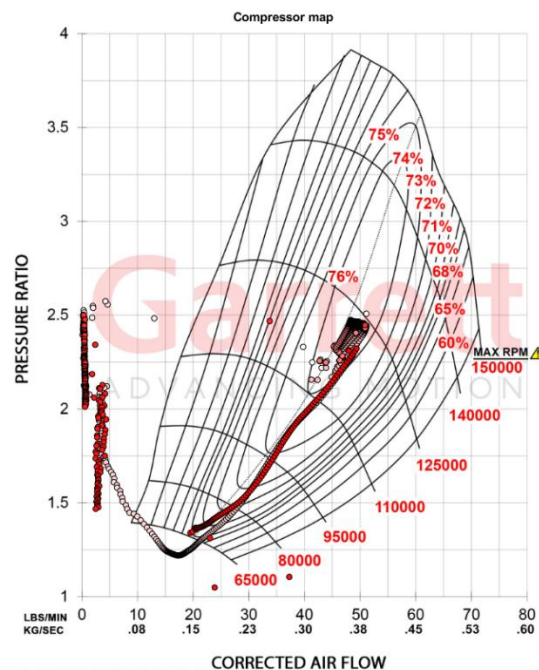


Figure 4-12 - Example of simulation control leading to compressor surge

The point colour evolves from white to red with simulation time. It can clearly be seen that a large number of points are operating in the high-efficiency zone of the

compressor map which corresponds to the full load engine operation but also that the compressor is surging at other moments of the simulation.

By cutting the fuel supply during deceleration events, an unusual control of the throttle valve is required so that turbocharger rotor speed can evolve in a manner that appears rational. However, this choice of simulation control leads to unrealistic compressor operating points. Surge can be tolerated for a short time if it allows the car to be faster but not for prolonged period of time as it is the case in Figure 4-12. This modelling approach also leads to unrealistic short-duration spikes in compressor outlet temperature at the beginning of deceleration events when the compressor starts surging due to temporary reduction in air mass flow rate. As a results, the intercooler outlet temperature also spikes for a short time. Implementing a recirculation valve for surge protection is a practical option on the physical engine but this 3D flow detail cannot be modelled readily in AVL Boost™ and the unrealistic temperature spikes remain in the modelling results.

Regarding the intercooler, it has been modelled with a very high efficiency of up to 95% (target outlet temperature of 40°C and a 35°C coolant temperature) which is not realistic for an onboard intercooler but possible in an optimised test cell. This choice of intercooling parameters was made in absence of more detailed data on the intercooler, inlet temperature and the precise cooling effect of GDI in the real Formula One™ engine.

With the intercooler model proposed by AVL Boost™, only the target outlet temperature, the coolant temperature and the target pressure drop can be defined which characterise the heat capture. However, the heat rejection side of the intercooler system depends on the air flow over the radiator in the sidepod and the pressure and flow rate of the coolant flow. The decision is taken here to assume that the combination of vehicle and engine speeds, alongside careful system design would enable the intercooler to control the air temperature on a moving race car to a constant outlet temperature in the same way it is handled in the simulation.

As investigating turbo lag is one of the objectives of this model, the priority has been put on the overall turbocharger behaviour, ensuring relevant rotor speed evolution, rather than on relevant compressor operation during deceleration phases.

For the 2026 Formula One™ transient engine model and with the chosen compressor map (Figure 4-11), the turbocharger mostly operates close to 120,000 RPM. If turbo lag were to be limited by only allowing the turbocharger rotor speed to drop to the 110,000 RPM iso-speed line, the compressor must be provided between 19.4 (at the surge line) and 28.1 kW (at the choke line). These values are far from the reported 65 kW peak MGU-H power in Chapter 3. Even though the compressor map is different for the 2014+ Formula One™ engine, it shows how simple it can be to control the turbocharger rotor speed during low exhaust enthalpy period using an electrically assisted turbocharger.

Another limitation of the transient model presented here concerns the realism of gear changes. Because of the large MGU-K moment of inertia and the way AVL

Boost™ models gear changes, a significant and sudden increase in vehicle speed can be observed as shown in Figure 4-13:

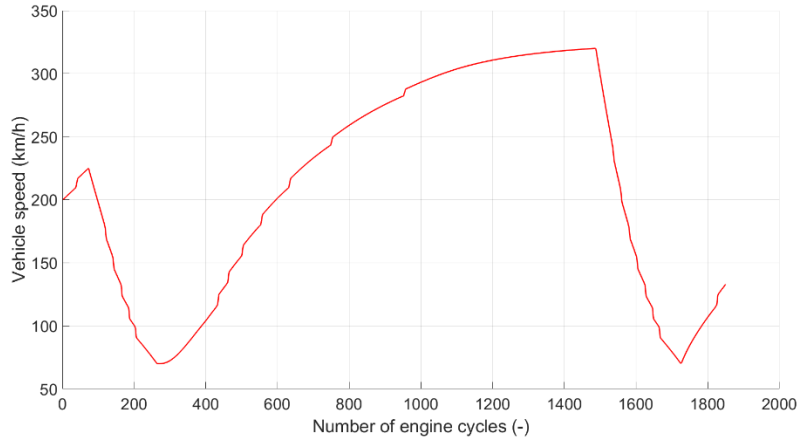


Figure 4-13 - Example of vehicle speed evolution with the transient 2026 engine model

Different solutions have been tried to solve this issue, unsuccessfully. However, the vehicle model and a relevant moment of inertia for the MGU-K are necessary in the transient model to properly model the evolution of engine load and engine speed, this is why they were kept despite this issue during gear change when the model features an MGU-K. Nevertheless, this affects all the cases run with this model in the same way. Thus, it does not impact the comparisons made between models.

4.6 Energy and exergy analysis MATLAB tool

A MATLAB code has been written to conduct energy and exergy analyses using results from AVL Boost™ simulations. The energy / exergy analysis conducted in this Doctoral Thesis only focuses on the efficiency with which the engine transforms fuel into useful work. Thus, the elements external to the cylinders, i.e. the turbocharger, the electric motors, the intercooler, are not considering in the analyses.

In addition, the energy and exergy analyses conducted here do not aim at investigating in depth the conversion processes happening within the cylinders, like the fuel/air mixing or the flame propagation. They only relate the input (fuel energy) to the outputs (power, friction losses, exhaust enthalpy, etc).

For energy analysis:

$$E_{in} = E_{out} \quad (4-10)$$

$$\dot{E}_{fuel} = \dot{E}_{power} + \dot{E}_{exhaust} + \dot{E}_{heat_transfer} + \dot{E}_{friction} \quad (4-11)$$

where $\dot{E}_{fuel}$ is the fuel energy flow, $\dot{E}_{power}$ the brake power produced by the engine, $\dot{E}_{exhaust}$ the exhaust enthalpy flow, $\dot{E}_{heat_transfer}$ the amount of heat

transferred from the cylinder to the engine cooling system and $\dot{E}_{friction}$ the power lost due to friction.

For exergy analysis [41]:

$$E_{x_{in}} - E_{x_{out}} = E_{x_{dest}} \quad (4-12)$$

$$\begin{aligned} \dot{E}_{x_{fuel}} + \dot{E}_{x_{air}} \\ = \dot{E}_{x_{power}} + \dot{E}_{x_{exhaust}} + \dot{E}_{x_{heat_transfer}} + \dot{E}_{x_{friction}} \\ + \dot{E}_{x_{dest}} \end{aligned} \quad (4-13)$$

where $\dot{E}_{x_{fuel}}$ is the fuel exergy rate, $\dot{E}_{x_{air}}$ the air exergy rate, $\dot{E}_{x_{power}}$ the brake power produced by the engine, $\dot{E}_{x_{exhaust}}$ the exergy rate of the exhaust gas, $\dot{E}_{x_{heat_transfer}}$ the exergy rate of heat transfer from the cylinder to the cooling system, $\dot{E}_{x_{friction}}$ the power lost in friction and $\dot{E}_{x_{dest}}$ the rate of exergy destruction.

$$\dot{E}_{x_{fuel}} = \dot{m}_{fuel} \times \varphi \times Q_{lvh} \quad (4-14)$$

$$\begin{aligned} \varphi = 1.0401 + 0.1728 \times \frac{H^r}{C^r} + 0.0432 \times \frac{O^r}{C^r} + 0.2196 \times \frac{S^r}{C^r} \times (1 \\ - 2.0628 \times \frac{H^r}{C^r}) \end{aligned} \quad (4-15)$$

where φ is the fuel exergy factor and H^r/C^r , O^r/C^r and S^r/C^r are ratios of mass fractions of hydrogen, oxygen, and sulfur with respect to carbon [160]. The fuel exergy factor is introduced as the fuel lower heating value does not fully represent the amount of work extractable from the oxidation of the fuel molecules. For instance, it does not account for the work potential (i.e exergy) of the hot exhaust gases.

$$\dot{E}_{x_{heat_transfer}} = \left(1 - \frac{T_0}{T_s}\right) \times \dot{E}_{heat_transfer} \quad (4-16)$$

where T_0 is the reference temperature of the environment (set here at 300 K) and T_s the wall temperature.

For inlet air and exhaust gas, their exergy is calculated using the following equations:

$$\dot{E}_{x_{mixture}} = \sum \dot{m}_i \times (\varepsilon_i^{therm} + \varepsilon_i^{chem}) \quad (4-17)$$

where $\dot{m}_i$ represents the mass flow of each species, and ε_i^{therm} and ε_i^{chem} are respectively the thermomechanical and the chemical specific exergy.

$$\varepsilon_i^{therm} = (h_i - h_{oi}) - T_0 \times (s_i - s_{oi}) \quad (4-18)$$

$$\varepsilon_i^{chem} = \frac{R}{M_i} \times T_0 \times \ln \left(\frac{y_i}{y_{oi}} \right) \quad (4-19)$$

$$h_i - h_{oi} = C_{p_i}(T_i) \times T_i - C_{p_{io}}(T_o) \times T_o \quad (4-20)$$

$$s_i - s_{oi} = C_{p_i}(T_i) \times \ln \left(\frac{T_i}{T_o} \right) - R \times \ln \left(\frac{P_i}{P_o} \right) \quad (4-21)$$

where the subscript “i” refers to the species under consideration in the mixture and “oi” in the reference state, i.e. the environment. h refers to the specific enthalpy and s to the specific entropy, R is the ideal gas constant (8.314 J/(K.mol)), M the molar mass, y the molar fraction, C_p the constant pressure heat capacity and P_0 the reference pressure of the environment (set here at 1 bar).

For the calculations concerning the intake air and exhaust gas, data needs to be measured at specific measuring points. The measuring points considered for both the 2014+ and 2026 engine models are presented in Table 4-6. For the exhaust, as the pipes of each side of the V never merge, a measuring point is selected on each side and the pressure and temperature used for calculations is the average over both sides. For mass flow and enthalpy, the sum of both sides is considered.

	2014+ (see Figure 4-4)	2026 (see Figure 4-7)
Intake	MP21	MP14
Exhaust	MP19 and MP20	MP15 and MP26

Table 4-6 - Measuring points considered on each engine model for energy and exergy analyses

The results of energy / exergy analysis are presented as bar diagrams in kW or % of fuel energy / exergy. An example is shown in Figure 4-14 where for each simulated engine speed, the repartition of the input fuel exergy between the different considered exergy categories is presented.

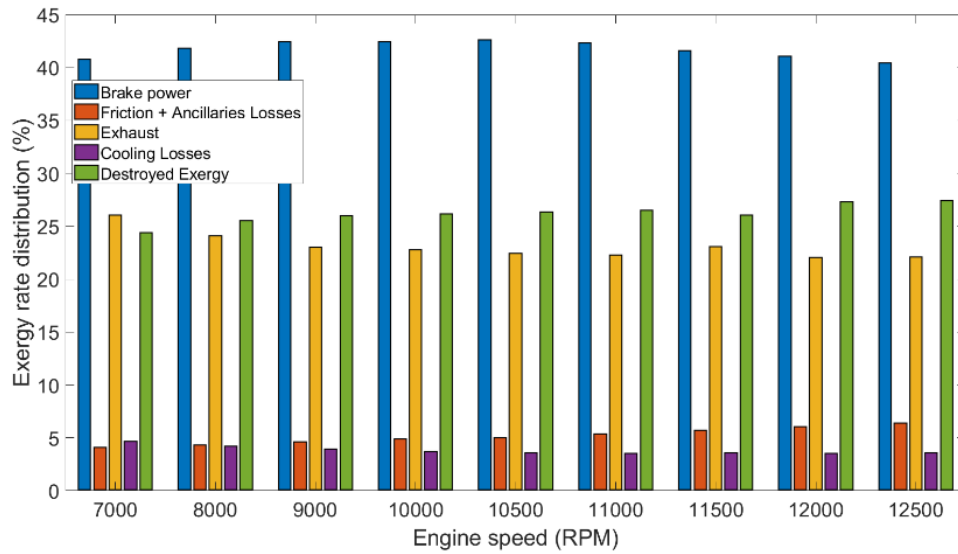


Figure 4-14 - Example of exergy analysis bar diagram

4.7 MATLAB results data extraction tool

To simplify the extraction of data from AVL Boost™ simulation results and use it for different purposes like automatic plotting or external analyses like exergy assessment, a MATLAB code has been developed to directly recover the information from AVL Boost™ .gid result files.

The MATLAB code is given the path leading to the files of interest, the variables that are to be extracted, coefficients to modify, if desired, the unit of the variable of interest and parameters for automatic plotting. The data extraction is then done by calling a specific function which contains the code to open the files as text files, extract the desired data and close the files. An example of how this code is used is shown in Figure 4-15:

```
folder_path = ["C:\Users\309430\OneDrive - Cranfield University\PhD\AVL_Boost\2014_regulations_V6_model\2014_regulations_V6_model\Closed\simulation.dir";
              "C:\Users\309430\OneDrive - Cranfield University\PhD\AVL_Boost\2014_regulations_V6_model\2014_regulations_V6_model\Open\simulation.dir"];

plot_authorization = 0;
legends = ["Closed WP", "Open WP"];
colors = [1,0,0;
          0,1,0;
          0,0,1;
          0,0,0];
font_size = 25; %Graph font size
line_width = 2; %Curve line width
x_label = "Engine speed (RPM)"; %x-axis label
fullscreen = 1;

file_name_engine = ["%E01_1.gid"%sE01_1.gid"%sE01_1.gid"%sE01_1.gid"];
list_engine = ["POWER_PP", "EFFECTIVE-EFFICIENCY_PP", "AIR-EXCESS-RATIO_SHF", "TRAPPED-FUEL-PERS_NB", "MASS-ASPIRATED-PERS_NB", "VOLUMETRIC-EFFICIENCY-INT_O", "PEAKCYLINDERPRESSURE_C", "WALLHEATFLOWERS_B", "MEAN-EFF-ETC_B",
               "POWER_PP", "EFFECTIVE-EFFICIENCY_PP", "AIR-EXCESS-RATIO_SHF", "TRAPPED-FUEL-PERS_NB", "MASS-ASPIRATED-PERS_NB", "VOLUMETRIC-EFFICIENCY-INT_O", "PEAKCYLINDERPRESSURE_C", "WALLHEATFLOWERS_B", "MEAN-EFF-ETC_B"];
multiplier_engine = [1,1,1,3600,3600,1,10^-5,10^-3,1;
                    1,1,1,3600,3600,1,10^-5,10^-3,1;
                    1,1,1,3600,3600,1,10^-5,10^-3,1];
ylabel_graph_engine = ["Engine power (kW)", "Effective efficiency (-)", "Lambda (-)", "Fuel mass flow (kg/h)", "Air mass flow (kg/h)", "Internal volumetric efficiency (-)", "Peak in-cylinder pressure (bar)", ...
                      "Wall heat transfer coefficient (W/(m^2.K))"];
[engine_results, separation_indexes] = graph_Boost(folder_path, file_name_engine, list_engine, multiplier_engine, [], plot_authorization, ylabel_graph_engine, colors, legends, font_size, line_width, 0, x_label, fullscreen, 0, []);
```

Figure 4-15 - Example of AVL Boost data extraction using the dedicated MATLAB code

This tool does not do calculations or bring insights but is presented because it has been extremely useful during this research project, led to a lot of time saving and a significant number of graphics presented in this Thesis have been plotted using it.

4.8 Engine efficiency chain

Across this thesis, different types of engine efficiencies are mentioned. For clarity, the different denominations, and the conversion processes they refer to are summarised in Figure 4-16.

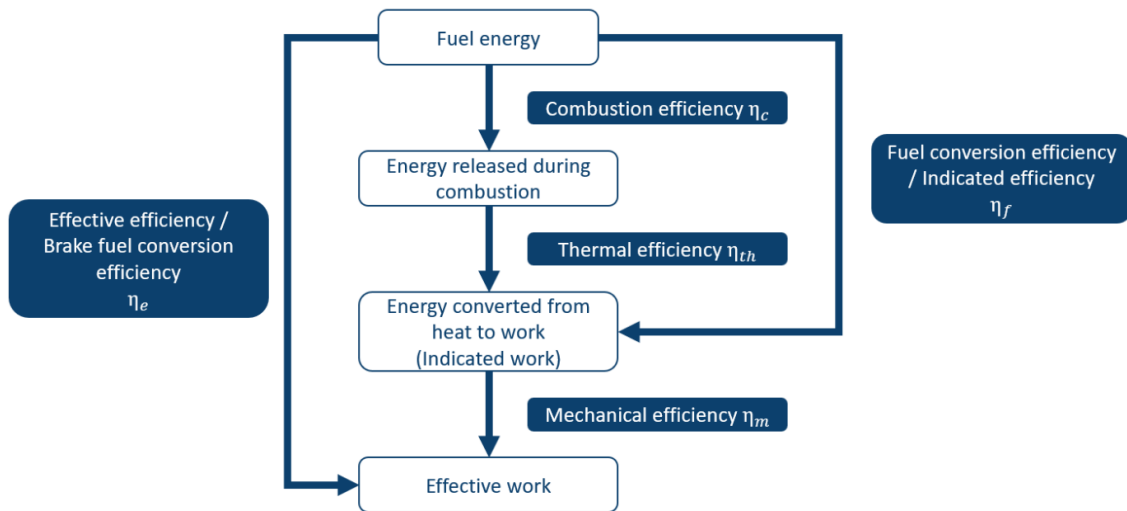


Figure 4-16 - Engine efficiency chain

5 In-depth investigation of the 2014+ Formula One™ engine operation using a Digital Twin

5.1 Introduction

As stated in Chapter 1, prior to 2014, the thermo-fluid-chemical fundamentals in the behaviour of the prevailing Formula One™ engines were predictable using the generalised equations governing internal combustion engines (ICE) which are widely understood by anyone familiar with high-speed engines (including motorcycles). The art of engine design and development was a highly evolving practice experienced by few, but the under-pinning science was more widely shared in the literature on account of it relating to more conventional automotive product.

Even if some aspects of these engines were kept secret by engine manufacturers in the period they were used, most details tend to be shared once a regulation-family is completed or when a manufacturer exits as seen in Chapter 2, for example the publications made by Honda and BMW [1]; [2]; [6]; [7]; [161].

Now, when looking back at the first era of Formula One™ turbocharged engines, their design architecture is easily read by most people with engine knowledge as the principles used for these engines have now been used for decades. Table 5-1 presents a list of the key technologies / techniques featured in that generation of Formula One™ engines and examples of associated experts in the domain:

Technology	Examples of centres of technology / experts
Port injection	Bosch, Denso, Magneti Marelli, TAG, university partners
Highly engineered fuels	Formula One™ petrochemical partners
Rich / stoichiometric combustion	OEMs, AVL, Horiba, University partners
Wastegated turbocharger	Garrett, KKK, Borg Warner, Holset, IHI Watson & Janota [162]
Intercooler technology	BEHR, Mezzo Technologies
Open loop engine control	Magneti Marelli, Bosch, Pi Research, TAG, Ford Electronics

Table 5-1 - Key technologies of the first generation of turbocharged Formula One™ engines (1977 - 1988)

For the 2014+ Formula One™ engine, little knowledge has been made publicly available, especially experimentally validated scientific / fundamental principles. However, through several years of the Cranfield F1™ Design Sprint, it has been possible to identify that key features of the engine required to reach the desired fuel conversion efficiency are high-charge-dilution, high in-cylinder pressure, short combustion duration and specific waste gate operation.

The computerised performance prediction models developed during the Cranfield F1™ Design Sprint never benefitted from data input from the industry (including key items like rate-of-heat-release curve or valve lifts) or from detailed uncertainty investigation as presented in Chapter 3. This discounts them as ‘Digital Twins’ contrary to the model developed for this research project and reported here because they are not sufficiently precise.

The aim of this chapter, building on the 2014+ Digital Twin presented in Chapter 3 and Chapter 4, is not about engine design but is about understanding the science behind the choices made by engine designers for this unique high-performance fuel / energy-limited Power Unit. The research has been to scientifically investigate the impact of each feature of the engine on the very high efficiency that is reached and to push the understanding further by carrying out analyses only possible using a numerical model / Digital Twin of the engine and/or which are not a priority for the industry when developing the engine.

When the 2014 regulations were published, having a detailed and definite opinion of how the engine would operate within this new Power Unit was difficult to imagine, even for engine specialists. Although it was understood that high fuel conversion efficiency would be key because of the limited fuel mass flow, the potential in terms of complex energy management, the level of efficiency that could be reached and the associated specificities of the combustion process were not easily predicted [163]; [164]; [165]. This chapter has the objective of clearly presenting a review of the key features of the 2014+ engine so that the reader can form a structured scientific understanding of the technologies that enable the high levels of efficiency and of the challenges and solutions to make the most out of a limited energy source.

Like Table 5-1, Table 5-2 represents a list of the key features of the 2014+ Formula One™ engines alongside a list of examples of specialists for each technology / technique. In contrast to Table 5-1, even if the technologies are referenced by the researchers or companies listed, they are not mainstream yet and certainly not combined in a unique Power Unit, especially for a high-performance application, as it is the case for the 2014+ Formula One™ Power Unit, showing that this set of regulations brought something new to the industry.

Technology	Examples of centres of technology / experts
High pressure direct injection	Bosch, Magneti Marelli, AVL
High octane number gasoline	Fuel suppliers
High geometric compression ratio	Mazda Zhaolei Zheng, Xudong Zhen, Zhi Wang
Passive pre-chamber / Lean combustion	Mahle Cédric Libert, Michael P. Bunce (Mahle), Ricardo Novella, Ponnya Hlaing
MGU-H / Electrically assisted turbocharger	Garrett

Technology	Examples of centres of technology / experts
MGU-H / Electrically assisted turbocharger (Follow-up)	Hui Xie, Kang Song
Waste gate on a turbine with full swallowing capacity	Garrett, KKK, Borg Warner
MGU-K as a torque filling device	OEMs (hybrid vehicles), Helix
MGU-K as a torque absorber device	OEMs (hybrid vehicles), Helix
Sophisticated intercooler	Reaction engine, PWR
Complex power unit energy management	-

Table 5-2 - Key technologies of the second generation of turbocharged Formula One™ engines

The following section of this chapter will present how these different technologies interact within the 2014+ engine, how the fuel energy conversion is maximised and what are the consequences on the engine operation by using the 2014+ Digital Twin and pushing AVL Boost™ quasi-steady state simulations to their limits.

5.2 Path to 50+% brake efficiency: how does the 2014+ Formula One™ engine reach very-high fuel conversion efficiency?

This section presents the impact of the different features of the 2014+ Formula One™ engine on effective (fuel conversion) efficiency and explains the rationale behind the adoption of each. To demonstrate the impact of each individual solution, different engine models based on the 2014+ Digital Twin have been created.

The ultimate model: the 2014+ Digital Twin

The ultimate model has all the features of the 2014+ Formula One™ engine that are known and identified during this Doctoral research project. It is given the name 'Digital Twin' as it is the best equivalent possible to the real engine considering the available data and its input dataset is the one presented and investigated in Chapter 3 and Appendix C.

Only the model possessing all the components of the 2014+ Formula One™ engine discussed in this thesis is qualified as a Digital Twin. If one component or more are missing, the model does not refer to the 2014+ Digital Twin anymore.

The Baseline model

The baseline model reported here is set up to be in the style of the first era of turbocharged Formula One™ engine: it has the same base-engine (plus intake and exhaust) geometry as the 2014+ Digital Twin (optimised for the 2014+ Digital Twin) but all the special features of the 2014+ Formula One™ engine are removed.

The baseline model does not have an MGU-H, has a 10:1 compression ratio, operates at $\lambda = 1$ with conventional (symmetrical) valve-lift profiles, has a 50 °CA combustion duration, is port injected and has conventional waste gate control (where the waste gate is used to balance compressor requirement and turbine power).

As the chapter unfolds, new elements will be progressively added to this baseline model to show the modifications brought by each design innovation (hyper dilution, short combustion duration, etc) until the 2014+ Digital Twin is completely “built back up”. The average effective efficiency of this baseline engine (34.9%) is that of a good but conventional spark-ignited engine.

Intermediate models

In this chapter, the baseline model is progressively improved by the addition of one performance design feature after another until the 2014+ Digital Twin model is “recreated” in the final step.

The order in which the performance design features are introduced has been chosen to be rational, to offer a clear and logical exposition of the enabling technologies of this 2014+ Formula One™ engine but other authors could have chosen a different order. This technique of starting from a baseline engine until the 2014+ Digital Twin is reached is used to give the reader insight of the different performance features.

Only nominated performance design features have been modified in this process. The geometry of the models does not get changed to re-maximise volumetric efficiency at any step in the process and the turbine swallowing capacity has not been modified to suit the exact operating conditions of each engine model. All the intermediate models feature the same geometrical data optimised for the 2014+ Digital Twin and presented in Chapter 4. The names associated with each intermediate model identify the latest additional performance feature implemented in the model compared to the previous one.

Table 5-3 presents the different engine models investigated during this chapter, the name they are referred to and that will appear in the graph’s legends, and if they possess the features of the 2014+ Digital Twin or not (Green cell: same settings as the 2014+ Digital Twin / Red cell: same setting as the baseline model):

Category	Model name	16:1 CR	High dilution	41 °CA combustion duration	Direct injection	Asymmetric valve profiles	MGU-H	Fully open waste gates
Baseline	Baseline	×	×	×	×	×	×	×
Intermediate models	Increased CR	✓	×	×	×	×	×	×
	High dilution	✓	✓	×	×	×	×	×
	Short combustion duration	✓	✓	✓	×	×	×	×
	GDI	✓	✓	✓	✓	×	×	×
	Asymmetric valve lift profiles	✓	✓	✓	✓	✓	×	×
2014+ Digital Twin	MGU-H	✓	✓	✓	✓	✓	✓	×
	OWG	✓	✓	✓	✓	✓	✓	✓

Table 5-3 - Summary of the properties of each engine model investigated for the 2014+ Formula One™ engine breakdown

The last two investigated engine models (referred to as MGU-H and OWG) both correspond to the 2014+ Digital Twin. They have all the special features of the 2014+ Formula One™ engine but they represent two different engine operating modes (CWG and OWG modes) that will be detailed later.

When referring to the 2014+ Digital Twin, the reader must only consider these two engine modes as shown in Table 5-3 and not any of the baseline or intermediate models.

2014+ performance study

As emphasised throughout this thesis, the operation of the 2014+ Formula One™ engine is significantly impacted by the fuel flow limit imposed by the regulations and which is shown in Figure 5-1.

Because the fuel flow is limited, maximising brake fuel conversion efficiency η_e is essential as suggested by Equation 5-1 and is the main focus for the design of the 2014+ Formula One™ engine and maximum power output.

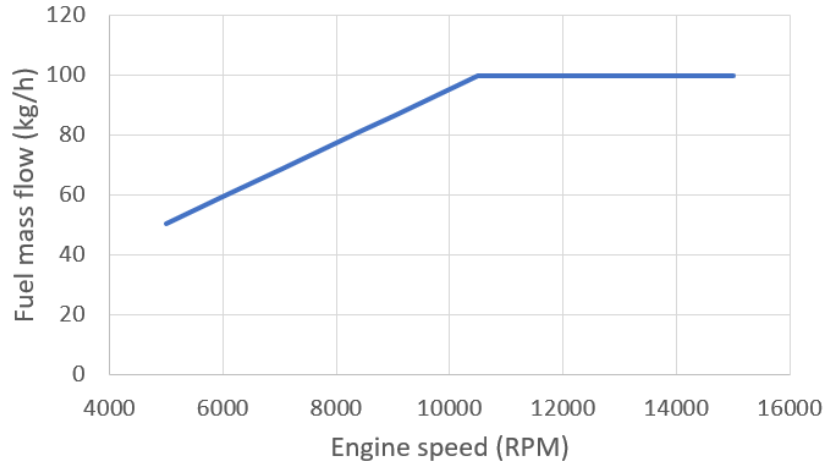


Figure 5-1 - 2014+ Formula One™ regulations fuel mass flow limit

$$P_b = \eta_e \dot{m}_f Q_{lvh} \quad (5-1)$$

In addition, because of the fuel flow being limited, as explained in Chapter 3, the shape of the power curve will be broadly that of the fuel flow rate assuming that the speed dependence on effective efficiency is weak compared to the speed dependency of fuel flow rate. By considering a range of viable effective efficiencies, Equation 5-1 and the fuel's lower heating value Q_{lvh} (here 44 MJ/kg), it is possible to forecast the performance envelope for the power curve, as shown in Figure 5-2 using two effective efficiencies considered as potential extremums for the 2014+ Formula One™ engine:

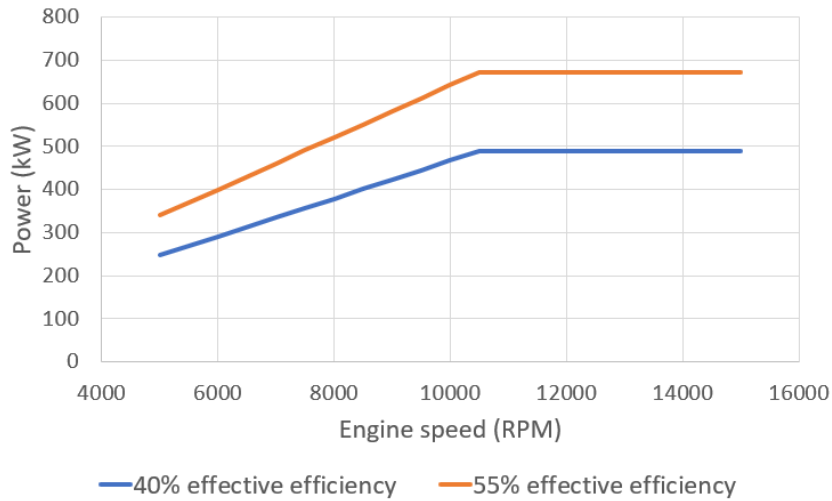


Figure 5-2 - Expected window for the engine power curve

As seen in Chapter 4, the effective efficiency in Equation 5-1 can be developed into:

$$P_b = \eta_m \eta_{th} \eta_c \dot{m}_f Q_{lvh} \quad (5-2)$$

Q_{lvh} depends on the fuel properties and $\dot{m}_f$ is limited by the regulations meaning that to increase power output, the efficiency of each step of the engine cycle must be maximised.

In the next sections of this chapter, the impact of the addition of each performance feature that combine to form the 2014+ Digital Twin is investigated. For each section, the model with the new feature is introduced and compared to that of the previous section following the order shown in Table 5-3. For example, first, the engine model with the increased compression ratio (referred to as Increased CR) is compared to the baseline model. Then, the following model, which operates at high dilution and also features the high compression ratio (High dilution model), is compared to the increased CR model and not the baseline one. By following this methodology, the impact of each solution can be investigated independently of the others.

5.2.1 High compression ratio

One well-known way to increase thermal efficiency η_{th} , i.e. the conversion of combustion heat into work, is to increase the engine's geometrical compression ratio. As discussed in Chapter 3 when looking at idealised engine cycles, a high compression ratio leads to a higher expansion ratio, higher pressures and increases the temperature difference between the beginning and the end of the expansion stroke leading to more work extracted from the hot exhaust gases as demonstrated by the diagrams in Figure 5-3.

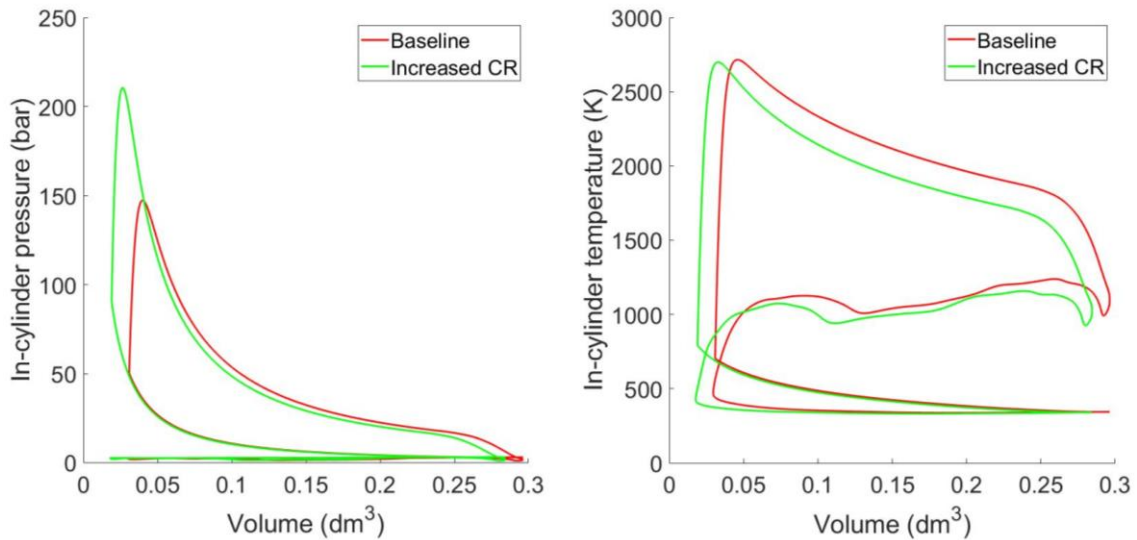


Figure 5-3 - PV and TV diagrams at 10,500 RPM - Baseline engine (Red curve: CR = 10:1) and increased compression ratio (Green curve: CR = 16:1) models

Using the baseline engine model created for this chapter which features a 10:1 compression ratio and modifying it to that of the 2014+ Digital Twin (16:1), both models are still operating at stoichiometry, allows to demonstrate the benefits of a higher compression ratio as shown in Figure 5-4 which represents the average

value over the simulated engine speed range (7,000 - 12,500 RPM) for twelve parameters and for both cases of interest.

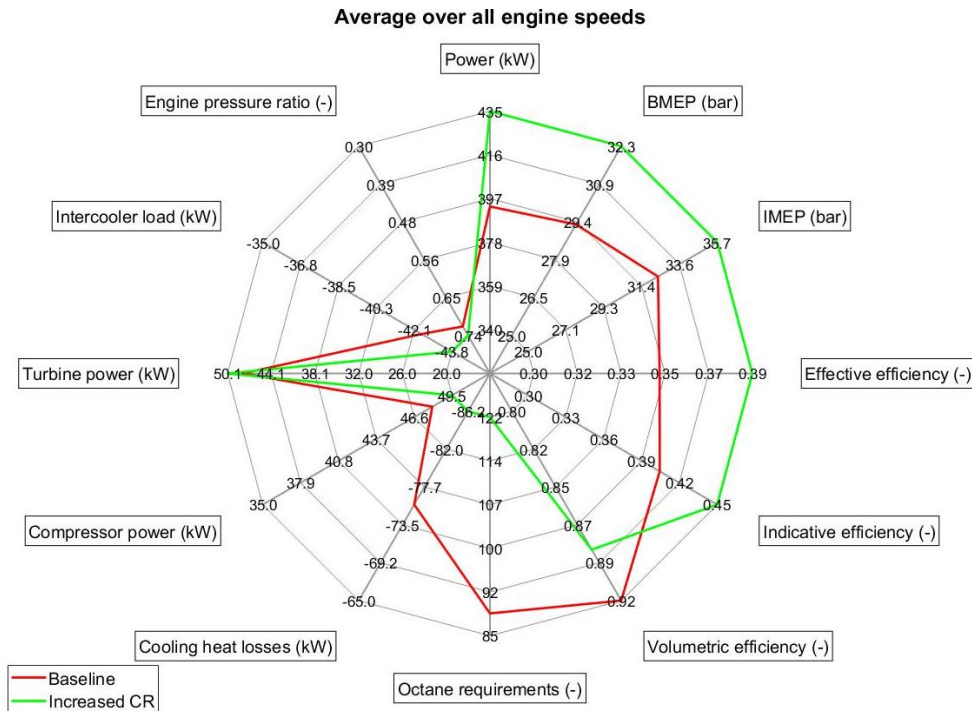


Figure 5-4 - Performance improvements brought by a higher compression ratio (Red curve: 10:1 CR / Green curve: 16:1 CR)

Every time volumetric efficiency is mentioned in this chapter, it refers to internal volumetric efficiency, i.e. volumetric efficiency for which the pressure and temperature references are taken at the plenum inlet (MP21) as presented in Chapter 4, rather than using atmospheric conditions.

The label “Engine pressure ratio” refers to the pressure ratio across the engine which is calculated by dividing the exhaust pressure by the inlet pressure (cycle average values). The aim is for it to be as low as possible, i.e. the inlet pressure is higher than that of the exhaust, as it facilitates engine breathing. In the models presented here, the intake pressure is measured at the reference point for the internal volumetric efficiency (at the plenum inlet, MP21) and the exhaust pressure is the average of the pressures in the secondary pipes on both sides of the engine (MP19 and MP20 in Figure 4-4 in Chapter 4), after the exhaust primaries have merged.

As shown by Figure 5-4, for which a value on the outside of the graphic is an improvement, the performance increase due to a higher compression ratio is significant. The indicated and effective efficiencies are respectively increased by 4.49 and 3.62 percentage points; this a massive increase.

But this efficiency increase comes with a price, the biggest consequence being the higher octane number requirements. The benefits from a higher compression ratio comes from the higher in-cylinder temperatures and pressures but these are also the causes of higher risks of knock. Equation 5-3 shows the integral

developed by Livengood et al. [166] to predict auto-ignition based on an ignition time model τ_i . Auto-ignition occurs when the integral reaches 1. Equation 5-4 shows the ignition time model proposed by Douaud [167] while Equation 5-5 is the empirical function used by AVL Boost™ which is derived from the two previous equations to predict the fuel's octane number requirements to cope with the simulated conditions.

$$\int_{t_{start}}^{t_{ignition}} \frac{1}{\tau_{ignition}} = 1 \quad (5-3)$$

$$\tau_{ignition}(p, T_u) = 0.01768 \times \left(\frac{ON}{100}\right)^{3.402} p^{-1.7} e^{\frac{3800}{T_u}} \quad (5-4)$$

$$ON = 100 * \left(\frac{1}{0.01768} \times \int_{t_{IVC}}^{t_{85\% MFB}} \left[\left(\frac{p}{p_{ref}}\right)^{1.7} \times \exp\left(-\frac{3800}{T_u}\right) \right] dt \right)^{\frac{1}{3.402}} \quad (5-5)$$

where t_{start} is the initial time of integration, $t_{ignition}$ is the time when auto-ignition occurs, ON is the fuel's octane number, p is the in-cylinder pressure, p_{ref} is the atmospheric pressure, T_u the temperature of the unburned zone in the two-zone combustion model, t_{IVC} the closing time of the intake valve and $t_{85\% MFB}$ the time at which 85% of the charge is burned.

These models rely on two-zone combustion models as the temperature of the unburned zone is required for the calculations. Such models allow to have an estimate of the knock sensitivity of an engine but cannot appreciate the complex temperature distribution of a real-world application which is very important for abnormal combustion phenomenon. Investigating auto-ignition delays could have been possible using Ansys Chemkin if it had worked for this project.

This equation shows that the higher the temperatures and pressures, the higher the results of the integral in Equation 5-5 and thus the higher the predicted necessary octane number to sustain the operating conditions.

Nevertheless, as discussed in Section 5.2.3, the coefficients used in Equation 5-5 are not appropriate for the 2014+ Formula One™ engine, as the predictions it produces are obviously too high. The pressure exponent (here set at 1.7) must be modified for the equation to give relevant predictions. But the principles behind the increase in octane number requirements with compression ratio described here are unchanged.

Because of the higher compression ratio, the clearance volume of the combustion chamber is reduced making harder to trap as much air as required for the combustion, i.e. reducing the volumetric efficiency. This is why more work is required from the compressor despite still operating at stoichiometry, which also increases the intercooler load.

Despite the similar in-cylinder temperatures, cooling losses increase with higher compression ratio. This is a consequence of the in-cylinder pressure dependency of the heat transfer coefficient in Woschini model, see Equation 5-6 (Equation already shown in Chapter 3 as Equation 3-7).

$$h_c = 3.2 \times B^{-0.2} \times p^{0.8} \times T^{-0.55} \times w^{0.8} \quad (5-6)$$

Higher in-cylinder pressures lead to higher heat transfer coefficient. Because the temperature delta between the mixture and combustion chamber walls and the heat transfer surface area are similar in both engine models but heat transfer coefficients are higher, it results in higher cooling losses for the increased CR case. It is assumed that the cooling systems will be adapted to keep cylinder wall and head surface temperatures consistent throughout the different cases.

As mentioned in Chapter 3, the 2014+ Formula One™ Technical regulations allows compression ratios up to 18:1 [8]; [155] but lower compression ratios are used [5] which, at first sight, seems counter-intuitive considering the performance benefits brought by higher compression ratios.

However, the choice of the compression ratio is a question of compromise. Indeed, it increases thermal efficiency but requires higher anti-knock properties from the fuel and results in higher stress for the engine. Thus, the 2014+ Formula One™ engine, which is already operating at the limit of knock [5], features a higher compression ratio than conventional gasoline engines but does not reach the 18:1 limit as the benefits from an even higher compression ratio must be overpowered by its drawbacks and requirements to operate reliably.

5.2.2 High dilution

The next very important objective for this fuel-limited engine is to ensure that all the fuel energy is released during the combustion, that means maximising combustion efficiency η_c . As discussed in Chapter 3, this is done by running the engine at least at stoichiometric ratio, i.e. ensuring that enough air molecules are trapped in the combustion chamber to oxidise all the fuel molecules.

The baseline and increased CR models used in the previous section are already operating at stoichiometric conditions, ensuring that no fuel is left unburned which would be detrimental for efficiency as this fraction of the injected fuel would be wasted.

However, by further increasing dilution (higher air/fuel ratio than stoichiometry) to match the evolution implemented in the 2014+ Digital Twin (Figure 5-5) thermal efficiency can be further improved. High air/fuel ratios bring two advantages which are higher in-cylinder pressures and lower temperatures as shown in Figure 5-6 where the high dilution model operates at the same air/fuel ratios as the 2014+ Digital Twin.

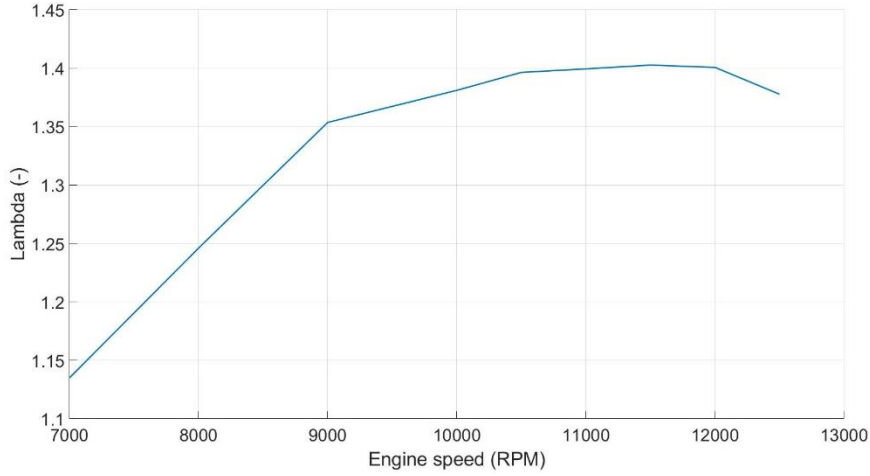


Figure 5-5 - Lambda evolution implemented in the 2014+ Digital Twin

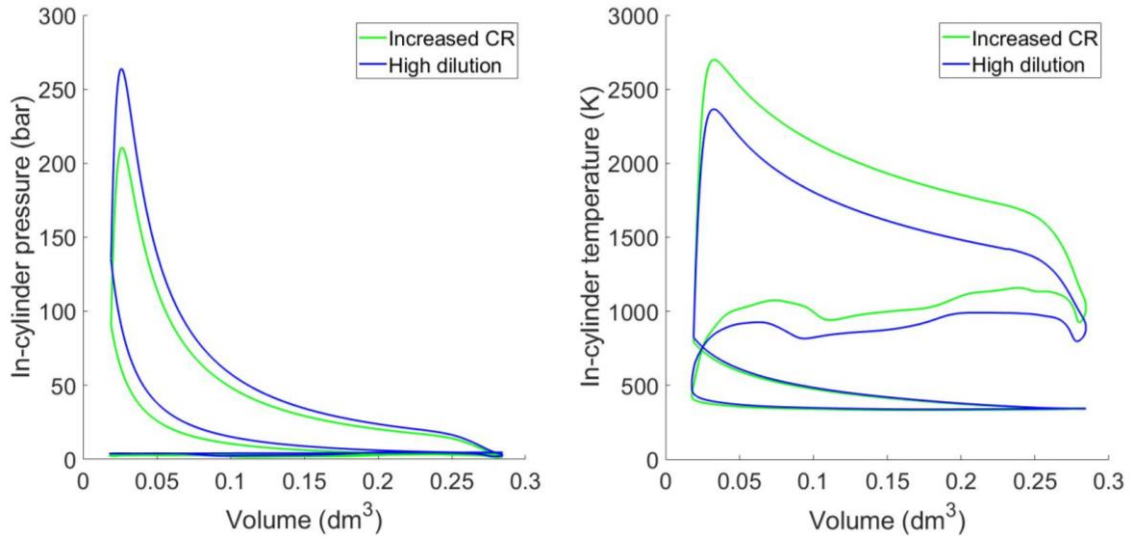


Figure 5-6 - PV and TV diagrams at 10,500 RPM – Increased compression ratio (Green curve: $\lambda = 1$ for all engine speeds) and high dilution (Blue curve: Lambda evolution of the 2014+ Digital Twin) models

As also mentioned in Chapter 3, the benefits of high dilution level are predictable by considering Otto's cycle thermal efficiency equation (already shown as Equations 2-1 and 3-4):

$$\eta_{th} = 1 - \frac{1}{CR^{\gamma-1}} \quad (5-7)$$

By increasing air/fuel ratios, the ratio of specific heat capacity of the mixture γ is raised leading to higher thermal efficiency.

$$W = \int p dV \quad (5-8)$$

By looking at the PV diagram from Figure 5-6, plotted using the simulation results from both models at 10,500 RPM, it can be noticed in the higher dilution case that because more air is introduced in the cylinder, the total trapped mass is increased, thus for the same evolution in volume, the pressure increases. This

results in higher compression load required from the engine (see Equation 5-8). However, this is an investment. In the expansion stroke, the reflection is reversed. Because the piston is moving downwards, dV is positive and the higher in-cylinder pressures leads to more work produced for the same volume evolution. The compression stroke requires 146 J/cycle more in the high dilution case but results in additional 247 J/cycle produced.

Thanks to the lower in-cylinder temperatures enabled by the high level of dilution, the delta in temperature between the mixture and the combustion chamber wall is reduced, reducing in consequence the cooling losses as shown in Figure 5-7.

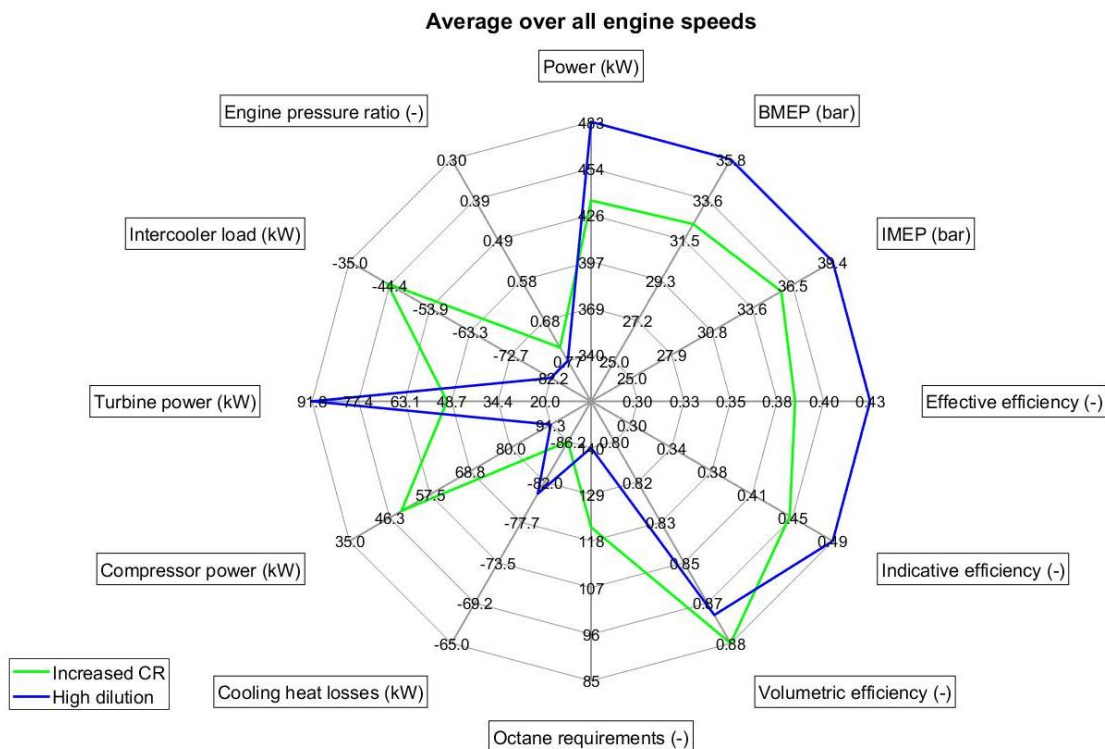


Figure 5-7 - Performance improvements brought by high dilution (Green curve: $\lambda = 1$ for all engine speeds / Blue curve: Lambda evolution of the 2014+ Digital Twin)

On average, the indicated and effective efficiencies are increased by 4.01 and 4.07 percentage points thanks to the higher air/fuel ratios. The effective efficiency increases more than the indicated one as they are not calculated on the same basis. The indicated efficiency, in addition to not considering mechanical losses, uses the amount of fuel trapped in the combustion chamber as a reference whereas the effective one is referenced to the amount of fuel injected.

However, as discussed in Section 5.2.4, fuelling control with port-injection is relatively poor meaning that all the injected fuel does not end up trapped in the cylinder, thus here, the indicated and effective efficiencies are not calculated on the same basis explaining why the effective efficiency increases more than the indicated.

When considering Equation 5-9 which is one of the governing equations for engine performance predictions (Equation 4-6 in Chapter 4), it can appear at first

sight that increasing air/fuel ratios is not the right solution because of the presence of (F/A) at the numerator of the equation:

$$P_I = \frac{\eta_{th}\eta_c\eta_v NV_{swept} Q_{lvh} \rho_a (F/A)}{2} \quad (5-9)$$

Nevertheless, as stated previously, this is the multiplication of all these factors that must be increased. In this studied case, the fuel/air ratio has relatively been decreased by 25% on average, however, this drop is compensated by relative increases in inlet air density ρ_a and indicated efficiency $\eta_{th} \times \eta_c$ by respectively 37.2 and 8.96%, leading to a higher power output.

Referring to the second law of thermodynamics, it can be readily seen that increasing air/fuel ratio does not appear to be an effective solution. The entropy variation for a transformation state 1 to state 2 can be described for an ideal gas and by considering constant specific heat capacity by the following equation [46]:

$$S_2 - S_1 = m[C_p \ln\left(\frac{T_2}{T_1}\right) - r \ln\left(\frac{P_2}{P_1}\right)] \quad (5-10)$$

Using the PV and TV diagrams shown in Figure 5-6 and Equation 5-10, it is possible to estimate the entropy creation during the combustion process of the increased CR and high dilution engine models.

Considering the theoretical exhaust composition of both cases at their respective air/fuel ratio, their average combustion temperature, their constant pressure heat capacities are estimated at 1,457 J/kg/K for the increased CR case and 1,389 J/kg/K for the high dilution one and their respective specific gas constant r is nearly identical because of very similar exhaust gas molar masses (~289.8 J/kg/K). As air/fuel ratio is different between both cases, the masses trapped in the cylinder are too. The increased CR case operates with 0.824 gram of mixture against 1.156 grams for the high dilution one.

Using Equation 5-10, it is estimated that the higher dilution case at 10,500 RPM causes the creation of 15.6 % more entropy (1.299 vs 1.501 J/K). This observation is consistent with the work done by Dunbar et al. [43] and by Farrell et al. [38]. As demonstrated by Dunbar et al. and as presented in Chapter 2, combustion irreversibility increases with air/fuel ratios because of the increased mixing loss and the fact that the lower combustion temperature results in a heat transfer at low temperature which increases availability destruction.

Nevertheless, considering the complete engine cycle, despite the higher combustion irreversibility, high dilution results in higher pressure and lower temperature which increase work production and lower the proportion of fuel exergy lost in heat cooling losses and exhaust flow as discussed in Section 5.3.

Because a higher air mass flow is necessary to operate at high air/fuel ratios, the compressor power requirement significantly increases, nearly doubling and so is the load on the intercooler to ensure the intake air is at the desired temperature. However, the increased air mass flow also leads to higher exhaust mass flow and thus enthalpy despite the lower combustion temperatures which allows the

turbine to recover enough power for the compressor. But compared to the previous engine model, the breathing of the engine is negatively impacted by the high pressure ratio across the engine which is reflected by the lower volumetric efficiency. Nevertheless, an additional benefit from running at high dilution level and having higher exhaust enthalpy will be presented in Section 5.2.6.

In addition as shown in Figure 5-7, running at higher air/fuel ratios requires improved detonation (knock) resistance highlighting the domination of pressure over temperature in the knock onset model used in AVL Boost™ (Equation 5-5). Even though increased dilution reduces in-cylinder temperature, the exposition to higher pressures requires higher octane numbers to operate safely under these conditions.

Another drawback from high dilution concerns speed and reliability of combustion. As visible in Figure 5-8 which shows the evolution of iso-octane laminar flame speed with equivalence ratio (inverse of lambda), flame speeds reduce very significantly as equivalence ratios are decreased (i.e. as lambda is increased). This means that from the mixture ignition, the flame will require longer to reach the combustion walls and complete the combustion, with higher risks of incomplete combustion .

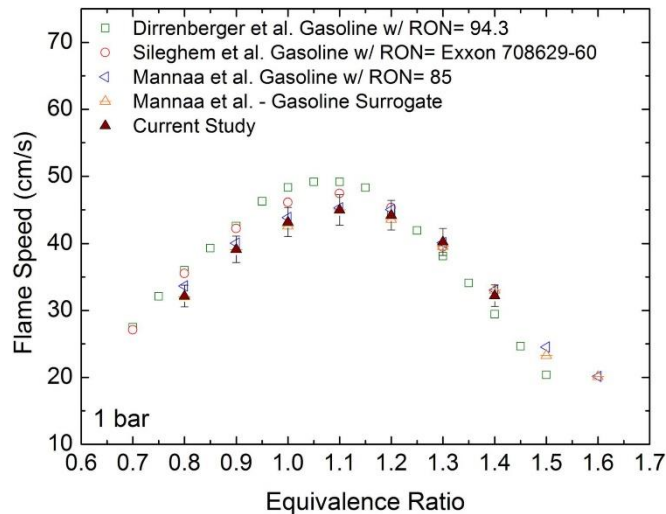


Figure 5-8 - Evolution of gasoline and gasoline flame laminar flame speed with equivalence ratio [49]

However, as explained in Chapter 3, achieving the highest fuel conversion efficiency requires a combustion as short as possible. In addition, lower flame speed makes the combustion less reliable, it increases the risks to quench the propagating flame, thus increasing the IMEP variability between cycles as suggested by Heywood and shown in Figure 5-9 [34].

The combustion duration in the two models compared in this section has been kept constant despite the increase in air/fuel ratios, the slower flame speed is thus not considered. However, as it will be further explained in the next section, high turbulence level helps to increase flame speed and thus reduce combustion duration. In the high dilution model, because more air is introduced during the same period of time and at higher pressure, it can be assumed that the in-cylinder turbulences would be higher, thus increasing the flame speed. Keeping the

combustion duration constant is thus defended as a reasonable approach in the absence of test data.

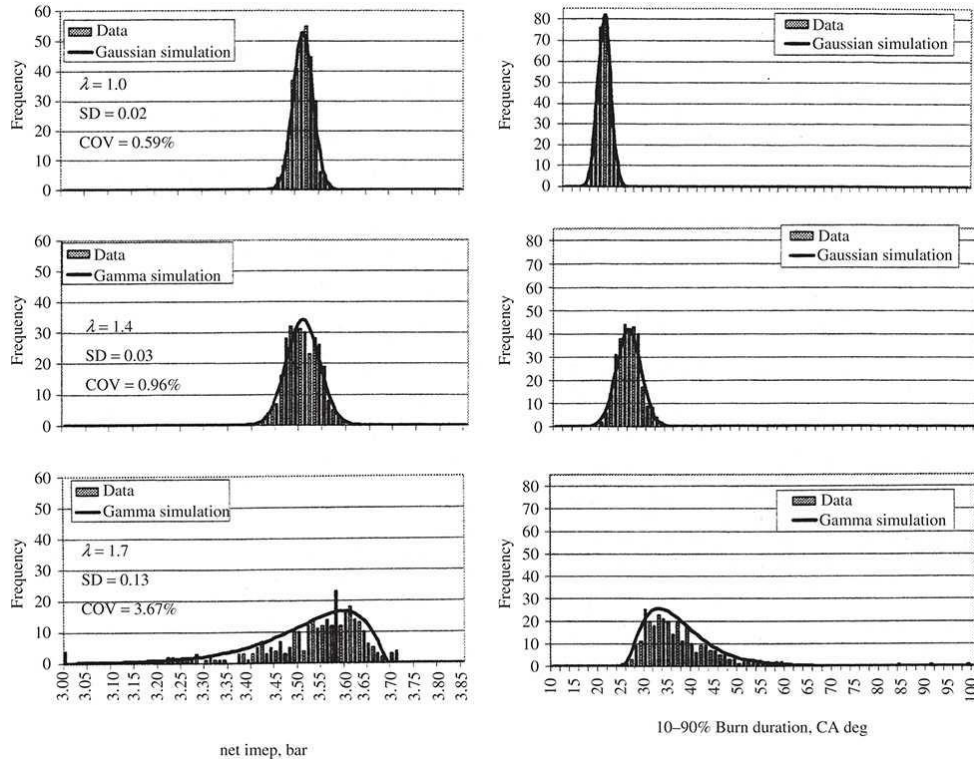


Figure 5-9 - Increase of combustion cyclic variability with dilution [34]

5.2.3 Pre-chamber, improved ignition reliability for lean mixtures and shorter combustion duration

To further improve the engine's thermal efficiency and take full advantages of the benefits brought by high dilution, a solution must be found to increase the flame speed which is low because of the high air/fuel ratios and thus shorten the combustion duration.

As explained by Sodré [168], the relationship between laminar flame speed and turbulent flame speed can be expressed by the following equation:

$$\frac{S_T}{S_L} = 1 + C \left(\frac{\rho_u}{\rho_b} \right)^{\frac{1}{2}} \left(\frac{u'}{S_L} \right) \left[1 - e^{-\frac{R_f}{r_c}} \right] \quad (5-11)$$

where S_T is the turbulence flame speed, S_L the laminar flame speed, C a non-dimensional coefficient relating to engine operating conditions and geometry, ρ_u the density of the unburned mixture, ρ_b the density of the burnt mixture, R_f the flame radius, r_c the parameter of order of turbulence integral scale in meter and u' the turbulence intensity in m/s.

The laminar flame speed represents the velocity of the flame in controlled conditions like theoretical combustion devices but the actual flame speed in the engine combustion chamber is the turbulent one. The difference is that the turbulence in the combustion chamber will wrinkle the flame front, increasing the surface area of contact between the flame front and the unburned charge, thus increasing the rate at which the unburned charge can be burned. Equation 5-11 further highlights the importance of turbulence intensity to increase the difference between laminar and turbulent flame speed. As the laminar flame speed is limited in the 2014+ Formula One™ engine because of the high dilution level, increasing turbulence during the combustion is one solution to cope with this reduction by increasing the turbulent flame speed and thus reduce the combustion duration to get as close as possible to a constant-volume combustion.

Using a pre-chamber which is common in engines running with natural gas is a very effective solution to increase in-cylinder turbulences during the combustion event and thus to increase the turbulent flame speed and reduce combustion duration.

In addition to increased turbulence intensity, the flame jets created by the pre-chamber also provide higher ignition energy for the main combustion chamber charge, each flame jet coming out of the pre-chamber is an ignition point, which creates distributed ignition sites and is important to reliably ignite the highly diluted mixture.

Because only one injector per cylinder is allowed [155], it is allocated in the main combustion chamber as only a small fraction of the fuel is dedicated to the pre-chamber. The pre-chamber featured in Formula One™ engine is thus a passive pre-chamber with no fuel directly sprayed in it.

The performance impact of a reduced combustion duration, allowed by the pre-chamber as an ignition system, is shown in Figure 5-10 and Figure 5-11. The combustion duration is reduced from 50 °CA to that of the 2014+ Digital Twin (41 °CA). The value is kept the same for all engine speeds (see Chapter 3)

Because the same amount of fuel energy is released in a shorter period, the peak in-cylinder pressure and temperature increase. From the PV diagram, this increase in pressure leads to a higher area under the curve in the high-pressure part of the cycle which combined with similar lower pressure parts of the cycle yields higher work extracted during the expansion stroke thus increasing the thermal efficiency.

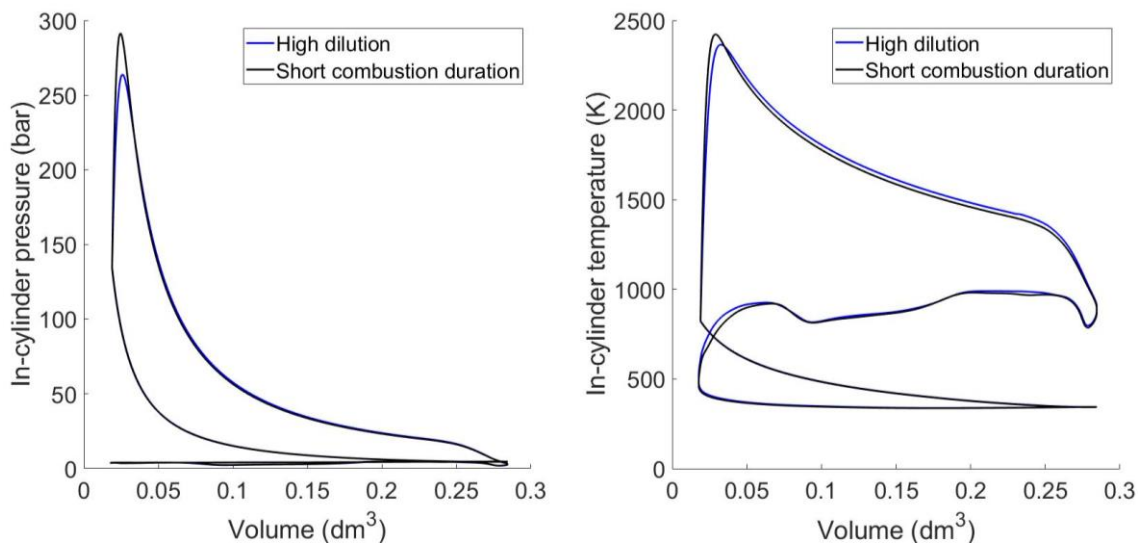


Figure 5-10 - PV and TV diagrams at 10,500 RPM - High dilution (Blue curve: 50 °CA combustion duration) and short combustion duration (Black curve: 2014+ Digital Twin combustion duration) models

On average, the reduction in combustion duration results in 0.58 and 0.57 percentage point improvements in terms of indicated and effective efficiencies respectively. These increases are small compared to the values presented in the previous sections, but it must be kept in mind that the pre-chamber enables the combustion duration to be shorter but also that it makes the ignition of the lean mixture reliable.

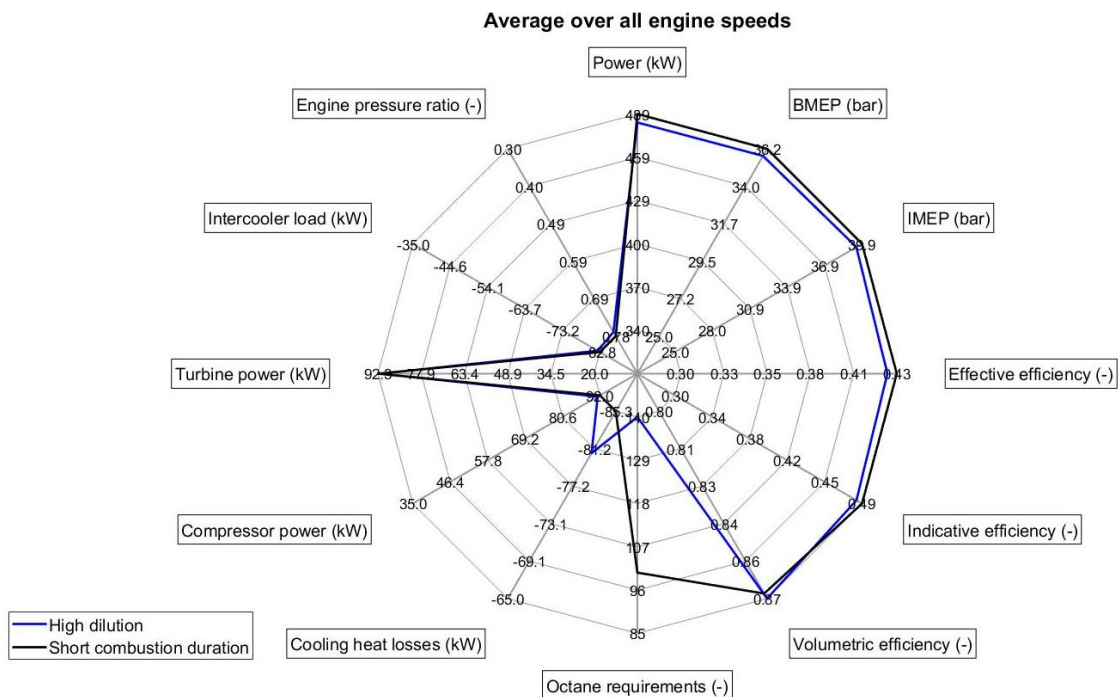


Figure 5-11 - Performance improvements brought by short combustion duration (Blue curve: 50 °CA combustion duration / Black curve: 2014+ Digital Twin combustion duration (41 °CA))

Without a pre-chamber, it would not be possible to operate at $\lambda = 1.4$ which, as seen in Section 5.2.2, brings a significant improvement in terms of efficiency. The

net impact of the pre-chamber use is thus greater than what is only due to shorter combustion duration.

Figure 5-10 shows the increase in peak in-cylinder pressure due to the shorter combustion duration and it gets very close the 300 bar which is considered the mechanical limit of what is possible in terms of mechanical reliability [5]. Further reducing the combustion duration could be investigated but would result in the 300 bar limit being exceeded which is deemed not practical for engine manufacturers from a reliability point of view.

Because of the higher in-cylinder temperatures during combustion, it can be seen in Figure 5-11 that cooling losses have increased by 5.6% but all the other performance parameters have similar values except for the octane number requirements which has significantly reduced with shorter combustion duration. This is due to the change in pressure exponent for the knock onset model presented below (Equation 5-5).

As seen in previous sections and seen in Figure 5-4 and Figure 5-7, running at high compression ratio and high dilution result in very high requirement on fuel's octane number to operate under these pressures and temperatures without knocking. In the 2014+ Digital Twin, if the Douaud [167] equation coefficients are used, a peak of 160 octane number is reached. However, it is known that the 2014+ Formula One™ engines are running a fuel with a RON number much closer to the 102 of conventional high-performance gasoline [5].

This highlights that Douaud's model with its default coefficients as used in AVL Boost™ is not appropriate to the 2014+ Formula One™ engine operation. The Douaud equation was developed using a Cooperative Fuel Research (CFR) engine which is closer to a conventional (albeit low speed) spark-ignited engine than to the high-performance engine under study in this research project. The model must be adapted to give more sensible predictions, more in line with the experience.

In addition to Douaud's model, the equations developed by Wayne [169], Hoepke [170], Chen [171] and Soylu [172] for knock onset prediction under different conditions have been investigated. To better understand the knock onset models, a sensitivity analysis was conducted on the coefficients of the formulae. Using simulation results from the 2014+ Digital Twin, each coefficient was varied by +/-10% and their influence on the auto-ignition delay (crank-angle at which the model predicts auto-ignition) is ranked. The results of the sensitivity analysis are shown in Figure 5-12:

	Douaud	Wayne	Hoepke	Chen	Soylu
Pressure exponent	1	1	2	1	2
Exponential coefficient	2	2	1	2	1
Constant C1	3	3	3	3	4
Additionnal parameter (octane number, lambda, x EGR, x propane)	4	4	4	4	3

Figure 5-12 - Sensitivity analysis results on knock onset models

The numbers in Figure 5-12 relate to the influence of each parameter on the auto-ignition delay predictions, 1 being the highest and 4 the lowest. Two parameters are more important than the others, namely the pressure exponent and the exponential coefficient (which brings temperature in consideration) which are either the most or second most influential parameters in the investigated models.

In all equations, the exponential contains the unburned zone temperature which means that the most influential parameters are those related to pressure and temperature which are known to be the main drivers when considering knock.

Regarding knock, the integral in Equation 5-5 shows that the longer the charge is exposed to higher pressures and temperatures, the more likely it is to auto-ignite, so time must have an influence on the models studied above. Time is not directly showed in the investigated auto-ignition delay formulae in Figure 5-12 but as all models are based on an integral over time, that is how this parameter is considered and it is a consistent and important factor in determining knock resistance.

By calculating auto-ignition timing for different engine speeds, i.e. different cycle duration, using the same pressure and unburned zone temperatures traces for all of them, it is possible to assess the influence that time has on the auto-ignition prediction. Using the in-cylinder pressure and unburned zone temperature results of the 2014+ Digital Twin at 10,500 RPM with Douaud's equation, the auto-ignition timing was calculated for different engine speeds. At 8,000 RPM, auto-ignition is predicted at 2 °CA ATDC, at 5 °CA ATDC for 10,500 RPM and at 8 °CA ATDC for 15,000 RPM.

This is in accordance with experience, the higher the engine speed, the later auto-ignition might happen, the knock onset model is thus relevant on this matter. Knock resistance is lower at lower engine speeds as one engine cycle takes longer to complete thus the charge is exposed to high pressure and temperature for longer. One of the engineering solutions to cope with this is to delay the ignition advance at lower engine speeds to reduce combustion temperatures and pressures and hence reduce the probability of knock. But this also reduces fuel conversion efficiency. Time is one parameter to consider for knock onset prediction but is not the reason for the unrealistic higher prediction made by Douaud's model for the 2014+ Digital Twin.

As seen with the sensitivity analysis, the pressure exponent and exponential coefficient can affect the results significantly and all the models studied are based on (some historic) datasets measured on engines very different from the current Formula One™ engine. Between a conventional engine and the 2014+ Digital Twin, the in-cylinder temperature evolution difference is less significant than that of the pressure. Considering iso-octane, its adiabatic flame temperature between $\lambda = 0.9$ and $\lambda = 1.4$ reduces by 12%, where as in-cylinder pressure in the 2014+ Formula One™ engine are three to four times higher than that of the conventional gasoline engine. These observations demonstrate that it is the 1.7 value used in Douaud's model for the pressure exponent which is most likely not to be well-adapted to the 2014+ Digital Twin operating conditions.

To select a new relevant value for the pressure exponent of Douaud's knock onset equation, the 2014+ Digital Twin has been run at different start of combustion and peak air/fuel ratio settings within a relevant window ([-10, -5, -3, -1, 5] °CA ATDC and [1.3, 1.35, 1.4, 1.45, 1.5] peak lambda) representing sensible operating conditions for the engine. For each combination of settings, Douaud's equation is used in combination of a dichotomy algorithm to determine the necessary value of the pressure exponent to reach an octane number requirement prediction matching the data provided by the industry.

The average pressure exponent value calculated over all cases is 1.485 which represents a 12.6% reduction compared to the conventional 1.7 used in Douaud's model. This new value is used for the 2014+ Digital Twin and for the intermediate models starting from the 'Short combustion duration' one and explains the significant reduction in octane number requirement observed in Figure 5-11. Figure 5-13 shows how impactful the change in pressure exponent is on the octane number prediction for the 2014+ Digital Twin and that the 1.485 value results are now relevant.

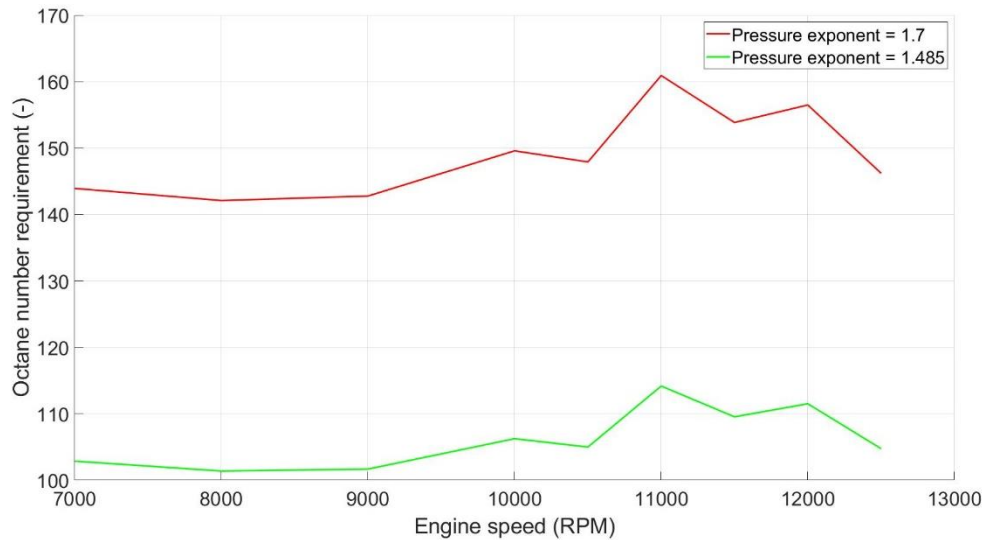


Figure 5-13 - Impact of the modification of the pressure exponent in Douaud's knock onset model on the octane number requirement predictions

The fact that the knock onset model had to be modified to give relevant results shows that because the 2014+ Formula One™ engine features an advanced ignition system which provides multiple points of ignition (pre-chamber) and a specifically developed fuel, its knock resistivity is increased, and the engine can run unusual operating conditions leading to high performance and efficiency. This fact is congruent with the observations made in the literature review (Chapter 2) regarding the advantages of pre-chamber ignition systems.

By allowing the engine to run lean mixtures, to achieve short combustion duration and reducing its knock sensitivity (thus enabling the use of high compression ratio), the pre-chamber is responsible for an overall 4.64 percentage points increase in effective efficiency.

5.2.4 Direct injection

The last section demonstrated how important the pre-chamber is for the 2014+ Formula One™ engine efficiency by making possible the ignition of lean mixture via multiple ignition points which also help to reduce combustion duration and the engine's sensitivity to knock but direct injection is one of the enabling technologies on which the 2014+ Formula One™ operation relies.

To operate, the pre-chamber requires the formation of a rich mixture in its volume to maximise ignition reliability while the main combustion chamber operates above stoichiometric ratio. This stratification of the charge would be very difficult, if not impossible, with a port-injection engine. Indeed, as only one injector per cylinder is allowed [155], and as most of the fuel must be injected in the main combustion chamber, this is where the injector is positioned, and it is not possible to directly inject fuel into the pre-chamber for it to be 'active'.

The charge that will ignite in the pre-chamber thanks to the spark plug must be forced in through the pre-chamber orifices during the compression stroke. For the pre-chamber to operate below stoichiometry, the creation of a rich mixture surrounding the pre-chamber while the rest of the combustion chamber mixture is lean charge is necessary. Otherwise, if the whole combustion chamber charge is homogeneous, the mixture forced into the pre-chamber would have the same composition to that of the main chamber, i.e. a lean composition, which is the opposite of what is required.

Given the turbulence in the cylinder inlet port, it would be impossible to precisely achieve such a stratification with a port injection. Direct-injection is thus primordial for the pre-chamber efficacy and the injector must be positioned so that the rich-mixture around the pre-chamber is created without impeding the rest of the mixture. If the inhomogeneity of the main combustion chamber charge is too high, the combustion can be negatively impacted.

As shown by Equation 5-9, maximising volumetric efficiency is one of the levers to increase power output. For a given engine speed, there is a maximum mass flow that can go through the intake valve. In a port-injection engine, as the fuel is injected before the valve, a fraction of this flow is dedicated to the fuel, limiting the air mass flow and thus the volumetric efficiency which only considers air mass flow in its equation (Equation 5-12). In a direct injection engine, the whole mass flow going through the intake valve is composed of air, thus for the same condition, more air is trapped in the cylinder, the volumetric efficiency is increased. Otherwise, the same amount of air can be introduced in the cylinder with less effort from the intake (less wave action tuning for NA engines or less compressor work for TC engines).

$$\eta_v = \frac{m_a}{\rho_a V_{swept}} = \frac{2\dot{m}_a}{\rho_a V_{swept} N} \quad (5-12)$$

where m_a is the mass of air trapped in the cylinder, ρ_a the inlet air density, V_{swept} the engine swept volume, $\dot{m}_a$ the air mass flow trapped in the cylinder and N the engine speed.

As the fuel is directly injected in the combustion chamber, under the form of liquid droplets, heat is extracted from the charge and the combustion wall for them to evaporate. This extraction of heat leads to lower in-cylinder temperatures and lower knock sensitivity during the compression stroke (see Figure 5-14). Plus, the lower pressures and temperatures reduces the work required to compress the charge.

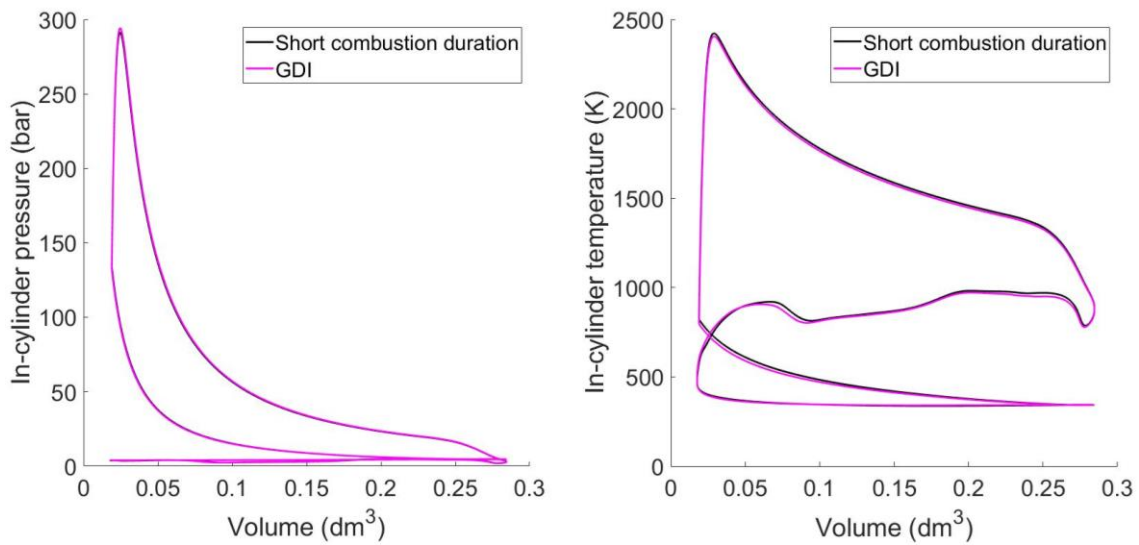


Figure 5-14 - PV and TV diagrams at 10,500 RPM - Short combustion duration (Black curve: Port injection) and GDI (Pink curve: Direct injection) models

In addition, direct injection maximises the amount of fuel trapped in the cylinder and usable for the combustion on the contrary to port injection as the risk of the fuel going through the valves (either through the exhaust valve during valve overlap or through the intake valve because of backflow) is minimised. If a maximum of the fuel injected is available for combustion, the fuel conversion efficiency is increased (for equal combustion efficiency). Figure 5-15 shows that despite injectors delivering the same fuel mass flows, the amount of fuel actually trapped in the cylinders varies significantly between port injection (short combustion duration model - Black curve) and direct injection (Pink curve).

As shown in Figure 5-14, despite the lower pressures and temperatures during the compression stroke, the conditions for the combustion and then the expansion stroke are very similar. This is the results of trapping more fuel in the cylinders. More fuel energy can be released during the combustion, leading to higher increase in pressure and temperature.

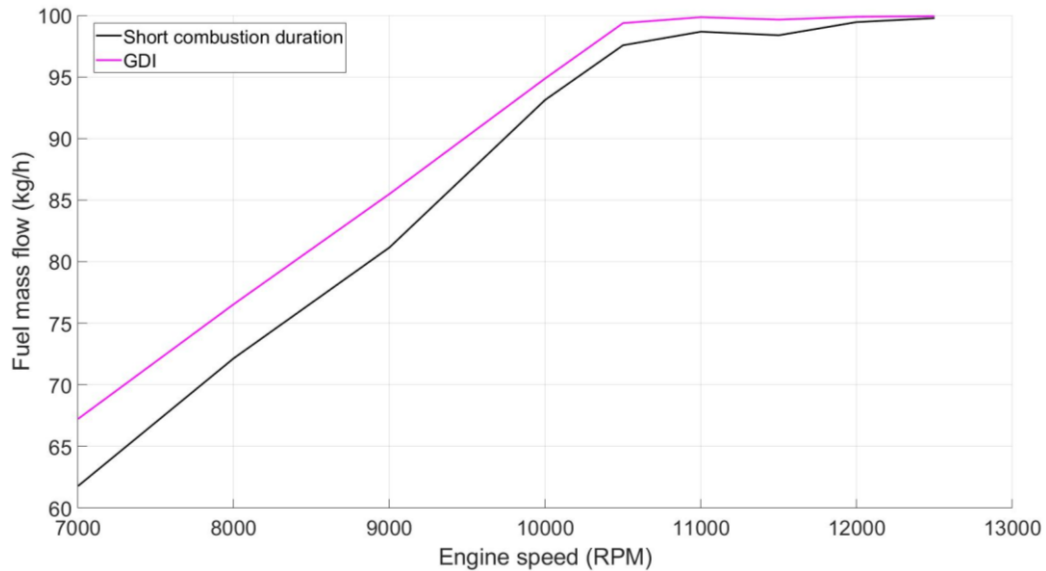


Figure 5-15 - Trapped fuel mass flow for port-injection (Black curve: Port injection) and direct injection (Pink curve: Direct injection)

The average simulation results shown in Figure 5-16 confirms, from a simulation point of view, the benefits of direct injection. The volumetric efficiency is increased, and the octane requirements are reduced. In addition, the combination of lower compression stroke work and more fuel being trapped in the cylinders leads to higher power and MEPs.

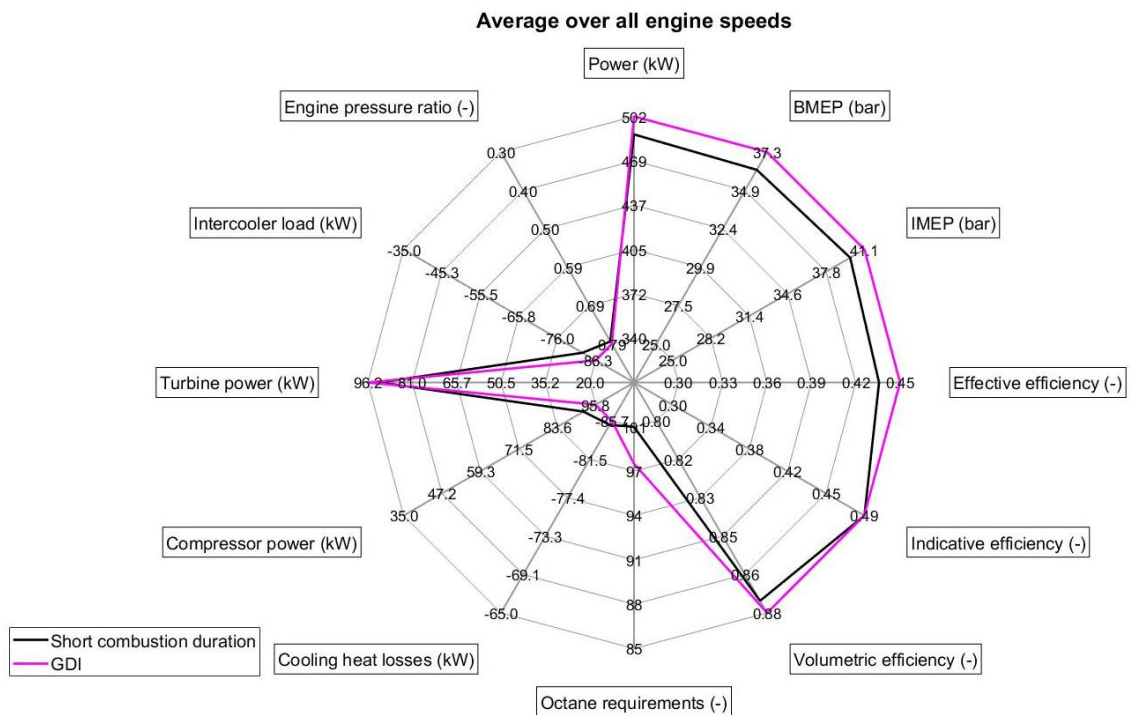


Figure 5-16 - Performance improvements brought by direct injection (Black curve: Port injection / Pink curve: Direct injection)

On average, the indicated and effective efficiencies are modified by -0.01 and 1.38 percentage points. The indicated efficiency does not improve despite the

higher IMEP which is unintuitive. But the higher IMEP figure is related to more fuel being burned, both trapped fuel mass flow and IMEP have increased by the same relative amount (in average 3 and 2.98% respectively) and as explained in Section 5.2.2, the indicated efficiency considers the amount of fuel trapped in the cylinders as a reference for calculations on the contrary to the effective efficiency which considers the injected fuel mass.

Because of the difference in injection technology, the trapped fuel masses considered for the calculations of the indicated efficiencies for both models are not the same (Figure 5-15) but as combustion properties are the same, the conversion of fuel energy into useful work is similar between both models. This is why indicated efficiency barely changes but the IMEP does. In real applications, direct injection could affect the combustion properties which is not modelled in AVL Boost™.

Because the effective efficiency considers the total amount of fuel injected as a reference, and not only the trapped fuel, changing the type of injection has not modified the quantity of fuel injected so as more BMEP is produced, thanks to the more fuel being effectively trapped and burned in the cylinders, the effective efficiency has increased.

It can be highlighted in Figure 5-16 that the compressor in the GDI model consumes slightly more turbine power than that in the previous model which is port injected. Once again, this is due to the difference in fuel flow. Both models operate at the same air/fuel ratios but because direct injection traps more fuel in the cylinder, more air is necessary to achieve similar dilution levels. As the air mass flow is higher, the intercooler load is also increased.

In the simulation presented here, AVL Boost™ models relies on the assumption of a 2-zone 0D homogeneous charge combustion which allows fast computation but also gives a low value assessment of direct injection as it cannot capture all the specificities of the combustion event allowed by this injection technology.

By placing the injector directly in the combustion chamber, it opens the possibility to complex fuel injection control that can lead to other types of combustion like HCCI or diffusion flame combustion.

With the tools available for this Doctoral research project, it is not possible to investigate how the use of a pre-chamber and direct injection can lead to a combination of combustion modes within a single combustion event (such as pre-mixed, diffusion, HCCI). Nevertheless, there is no doubt that if there are performance benefits to use direct injection in such fashion, especially to reduce combustion duration, engine manufacturers are exploiting it. What is sure is that direct injection is a mainstay of the 2014+ Formula One™ engine.

Moreover, the 2014+ Formula One™ regulations allow a maximum injection pressure of 500 bar and as seen in the literature review, high injection pressure, in addition to improving fuel preparation, creates small-scale turbulence which promotes faster flame speed and reduced combustion duration. There is also no doubt that high injection pressures were exploited by engine manufacturers to further improve combustion characteristics in combination with the pre-chamber.

5.2.5 Asymmetric valve lift profiles (not Millerisation)

The models previously presented in this chapter are provided with a set of valve lift profiles nearly identical for the intake (pink curve) and exhaust (black curve) as shown in Figure 5-17. However, it is known from private conversation that the 2014+ Formula One™ engine is equipped with asymmetric valve lift profiles with an intake valve lift profile reduced in lift and duration (brown curve) while the exhaust valve stays conventional [5] (see Figure 5-17).

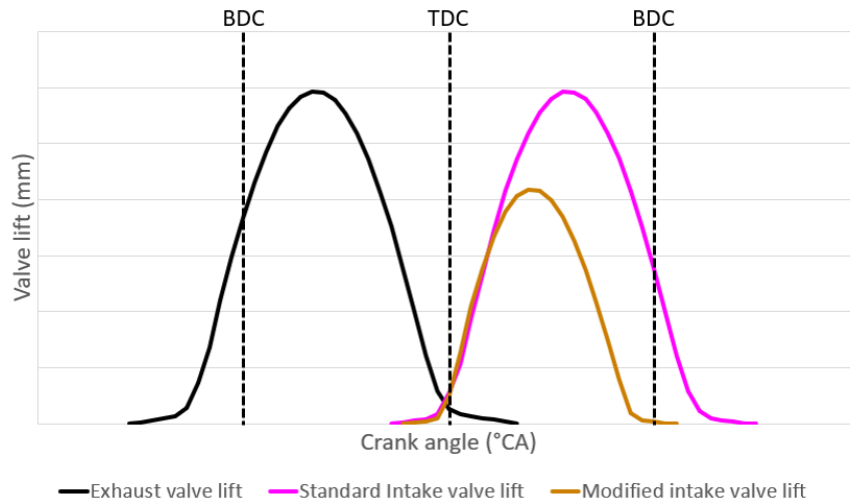


Figure 5-17 - Valve lift profile comparison

The drawback from the conventional intake valve lift profile is that it closes after Bottom Dead Centre (BDC), i.e. during the compressor stroke, which can cause back flow while the piston travels upwards. Thus, to trap as much air as desired, high boost pressure are required to compensate for the back flow. In consequence, the compressor consumes more work which requires more power from the turbine leading to higher back pressure which negatively impacts the breathing capacity of the engine. This type of valve is used as for the engine speeds for which the valve timing is optimised, the action of wave tuning leads to more air trapped right before the valve closes thanks to the high-pressure wave in the intake port and results in higher volumetric efficiency. However, this is advantageous only for the engine speeds for which the system is optimised.

By allowing the intake valve to close earlier (early IVC), closer to BDC, the back flow can be limited for the engine speeds that were compromised by the conventional profile but the engine speeds that were previously optimised cannot any more benefit from the late high-pressure wave. Overall, thanks to the reduction in back flow, the boost pressures required from the compressor are lowered.

Because IVC is setting earlier in the cycle and that valve overlap cannot be extended too much to avoid large amount of air travelling directly from the intake to the exhaust ports, the period of intake valve opening is decreased. Because of the high speeds at which the engine is run, the valves face extreme values of acceleration. If the valve opening duration decreases but the maximum valve lift stays constant, the valve acceleration increases and so does its stress. To ensure

the reliability of the valve, the maximum lift value must also be decreased to keep valve acceleration under control.

Such a modification of the intake valve lift profile is only possible for highly turbocharged engine in which the high boost pressures make up for the reduced air intake window.

As can be seen in Figure 5-18, the average volumetric efficiency is significantly increased using asymmetric valves (15.7 percentage points) leading to a reduction in the compressor power and intercooler load. By consequence, the turbine produces less power (the waste gate operates at greater angles), and the back pressure is reduced. However, because the intake pressures reduce more than the back pressures, the average pressure ratio across the engine increases.

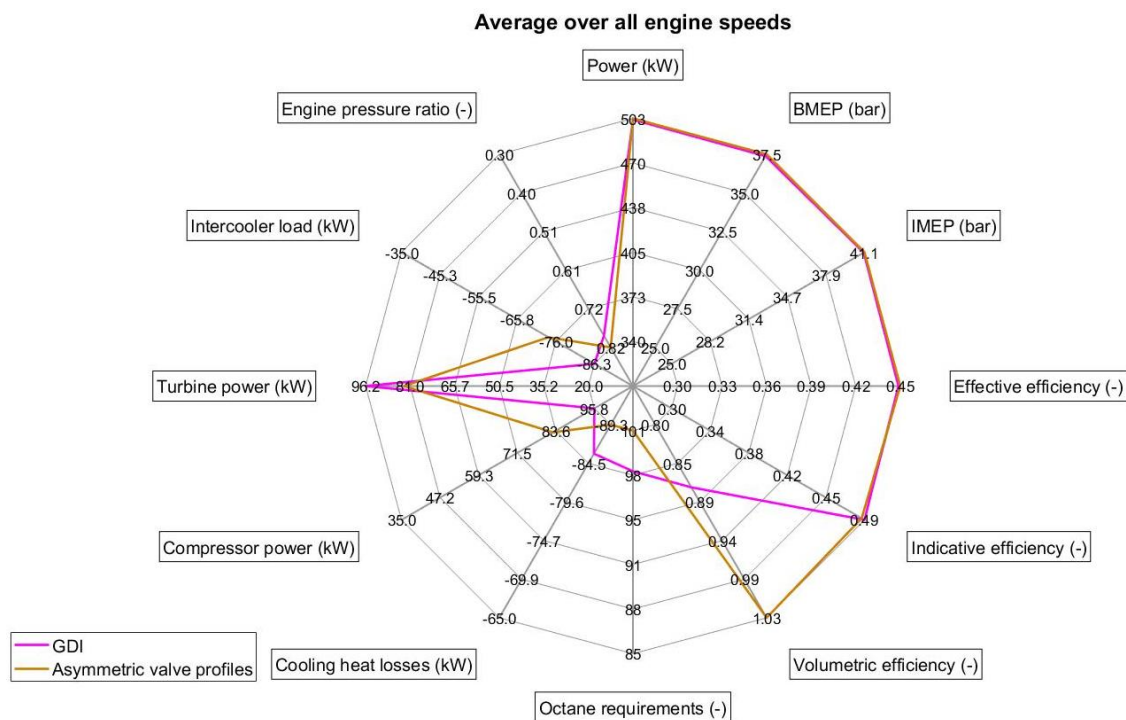


Figure 5-18 - Improvements brought by asymmetric valve profiles (Pink curve: Conventional intake valve lift profile / Brown curve: Modified intake valve lift profile, early IVC)

By reducing back flow, the modification of the intake valve profile also leads to more fuel being trapped in the cylinder, especially at low engine speeds, as fuel injection starts during the intake stroke. Overall, as seen in the previous section with direct injection, it results in increased average IMEP and BMEP (especially increased at low engine speeds) but the indicated and effective efficiencies evolve by -0.28 and 0.12 percentage point respectively for the same reason as mentioned in Section 5.2.4.

At first sight, the earlier IVC could point to the use of Millerisation (see Chapter 2 for the description of a Miller cycle) but the effects observed here do not meet the principles of this method. With the shortened intake valve profile, IVC occurs closer to BDC than for the conventional profile, meaning that the compression

lasts longer. This results in higher temperatures and pressures and thus higher octane requirements as the charge is exposed to harsher conditions.

For it to be authentic Millerisation, IVC should occur much before BDC so that the expansion before the compression could cool the trapped air, reducing octane requirements, and decrease the effective mechanical compression ratio which is not the case with the intake valve used here.

In [77], Rosetti et al. explain how the intake geometry of the 2014+ Formula One™ engine can be adapted to either tune, de-tune, or anti-tune the engine, the implications regarding the flow through the intake valve and how it can be used to improve the knock resistance of the engine. Such methods have not been investigated in this research work, but it shows how much valve flow can have an impact on engine behaviour.

5.2.6 MGU-H and unconventional waste gate operation

One very important component of the 2014+ Formula One™ Power Unit is the MGU-H, the electric motor connected to the rotor of the turbocharger. The presence of the MGU-H allows one to effectively semi-decouple the compressor from the turbine because the power of the compressor is no longer solely dependent on the turbine, an additional degree of freedom is added.

Indeed, as the MGU-H can provide and recover energy, the behaviour of the compressor can be precisely controlled independently of the conditions at the turbine. Two uncommon operating modes made possible by the MGU-H are presented in these sections. These two modes are extremes, a range of additional operating conditions will exist in between them. With the addition of the MGU-H to the model presented in the previous section, the 2014+ Digital Twin, i.e. the model of the 2014+ Formula One™ engine proposed in this thesis, is obtained.

As it will be discussed in Chapter 7, an electrically assisted turbocharger is valuable to maximise fuel conversion efficiency and acceleration during transient events, i.e. sudden change in load demand. Nevertheless, even with steady-state simulations, benefits provided by the MGU-H can be highlighted.

In a conventional turbocharged engine, the turbocharger is selected so that its compressor map is wide enough to cover the engine air requirement and that high boost pressures are provided at low engine speed, which maximise low-speed engine torque and the turbine is selected to allow the compressor to work at the target operating points. The system size is minimised as much as possible so that moment of inertia and turbo lag are reduced. At high rotor speed and mass flow, not to exceed the turbine swallowing capacity and thus restricting the engine high speed performance, a waste gate can be used to divert a fraction of the exhaust flow from the turbine.

However, the fraction of the exhaust which goes through the waste gates is wasted energy which cannot be recovered by the turbine. For a fuel-limited engine, energy losses must be minimised to make the most out of every gram of fuel burned.

This is why, thanks to the MGU-H, which decouples the compressor need and the turbine power, the 2014+ Formula One™ engine features a turbine capable of swallowing the whole engine mass flow at any engine speed [5]. The engine can operate without any waste gate as suggested in Chapter 2. By keeping the waste gates closed at all engine speeds, the energy recovered by the turbine is increased by 50% in average (Figure 5-19) and it is not anymore equal to the compressor required power as the fraction of the recovered energy not necessary to power the compressor is absorbed by the MGU-H to recharge the ES.

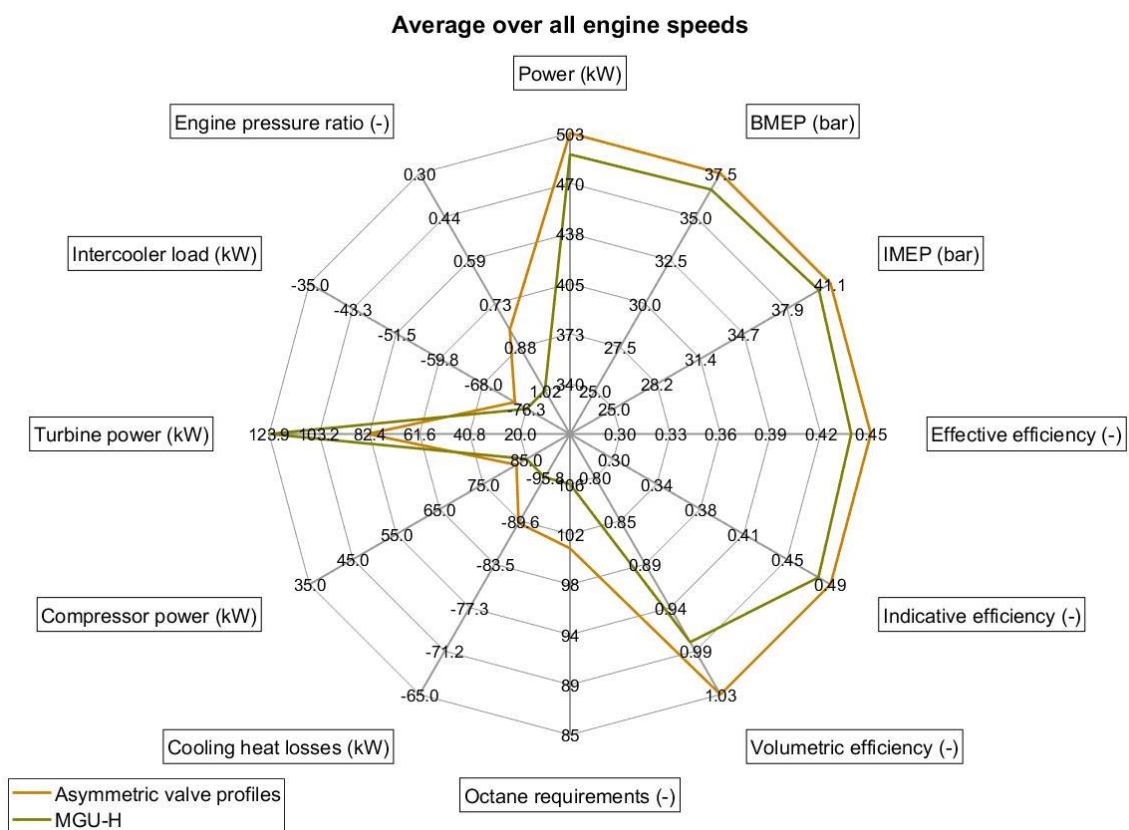


Figure 5-19 - Modifications brought by the addition of the MGU-H (Brown curve: No MGU-H, normal waste gate operation / Green curve: MGU-H and waste gate closed)

Nevertheless, recovering this additional energy comes with a price. Because a large mass flow must go through the turbine, the back pressure is increased leading to reduced breathing capacity, i.e. lower volumetric efficiency. As visible in Figure 5-20, this mode leads to a significant increase in pressure ratio across the engine which becomes higher and higher as engine speed and engine mass flow increase. Above 10,000 RPM, the pressure ratio is even higher than 1 meaning that the pressure at the exhaust is higher than that at the intake.

Because of the high pressure ratios across the engine, engine scavenging is also more challenging.

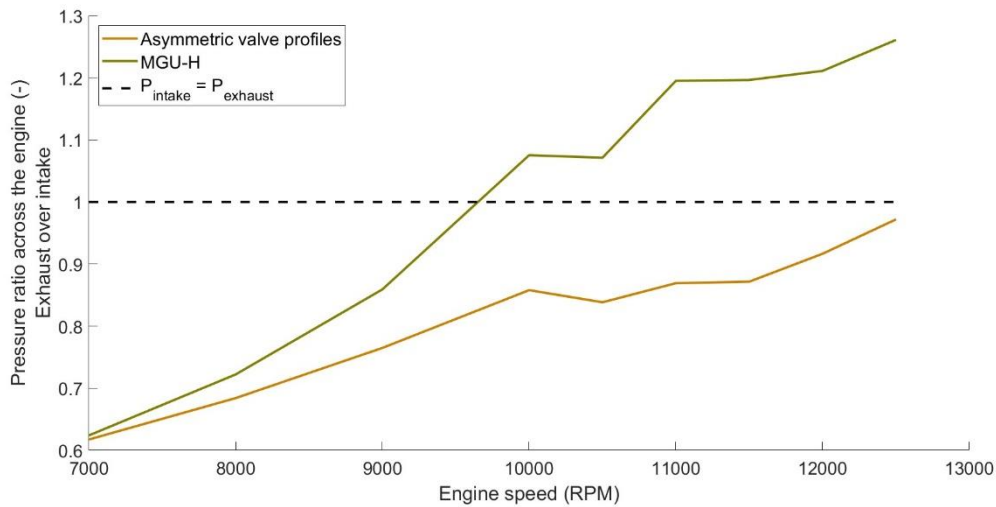


Figure 5-20 - Pressure ratio across the engine with and without the MGU-H (Brown curve: No MGU-H, normal waste gate operation / Green curve: MGU-H and waste gate closed)

Both models are operating at the same air/fuel ratios, leading to similar IMEP produced in the closed-valve part of the engine cycle which is confirmed by the very similar PV diagrams shown in Figure 5-21.

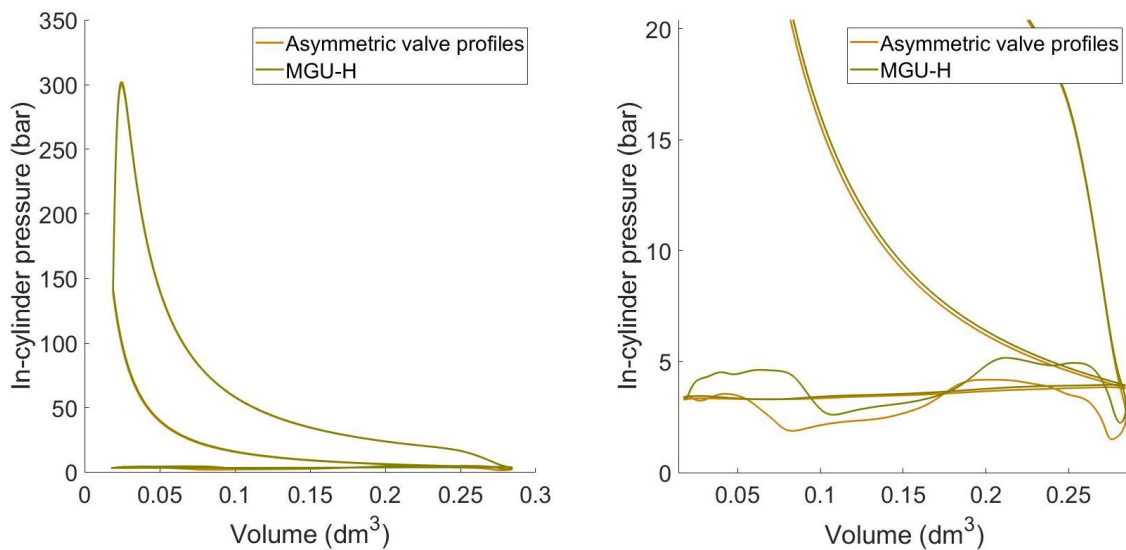


Figure 5-21 - PV diagrams (zoomed on the right graph) at 10,500 RPM – Asymmetric valve profiles (Brown curve: No MGU-H, normal waste gate operation) and MGU-H (Green curve: MGU-H and waste gate closed) models

Nevertheless, the low-pressure part of the cycle is negatively impacted by the lower volumetric efficiency caused by the higher exhaust back pressure. This leads to higher pumping losses and thus lower overall IMEP values. In average, the indicated and effective efficiencies are reduced by -1.06 and 1.14 percentage points respectively.

This high back pressure engine mode can seem absurd because of the reduction in power output compared to an engine without MGU-H. However, the engine is now part of wider system, the Power Unit, meaning that the ICE is not any more

the only source of power of the car. To optimise the operating of the entire system, it must be ensured that the ES is kept at an acceptable level so that the MGU-K can be used.

At 10,500 RPM, the MGU-H in the 2014+ Digital Twin recovers 47.5 kW for a 14.4 kW engine power reduction (Figure 5-22). By accepting to momentarily sacrifice engine power, additional energy can be recovered to later power the 120 kW MGU-K and maximise the total power output of the Power Unit.

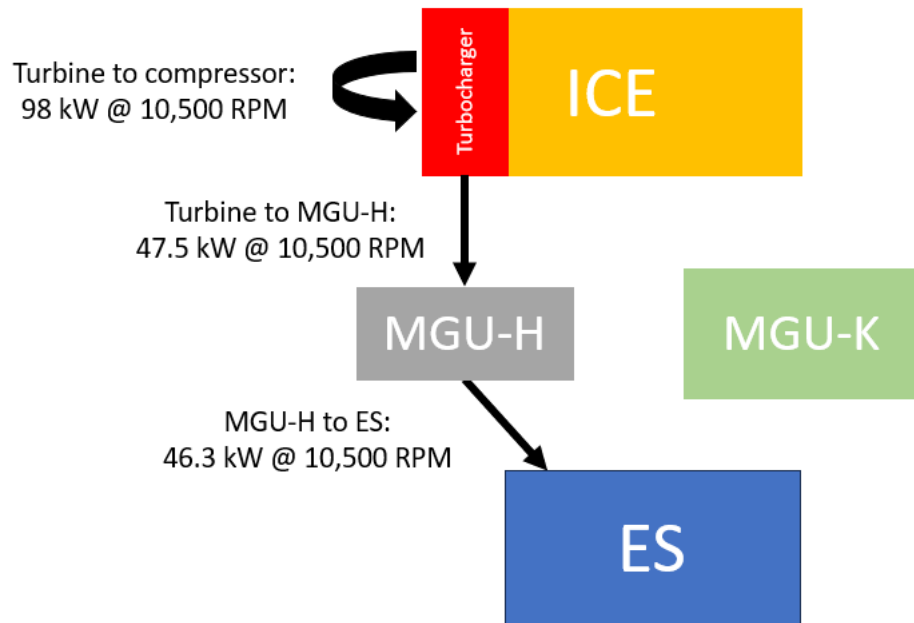


Figure 5-22 - Schematic of the 2014+ Digital Twin behaviour at 10,500 RPM in CWG-recharge mode

However, this view of the addition of the MGU-H is a bit reductive as it must be kept in mind that in the 2014+ regulations, it is also possible to transfer an unlimited amount of energy between the MGU-H and MGU-K (see Figure 1-4), energy that can then be used by the MGU-K to add power to the crankshaft, which increases the brake efficiency of the engine and help accelerate the car (Figure 5-23). By doing so, no energy goes in or out of the ES, its SOC is stable, but the power of the car is increased.

Compared to the CWG recharge mode, the CWG power mode adds an additional 45.1 kW (0.95 efficiency considered between the energy recovered by the MGU-H and the one delivered by the MGU-K) to the crankshaft which increases the efficiency of the engine in average by 3.09 percentage points (from 43.54 to 46.64%) without any modifications to the engine operating, the indicated power is unchanged.

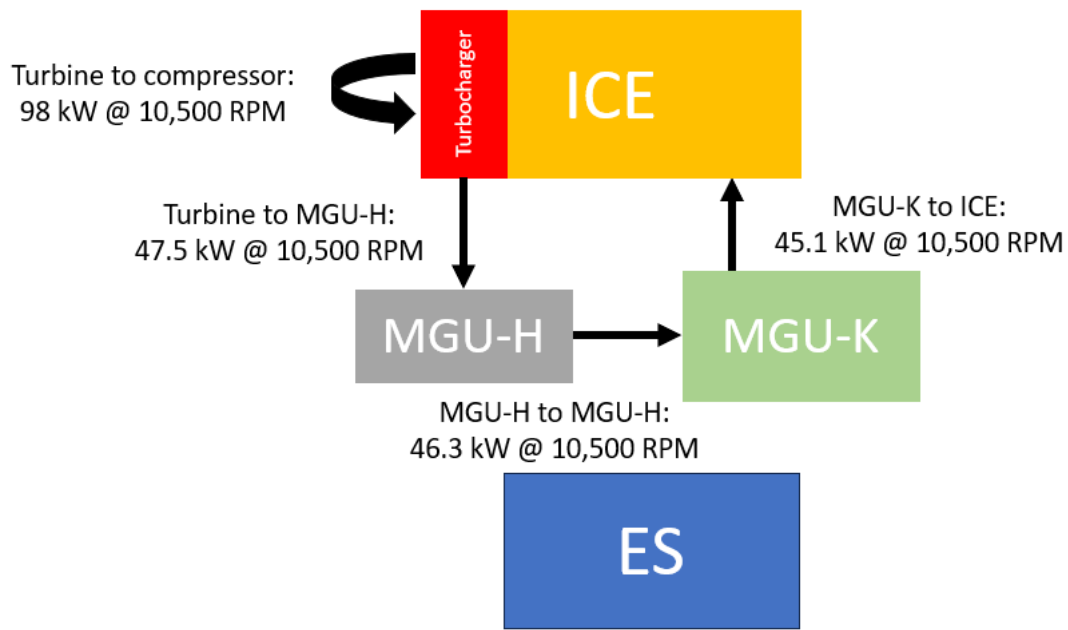


Figure 5-23 - Schematic of the 2014+ Digital Twin behaviour at 10,500 RPM in CWG power mode

In the same way that the MGU-H allows the turbocharger to operate with the turbine recovering more energy than needed by the compressor, the opposite is also possible.

As previously discussed, the biggest drawback associated with the closed waste gate (CWG) mode is the significant increase in back pressure which results in lower volumetric efficiency and lower output performance from the engine only point of view (in CWG recharge mode).

To reduce the back pressure, the engine can operate in an opposite mode in which the waste gates are fully open (OWG mode). By doing so, a large proportion of the exhaust mass flow is diverted from the turbine which reduces back pressure, improving the breathing of the engine, but also reduces the power produced by the turbine. The waste gates are used as pressure release valve. In this mode, because the engine is still operating at high dilution levels and that the turbine cannot provide the compressor with enough power, the MGU-H steps in and provides the difference.

Figure 5-24 demonstrates the breathing capacity improvement enabled by opening the waste gates. The back pressure drops so significantly that the pressure across the engine is always favourable ($P_{\text{intake}} > P_{\text{exhaust}}$), leading to a volumetric efficiency always above 1 and much improved at high engine speeds compared to the CWG mode.

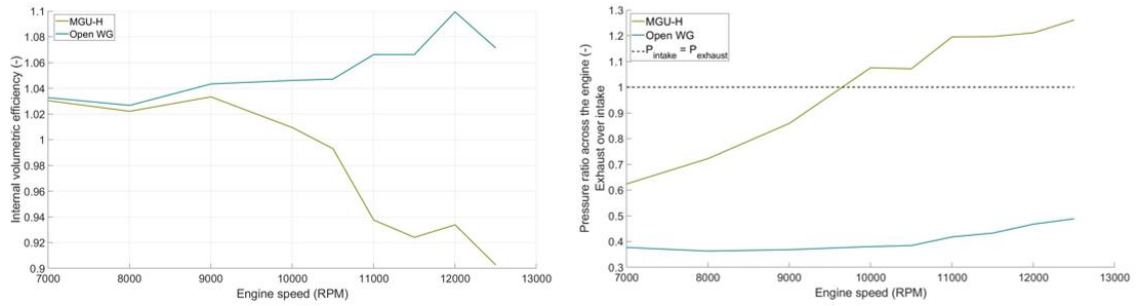


Figure 5-24 - Internal volumetric efficiency (left) and pressure ratio across the engine (right) evolutions with engine speeds for CWG and OWG modes of the 2014+ Digital Twin

It must be noted that the left graph in Figure 5-24 is the same as the one shown in Figure 3-3 (Chapter 3), as the two modes presented in this section are the two modes studied with the complete 2014+ Digital Twin.

As visible in Figure 5-25, the low-pressure part of the engine cycle is drastically changed between the CWG and OWG mode because of the exhaust back pressure drop. This reduction leads to much lower in-cylinder pressures during the exhaust stroke and as the compressor still provides the engine with high-pressure air, work is produced during the gas-exchange phase of the engine cycle.

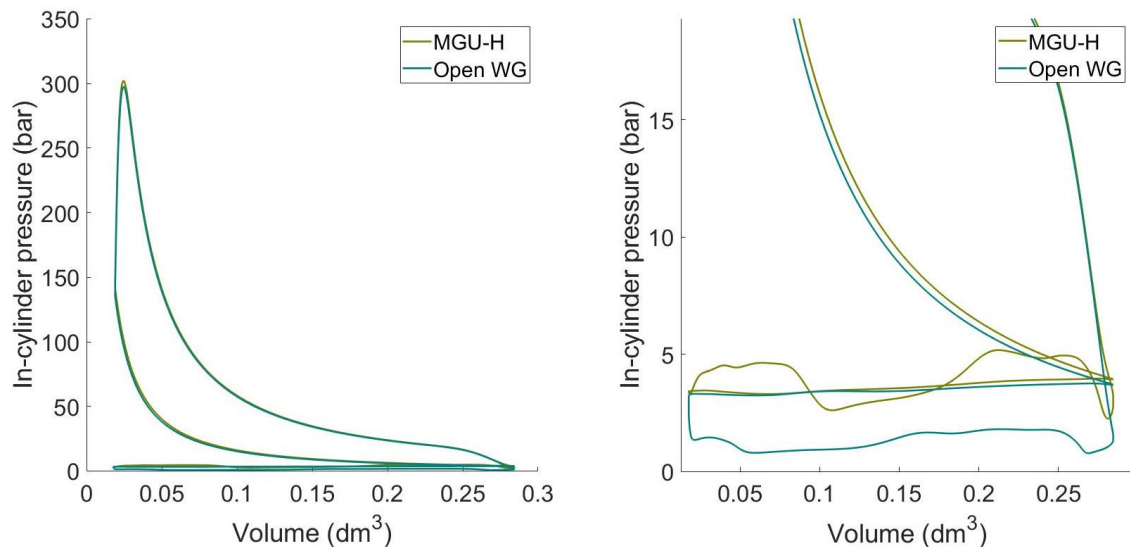


Figure 5-25 - PV diagrams (zoomed on the right graph) at 10,500 RPM - MGU-H (Light green curve: Waste gates closed, excess turbine power recovered by the MGU-H) and Open Waste Gate (Dark green curve: Waste gates open, the MGU-H supplies most of the power required by the compressor) models

It can also be noticed in Figure 5-25 that pressures are slightly reduced in the high-pressure part of the cycle for the OWG mode, including peak pressure which reduce the stress on the engine and octane requirement. Because the volumetric efficiency of the engine is increased by the opening the waste gates, lower boost pressures are required from the compressor to achieve the same air/fuel ratios which results in lower in-cylinder pressure at Inlet Valve Closing (IVC) and for the rest of the cycle.

Figure 5-26 shows the performances of the OWG mode compared to that of the CWG recharge mode and that of the last engine model without MGU-H and with a conventional waste gate operation. Compared to the two other models, the better breathing of the OWG mode resulting from the significantly lower pressure ratio across the engine yields improvements for most of the performance indicators shown in Figure 5-26.

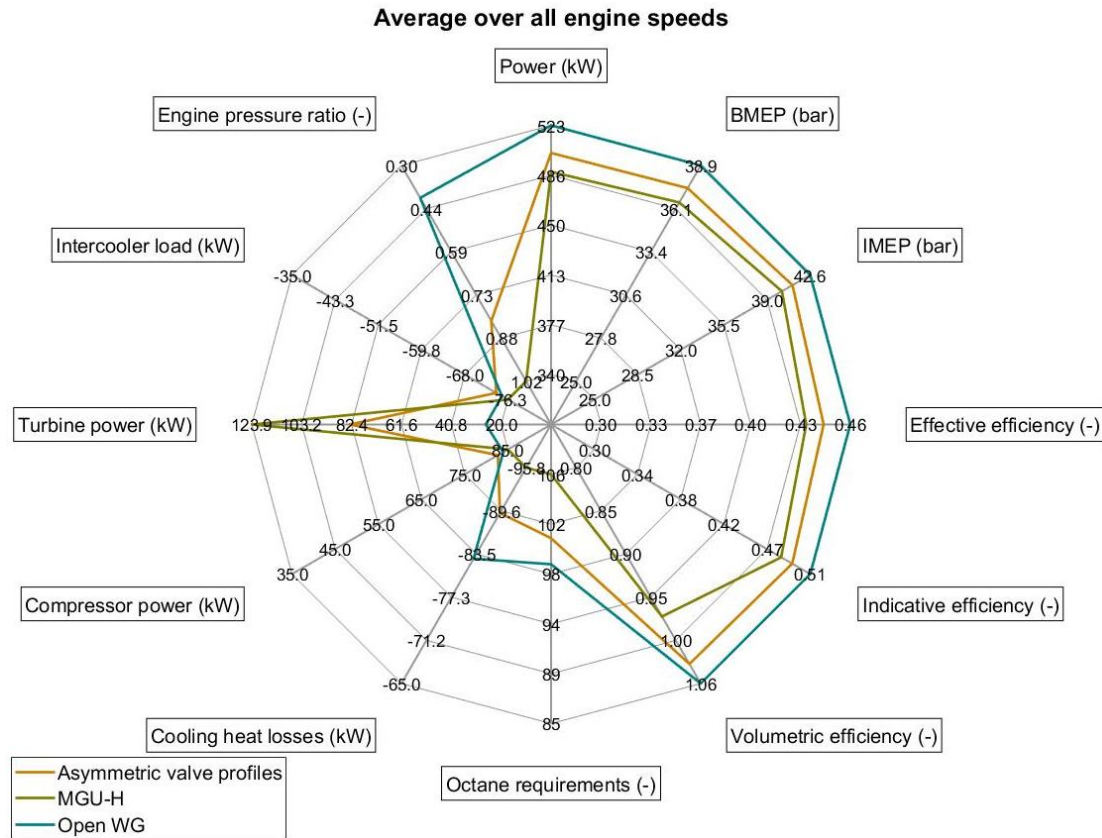


Figure 5-26 - Improvements brought by fully opening the waste gates (Brown curve: No MGU-H, conventional waste gate operation / Light green curve: MGU-H, closed WG / Dark green curve: MGU-H, open WG)

Respectively, the indicative and effective efficiencies are increased by 2.8 and 2.9 percentage points by opening the waste gates. The average indicated efficiency is now, finally, above 50%. In addition, because of the lower in-cylinder pressures, octane requirements and cooling losses are reduced.

Nevertheless, this mode is not sustainable for long periods. Indeed, to properly operate with the waste gates open, the power provided by the MGU-H is necessary to ensure the compressor reaches the target air mass flow and pressure. Thus, energy must be taken out of the battery to power the MGU-H and the compressor without any energy recovered to replenish the ES.

At 10,500 RPM, 69.8 kW is required from the MGU-H to operate the engine in the desired conditions (Figure 5-27). The MGU-H power in both CWG and OWG modes are shown in Figure 5-28. A positive value means that the MGU-H provides energy and the negative one that it recovers energy to recharge the ES.

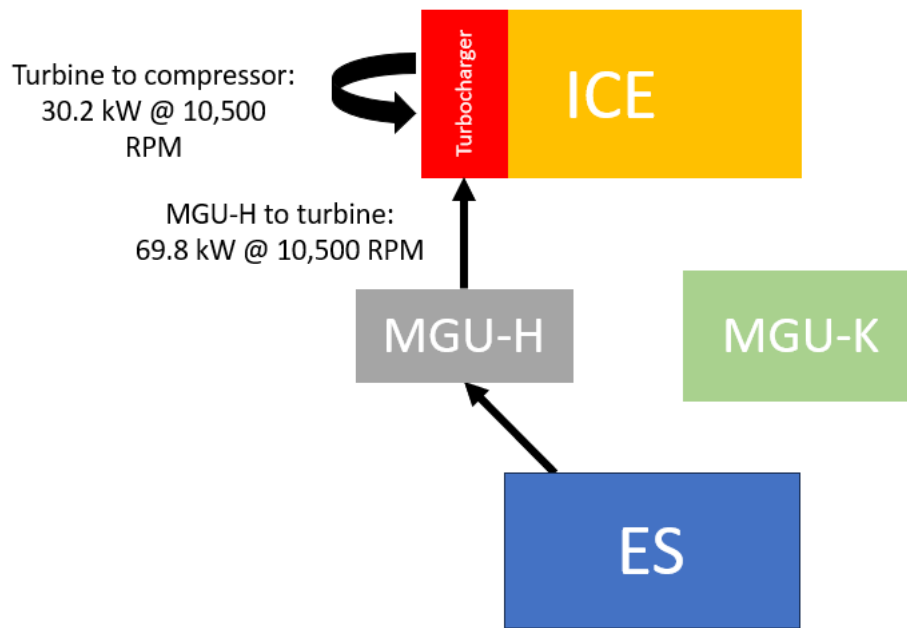


Figure 5-27 - Schematic of the 2014+ Digital Twin behaviour at 10,500 RPM in OWG mode

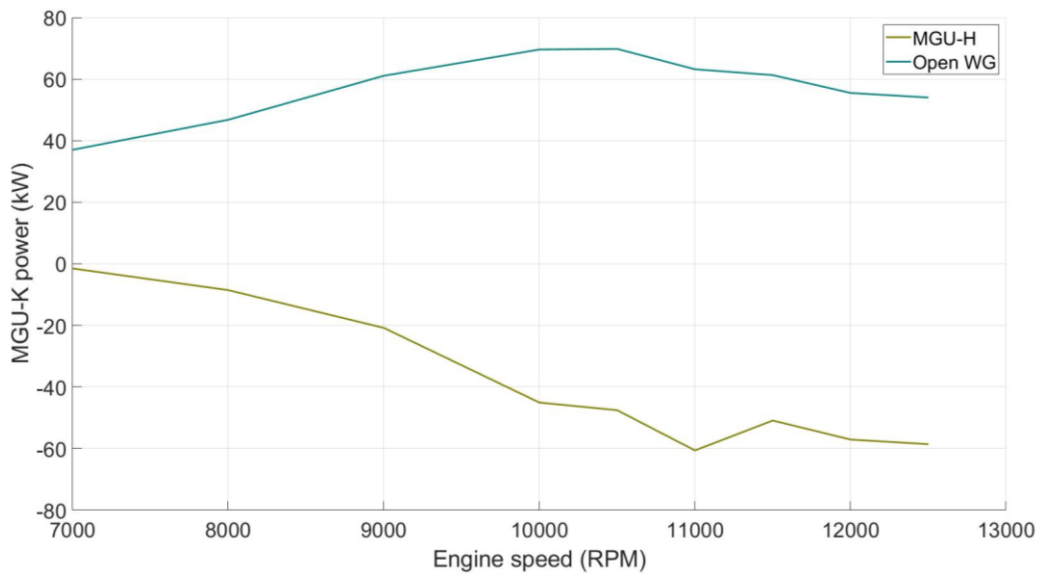


Figure 5-28 - MGU-H power evolution with engine speed for CWG and OWG modes

In addition, in terms of effective efficiency, the OWG mode is less efficient than the CWG power mode (46.44 vs 46.64%) because despite the engine power improvement, the MGU-H is providing energy and not recovering so the MGU-K cannot be used to artificially increase the brake power output.

If the engine and the MGU-K are used at their full power, this OWG engine mode is the most powerful one, as the engine output is maximised but can only be used for short periods in specific conditions where battery management is not crucial (qualifications for example) or when the racing conditions are suitable (attacking or defending from a competitor).

The link between the MGU-K and the MGU-H allowed by the regulations [155] and which makes energy transfer from one to the other possible is essential for the optimisation of energy management and is at the heart of this Power Unit. As shown in this part, when the energy recovered by the MGU-H is transferred to the crankshaft using the MGU-K, the brake efficiency is increased by more than 3 percentage points and lead to the most efficient mode of this Power Unit.

Although not studied in this Doctoral study, there is undoubtedly a compromise between MGU-H and ICE performance levels. If the MGU-H power input/output were to be increased, the penalty on the engine breathing would be higher but because recovered energy would be higher, the overall efficiency of this engine could be increased by doing so. However, there would be a limit to this which is where the additional energy recovered does not overcome the additional pumping losses faced by the engine.

5.2.7 Summary and conclusions

The previous sections of this chapter presented the different performance enhancement features known to be adopted for the 2014+ Formula One™ engine to maximise fuel conversion efficiency and thus produce the maximum power output from a limited source of energy and demonstrated from a scientific perspective, with the support of the 2014+ Digital Twin, how each solution influences the behaviour of the engine.

By running at higher air/fuel ratios than stoichiometry, combustion efficiency is improved by introducing enough air in the combustion chamber to oxidise every trapped fuel molecule to potentially release all the injected fuel energy. High dilution and high compression ratio then maximise thermal efficiency (conversion of heat into useful work) by increasing in-cylinder pressure and keeping temperatures under control which minimises the amount of heat lost to the cooling systems and to the exhaust and thus maximised the amount of work that can be produced.

The use of a pre-chamber, made possible by direct injection, provides distributed high ignition energy, and increased in-cylinder turbulence to maximise the combustion reliability of the lean mixture and to increase turbulent flame speed to achieve short combustion duration. Both the pre-chamber and direct injection help to reduce engine knock sensitivity and thus reduce the octane requirements to run at the extreme conditions described previously compared to a conventional gasoline engine.

In addition, direct injection improves fuelling control and increases the amount of fuel effectively trapped in the cylinder, resulting in higher power output. It also improves internal volumetric efficiency, as does asymmetric valve profiles.

The presence of the MGU-H on the turbocharger rotor allows to decouple the compressor needs and the turbine power which leads to a specific strategy for waste gate operation. By keeping the waste gates shut at any engine speed, the engine pumping losses are increased but the amount of enthalpy recovered from

the exhaust gas is maximised, allowing either to recharge the ES via the MGU-H or to transfer the energy to the MGU-K to significantly increase the engine brake efficiency.

On the other hand, the engine can be operated with the waste gate fully open to minimise back pressure and thus significantly increase indicated efficiency by powering the compressor using the MGU-H. But this mode can only be use for short periods as it draws energy out of the ES and proves to be less powerful than the CWG power mode which does not require any energy from the ES.

The impact of each solution on efficiency and a summary of their benefits are presented in Table 5-4:

Solution	Effective efficiency increase in percentage points compared to the previous model	Benefits
High compression ratio	+3.62	Results in higher in-cylinder pressures and a higher expansion ratio which improves thermal efficiency
High dilution	+4.07	Allows to maximise combustion efficiency and to run at high pressures and low temperatures resulting in less cooling and exhaust losses and thus higher output power despite higher combustion entropy creation (higher thermal efficiency)
Pre-chamber and shorter combustion duration	+0.57	<p>The pre-chamber makes possible and reliable the ignition of the lean mixture thanks to multiple ignition points and increases in-cylinder turbulence to overcome low laminar flame speed due to high air/fuel ratios resulting in shorter combustion duration</p> <p>In addition, alongside direct injection, it reduces the engine knock sensitivity</p>
Direct injection	+1.38	Improves fuelling control, i.e. results in more injected fuel effectively trapped in the cylinders and allows the specific stratification necessary for the pre-chamber operation
Asymmetric valve profiles	+0.12	Allowed by high-pressure turbocharging and results in higher volumetric efficiency by limit intake valve back flow

Solution	Effective efficiency increase in percentage points compared to the previous model	Benefits
MGU-H introduction (CWG recharge mode)	-1.15	Detrimental in terms of engine power but allows all the engine mass flow to go through the turbine to recover larger amount of energy from the exhaust gas and recharge the ES
MGU-H introduction + use of the MGU-H / MGU-K energy link (CWG power mode)	+3.09	The energy recovered by the MGU-H is transferred to the crankshaft via the MGU-K leading to a significant increase in effective efficiency as additional power is brought to the crankshaft without any energy drawn out of the ES
Open Waste Gate mode	-0.2	<p>Allowed by the MGU-H which powers the compressor instead of the turbine</p> <p>The back pressure drops drastically by fully opening the waste gates resulting in significantly lower pumping losses and higher indicated efficiency</p> <p>This mode is not sustainable as it draws energy out of the ES, has a lower effective efficiency than the CWG power mode but has the highest total PU power (ICE + MGU-K at peak power)</p>

Table 5-4 - Summary of the key solutions enabling the 2014+ Formula One™ engine high efficiency

The impressive difference between a conventional high-performance engine running at stoichiometric conditions, with a 10:1 compression ratio, 50 °CA combustion duration, port-injection and a conventional turbocharger compared to the 2014+ Formula One™ engine Digital Twin in OWG mode is shown in Figure 5-29.

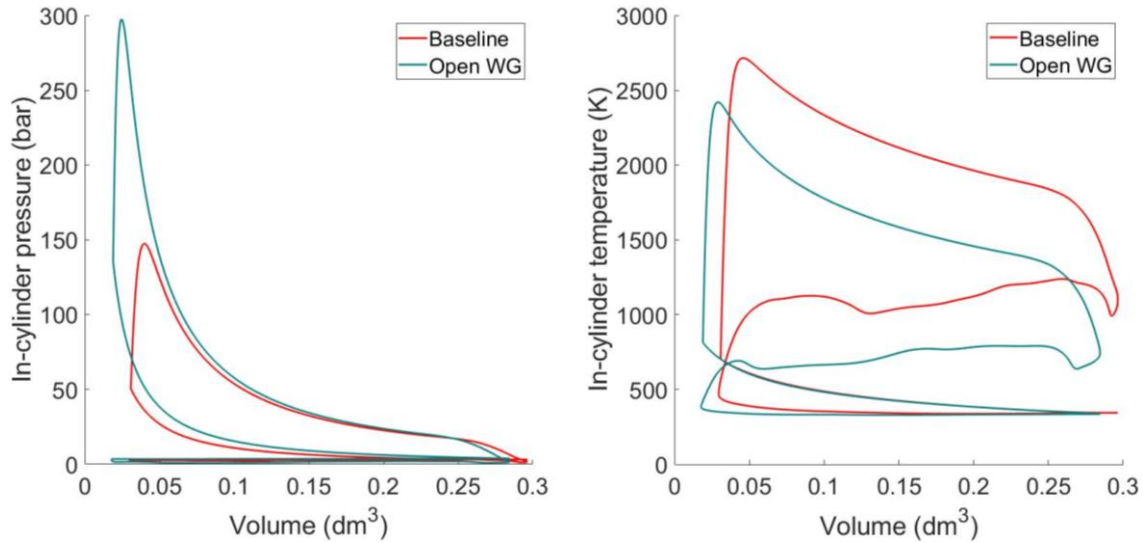


Figure 5-29 - PV and TV diagrams at 10,500 RPM - Conventional (Red curve: Base engine model) and the 2014+ Digital Twin in its highest performance mode (Green curve: 2014+ Digital Twin in OWG mode)

The important difference between the two engines comes from the capacity of the 2014+ Formula One™ engine to operate at very high pressures and controlled temperatures without knocking (at the limit of knocking) thanks to the solutions previously presented. By doing so, the entropy created during the engine cycle is limited (Equation 5-10) and the surface area of the PV diagrams is significantly increased which results in higher power production.

Figure 5-30 and Table 5-5 show respectively the power and effective efficiency curves of the CWG and OWG modes compared to the expected power window defined with potential effective efficiency extremums of 40 and 55% presented in Figure 5-2 and the peak performances of each engine mode.

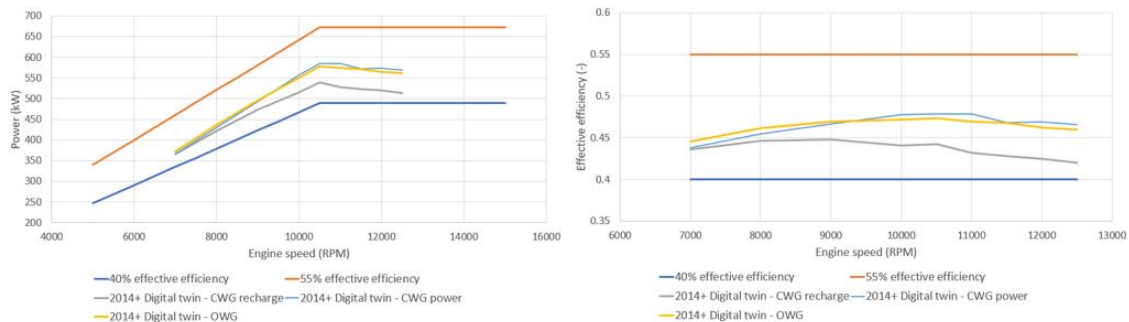


Figure 5-30 – 2014+ Digital Twin power and effective efficiency curves for the CWG and OWG modes compared to the expected power window from Figure 5-2

Engine mode	Peak indicated efficiency (%)	Peak effective efficiency (%)	Peak engine power (kW)	Peak power unit power (hp) ICE + MGU-K
CWG recharge	48.9 (@ 9,000 RPM)	44.8 (@ 9,000 RPM)	540 (@ 10,500 RPM)	885 (@ 10,500 RPM)
CWG power	48.9 (@ 9,000 RPM)	47.9 (@ 10,500 RPM)	585 (@ 10,500 RPM)	885 (@ 10,500 RPM)
OWG	51.6 (@ 10,500 RPM)	47.3 (@ 10,500 RPM)	578 (@ 10,500 RPM)	936 (@ 10,500 RPM)

Table 5-5 – 2014+ Digital Twin performance predictions

Because of the fuel flow limit, the power curves have very similar shape to what was expected as demonstrated by Figure 5-30. For all engine speeds and for both modes, the power curves are between both limits meaning that the effective efficiency of the 2014+ Digital Twin is always between 40 and 55%.

In CWG power mode, it can be observed that until 9,000 RPM, the power recovered by the MGU-H and provided to the crankshaft helps to improve the efficiency compared to the CWG recharge mode but the penalty in IMEP-BP is not overcome by the increased effective efficiency meaning that the OWG is the most efficient. But from 10,000 RPM, the observation changes and the CWG power mode, once the MGU-H can recover more amount of energy, becomes the most powerful mode and thus, without requiring energy from the ES to operate.

The peak engine power figures are achieved at 10,500 RPM, when the 100 kg/h fuel flow limit is reached, then the power slightly decreases linearly. One reason being that for the highest engine speeds, as shown in Figure 5-5, air/fuel ratios are reduced, and so is the efficiency in consequence, as it becomes more and more difficult to ensure the reliability of the lean combustion as engine speed increases. Another reason is the increase of FMEP with engine speed. Even though the fuel conversion efficiency was kept constant meaning that the IMEP would be stable too, the effective power would decrease because of the increase in friction losses with engine speed.

The 2014+ Formula One™ engine is mostly operated in a narrow engine speed window between 9,000 and 12,500 RPM where it spends in average 85% of a lap (value depending on the track) [5]. This characteristic is due to the shape of the power curve. The fuel flow restriction limits the power that can be produced below 10,500 RPM (before the fuel flow limit is reached) and the increase in friction with engine speed and the combustion stability limit performances above 12,500 RPM.

When the 2014+ Power Units were first introduced [8], the regulations stated a maximum engine speed limit of 15,000 RPM which since has been removed in 2022 [119] as no engine manufacturer was running its engine close to the limit because of the reasons listed above and the fact that turbocharged engines are

far less reliant on engine speed to improve engine breathing than naturally-aspirated engines.

The CWG power mode has the highest engine power but not the highest PU power as a fraction of the MGU-K power is used to transfer energy from the MGU-H to the crankshaft. In OWG mode, when the MGU-K is activated, the power is increased by the full power of the MGU-K, i.e. 120 kW, instead of 74.9 kW in CWG power mode.

As discussed in Chapter 3, with Equation 5-1 (governing equation for brake power), if a fuel's lower heating value of 44 MJ/kg and the fuel flow regulation limit are considered, a 51.2% effective efficiency is required to reach a PU total power of 1,000 hp with a 120 kW MGU-K. By accounting for 5% friction losses, it means the engine indicated efficiency must reach 53.2%.

In [13], it is stated that the engine achieves “in excess of 50% brake thermal efficiency” and in [14], it is reported that the PU “is producing in excess of a thousand horsepower” and the engine reaches over 50% thermal efficiency. These numbers are thus coherent with the predictions of the governing equations.

When stating these numbers, it is believed that the engine manufacturers are referring to their maximum performance mode which corresponds to the Closed Waste Gate power mode of the 2014+ Digital Twin. Looking at the numbers given in Table 5-5, the 2014+ Digital Twin in CWG power mode does not reach the values necessary for the PU to produce 1,000 hp. An additional 3.3 percentage points are necessary to reach 51.2% effective efficiency apparently required for 1,000 hp [5].

Considering the indicated efficiency, the 2014+ Digital Twin shows that values in excess of 50% are achievable, confirming that the manufacturer claims are plausible beyond governing equations using 0D/1D engine simulations. By accounting for the 0.7 bar IMEP uncertainty on the 2014+ Digital Twin predictions because of the inherent AVL Boost™ modelling error and the uncertainty linked to the input dataset (see Chapter 3), the peak indicated efficiency prediction potentially reaches 52.4%.

As previously stated in Chapters 3 and 4, the aim of the 2014+ Digital Twin is not to match the exact performance of the real engine but to inspire confidence in its ability to faithfully exhibit similar performance trends. Due to the nature of the 2014+ Digital Twin input dataset which is not a complete dataset from an engine manufacturer, the difference between its predictions and the manufacturer claims are not surprising.

Nevertheless, this 2014+ Digital Twin, whose behaviour is coherent with the feedback received from the industry, predicts performances of the same order of magnitude than the numbers given by the industry in its communication to the public showing that these values are reachable from a scientific point of view with the conditions the engine is running under. In addition, this reinforces the fact that the impact of each solution featured in the 2014+ Formula One™ engine and presented in this chapter on the engine performance are relevant.

The difference between the manufacturer claims and the 2014+ Digital Twin can be attributed to the reasons stated in Chapter 3, i.e. inaccuracy of the friction model, of the selected air/fuel ratios, modification of the combustion characteristics with engine speed for instance but also maybe from the inaccuracy of the elements on which the model was based and that were not questioned (ROHR profile, cam profiles, compression ratio value).

5.3 Energy and exergy analyses of the 2014+ Formula One™ engine

As presented in Chapter 2, energy and exergy are two different thermodynamic concepts that can be applied to a system, in this case an internal combustion engine, to investigate how the input (fuel) energy is transferred, both concepts are complementary.

Because exergy (also referred to sometimes as ‘availability’ but ‘exergy’ is preferred here) is based on the second law of thermodynamics, it quantifies irreversibility, i.e. when entropy is created and when useful energy is ‘lost’ (in as much as it cannot be reclaimed) during energy transfer or transformation.

Carrying out an exergy analysis on an engine is thus useful as it enables to identify which parts of the energy conversion process creates the more entropy / destroys the most exergy. By considering both the energy and exergy analysis, it is then possible to take actions to reduce these losses and increase the overall efficiency of the engine. It is even more revealing in Formula One™ as it is a sport where marginal gains matter and minimising irreversibility is key because of the limited fuel flow. So, all components of efficiency must be studied to look for any opportunity to extract as much as possible of the fuel energy.

To demonstrate the utility of exergy analysis and support the analyses of the 2014+ Formula One™ engine Digital Twin made in Section 5.2, the entropy created by idealised pressure-limited cycles can be investigated using Equation 5-10.

The cycles are limited to 300 bar which is taken to be a reasonably practicable limit for an engine that will meet the durability and reliability required for the prevailing 2014+ Formula One™ regulations. The quantity of fuel has been adapted using a dichotomy algorithm so that all cycles produce the same IMEP within 10^{-3} bar. The first cycle operates at stoichiometry with a 10:1 compression ratio, the second cycle at stoichiometry and 16:1 CR and the last cycle at $\lambda = 1.4$ and 16:1 CR.

The PV diagrams calculated for these three pressure-limited cycles are shown in Figure 5-31. Because all cycles produce the same IMEP, the area under each curve is the same. Both cycles running at 16:1 compression ratio are reaching the 300 bar limit whereas the 10:1 case peak pressure is of 257 bar.

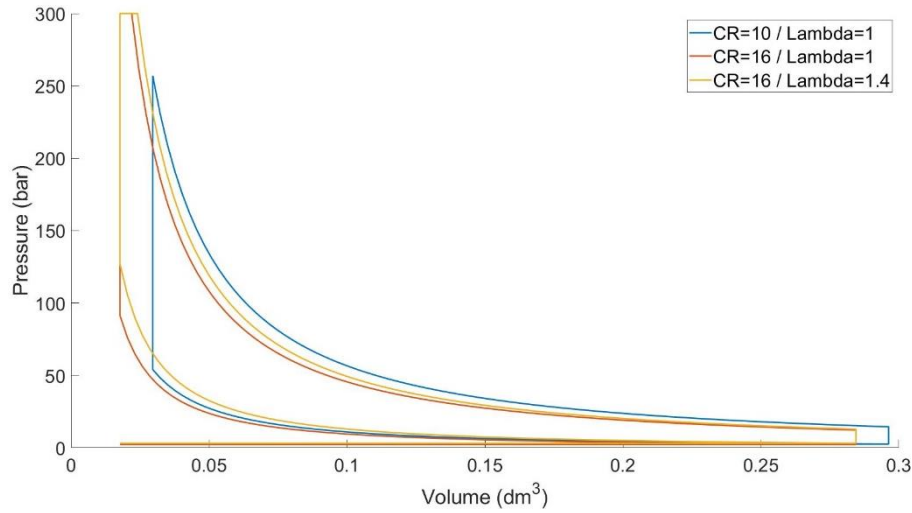


Figure 5-31 - PV diagram - Idealised pressure-limited cycles

All cycles produce the same IMEP but not from the same mass of fuel in consequence of difference in fuel conversion efficiency. In order, the cycles yield 40.8, 46.7 and 49.7% fuel conversion efficiency. The differences can be explained by looking at the entropy generated by each cycle as shown in Figure 5-32.

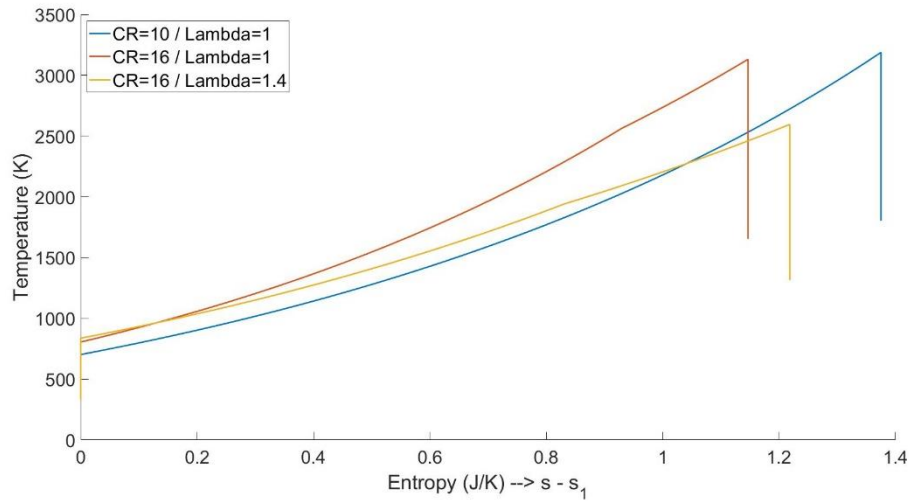


Figure 5-32 - TS diagram - Idealised pressure-limited cycles

By increasing compression ratio from 10:1 to 16:1, less entropy is created. Remembering Equation 5-10, if pressures are increased but temperatures remain similar as it is the case with the increased compression ratio, it results in lower entropy creation as confirmed by Figure 5-32.

$$S_2 - S_1 = m \left[C_p \ln \left(\frac{T_2}{T_1} \right) - r \ln \left(\frac{P_2}{P_1} \right) \right] \quad (5-10)$$

Increasing air/fuel ratio leads to lower in-cylinder temperature but causes more entropy creation, confirming what was observed in Section 5.2.2, because the combustion heat transfer is realised at lower temperatures and affects a higher

mass. Fortunately, the higher entropy creation is outweighed by the increased in-cylinder pressures leading to higher work and thus higher fuel conversion efficiency.

This study using idealised pressure-limited cycles is interesting as it demonstrates that the same IMEP can be reached via different cycles which do not require the same input. Compared to the $CR = 10 / \lambda = 1$ case, the two other cycles require respectively 11.7% and 15.2% less fuel. It again justifies why Formula One™ engine manufacturers are running their engine with lean mixtures to make the most out of their limited source of energy; solution which could also be useful in a non-fuel-limited engine to minimise the necessary onboard fuel mass.

Referring to the second law of thermodynamics is a useful tool when it comes to margin gains and trying to increase the understanding of the engine to further improve its efficiency.

5.3.1 Energy and exergy analyses of the CWG and OWG modes

Using the equations presented in Chapter 4, the energy and exergy distributions of the 2014+ Digital Twin can be analysed. In this part, only the thermodynamic behaviour of the engine is considered meaning that potential use of the MGU-K to increase the engine power output is not considered. This is why, the CWG mode studied here is the CWG recharge one and not the CWG power.

The energy distribution in terms of percentage of fuel energy for the Open Waste Gate (OWG) mode of the 2014+ Digital Twin is shown in Figure 5-33.

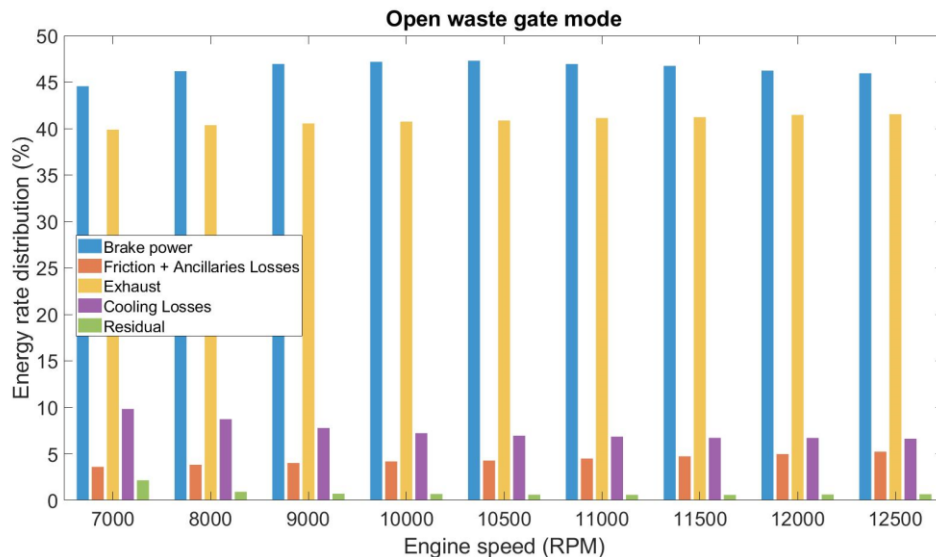


Figure 5-33 – 2014+ Digital Twin energy distribution - OWG mode

It must be highlighted in Figure 5-33 that a category of energy has been added compared to Equation 4-11 (Chapter 4). This is “Residual energy”. Indeed, when only considering brake power, friction and ancillaries losses, exhaust enthalpy

and cooling losses, the sum total is not exactly equal to the fuel energy. In average, the residual energy represents 0.83% of the fuel energy for the OWG engine mode.

The residual energy can be attributed to several factors which are not accounted for in the equations used to calculate energy distribution and create Figure 5-33. For instance, blow-by losses or the exhaust pipe radiation are contributing parameters of the residual energy. Because enthalpy flow is not measured directly in the cylinder outlet port but in the exhaust secondaries when exhaust pipes of each bank have merged, heat is transferred from the hot exhaust gas to the pipe and then radiated to the environment and this heat transfer is not exactly measured for the present energy and exergy studies.

However, as this residual energy contribution is relatively low, lower than 1% of the fuel energy, not investigating it in detail to break down the contribution of each factor to it will not affect the conclusions drawn from the energy analysis.

As seen in Figure 5-33, most of the fuel energy provided to the 2014+ Digital Twin in OWG mode is converted into brake power and exhaust enthalpy. The heat transfer to the coolant and friction / ancillaries losses are relatively low with a maximum of 9.8% and 5.2% respectively.

Heywood [34] reports the values shown in Table 5-6 for the energy distribution in conventional gasoline and Diesel engines.

Engine type	Power	Exhaust	Heat transfer to cooling system (oil + cooling medium)	Exhaust enthalpy losses due to incomplete combustion
Gasoline	25-30	25-35	22-35	6-15
Diesel	35-40	22-35	22-41	1-2

Table 5-6 - Average energy distribution for conventional gasoline and Diesel engines

By comparing the values displayed in Figure 5-33 and that of Table 5-6, the specificity of the 2014+ Formula One™ can be highlighted further. With an average effective efficiency of 46.4% in OWG mode, the 2014+ Digital Twin is much more efficient than a conventional gasoline engine and converts a higher percentage of fuel energy into work than Diesel engines.

By running with lean mixtures and as explained in Section 5.2, the 2014+ Formula One™ engine maximises combustion efficiency whereas for a conventional gasoline engine, up to 15% of the fuel energy is wasted by running at air/fuel ratios lower than stoichiometry.

By making the assumptions for this comparison that friction / ancillaries losses are fully transformed in heat, the total cooling losses are in average 15% for the 2014+ Digital Twin which is twice lower than the numbers shown in Table 5-6. As discussed previously, engine manufacturers have pushed hard to minimise

friction losses and maximise output power which ultimately reduce the need for cooling.

In addition, because the engine runs at high dilution, the in-cylinder temperatures are significantly reduced compared to a more conventional engine as demonstrated in Figure 5-29, and because convective heat transfer depends on the temperature difference between the combustion chamber wall and the gas (see Equation 5-13), heat transfer is limited and so is the need for cooling in terms of percentage of fuel energy.

$$E_{heat\ transfer} = h_c A_{HT} (T_{gas} - T_{wall}) \quad (5-13)$$

where h_c is the heat transfer coefficient, A_{HT} the heat transfer surface area, T_{gas} the gas temperature and T_{wall} the wall temperature.

In addition, the 2014+ Formula One™ engine runs a larger bore than its stroke (80 vs 53.052 mm) whereas it is generally the opposite for more conventional road car engines. However, as shown by Figure 5-34, for a given volume and if modelled as a conventional cylinder without considering the exact shape of a combustion chamber to give a rough estimation, the total cylinder surface area (liner + head + piston) does not evolve linearly. There is a range of bore values which minimises the cylinder surface area and thus further limits heat transfer as suggested by Equation 5-13. The dimensions of the 2014+ Formula One™ engine (shown in Figure 5-34 by the orange point) results in a cylinder surface area only 2% higher than the minimum value for this specific 0.266667 L (1.6 L / 6) volume.

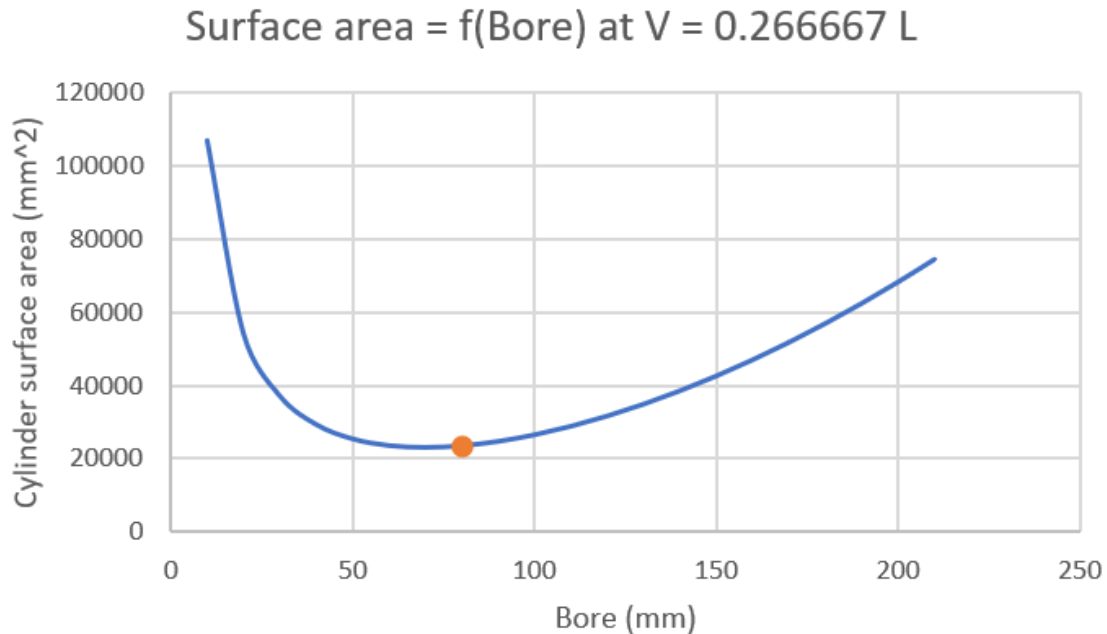


Figure 5-34 - Evolution of the cylinder surface area with bore for a given volume

Despite only representing a relatively small proportion of the fuel energy, the average cooling load over the engine speed range used on the track is of 82 kW which when added to the 75 kW cooling required by the intercooler still represents

a very significant amount of heat to remove from the car. This highlights the need for high performance heat exchangers as bigger sidepods to allow for higher air mass flow and bigger exchangers result in higher drag penalty.

If the 2014+ Formula One™ engine was behaving like the engines described in Table 5-6 with about 33% of the fuel energy being lost to the cooling system, this latter would need to be designed to remove 403 kW out of the engine which is extremely high and the engine would produce significantly lower power.

As shown in Figure 5-33, after brake power, most of fuel energy is converted into exhaust enthalpy. As shown by Equation 5-14, enthalpy flow depends on mass flow, the specific heat capacity at constant pressure and temperature. As the 2014+ Formula One™ is operating at high air/fuel ratios, temperatures are reduced compared to a conventional engine but for the same reason, the engine mass flow is increased significantly as large amount of air is required to operate at the targeted air/fuel ratios. The combination of lower temperatures but higher mass flow results in increased exhaust enthalpy, higher than that of conventional engines as suggested by Table 5-6. Because of the operating conditions of the engine, the heat released by the combustion which is not transferred to the cooling systems compared to a conventional engine is converted into useful work and increased exhaust enthalpy.

$$\dot{H} = \dot{m}C_pT \quad (5-14)$$

where $\dot{H}$ is the enthalpy flow, $\dot{m}$ the engine mass flow, C_p the constant pressure heat capacity of the mixture and T the mixture temperature.

Having a high exhaust enthalpy flow is important for the operation of the 2014+ Formula One™ Power Unit as a larger proportion of this energy can be recovered by the turbine thanks to the MGU-H, thus maintaining the ES state of charge high and maximising the opportunities to re-deploy this energy using the MGU-K. Once more, all the power produced by this Power Unit comes directly (engine power) or indirectly (MGU-K power) from fuel whose mass flow is limited. This is thus primordial to minimise energy waste.

The exergy analysis conducted on the 2014+ Digital Twin in OWG mode is shown in Figure 5-35 as exergy distribution for each investigated engine speed.

As for energy, most of the fuel exergy is converted into useful work with an average 43.5% efficiency which is 2.9 percentage points lower than the average energy effective efficiency (46.4%). This exergetic efficiency value is within the upper limit of the range observed by Farrell et al. [38] (between 35% and 42%) but their maximum value was achieved with an engine running in ultra-lean condition (up to $\lambda = 5$) whereas for the study reported here it was $\lambda < 1.5$.

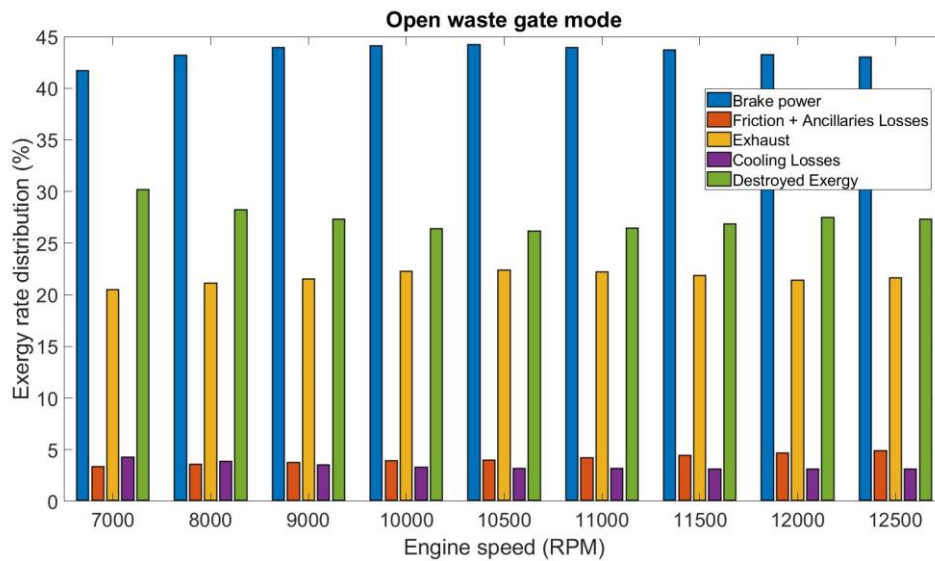


Figure 5-35 – 2014+ Digital Twin exergy distribution - OWG mode

After accounting for brake power, most of the fuel exergy is destroyed with an average of 27.4%. This amount represents all the irreversibility faced by the engine during the different processes it goes through, except friction, which is accounted for separately. Whatever is done with the engine, this component of the fuel exergy cannot be recovered and converted into work. Exergy destruction mainly represents the availability lost during combustion (which is increased with air/fuel ratios) and heat exchange.

Regarding the fluid flow losses, it is believed that they are already quite low on the engine under study as it is turbocharged, runs with lean mixture, and is mostly operated at full load so, there is barely any throttling. To further reduce fluid flow / pumping losses, volumetric efficiency must be increased which is possible within the 2014+ Formula One™ regulations using variable length trumpets. Another solution is the use of variable valve timing, but it is currently forbidden by the regulations and considering the stress already faced by the valves of these high-performance engines, this solution is very difficult to implement as it adds weight and complexity to the system.

The pre-chamber ignition system used by the 2014+ Formula One™ engine relies on high-speed jets of hot gas coming out of the pre-chamber which result in high turbulence intensity. This improves the ignition reliability of the lean mixture by providing more ignition points and thus ignition energy and increases flame speed, but turbulence results in more irreversibility than a conventional ignition system, i.e. increases the amount of exergy destruction during combustion. Nevertheless, the pre-chamber allows the engine to run with lean mixtures which outweighs the drawbacks linked to higher combustion irreversibility.

It can be noticed in Figure 5-35 that exhaust exergy represents a lower amount of fuel exergy (21.7%) than that of exhaust enthalpy flow for fuel energy (40.9%). This demonstrates that at the maximum, from the energy carried by the exhaust gas approximately only half of it can be converted into useful work because of

the inevitable irreversibility that will be associated with the energy transformation from heat to work.

Nevertheless, the available energy in the exhaust gas is much higher than that of the cooling system which represents on average only 3.4% of the fuel exergy. It is thus more interesting to try to recover energy from the exhaust gas using the turbine and the MGU-H rather than implementing energy recovering cycle, like Organic Rankine Cycle, using the heat from the cooling system.

Figure 5-36 and Figure 5-37 show a comparison for the energy and exergy distribution of both 2014+ Digital Twin modes, i.e. Open Waste Gate (OWG) and Closed Waste Gate (CWG).

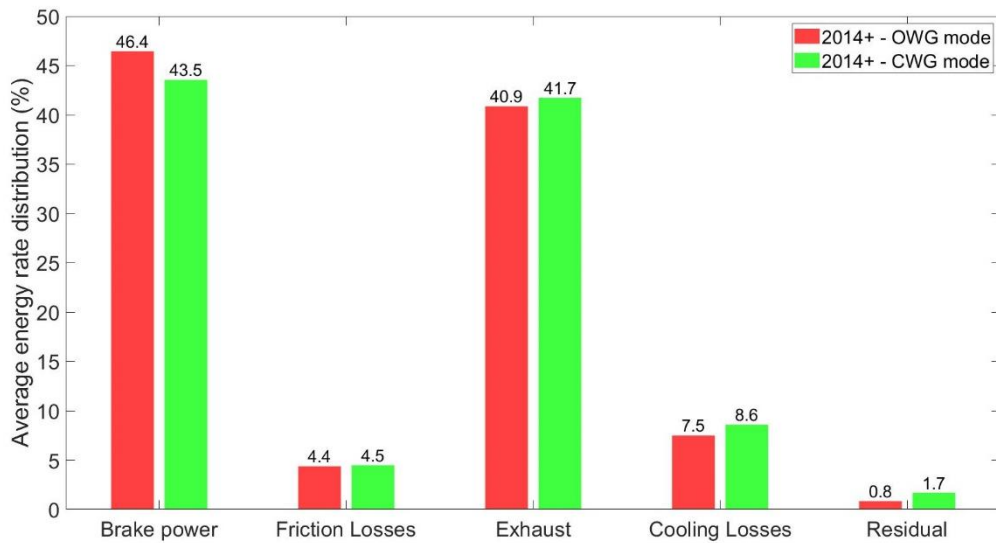


Figure 5-36 – 2014+ Digital Twin energy distribution - OWG and CWG comparison

Confirming the lower efficiency of the CWG mode seen in Section 5.2, the fraction of fuel energy and exergy converted into useful work is lower for the CWG mode. Because of its lower volumetric efficiency, the CWG mode operates at higher boost pressures resulting in higher in-cylinder pressures. In consequence, residual losses are increased (blow-by losses are increased with pressure) and so are convective heat transfer coefficients which increase with pressure according to Woschini model (see Equation 5-6). With higher heat transfer coefficients, heat transfer is higher as confirmed by Figure 5-36.

Similar amount of fuel energy is converted into friction and exhaust enthalpy for both modes.

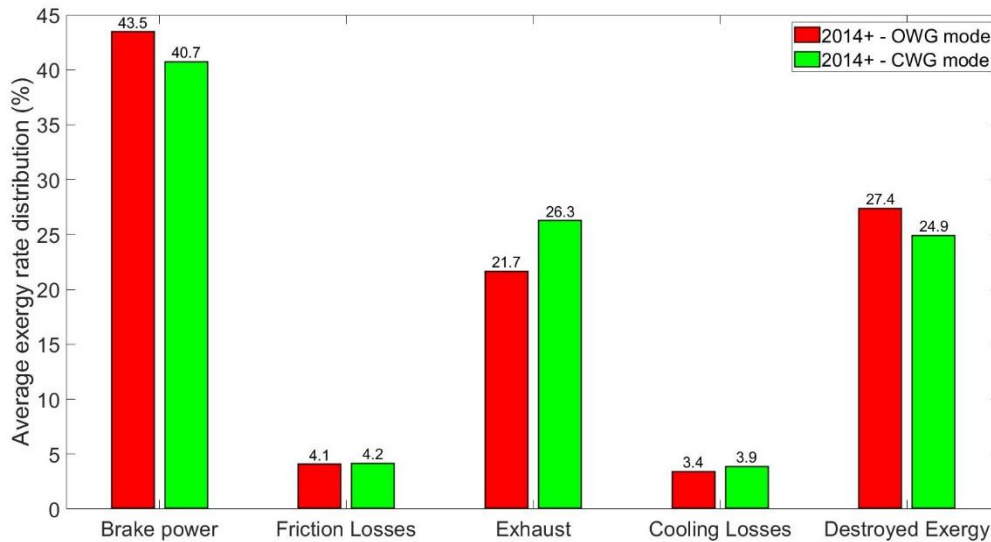


Figure 5-37 – 2014+ Digital Twin exergy distribution - OWG and CWG comparison

Regarding exergy, it is interesting to note that despite being less efficient, the CWG model destroys less exergy mainly because of the higher exhaust exergy which is due to the high pressure and temperature conditions in the exhaust system as the waste gates are closed. The higher the flow temperature, the more work can be extracted out of it as shown by Equation 4-16 in Chapter 4.

The high pressures and temperatures in the exhaust system result in higher energy available in the exhaust gas which is interesting considering that this CWG mode is made to recover as much energy as possible using the MGU-H to re-charge the ES.

5.3.2 Energy and exergy distribution of the 2014+ Formula One™ engine compared to that of an engine from the first era of Formula One™ turbocharged era

Using the AVL Boost™ model of the 1988 Honda RA168E built using the information provided by Otobe et al. [6] and validated against the engine power curve from the same article, it is possible to compare the energy and exergy distributions of both Formula One™ turbocharged eras.

The results of the energy and exergy analyses conducted on both engines are shown in Figure 5-38 and Figure 5-39.

As previously mentioned, the RA168E engine is similar to the engine model used in the Section 5.2 of this chapter and stripped off the 2014+ Formula One™ special features (referred to as “Baseline engine”). It is operating at air/fuel ratios lower than stoichiometry, with a 9.4:1 compression ratio, port-injection, and a conventional turbocharger.

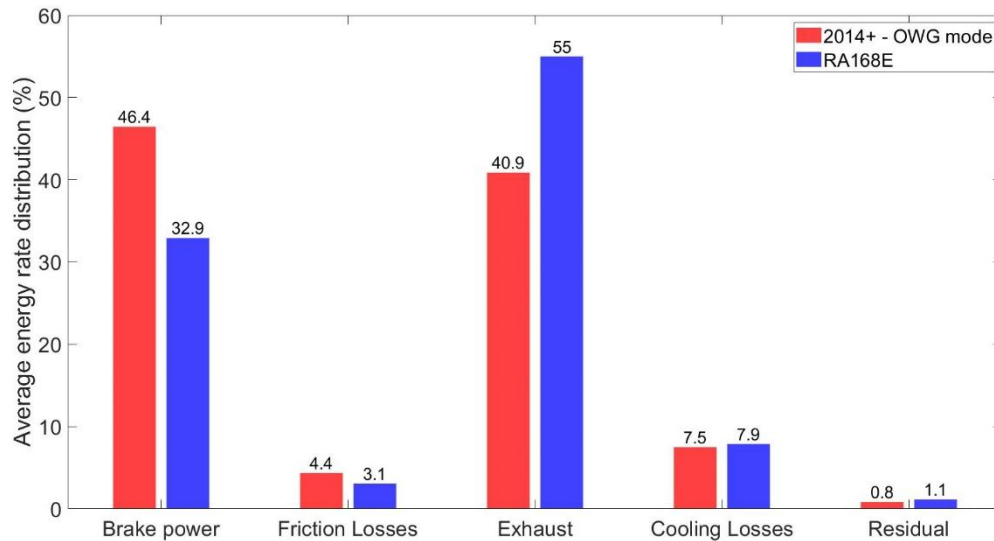


Figure 5-38 – Energy distribution – OWG mode 2014+ Digital Twin and Honda RA168E comparison

As seen in Section 5.2, all these special features are the enabling innovations for the 2014+ Formula One™ engine efficiency, thus when not implemented, there is a significant difference in effective efficiency as shown in Figure 5-38 with 13.5% less of the fuel energy being transformed into useful work for the RA168E compared to the 2014+ Digital Twin,.

As the RA168E engine is operating at air/fuel ratios lower than stoichiometry, in-cylinder temperatures are significantly higher with an average difference of 257 K over the simulated engine speed range according the AVL Boost™ models. But because of a low compression ratio and a 2.5 bar (absolute) boost pressure limit, in-cylinder pressures are significantly lower than that of the 2014+ Digital Twin (185 vs 300 bar).

As heat transfer coefficients increase with in-cylinder pressure but reduce with increasing temperature (Equation 5-6), their values for the RA168E engine are lower than that of the 2014+ Digital Twin. But because of the increased in-cylinder temperatures, the reduction in heat transfer coefficient is compensated which leads to similar levels of cooling losses for both engines.

Because of the lower in-cylinder pressures, the friction losses of the RA168E engine are lower than that of the 2014+ Digital Twin.

Regarding exhaust enthalpy, more than half the fuel energy of the RA168E engine is transformed into exhaust enthalpy. Despite the lower engine mass flow compared to the 2014+ Formula One™ engine, the higher in-cylinder temperatures and differences in exhaust gas composition because of the air/fuel ratios at which the RA168E operates result in significantly higher exhaust gas temperature and heat capacity leading to very high exhaust enthalpy.

As a turbocharged engine, the RA168E recovers a fraction of the exhaust enthalpy using its turbine but as it does not feature an MGU-H, the waste gates are operated as on a conventional turbocharger, i.e. a fraction of the exhaust flow is diverted from the turbine at high engine speeds not to exceed the limited turbine

swallowing capacity, meaning that the energy contained in the diverted flow is wasted.

Looking at the exergy analysis shown in Figure 5-39, it can be noticed that the RA168E fuel conversion into useful work results in a significantly higher proportion of destroyed availability (10 percentage point difference).

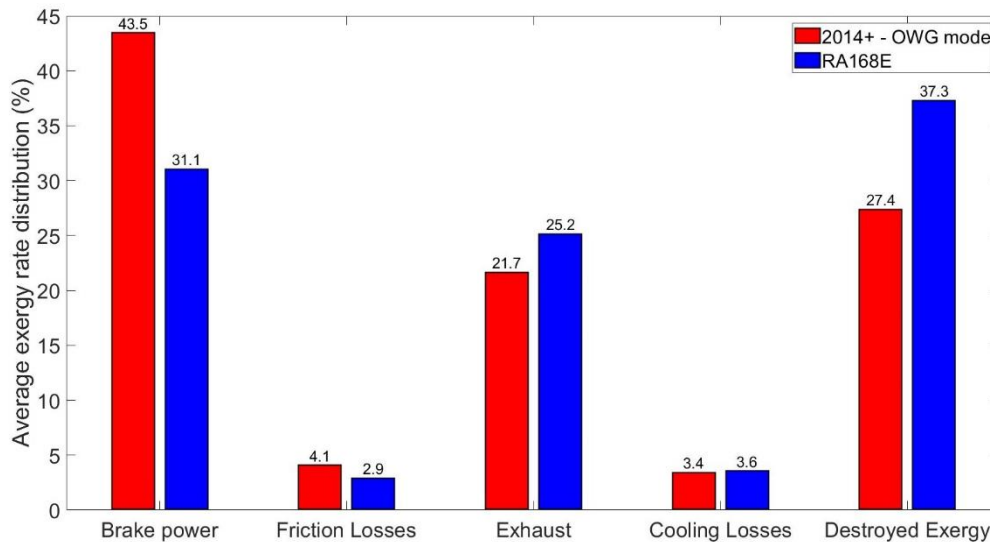


Figure 5-39 - Exergy distribution - OWG mode 2014+ Digital Twin and Honda RA168E comparison

As stated previously, the RA168E does not possess any of the special features of the 2014+ Formula One™ engine which make it efficient, i.e. it does not feature the solutions increasing the fraction of fuel energy transformed into useful work which are a high compression ratio and high dilution made possible by the pre-chamber ignition system and direct injection.

First, as the RA168E is operating with air/fuel ratios lower than stoichiometry ($\lambda = 0.87$), more than 10% for the fuel cannot be oxidised, i.e. the work potential contained in this fraction of the fuel is wasted, which is seen as exergy destruction. In addition, like lean mixtures, overly rich ones reduce adiabatic flame temperatures. The combustion thus happens at sub-optimal temperature which, as discussed previously, increases combustion irreversibility.

The energy analysis shown in Figure 5-38 highlighted that a large fraction of the fuel energy in the RA168E is converted into exhaust enthalpy. However, the conclusion given by the exergy analysis and Figure 5-39 is that only a small amount of this energy is recoverable, representing only 25% present of the fuel exergy. As explained in Chapter 2, from the second law of thermodynamic point of view, heat is energy of lower quality than work.

Indeed, a large fraction of the fuel energy goes into exhaust enthalpy which is recoverable using the turbine but as it is heat and not work, a smaller fraction is recoverable and results in higher exergy destruction.

The strength of the 2014+ Formula One™ engine compared to its predecessor is to run at higher air/fuel ratio than stoichiometry to release the whole injected fuel

energy (which is further helped by direct injection for better fuelling control) and its ability to operate at high pressure and low temperature to minimise the amount of energy transformed into heat, an energy of lower quality than work (not directly useful), resulting in less exergy wasted during heat transfer for cooling losses and in the exhaust. By minimising the fraction of fuel energy converted into heat, the conversion of fuel energy into work is increased and in consequence effective efficiency is higher.

5.4 Evolution of turbocharger operating characteristics with varying engine load and influence of the fuel flow limit on the engine behaviour

As already briefly discussed in this thesis, the 2014+ Formula One™ engine operates for 85% of the time, in average, in a narrow engine speed window between 9,000 and 12,500 RM. This is a consequence of the fuel flow limit imposed by the regulations which prevents the engine from increasing output power with engine speed and of the 8-gear gearbox featured in current Formula One™ cars.

Using the results of the 2014+ Digital Twin, it is useful to investigate how this fuel flow limit impacts the turbocharger behaviour and how the compressor operating points evolve with engine modes, load, or engine speed.

In this study, engine load is considered linearly linked to the fuel flow limit. 70% load was simulated with the 2014+ Digital Twin using 70% of the maximum fuel mass flow. Compressor boost pressures and MGU-H power are adapted for each case so that the air/fuel ratio evolution is always the same and that the turbocharger is balanced ($P_{compressor} = P_{turbine} + P_{MGU-H}$ with P_{MGU-K} positive or negative).

To simulate part-load conditions, adding a throttle body to the 2014+ Digital Twin was tested but for the tested load (between 70% and 100%), it proved not to make any difference on the results so the 2014+ Digital Twin without throttle body is used here.

According to data provided by the sponsor of this doctoral study [5], the engine operates for 72% of the time above 70% load demand, this is why only loads between 70% and 100% are investigated (see Figure 5-40).

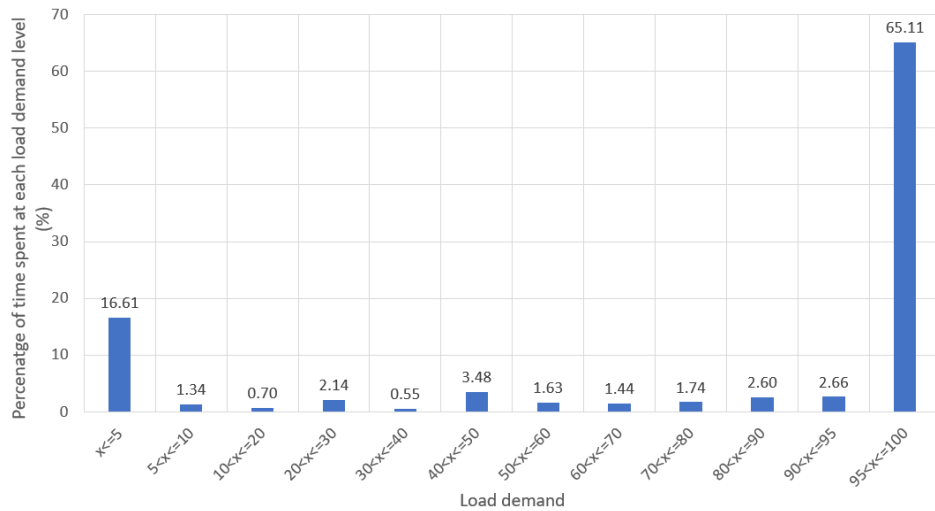


Figure 5-40 – Load demand distribution over a lap (x = Load demand)

In addition, the engine operation below 5% (i.e. no load, no torque demand from the driver during braking event) and above 95% load (full throttle) corresponds to 81.7% of the lap time, so except for 18.3% of the lap, the engine is either operated at full throttle or at no load. Part load operation below 70% only represents 11.3% of the lap time. This part is important for the engine transient operation (“tip-in” manoeuvre), but first, the focus is on the engine and turbocharger operation at high load ($> 70\%$).

Using the data from five different tracks with different specificities covering the spectre of tracks appearing in the Formula One™ calendar, the normalised average engine speed distribution is calculated and shown in Figure 5-41.

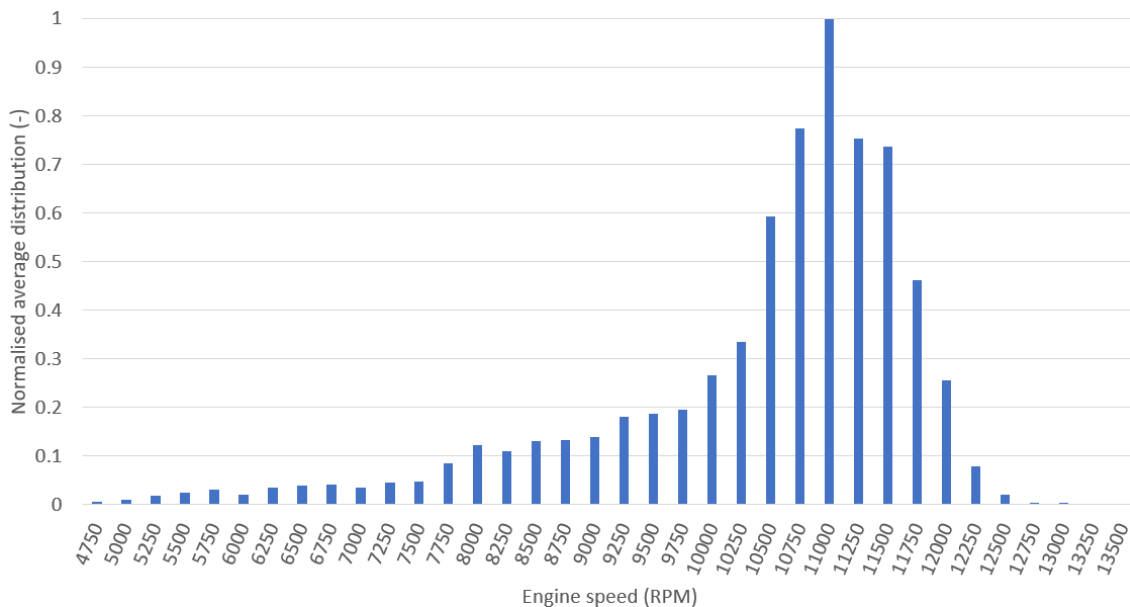


Figure 5-41 - Normalised average engine speed distribution over a lap

As previously discussed, the distribution is concentrated in the [10,500 – 12,000] RPM window with 66% of the lap time spent at these speeds. If the window is extended to [9,000 – 12,500] RPM, 86.3% of the lap is covered. The lowest

engine speeds, below 8,000 RPM, are only used for low-speed corners at low gears (8.2% of the lap time in average). During full-throttle operation, the 8-gear gearbox allows engine speed to be kept between 10,000 and 12,000 RPM.

To analyse the compressor operating point variation with load and speed for the 2014+ Digital Twin, the compressor map of the Garrett G40-1150 71mm turbocharger [173] was chosen but pressure ratios of the original compressor map have been raised by 1 bar to match the 2014+ Formula One™ engine needs. The map is only used for visualisation and is not implemented in the 2014+ Digital Twin which features a simple model of the turbocharger, as explain in Chapter 4, that does not require a compressor map to operate. As a forward note, the transient model of the 2026 regulations engine does have a complete compressor map.

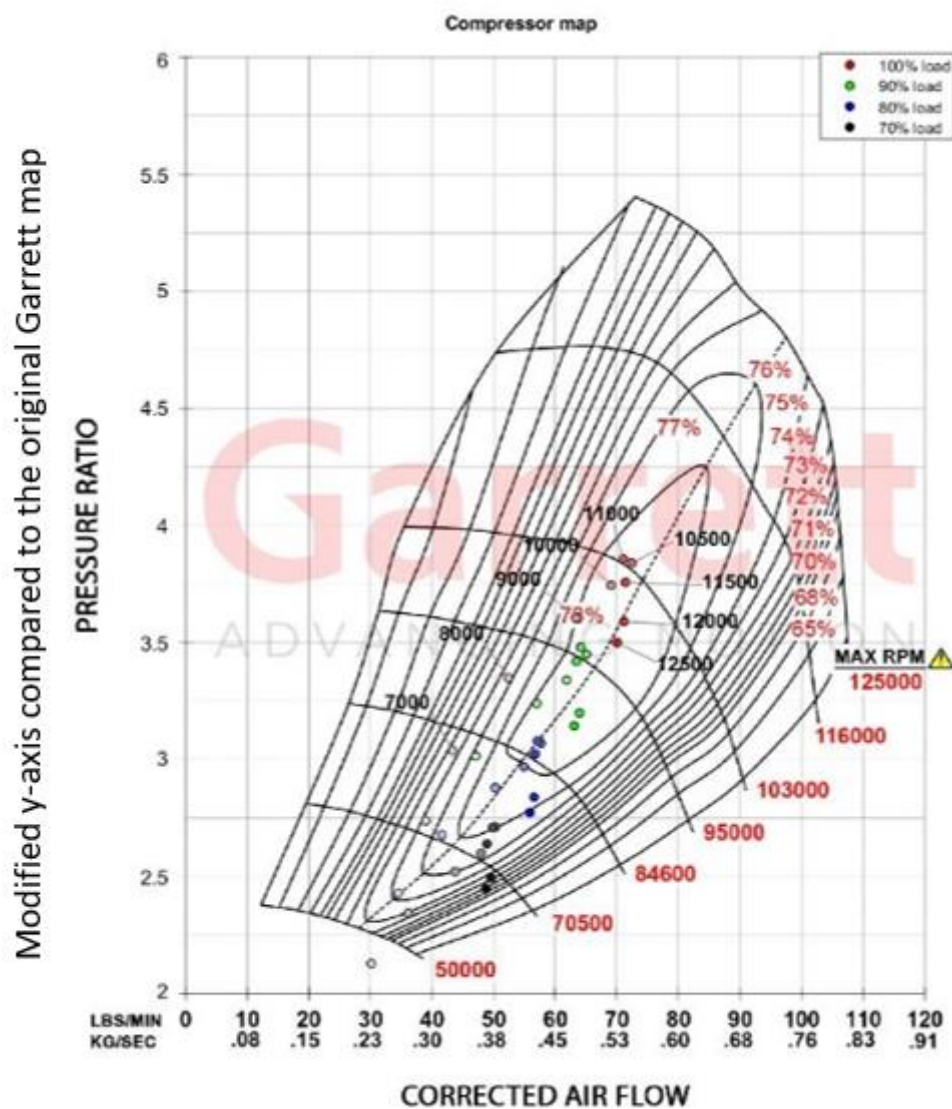


Figure 5-42 – 2014+ Digital Twin compressor operating points vs engine load and speed

The variation of the 2014+ Digital Twin compressor operating points with engine speed and load is shown in Figure 5-42. The 2014+ Digital Twin is simulated between 7,000 and 12,500 RPM which represent more than 95% of the engine speed distribution so relevant conclusions can be drawn out of the following figures. The different colours refer to different loads and within a same load case, the point colour evolves from white to the specific case colour with engine speed. The Closed Waste Gate (CWG) mode of the 2014+ Digital Twin has been used for the following figures and for all loads.

First, with the compressor map chosen, it can be highlighted that most operating points are located in the middle of the graph, i.e. in the highest efficiency zone. As the turbochargers featured on the 2014+ Formula One engines are specifically developed for this application, with some engine manufacturers developing their turbocharger in-house, this observation is considered likely to hold true for manufacturers' engines.

As load reduces, the operating points move to the bottom left of the compressor map. Indeed, when the driver demands less torque, as less fuel is injected, thus less air is required to operate at the same air/fuel ratios than at 100% load. Regarding compressor pressure ratio, as the engine internal volumetric efficiency is relatively constant as load varies, as there are no throttle losses and as the required air mass flow reduces, the required boost pressures to reach the target air/fuel ratios decrease.

For a given engine load, the evolution of the operating points is closely linked to the fuel flow limit and the choices made for air/fuel ratios. Between 7,000 and 10,500 RPM (see points in Figure 5-42 for 100% load), the fuel flow increases and so does target lambda (see Figure 5-5) which leads to increasing air flow demand and thus boost pressure, the compressor operating points are evolving linearly.

From 10,500 RPM, the fuel flow limit, i.e. 100 kg/h, has been reached and target air/fuel ratios do not vary as much as previously for low engine speed, so the air mass flow demand is relatively constant. This is why darker coloured points, which represent the highest engine speeds, are located around the same abscissa value. However, because engine speed increases, piston speed does too, making it easier for air to be trapped in the cylinder thanks to inertia effects. So, despite the engine volumetric efficiency decreasing with engine speed (see CWG mode, i.e. the model referred to as "MGU-H" in Figure 5-24), lower boost pressures are required as engine speed increases as shown in Figure 5-42.

Now considering the conditions the 2014+ Digital Twin is operated at and focusing on load to begin with, Figure 5-40 tells that for more than 70% of a lap time, the engine is used above 70% load, i.e. all the points shown in Figure 5-43 represents 70% of the compressor operating points in terms of time.

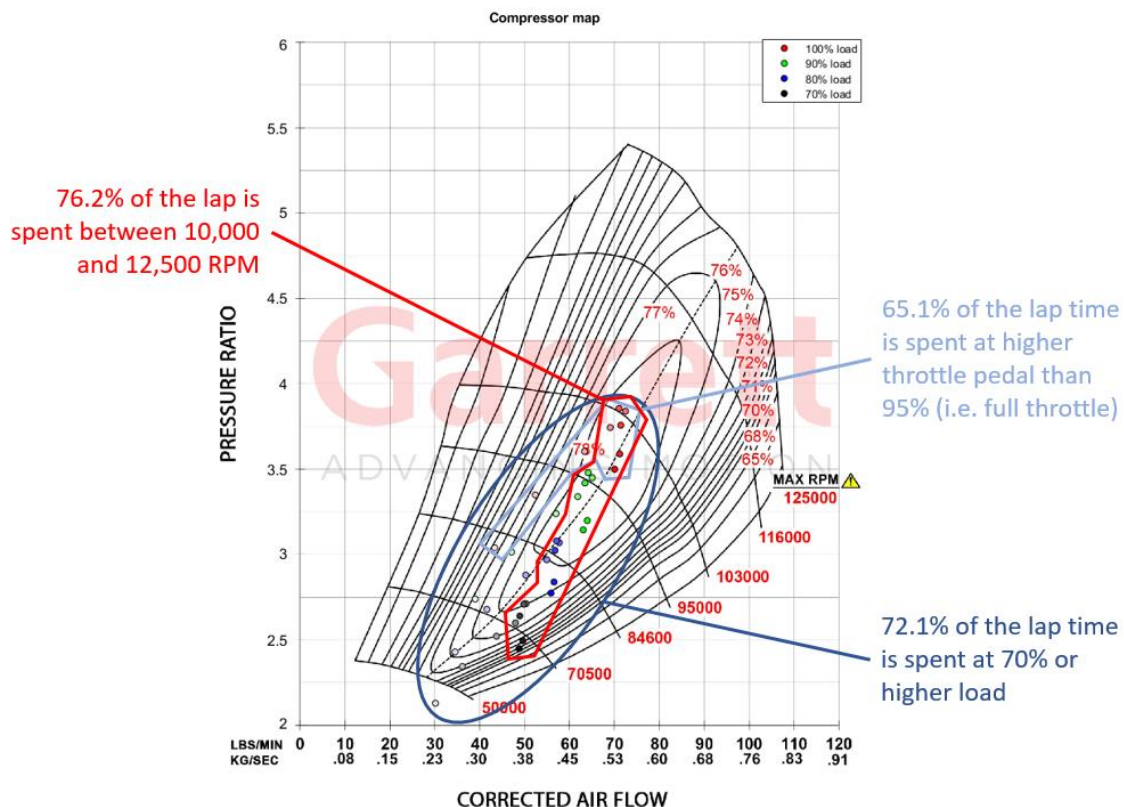


Figure 5-43 – 2014+ Digital Twin compressor operating points vs engine load and speed (Focus on the engine operating points)

However, it is also known from the load demand distribution (Figure 5-40) that 65.1% of the lap time is spent at 95% or higher load demand which corresponds to the 100% points in Figure 5-43 (red dots circled in light blue).

Now considering engine speed distribution around a lap and not engine load any more, Figure 5-41 shows that the engine is operated between 10,000 and 12,500 RPM for 76.2% of the lap time, this corresponds to the points circled in red in Figure 5-43. These points are also gathered in the high-efficiency zone of the compressor map independently of the engine load.

However, if the analysis is pushed further and the actual engine operating points (combination of engine speed and load at a given moment in time) are considered, it shows that when the engine operates above 95% load, engine speed only varies between 10,000 and 12,500 RPM. The engine operates in this window 65.1% of the time, meaning that for two thirds of the lap time, the compressor is operating within the very narrow zone circled in black in Figure 5-44 which is at the very heart of the highest efficiency zone of the compressor map.

As the compressor is operated in this zone for most of the lap (two thirds of the lap), it makes sense to optimise the compressor map so that these points are in the highest efficiency zone. This means that for a given compressor operating point, the power required by the compressor is minimised and a bigger fraction

of the power produced by the turbine can be recovered by the MGU-H and thus recharging the ES.

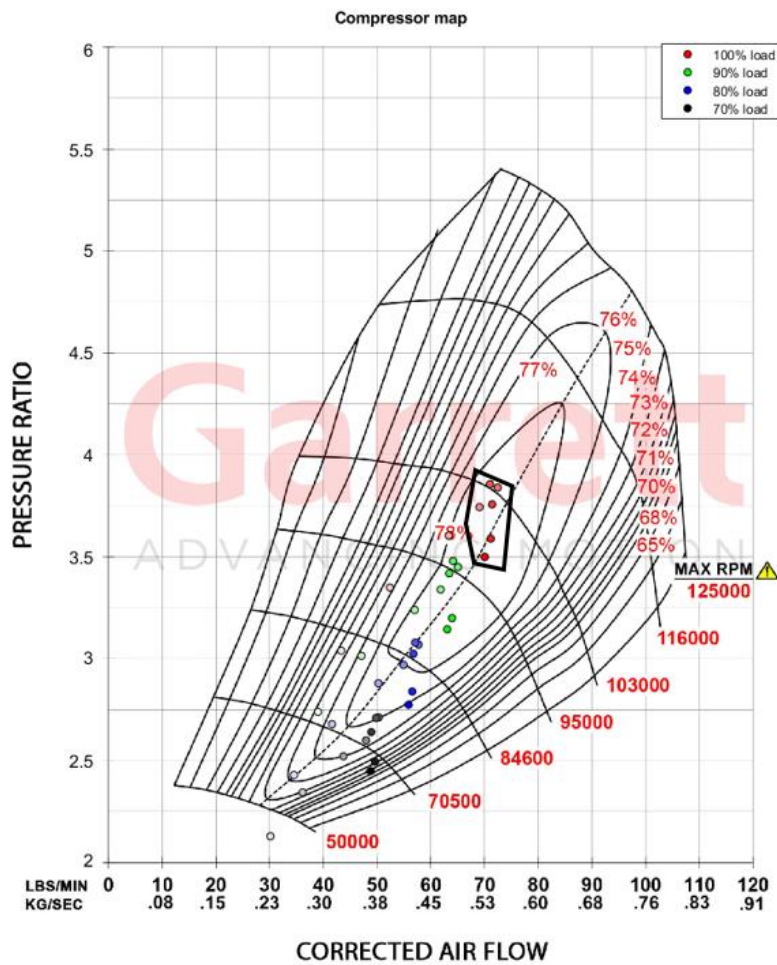


Figure 5-44 – 2014+ Digital Twin compressor operating points vs engine load and speed (Focus on the most used compressor points)

The fact that the 2014+ Formula One™ engine only operates in a narrow engine speed window due to the shape of the fuel flow limit combined with the specific operation of a Formula One™ engine which is mostly used at 100% load leads the compressor to be operating for 65% of a lap time in a very narrow high efficiency zone.

The power recovered by the MGU-H when the engine is operating in CWG mode is directly linked to the compressor need and the turbine recovery. As seen in Figure 5-42, air mass flows and boost pressures reduce with load and so does the compressor power. However, because less fuel is burned and less air is necessary, the exhaust enthalpy is also reduced with engine load.

As shown by Figure 5-45, the MGU-H power follows the same trend. Even at lower load, the turbine produced more power than necessary for the compressor, allowing the MGU-H to recover energy (hence the negative values) but less and less power is available for the MGU-H in consequence of the reduced exhaust enthalpy.

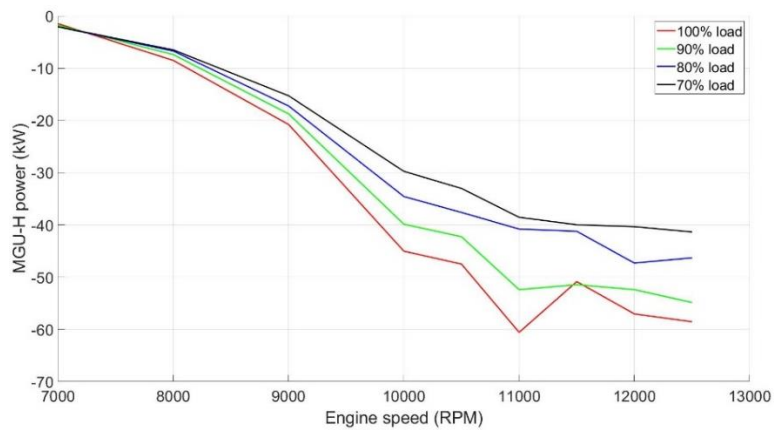


Figure 5-45 - MGU-H power evolution with engine load

When the 2014+ Formula One™ Power Unit switches from CWG to OWG mode, the compressor operating points are different. Air/fuel ratios are the same for both modes, but as volumetric efficiency is significantly improved in OWG mode (see Figure 5-24), lower boost pressures are required as shown in Figure 5-46.

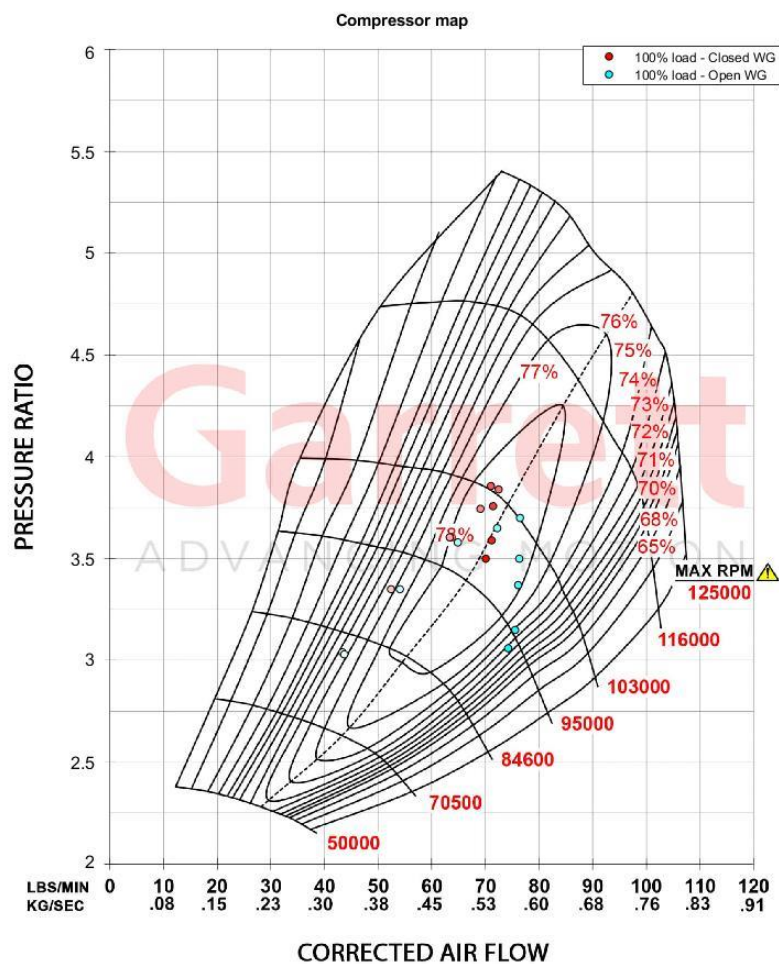


Figure 5-46 – 2014+ Digital Twin compressor operating points at 100% load – CWG (red) vs OWG (light blue) modes

This leads to operating points in slightly less optimal efficiency zone, especially at high speed, but as this mode can only be used for short periods, it seems more appropriate to prioritize high efficiency for the CWG mode.

It can also be noticed that despite similar air/fuel ratios between both modes, the OWG mode requires higher compressor mass flow. As previously explained, thanks to the open waste gates, back pressure is significantly reduced. This improves the engine breathing and leads to higher internal volumetric efficiency. Nevertheless, the engine breathing is so much improved that during the period of valve overlap, where the intake and exhaust valves are open at the same time, part of the high-pressure inlet air goes directly to the exhaust. This phenomenon happens but is limited in the CWG mode because of the higher exhaust pressure. Thus, to match the target air/fuel ratios, the compressor mass flow increases to compensate for the air that goes straight from the intake to the exhaust.

Even though this part relies on compressor points extracted from steady-state simulations which can differ from fully transient operating points, it allows to visualise how the compressor operation changes with load and to demonstrate that in addition to have implications on the engine performances, the fuel flow limit imposed the regulations also leads to a specific turbocharger operation. As the engine operates in a narrow engine speed window and mostly at full load, the compressor operating window is also very narrow, and its behaviour can be optimised for this specific zone.

5.5 NO_x production rate of the 2014+ Formula One™ engine compared to conventional road car engines

As presented in Chapter 2, in addition to fuel conversion efficiency, the future of internal combustion engines is also linked to their capacity to reduce their emissions, of which NO_x is part of.

In case NO_x emissions become regulated in the future, it is interesting to assess how much NO_x is produced by the current Formula One™ engine compared to that of a conventional engine.

The comparison will be done using an AVL Boost™ model of the Honda F20C engine validated against experimental data for torque [5]. Despite the NO_x emissions validation done by AVL for Boost™ for a given engine at different operating points (see Chapter 3, [151]), the NO_x production values predicted by the software for both the 2014+ Digital Twin and the F20C are not taken as exact considering the lack of validation data for these specific engines, but are used for comparing order of magnitude.

The absolute and specific NO_x productions of both engines are respectively shown in Figure 5-47 and Figure 5-48:

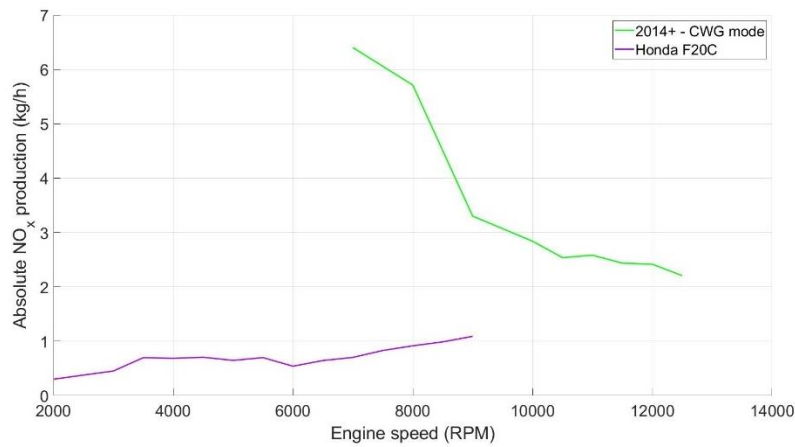


Figure 5-47 – NO_x production – 2014+ Digital Twin and Honda F20C comparison

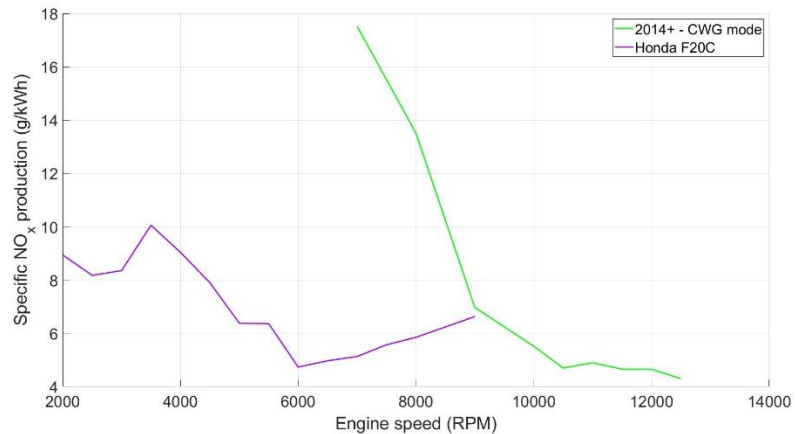


Figure 5-48 - Specific NO_x production – 2014+ Digital Twin and Honda F20C comparison

As shown by Figure 5-47, it can be noticed that the 2014+ Digital Twin predicts that the 2014+ Formula One™ engine produces in average 4.4 times more NO_x than a conventional engine. However, the 2014+ Digital Twin operates with, in average, 5 times more air flow and produces 3 times more power as it operates with 2 times more fuel flow and at high dilution level whereas the F20C operates at stoichiometry or below.

As the Formula One™ engine is a high-performance engine, it burns large amount of fuel, requiring larger air mass flow that a conventional engine. The higher the air mass flow, the higher the NO_x production as there are more N₂ and O₂ that can react together.

However, by normalising the NO_x production by the power produced by the engine to calculate the specific NO_x production as shown in Figure 5-48, it can be highlighted that the 2014+ Formula One™ Digital Twin predicts that the engine produces similar amount of NO_x to that of the F20C relatively to their power output, especially in the range the engine is the most used, i.e. above 9,000 RPM.

The 2014+ Formula One™ engine produces large amount of NO_x because of its high air mass flow. Nevertheless, thanks to its lean operation which also leads to higher fuel conversion efficiency, the in-cylinder temperatures are reduced leading to less NO_x production than if operated at stoichiometric air/fuel ratios. What Figure 5-48 suggests it that if the Formula One™ engine was scaled down to produce similar power output to that of the F20C, it would also produce similar amount of NO_x.

Nevertheless, the big differences between a Formula One™ engine and a conventional road engine is after-treatment systems. The latter can be fitted with a range of different after-treatment systems depending on the engine type to transform NO_x into less harmful molecules via chemical reactions.

On the contrary, the Formula One™ engine emits directly into the atmosphere the NO_x it produces. After-treatment systems are not a really viable solution for Formula One™ due to the increase weight and the additional back pressure created by such system which limits the engine efficiency as it compromises its breathing. For Formula One™ engines, the solution to reduce NO_x must come from the combustion directly.

The NO_x formation mechanism being highly influenced by temperature to initiate dioxygen dissociation to start the reaction process, the combustion temperature must be reduced to decrease NO_x but without compromising combustion and thus fuel conversion efficiency.

5.6 Conclusions on the insights brought by the 2014+ Digital Twin

By removing its special features (high compression ratio, high dilution, pre-chamber ignition system, direct injection, asymmetric valve profile, MGU-H, unconventional waste gate use) from the 2014+ Digital Twin to obtain a baseline engine model and then progressively adding them back until the 2014+ Digital Twin was built back up, this chapter explained from a scientific point of view the impact of each solution on performance and the thinking behind their adoption by the engine manufacturers. The pre-chamber ignition system is very important as it allows reliable and rapid lean combustion, but the enabling technology is direct injection.

Two engine operating modes are proposed in this chapter with different waste gate use, either closed or open. Both modes are extremes and showed what is possible if recharging or maximum performance are the main objectives. These modes also demonstrate the importance of the engine being part of a wider system, the Power Unit. This can either be used to improve engine efficiency further or the engine can be used to contribute maintaining the energy storage system state of charge high.

The analysis of the 2014+ Formula One™ Digital Twin was then pushed further by investigating its behaviour from the second law of thermodynamics, i.e.

exergy. Despite resulting in more entropy creation, the lean mixture used by this engine allows the engine to operate at high pressure and low temperature resulting in less exergy loss under the form of heat and a higher fraction of the fuel exergy being converted into useful work.

It was shown that the fuel flow shape imposed by the regulations has several impacts on the engine operation. Engine peak power is achieved around 10,500 RPM, speed at which the fuel flow limit is reached, before progressively decreasing because of power output being limited by the limited fuel flow, increasing friction losses and decreasing air/fuel ratio to maintain combustion stability at high speed. This results, in addition to the 8-gear gearbox and to the specific use of a Formula One™ engine (most of the time at 0 or 100% load), in the engine being for most of a lap used in a very narrow engine speed window located around peak power.

Accordingly, the compressor spends most of the lap in a narrow zone of its map which can be optimised to correspond to the highest efficient zone of the compressor map.

Finally, it was shown that despite the 2014+ Formula One™ engine producing large amount of NO_x compared to a conventional gasoline engine, if power output is considered, the specific NO_x production of both engines is similar thanks to the high dilution used in the 2014+ Formula One™ engine which limits in-cylinder temperature and thus O₂ dissociation.

6 Assessment of the impact of the 2026 Formula One™ Power Unit regulations on engine performance

As explained in Chapter 1, new Formula One™ Power Unit will be introduced in 2026 [15] with an unchanged engine architecture but the MGU-H being removed, the MGU-K power being raised from 120 to 350 kW, a reduced fuel flow limit now expressed in terms of energy rather than mass and advanced sustainable fuels.

The aim of this Chapter is to assess the impact on engine performance of these new regulations and investigate how these changes can bring opportunities for the engine manufacturers to modify the behaviour of their engine and further improve fuel conversion efficiency of these fuel-limited engines.

6.1 Evaluation of the performance differences between the 2014+ and 2026 Formula One™ Power Unit Regulations

To enable easy assessment of the differences between the 2014+ and 2026 Formula One™ regulations, both steady-state models presented in Chapter 4, and whose detailed input dataset are described in Appendix C, are used. As the 2026 engines are currently under development, the information about their geometry, operating conditions or output performance are not yet known. Chapter 4 discusses the choices that had to be made regarding the 2026 engine model input dataset and this chapter aims at giving a look at engine performance prediction. The term 'Digital Twin' is not used for the 2026 engine model as it did not benefit from any input or feedback from the industry regarding its input dataset or performance predictions considering that the 2026 Formula One™ engine is currently being developed by the Power Unit manufacturers.

The performance of the 2026 Formula One™ engine model will be compared to that of the 2014+ Digital Twin in CWG (Closed Waste Gate) mode. This is the most efficient mode and is more representative of the 2014+ Formula One™ engine operation as the OWG mode is only usable for short periods. Unless explicitly mentioned, the comparison between the 2026 model and the 2014+ Digital Twin is done with the CWG recharge mode.

For the engine operation in steady-state conditions, the biggest difference between both regulations comes from the fuel flow limit. As stated, the fuel flow limit went from a mass flow limit to an energy flow limit. Fuel suppliers have some liberty regarding their fuel energy density but the regulations state lower heating value Q_{lvh} must be between 38 and 41 MJ/kg. As maximising this value will result in less fuel to carry onboard, 41 MJ/kg is the value selected for the 2026 engine model.

Using the information for the 2014+ regulations and that of the 2026, the fuel flows can be compared (Figure 6-1):

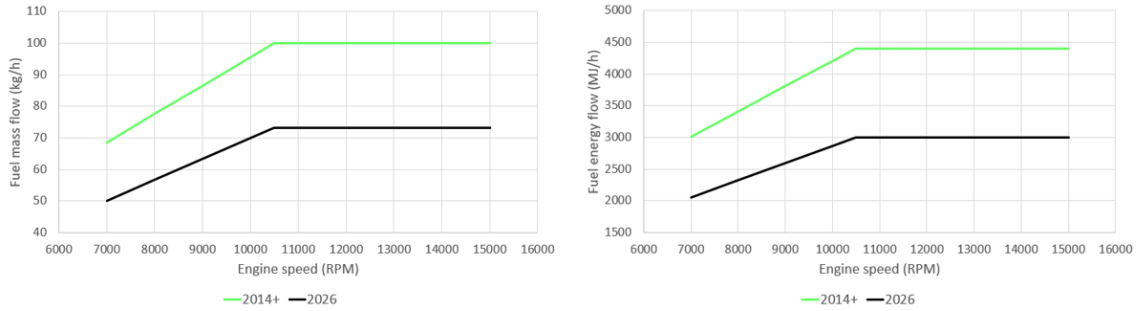


Figure 6-1 - Fuel mass and energy flow - 2014+ vs 2026 comparison

The fuel energy flow limit of 3000 MJ/h with a 41 MJ/kg Q_{lvh} results in a significant reduction in fuel mass and energy flows compared to that of the 2014+ engine. The mass flow is reduced by 26.8% and the energy flow by 31.8%. Only considering this factor, if the fuel conversion efficiency of the 2026 engine is kept at the same level to that of the 2014+ Digital Twin, the engine output performance will inevitably be reduced by a similar order.

As a first approach, both engines are simulated at the same air/fuel ratios to allow the analysis of the fuel flow reduction effect on the engine behaviour (Figure 6-2).

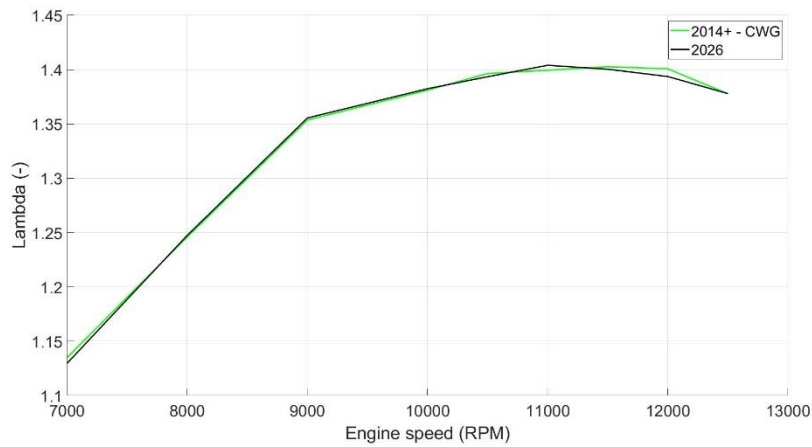


Figure 6-2 - Lambda evolution with engine speed - 2014+ vs 2026 comparison

Figure 6-3 shows the performance evolution between both sets of regulations. As expected with the reduced fuel flow mass, output torque and power are significantly reduced. In average, BMEP, torque and power have decreased by 31.3% which corresponds to the reduction in fuel energy flow. However, by considering the maximum brake power possible with 2014+ Digital Twin in CWG power mode, the decrease reaches 35.9% as the 2026 engine is not capable of exploiting the exhaust enthalpy to directly increase the engine power output (not MGU-H and thus no energy link between the MGU-H and MGU-K) with its conventional turbocharger.

As the geometry of the engine is unchanged, i.e. swept volume, camshafts, combustion chamber (bore and stroke) and effective compression ratio are the

same for both engine models, the reduction in BMEP for 2026 demonstrates that less power is produced from the same volume and that the 2026 engine is less stressed than its 2014+ counterpart. The engine is now closer to more conventional high-performance road car engine BMEP level. For instance, Honda's K20C1 engine featured in the Honda Civic Type R yields 21.75 bar BMEP [174].

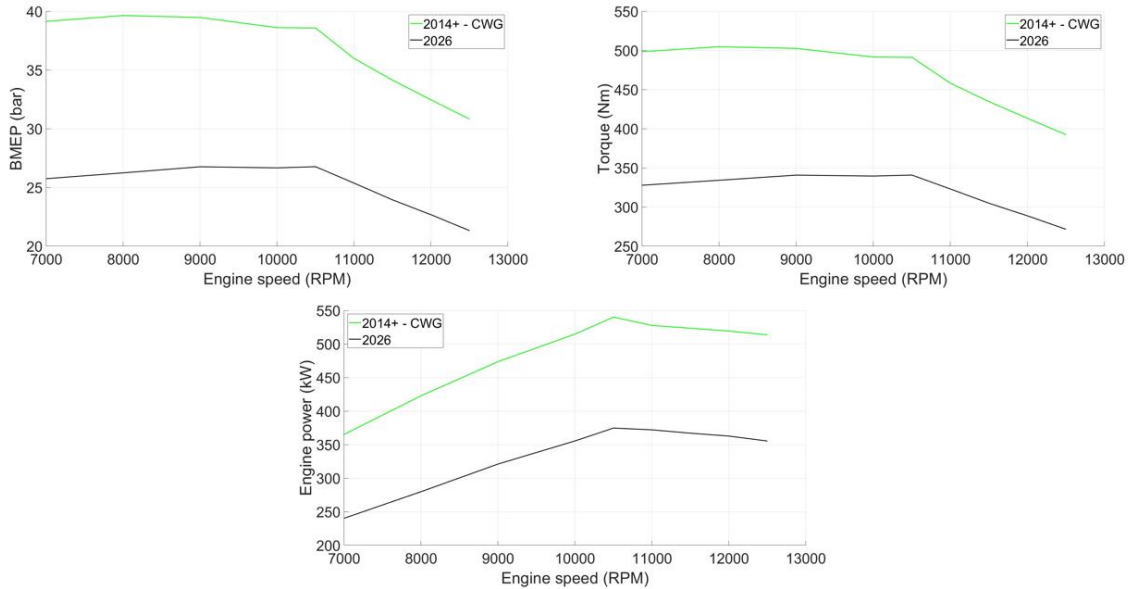


Figure 6-3 – BMEP (top left), torque (top right) and power (bottom) curves – 2014+ vs 2026 comparison

As for the 2014+ Formula One™ engine, the constant fuel flow limit imposed from 10,500 RPM limits the power production of the 2026 Formula One™ engine and results in a decrease as higher engine speeds are reached because of the higher friction losses.

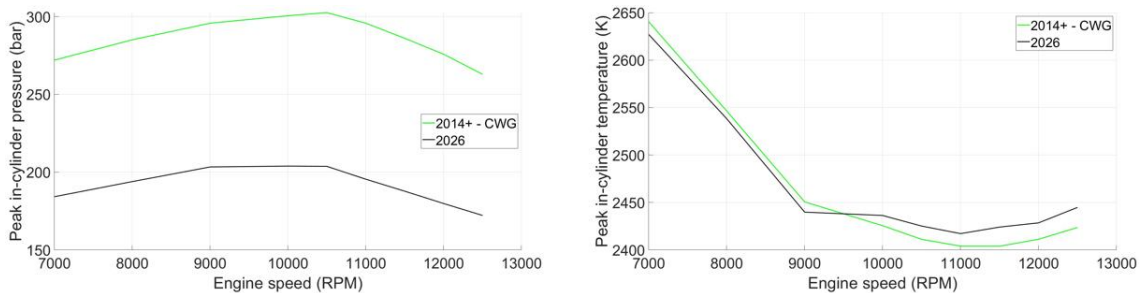


Figure 6-4 – Peak in-cylinder pressure and temperature curves – 2014+ vs 2026 comparison

Figure 6-4 further demonstrates that the stress on the 2026 engine has been reduced compared to that of the 2014+ engine. As both engines are operating at similar level of dilution and as their fuels are close in composition (2026 fuel: 80% 2014+ fuel + 20% ethanol), the peak in-cylinder temperatures are close. However, again, due to the reduction in fuel mass flow, the peak in-cylinder pressures are reduced by about 30%, going from 300 to 205 bar. This is a very significant change.

As explained in Chapter 5, the optimisation of the combustion process for the 2014+ Formula One™ engine is constrained by the engine being pressure limited. For the 2026 Formula One™ regulations, as the pressures are greatly reduced, there is room for the combustion to be pushed further as an increase in in-cylinder pressure will not result in the engine exceeding its material mechanical limits. This opens a lot of possibilities for engine manufacturers.

In terms of effective efficiency (Figure 6-5), both engines operate in the same range with values varying between 42 and 45%. Peak efficiencies are achieved at different engine speeds, but it must be remembered that the 2014+ Digital Twin breathing capacity in CWG mode is limited due to high exhaust back pressure. However, on the contrary to the 2014+ Digital Twin, the 2026 model cannot benefit from the efficiency boost linked to the use of an MGU-H (47.9 peak effective efficiency for the 2014+ Digital Twin in CWG power mode).

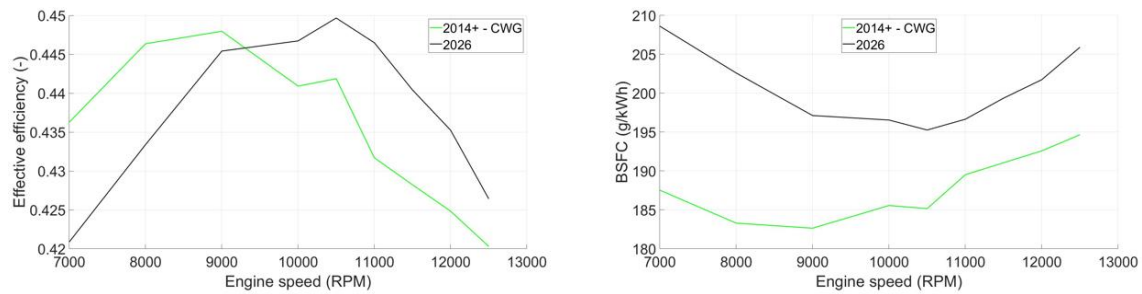


Figure 6-5 - Effective efficiency and BSFC curves - 2014+ vs 2026 comparison

As shown in Figure 6-6, the internal volumetric efficiency and pressure ratio across the engine are much more favourable for the 2026 engine than for the 2014+ Digital Twin. This is due to the different operation of the waste gates between both engines. As it features a conventional turbocharger, the 2026 Formula One™ turbine swallowing capacity is not selected to be able to swallow the whole engine mass flow but is chosen to prioritise torque at median engine speed and reduce turbo lag meaning that its waste gates are used to divert a fraction of the exhaust flow at higher engine speed to prevent back pressure build up.

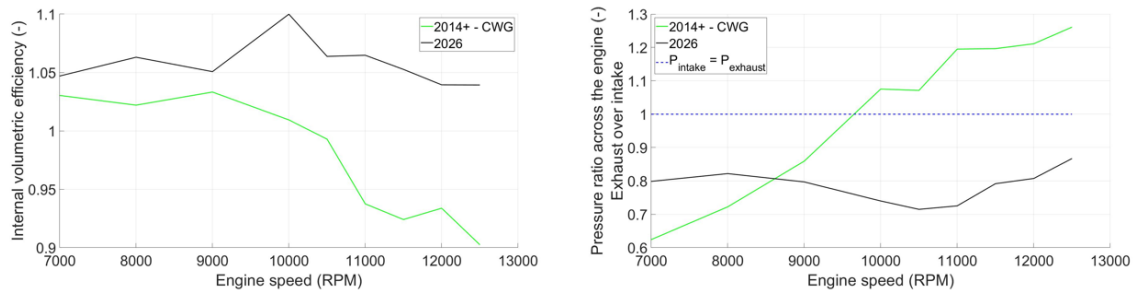


Figure 6-6 - Internal volumetric efficiency and pressure ratio across the engine curves - 2014+ vs 2026 comparison

If compared to the OWG mode of the 2014+ Digital Twin which is not limited by back pressure, both engines achieve peak efficiency at 10,500 RPM, i.e. when the fuel flow limit is reached. Nevertheless, the OWG mode of the 2014+ Digital

Twin yields a higher efficiency (47.3% maximum) thanks to higher in-cylinder pressure and even lower pumping losses because of the lower back pressure.

With the presence of the MGU-H in the 2014+ Digital Twin, extreme turbocharger operating modes are possible (OWG and CWG) and lead to extreme values of exhaust back pressure, respectively very low and very high. The removal of the MGU-H for the 2026 Formula One™ regulations means that the conventional turbocharger is used with a conventional wate gate operation. By doing so, the pressure ratio across the 2026 engine model is relatively constant and in between that of the two modes of the 2014+ Digital Twin.

Despite similar effective efficiency levels (as the 2014+ Digital Twin CWG recharge mode is considered), the 2026 engine's BSFC is higher than that of the 2014+ Digital Twin in CWG mode (Figure 6-5). This difference does not result from a difference in the engine operation but a difference in fuel properties. As the 2026 engine model is modelled with a 41 MJ/kg fuel lower heating value against 44 MJ/kg for the 2014+ Digital Twin, similar efficiency results in higher BSFC as the fuel contains less energy.

As the 2026 engine model is simulated at the same dilution level as the 2014+ Digital Twin and because of the fuel flow reduction, the required air mass flow is also reduced as demonstrated in Figure 6-7. Combined with the internal volumetric efficiency being better than that of the 2014+ Digital Twin (Figure 6-6), lower boost pressures are required from the compressor to reach the target air/fuel ratios.

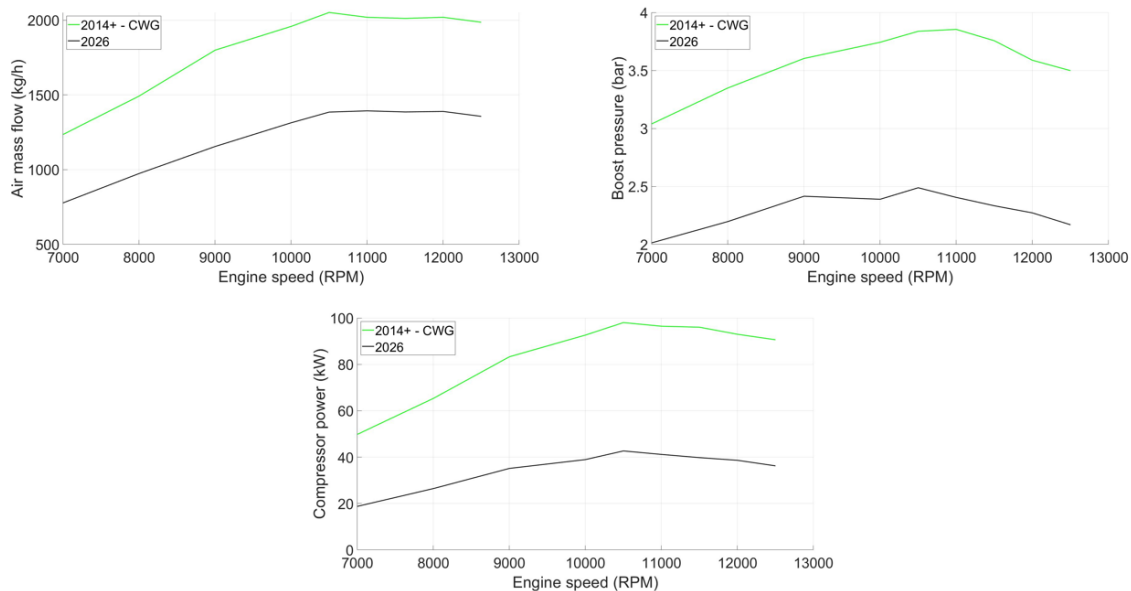


Figure 6-7 - Air mass flow (top left), boost pressure (top right) and compressor power (bottom) curves - 2014+ vs 2026 comparison

As a consequence, the power required by the compressor to supply the engine with the right air mass flow at the target pressure is significantly reduced compared to the 2014+ Digital Twin (-58.7%).

Regarding boost pressure, the 2026 Formula One™ regulations [15] impose a maximum engine inlet air pressure of 4.8 bar absolute. This limit is very different from the predictions made by the AVL Boost™ engine model created for this project (Figure 6-7). The engine is very unlikely to be so significantly different to the 2026 model presented here that an additional 2 bar boost pressure is required for its operation. Considering the reduced fuel flow, that would lead to air/fuel ratios way beyond the operating limit of a passive pre-chamber and even with a low volumetric efficiency, this level of boost during normal operation is unlikely for steady state operation.

However, the transient operation of the engine might lead to the turbocharger producing higher pressure, thus maybe justifying the boost pressure limit imposed by the regulations.

Compared to the 2014+ Formula One™ engine, a smaller turbine will be allowable for the 2026 regulations due to the different turbocharger philosophy. As the 2026 Formula One™ engine does not feature an MGU-H to absorb excess turbine power, the compressor and turbine powers must be balanced, and the turbine is not required to be able to swallow the whole engine mass flow. The objective for the 2026 turbine is to have the smallest turbine capable of supplying enough power for the compressor to minimise the turbocharger rotor moment of inertia and thus turbo lag which will be very important for on-track performances. Nevertheless, the 2026 Formula One™ regulations set a window for the different dimensions of the turbocharger rotor (wheel and inducer diameters and maximum distance between compressor and turbine)

In addition, as shown in Figure 6-6, the opening of the waste gates at high engine speeds will limit back pressure build up and improve engine breathing.

The reduction in fuel flow also influences engine heat rejection (Figure 6-8). As previously observed with Woschini heat transfer coefficient model (Equation 5-6), the reduced in-cylinder pressures observed in Figure 6-4 result in lower heat transfer coefficient. Combined with the fact that less energy is released during the combustion due to the reduced fuel flow, the cooling losses of the engine are also significantly reduced.

The operation of compressor for the 2026 Formula One™ regulations demands lower air mass flow and lower boost pressures. At equivalent compressor efficiency, the compressor outlet temperature will be lower for the 2026 engine than that of the 2014+ Digital Twin due to the lower boost requirements. Combined with lower air mass flow, the intercooler load is reduced by more than half (-60.6%).

When compared with that of the 2014+ Digital Twin, the 2026 engine model cooling load (engine + intercooler) is significantly reduced. A reduction in cooling losses is an advantage as it results in less air mass flow requirements for the different heat exchangers placed on the car and thus less drag.

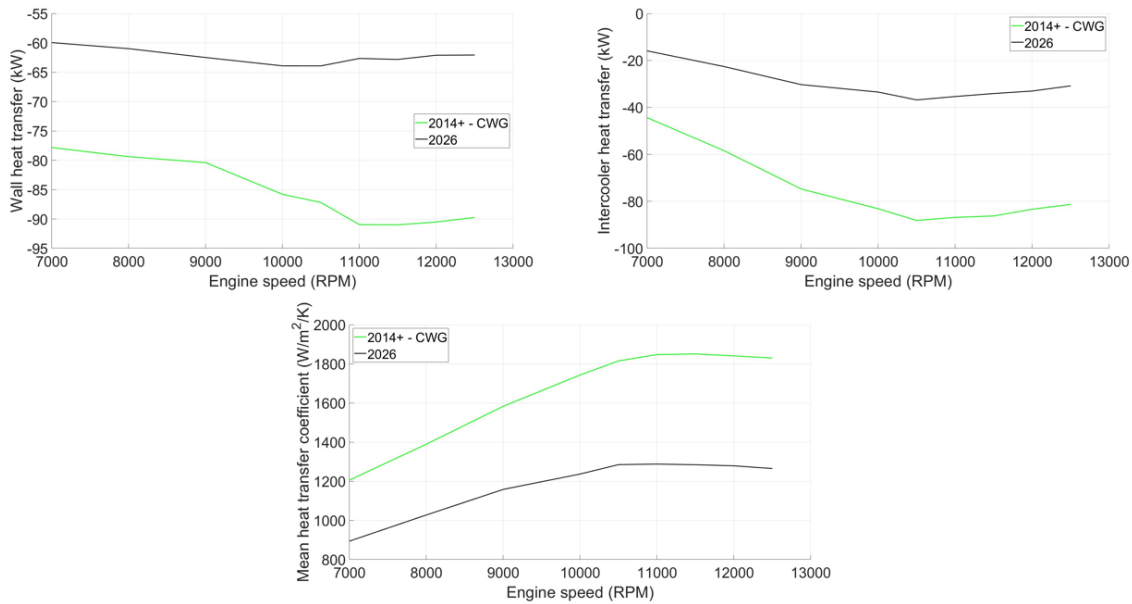


Figure 6-8 - Cooling losses (top left), intercooler load (top right) and combustion chamber wall heat transfer coefficient (bottom) curves - 2014+ vs 2026 comparison

Nevertheless, it must be highlighted that this research project only focuses on the internal combustion engine. As the MGU-K power and Energy Storage size will be increased in the 2026 Formula One™, their cooling needs will also go up. However, considering the efficiency of electric systems, which is much higher than that of an ICE, it can be assumed that the total cooling needs of the 2026 Formula One™ Power Unit will decrease. As stated previously, this leads to less drag but also a lighter cooling system.

As discussed in Chapter 5, the NO_x production of an engine is directly related to its air mass flow. As the air mass flow will be reduced, the absolute NO_x production of the 2026 Formula One™ is also expected to decrease, as suggested by Figure 6-9.

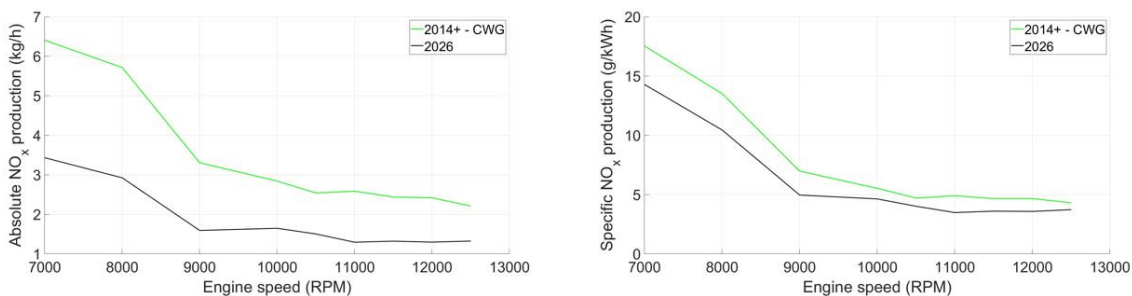


Figure 6-9 - Absolute and specific NO_x production - 2014+ vs 2026 comparison

As observed in Figure 6-4, the fact that both engines are operating at similar air/fuel ratios results in similar in-cylinder temperatures. Because NO_x production is significantly influenced by temperature, when the specific NO_x production is considered, both the 2014+ and 2026 engine models produce similar specific emission levels.

As seen in Figure 6-3, the fuel flow reduction leads to reduced output performance for the 2026 Formula One™ engine model. To compensate for this loss of thermal power, the output MGU-K power has nearly been tripled compared to the 2014+ Formula One™ regulations (120 vs 350 kW). Figure 6-10 shows the total power unit power and torque (ICE + MGU-K) for both regulations in their maximum Power Unit performance mode.

For the 2014+ Digital Twin, this corresponds to the OWG mode plus the 120 kW MGU-K whereas for the 2026 engine model, it is represented by both the engine and the MGU-K being operated at full load.

Due to its very high power (350 kW), because all the electric energy used by the Power Unit comes from regeneration only (i.e. indirectly from fuel energy) and because the amount of energy available around a lap is reduced, the 2026 MGU-K will have to be used carefully to ensure that the ES state of charge is kept at a reasonable level, ensuring that it can be used when it matters the most (at the beginning of an acceleration, in the limit of tyre grip), throughout the lap / race.

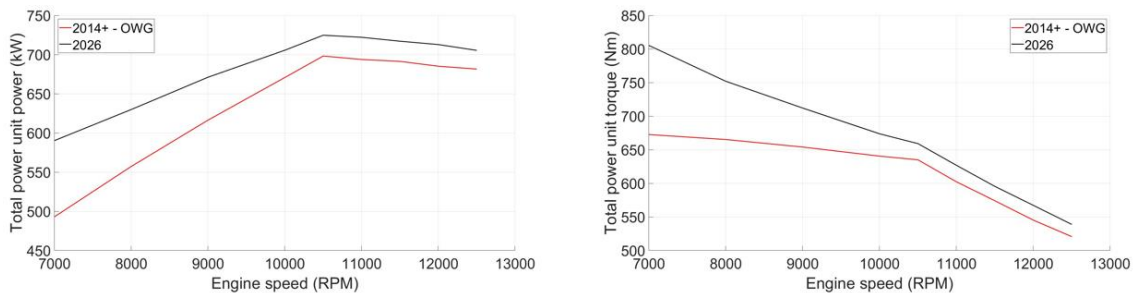


Figure 6-10 - Total power unit power and torque curves (ICE + MGU-K) - 2014+ vs 2026 comparison

This is why the 2026 PU operating at full load on both the engine and MGU-K is not believed to be a sustainable mode for a whole race but on the contrary is a maximum performance mode as is the OWG mode for the 2014+ Digital Twin.

According to the 2014+ Digital Twin and 2026 model predictions, the total Power Unit maximum performances for the 2026 Formula One™ regulations will be higher than that of the 2014+ (724.7 vs 698.1 kW).

The most significant difference comes for the low engine speed performance thanks to the higher MGU-K torque and power available at low speeds.

Regarding peak power, the difference can be explained by the engine models predicting an ICE power reduction of 203 kW because of the 31.8% fuel energy flow reduction, whereas the MGU-K power increases by 230 kW.

As for the 2014+ Digital Twin, energy and exergy analyses were conducted on the 2026 engine model, average results are shown in Figure 6-11 and Figure 6-13.

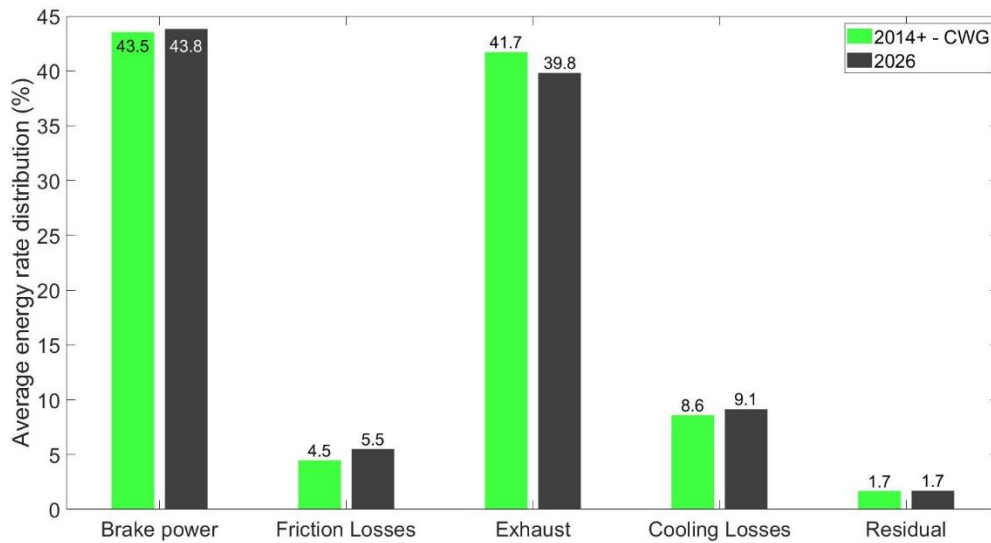


Figure 6-11 - Average energy distribution - 2014+ vs 2026 comparison

The first thing that can be noticed is that both engines have similar distributions which is not very surprising considering that they both are fuel-limited engines and are running at the same air/fuel ratios. Nevertheless, as shown in Figure 6-12 for 10,500 RPM, when the energy distribution is looked at in terms of power rather than percentage of fuel energy, the values referring to the future engine generation are lower.

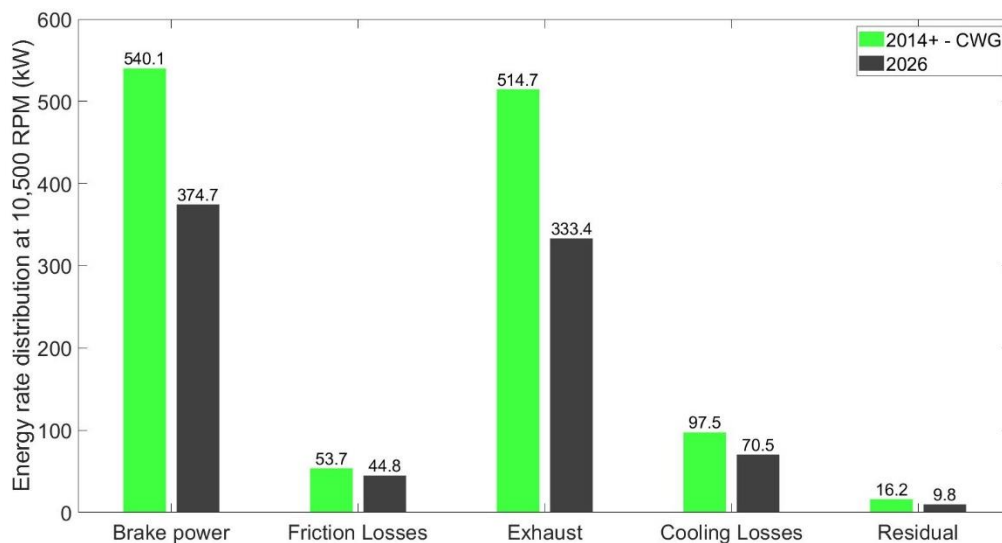


Figure 6-12 - Energy distribution at 10,500 RPM - 2014+ vs 2026 comparison

Focusing on the energy distribution, the 2026 engine is slightly more efficient (+0.3 percentage point) on average than its 2014+ counterpart. As seen in Figure 6-5, the effective efficiency is lower at low engine speed but is better at high speed thanks to its better breathing capacity resulting in this small average efficiency improvement.

The friction losses of the 2026 engine accounts for a higher percentage of the fuel energy suggesting that, with the friction model employed for these models,

friction predictions do not evolve linearly with power output. The 2026 engine shares the same geometry as the 2014+ Digital Twin for which there are inherent friction losses, which represent a fraction of total friction losses and do not depend on the engine operation, just on geometry. Thanks to lower in-cylinder pressures, total friction losses (inherent + operational) reduce (-16.6% at 10,500 RPM) but the reduction is smaller than that of the fuel energy flow (-31.8%) due to the inherent friction losses, leading to total friction losses corresponding to a higher fraction of the fuel energy.

Similarly, despite reduced cooling losses (Figure 6-8 and Figure 6-12), a higher proportion of the fuel energy goes into the cooling system for the 2026 engine compared to the 2014+ Digital Twin. Again, the engine geometry is the same, so the heat transfer surface area is unchanged. Because both engines operate at similar dilution level, the temperature difference between the mixture and the combustion chamber wall is almost unchanged. The difference thus comes from the evolution in heat transfer coefficient (Equation 6-1: Woschini heat transfer coefficient model, previously shown as Equations 2-1, 3-4 and 5-6).

$$h_c = 3.2 \times B^{-0.2} \times p^{0.8} \times T^{-0.55} \times w^{0.8} \quad (6-1)$$

Between the two engines, all parameters of Equation 6-1 are unchanged except for the pressure. The peak in-cylinder pressure (Figure 6-4) reduces by 33.1% on average but because of the 0.8 exponent in Woschini model, the heat transfer coefficients only decrease by 28.8% which explains why the heat losses represent a higher fraction of the fuel energy for the 2026 engine model than for the 2014+ Digital Twin.

As the residual energy is the same for both models (Figure 6-11) and because of the discussed variation on power, cooling losses and friction, the amount of energy going into the exhaust gases is reduced for the 2026 engine.

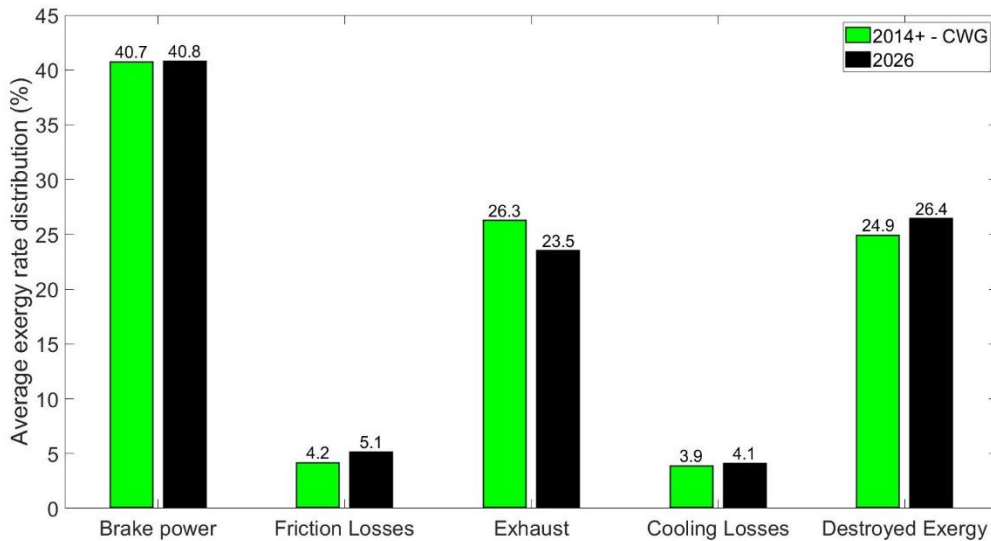


Figure 6-13 - Average exergy distribution - 2014+ vs 2026 comparison

Regarding the exergy distribution (Figure 6-13), the observations are very similar to that of the energy analysis. It can be highlighted that the 2026 engine model destroys a higher fraction of the fuel exergy. First, for this exergy analysis, it was considered that the fraction of exhaust flow diverted from the turbine (using the turbine to total parameter presented in Chapter 4) is destroyed exergy as it will not go through the turbine to be recovered and second, as discussed previously for the energy analysis, most of the losses have less decreased than the fuel exergy thus representing a higher percentage.

Table 6-1 shows a summary of the relative evolution between the 2014+ and 2026 Formula One™ regulations:

Parameters	Average relative evolution between the 2014+ (2014+ Digital Twin in CWG mode) and 2026 Formula One™ regulations (%)
Fuel mass flow	-26.8
Fuel energy flow	-31.8
Air mass flow	-33.1
Engine power	-31.3 (in CWG recharge mode) -35.9 (in CWG power mode)
Total PU peak power	+3.8
BMEP	-31.3 (in CWG recharge mode) -35.9 (in CWG power mode)
FMEP	-8.7
Effective efficiency	+0.7 (in CWG recharge mode) -6.0 (in CWG power mode)
BSFC	+6.6
Engine cooling losses	-27.6
Intercooler load	-60.6
Exhaust enthalpy	-34.9
Internal volumetric efficiency	+8.6
Pressure ratio across the engine	-18.6
Boost pressure	-35.8
Compressor power	-58.7
Peak in-cylinder pressure	-33.1

Parameters	Average relative evolution between the 2014+ (2014+ Digital Twin in CWG mode) and 2026 Formula One™ regulations (%)
Absolute NO _x production	-45.8
Specific NO _x production	-21.1
Average destroyed exergy fraction	+6.0

Table 6-1 - Average relative evolution between the 2014+ and 2026 Formula One™ regulations

6.2 Effect of combustion characteristic modifications on engine performance and assessment of the most effective parameter to increase fuel conversion efficiency

As seen in Figure 6-4 of Section 6.1, the in-cylinder pressures of the 2026 Formula One™ engine are significantly reduced compared to that of the 2014+ Digital Twin, thus the engine is not pressure-limited any more.

As discussed in Chapter 5, the fact that the 2014+ Formula One engine is running at its mechanical limits, 300 bar is reached (when the connecting rod deformation is not accounted for), means that the combustion cannot be pushed further at the risk of increasing even more the peak in-cylinder pressure and exceeding reliability limits. But as the 2026 Formula One™ will not face the same limit, opportunities to enhance the fuel energy conversion into work via the combustion are created.

It was observed in Chapter 5 that increasing compression ratio is an effective way to increase in-cylinder pressures and efficiency. Nevertheless, the 2026 engine proposed here is already running with a 16:1 compression ratio which is the maximum allowed by the 2026 Formula One™ Power Unit regulations [15].

The fuels used for the 2026 Formula One™ regulations will be fully engineered meaning that chemical engineers will try to adapt the composition of their fuel to influence the combustion process in a specific way. If using Ansys Chemkin for the investigations of interests for this Thesis had been possible, looking at the influence of different fuel composition on combustion properties would have been one of the objectives.

However, using AVL Boost™, it is possible to assess how to improve the fuel conversion process by modifying key parameters, namely start of combustion (also referred to as combustion advance), combustion duration and air/fuel ratio.

The idea behind the study presented in this section is to quantify the impact of the aforementioned parameters on the 2026 Formula One™ engine performances. By doing so, it will be possible to identify the main levers to improve performances and emissions without modifying the engine geometry and assess the potential avenues that engine manufacturers might follow to improve

their engine efficiency by taking advantages of the reduction in peak in-cylinder pressure. In addition, the study will highlight the trade-offs required to reach higher fuel conversion efficiency.

To quantify the impact of each parameter, three sets of simulations were run, in which only one parameter is changed at the time. The values investigated for all three parameters are:

- Start of combustion (SoC): [-10, -5, -3, -1, 5] °CA ATDC,
- Combustion duration (DOC): [30, 35, 41, 45, 50] °CA,
- Lambda evolution: [-0.1, -0.05, 0, +0.05, +0.1] compared to the lambda evolution shown in Figure 6-2. The shape of the rate-of-heat-release (ROHR) curve is kept constant for all cases.

Based on simulations run with these settings, the average impacts over all engine speeds on key performance indications of reference changes (+1 °CA combustion advance, -1 °CA combustion duration, +0.05 lambda evolution) are calculated and reported in Figure 6-14. As in Chapter 3, the results are presented using a spider diagram. Some axes are reversed, the green zone showing an improvement when the studied changes are applied and the contrary for the red zone.

The performance indicators shown in Figure 6-14 will be discussed clockwise.

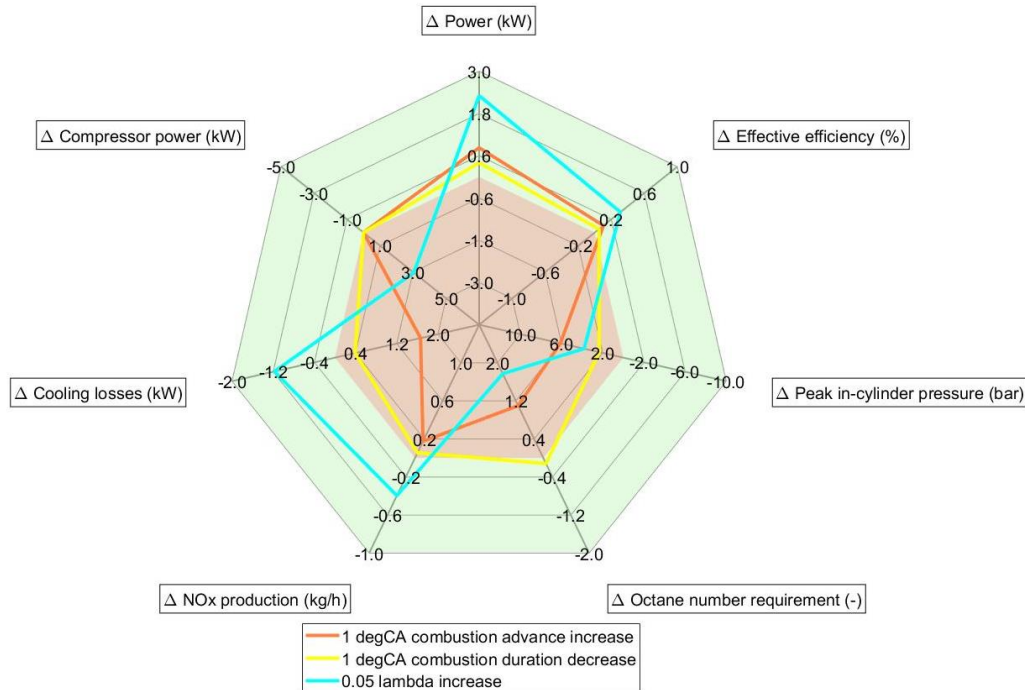


Figure 6-14 - Average impact of combustion advance, combustion duration and lambda evolution on the 2026 Formula One™ engine performances

As shown by the power and effective efficiency axes, all three reference modifications lead to improved output performances which is not surprising as they have been chosen to do so.

Increasing lambda is the modification that yields the best improvement, 2.8 and 5.7 times greater for the power increase than that of the combustion advance increase and combustion duration decrease respectively.

Nevertheless, the lambda increase case has only the second highest peak in-cylinder pressure increase. All modifications take advantage of the reduction in in-cylinder pressure between the 2014+ and 2026 Formula One™ regulations but this shows that the efficiency improvement is not only linked to the pressure change.

Regarding the predictions for octane number requirements, increased combustion advance and lambda evolution result in higher required knock resistance. The higher dilution needs the highest increased in octane number as the charge is also exposed to higher pressure during the combustion stroke whereas higher pressures are only faced during the combustion event when the combustion advance is increased. On the contrary to the two other parameters, the reduction in combustion duration results in lower octane number need as the charge is burnt faster, i.e. the unburned mixture is exposed to spontaneous ignition prone conditions for a shorter period.

As observed in Chapter 5, increasing mixture dilution helps to keep in-cylinder temperature under control. This explains why the NO_x production of the 0.05 lambda increase case is reduced meanwhile that of the two other cases are increased due to higher temperatures reached during the combustion event.

It was previously mentioned when looking at peak in-cylinder pressure that efficiency increase is not only linked to higher pressures, and this is because cooling losses are also of importance. As observed in Chapter 5 when conducting exergy analysis, higher air/fuel ratios result in higher exergy destruction during combustion but overall yield higher efficiency because of reduced losses to the cooling systems or to the exhaust, which is also shown in Figure 6-14.

By increasing combustion advance or decreasing combustion duration, in-cylinder temperatures are raised which, as demonstrated by Dunbar et al. [43], leads to lower exergy destruction during the combustion event as the heat transfer between the species within the combustion chamber occurs at higher temperature but the benefits of this non exergy destruction is balanced by the higher heat transfer to the coolant and the higher exhaust enthalpy.

Finally, to enhance effective efficiency, the increased dilution case requires more air to be trapped in the cylinders, i.e. higher boost pressures, which increases the compressor power demand that must be met by the turbine. In addition, the load on the intercooler will be increased by 2.7 kW on average, due to the higher mass flow, which outweighs the reduction in cooling losses. The two other investigated changes operate at unchanged air/fuel ratios, so the compressor power is also constant.

An exergy analysis was carried out on engine models featuring the modifications studied in Figure 6-14. The results of this analysis are shown in Figure 6-15.

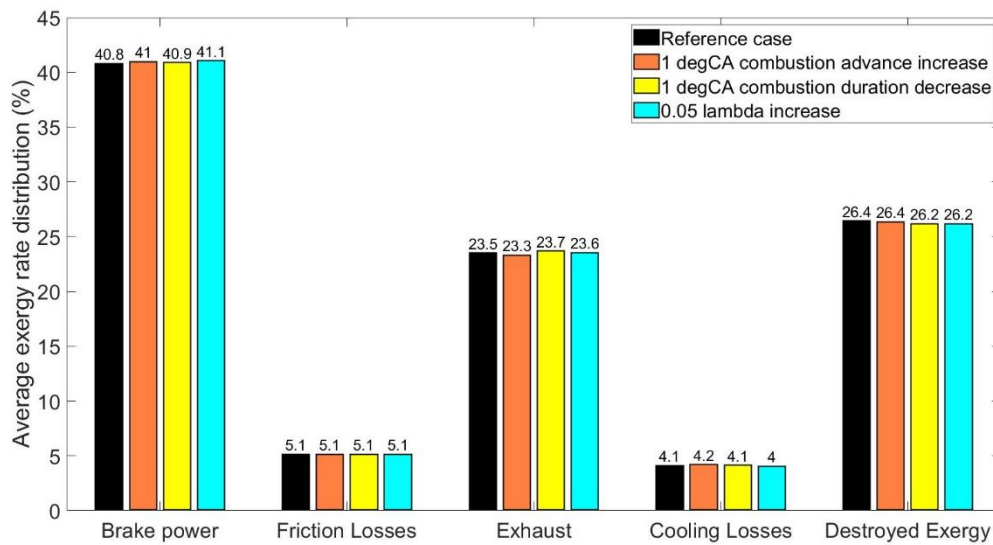


Figure 6-15 - Average exergy repartition of reference combustion advance, combustion duration and lambda evolution modifications compared to the reference 2026 engine model

All modified cases are close to the reference case which is unsurprising considering that they only result in small efficiency improvements as shown at the “brake power” abscissa. Due to the small in-cylinder pressure changes, the friction losses are the same for all cases. Thanks to the lower in-cylinder temperature, the increased lambda evolution case loses less exergy to the cooling system but slightly more to the exhaust compared to the reference case because of the higher mass flow. Despite the higher entropy creation during combustion, this case yields the lower overall exergy destruction thanks to the lower temperatures which limit heat transfer and promotes work production.

The modifications on combustion duration and advance do not result in as large an improvement because of higher in-cylinder temperatures which promotes heat transfer and losses to the exhaust.

However, the reference modifications studied in Figure 6-14 do not yield the same performance improvement making difficult to quantify their effectiveness in increasing efficiency. Thus, to complete the study, the results from Figure 6-14 are used to calculate “exchange rates”, i.e. by how much each parameter must be modified to reach a 1 percentage point effective efficiency improvement and what are the consequences on the other engine outputs. The exchange rates are shown in Figure 6-16.

Effective efficiency increase by 1%						
		Advantage / Cost				
		Peak in-cylinder pressure	Octane number requirement	NOx production	Cooling losses	Compressor power
Parameter	Modification required	[bar]	[-]	[kg/h]	[kW]	[kW]
Combustion advance (deg CA)	9.51	58.1	10.4	1.7	15.8	0.0
Combustion duration (deg CA)	-19.22	44.9	-2.3	1.1	7.5	0.0
Lambda evolution (-)	0.16	12.3	5.7	-1.3	-3.8	9.8

Figure 6-16 - Exchange rates to improve effective efficiency by 1 percentage point using combustion advance, combustion duration and lambda evolution

The values shown in Figure 6-16 should not be considered as the exact results that would be observed on a real engine if the calculated modifications were applied. Indeed, the results displayed in Figure 6-14 and used for the exchange rates are only valid for the range of parameter values tested here and are average values which do not model the non-linear evolution of engine performance with the modification of the input dataset. However, such an analysis is interesting to assess which of the three investigated parameters is the most effective lever to improve engine efficiency.

According to Figure 6-16, to reach a 1 percentage point effective efficiency increase, combustion advance must be increased by 9.51 °CA but does not improve any of the other parameters of interest. By further advancing combustion start, the peak in-cylinder pressure is significantly increased (~+25%) which is acceptable because the engine does not operate close to its pressure limit but the higher the pressures, the higher the engine stress and the more concerns for reliability. As previously mentioned, increasing combustion advance leads to higher temperatures resulting in higher octane number requirement, NO_x production and cooling losses.

As the calculations are based on average value, the 9.51 °CA proposed here is over-estimated as combustion advance does not have a linear effect on effective efficiency. In addition, the simple “exchange rate” method used here shows its limit as a 9.51 °CA increase results in a -10.51 °CA ATDC combustion start which is not realistic as the engine model run with a -10 °CA ATDC combustion start for the creation of Figure 6-14 proved to yield lower performance than the -5 °CA ATDC case. In reality, the 1% efficiency improvement would be achieved by a value between -10 and -5 °CA ATDC.

However, the “exchange rate” analysis showed that a significant increase in combustion advance is required to improve effective efficiency by 1 percentage point and that it is accompanied by a large increase in in-cylinder pressure, in octane number requirement from the fuel and in cooling losses.

Regarding combustion duration, it must be decreased by 19.22 °CA for effective efficiency to be increased by 1 percentage point. The consequences of this modification are less harsh than that for the combustion advance modification but still result in higher in-cylinder pressures, NO_x production and cooling losses.

However, the octane number requirement is reduced for the reasons previously mentioned when looking at Figure 6-14.

Nevertheless, the 19.22 °CA reduction corresponds to the combustion duration nearly being divided by 2 which seems rather unrealistic. The 41 °CA combustion duration used for the engine models presented in this research work is already lower than that of conventional gasoline engine thanks to the pre-chamber ignition system. Further reducing the combustion duration will require a fuel with a significantly higher laminar flame speed than conventional gasoline-like fuels, like hydrogen [175].

Using higher dilution to improve effective efficiency by 1 percentage point requires lambda to be raised by 0.16. This results in a relatively small in-cylinder pressure increase, reduced cooling losses and NO_x production but requires higher fuel knock resistivity and higher compressor power. If higher compressor power is needed, more energy must be recovered by the turbine which results in higher back pressure. Increased air/fuel ratio also comes with higher intercooler load. In addition, the potential modification of the Rate of Heat Release curve shape with change in air/fuel ratios is not captured with the present study.

As seen in Figure 6-16, modifying the combustion properties (advance or combustion) leads to higher efficiency but also to a much greater increase in in-cylinder pressure compared to increasing dilution as it requires significant modification to reach the 1 percentage effective efficiency improvement target. It is costly to get performance out of solutions which increase in-cylinder temperature. Despite not being pressure-limited, the lower in-cylinder pressure of the engine, the lower the stress it faces, and the lighter components can be.

Even though combustion properties will be optimised to maximise the engine effective efficiency and will be influenced by the level of dilution run on the engine, this “exchange rate” analysis suggests that increasing air/fuel ratios is the most interesting solution to pursue and is investigated in Section 6.3.

In addition, from the energy and exergy analyses conducted in Chapter 5, increasing dilution seems an obvious solution to try for engine manufacturers now that the engine is not pressure limited any more. It leads to higher fuel conversion efficiency, reduce exergy lost to the cooling system and maintain high exhaust enthalpy thanks to the increased mass flow. However, increasing air/fuel ratio will be strongly linked to the capacity to maintain reliable and efficient combustion using the passive pre-chamber and the engineered fuel properties.

All three parameters investigated here are closely linked to the advanced sustainable fuel properties that will be created by the Formula One™ fuel suppliers. Combustion advance will depend on the fuel knock properties, combustion duration on its laminar flame speed and increased dilution on most of the fuel combustion properties (laminar flame speed, ignitability limit, mixing properties for example). This shows the importance of fuel development for the 2026 Formula One™ regulations.

6.3 Requirements for the 2026 Formula One™ engine to run at $\lambda = 1.6$

As established in the previous section, increasing air/fuel ratio is an effective solution to take advantage of the reduction in peak in-cylinder pressure between the 2014+ and 2026 Formula One™ regulations to increase fuel conversion efficiency. This section aims at quantifying what it would take engine manufacturers to go from $\lambda = 1.4$ to $\lambda = 1.6$.

The value 1.6 has been chosen as it is reported in the literature to be the limit of a passive pre-chamber [60]; [61] and as still only one injector is allowed per cylinder by the future set of regulations, the engine will not switch to an active pre-chamber that could extend this limit. In addition, it seems that $\lambda = 1.6$ is the target for some engine manufacturers [5].

As reported in Table 4-2, the fuel adopted for the 2026 model is an E20 which matches relatively well the fuel characteristics described in the 2026 Formula One™ regulations. Even though the composition of the 2026 fuels is likely to be more sophisticated than that, Wouters et al. [176] showed that synthetically produced gasoline blended with 20% ethanol in volume can successfully burn up to $\lambda = 1.6$ under the limit of IMEP variation, demonstrating that $\lambda = 1.6$ seems to be an acceptable target for the 2026 engine.

$\lambda = 1.4$ and $\lambda = 1.6$ refer to the maximum value reached in the lambda evolution with engine speed. For this study, the lambda evolution of the 2026 engine model presented in Section 6.1 (which is the same as the 2014+ Digital Twin) has been raised by 0.2 for all engine speeds as demonstrated in Figure 6-17. Each engine model will be referred to as their maximum lambda value in the following sections.

The study presented in this section looks at the requirements for the engine to run at $\lambda_{max} = 1.6$ and what the consequences on the engine outputs are, but it is assumed that the combustion process has been optimised accordingly. This means that the combustion properties of the $\lambda_{max} = 1.6$ engine model are the same to that of the reference engine. Despite the higher lambda, it is assumed that the combustion duration can be kept constant using different fuel composition (to influence laminar flame speed) and the pre-chamber ignition system to maximise flame speed. As the pre-chamber technology will still be passive, the ROHR curve is also kept the same.

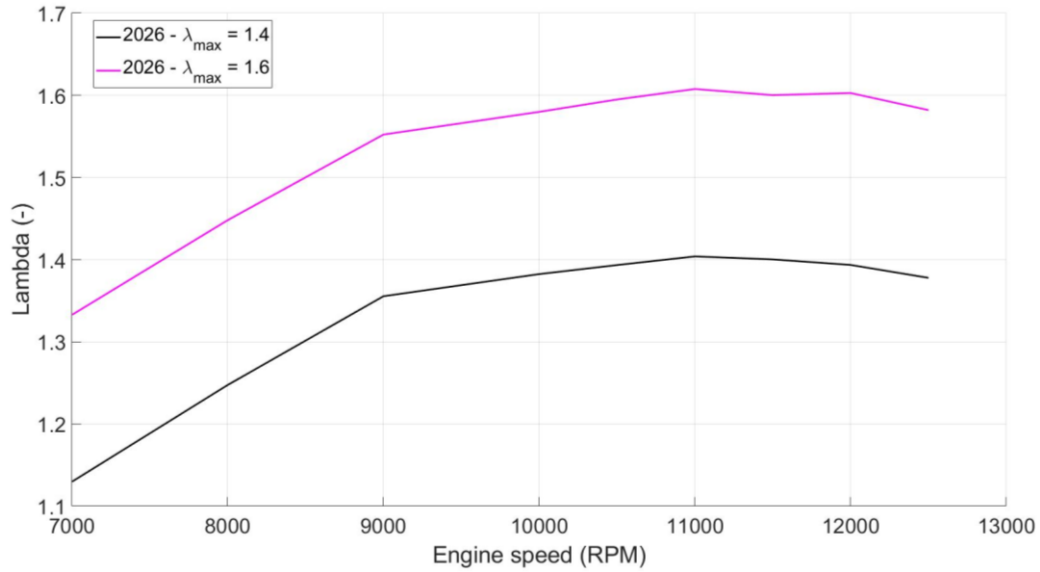


Figure 6-17 - Lambda evolution with engine speed – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

The first thing to look at when increasing lambda is the compressor. As shown in Figure 6-18, because the fuel flow is the same for both investigated models and because air/fuel ratio are raised, more air needs to be trapped in the cylinders.

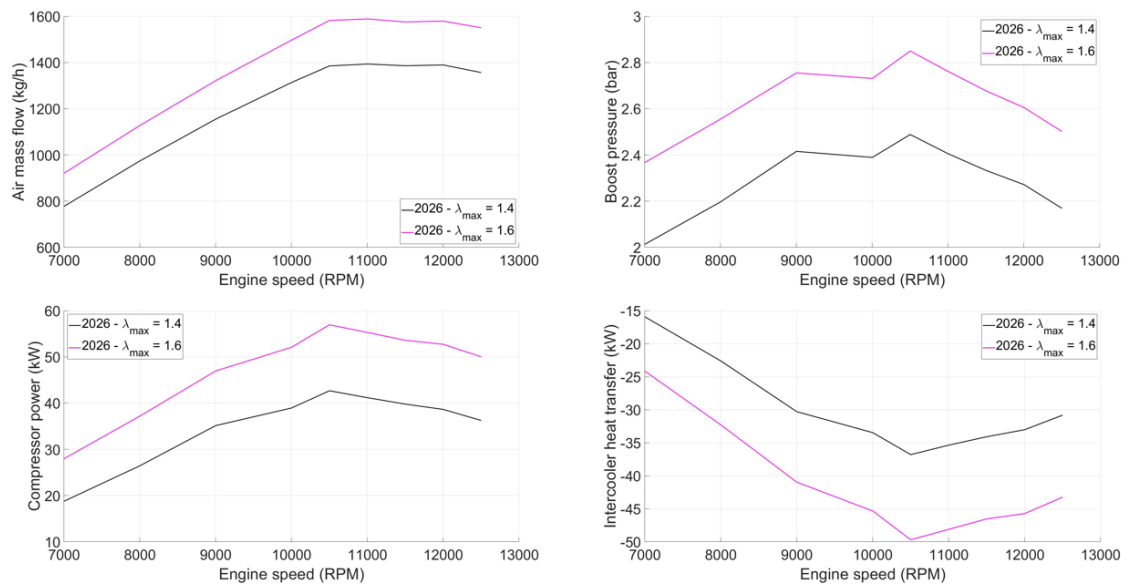


Figure 6-18 - Air mass flow (top left), boost pressure (top right), compressor power (bottom left) and intercooler load (bottom right) curves – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

In consequence and as the internal volumetric efficiency is relatively constant (Figure 6-19), boost pressures also need to be increased leading to higher compressor power, +12.8 kW in average which represents a relative increase of 37.1%.

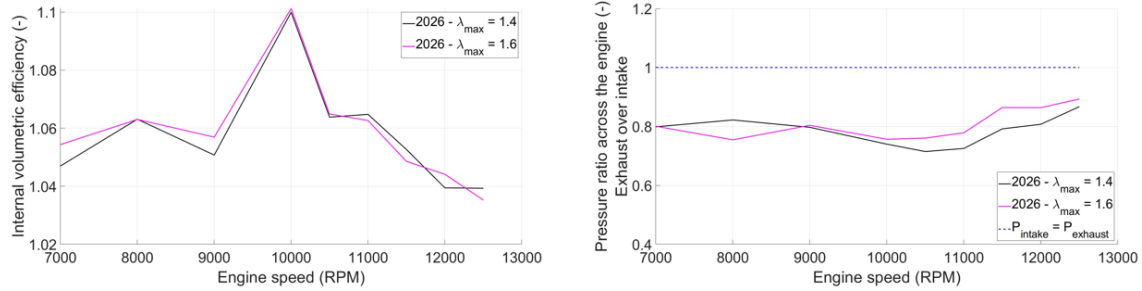


Figure 6-19 - Internal volumetric efficiency and pressure ratio across the engine curves – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

As mentioned in Section 6.2, higher air mass flow and boost pressure lead to higher intercooler load. Going from $\lambda_{max} = 1.4$ to $\lambda_{max} = 1.6$ requires an additional 11.5 kW of heat (+39.1%) on average to be removed from the inlet air mass flow using the intercooler.

With higher dilution, because the compressor demands more power, the turbine must also recover more power. This means that the turbine must be more restrictive, thus increasing the exhaust system pressure.

However, as shown in Figure 6-19, despite the back pressure increasing due to the more restrictive turbine, as the inlet pressure is also raised to reach the target air/fuel ratios, the pressure ratio across the engine and the internal volumetric of both engine models are very similar.

This means that despite the increase in back pressure, the breathing capacity of the engine is not significantly impacted. In addition, the pressure ratio across the engine stays for all engine speeds below 1 meaning that the inlet pressure is always higher than the back pressure, making it easier to transfer engine mass flow from the inlet to the exhaust (i.e. minimising pumping losses).

As expected with the increased air/fuel ratios and as shown in Figure 6-20, the performances of the engine have been raised. On average, the BMEP has been increased by 0.58 bar and the power by 7.38 kW which represents a 2.3% relative increase compared to the $\lambda_{max} = 1.4$ model.

With this increase, the total peak power of the 2026 Formula One™ Power Unit, i.e. when also considering the 350 kW MGU-K, goes up to 731.8 kW (981.3 hp) compared to 724.7 kW for $\lambda_{max} = 1.4$.

The 2.3% relative increase in BMEP and power is logically also observed for the effective efficiency which represents a 1 percentage point increase. It is interesting to compare this observation to the “exchange rate” analysis conducted in Section 6.2. The analysis predicted that a 0.16 increase in lambda was necessary to increase effective efficiency by 1 percentage point. The $\lambda_{max} = 1.6$ engine model presented here shows that the increase must be slightly higher, i.e. 0.2.

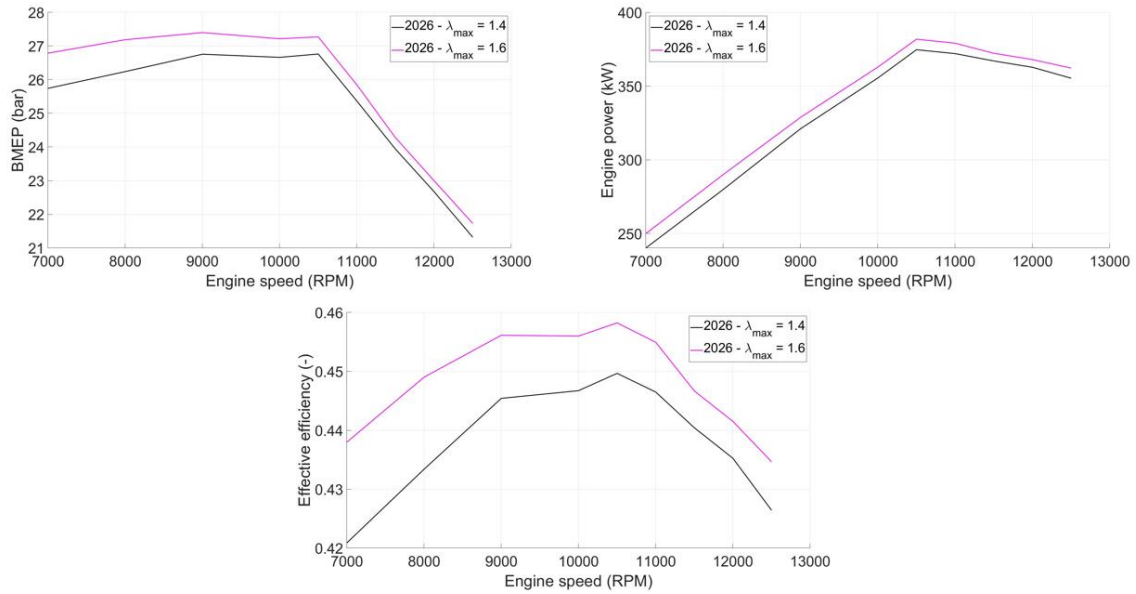


Figure 6-20 - BMEP (top left), power (top right) and effective efficiency (bottom) curves – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

This shows once more that the “exchange rate” analysis cannot capture all the engine performance non-linear evolution with changes in input dataset with a prediction 20% off the actual necessary value.

Despite the average effective efficiency increase being 1 percentage point, it can be observed in Figure 6-20 that the increase is higher at low engine speed than at high speed. This can be understood by considering Figure 6-21 which shows the evolution of indicated efficiency with lambda for an Otto cycle (idealised constant-volume combustion cycle). As lambda increases, efficiency keeps increasing but at a slower rate.

As the air/fuel ratios used at low engine speed are lower than that at high speed, the same increase in lambda will have greater impact on performance at low engine speed as confirmed here, with the $\lambda_{max} = 1.6$ engine model simulation results.

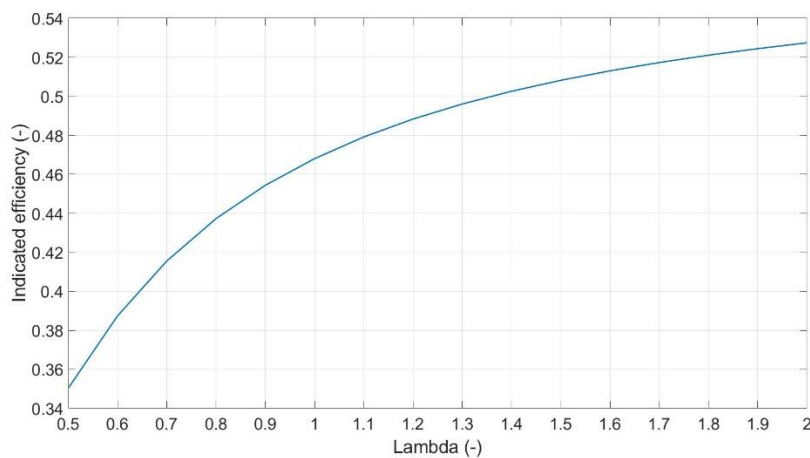


Figure 6-21 - Otto cycle indicated efficiency evolution with lambda

As discussed in Section 6.2, the idea of increasing air/fuel ratio is made possible thanks to the 2026 Formula One™ engine not being pressure-limited any more. As shown in Figure 6-22, peak in-cylinder pressures are raised by 14.3 bar in average in consequence of higher air/fuel ratios. In addition, the higher dilution means that the combustion heat is diluted in a bigger mass leading to lower peak in-cylinder temperature by 158.6 K on average.

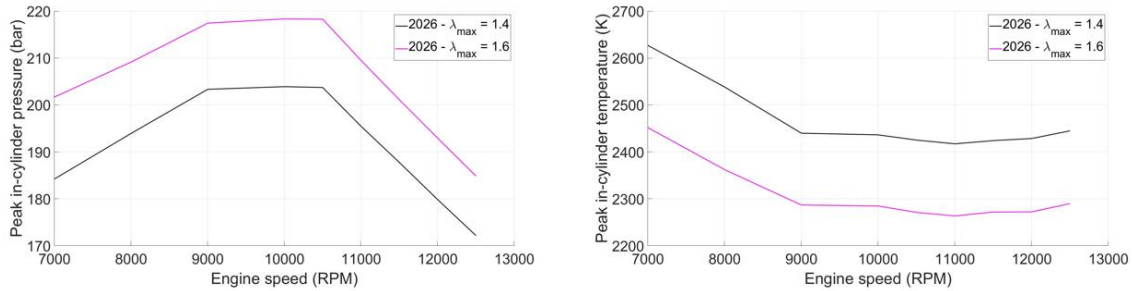


Figure 6-22 - Peak in-cylinder pressure and temperature curves – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

The ability to operate at higher pressure while reducing temperature is the strength of high dilution as it minimises heat transfers and maximises work production.

The reduction in heat transfer through the combustion chamber wall is shown in Figure 6-23 alongside the evolution of exhaust enthalpy flow. On average, cooling losses are reduced by 3.87 kW. However, as mentioned when looking at Figure 6-18 and compressor behaviour, the intercooler load has increased by 11.5 kW by switching from $\lambda_{max} = 1.4$ to $\lambda_{max} = 1.6$. Overall, running at higher air/fuel ratios will increase the cooling needs of the 2026 Formula One™ engine.

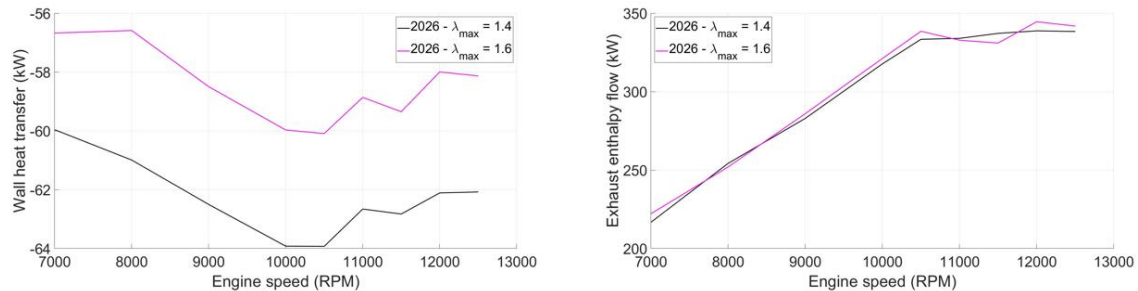


Figure 6-23 - Cooling losses and exhaust enthalpy flow curves – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

Interestingly, the exhaust enthalpy flow is very similar for both dilution levels. Despite the lower exhaust temperature, the higher air mass flow of the $\lambda_{max} = 1.6$ case keeps exhaust enthalpy flow high.

This observation is important considering that the turbine must recover more power from the exhaust gases to allow the compressor to provide higher air mass flow to the engine. The turbine must recover more energy from a similar energy flow which justifies the need for a more restrictive turbine. If the enthalpy would have decreased, the turbine would have to be even more restrictive with a penalty

in back pressure. The opposite would have been true if exhaust enthalpy had increased.

Another consequence of the lower in-cylinder temperature thanks to the higher dilution is the reduction in NO_x production, both absolute and specific, by over 70% for both (Figure 6-24). With the future of the sport being focussed on sustainability, implementing regulations which promotes higher dilution to achieve higher performances seems an efficient solution to meet both higher efficiency and lower emissions objectives even if engine manufacturers are solely focussed on performances.

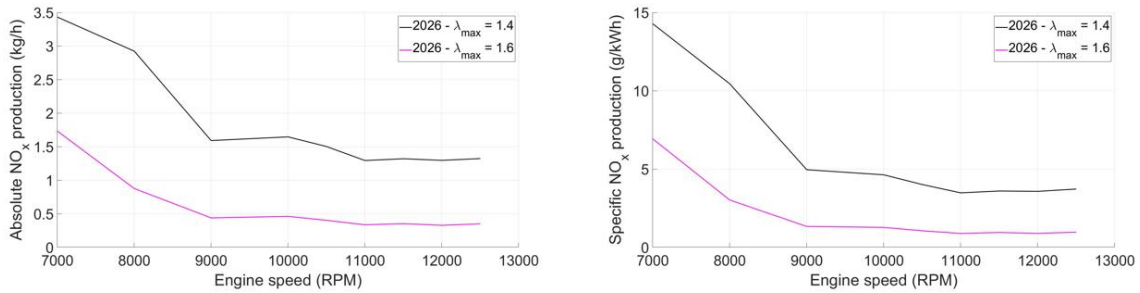


Figure 6-24 – Absolute and specific NO_x production – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

As for the previously investigated engine models, energy and exergy analyses were conducted on the $\lambda_{max} = 1.6$ case and compared to the $\lambda_{max} = 1.4$ results in Figure 6-25 and Figure 6-26:

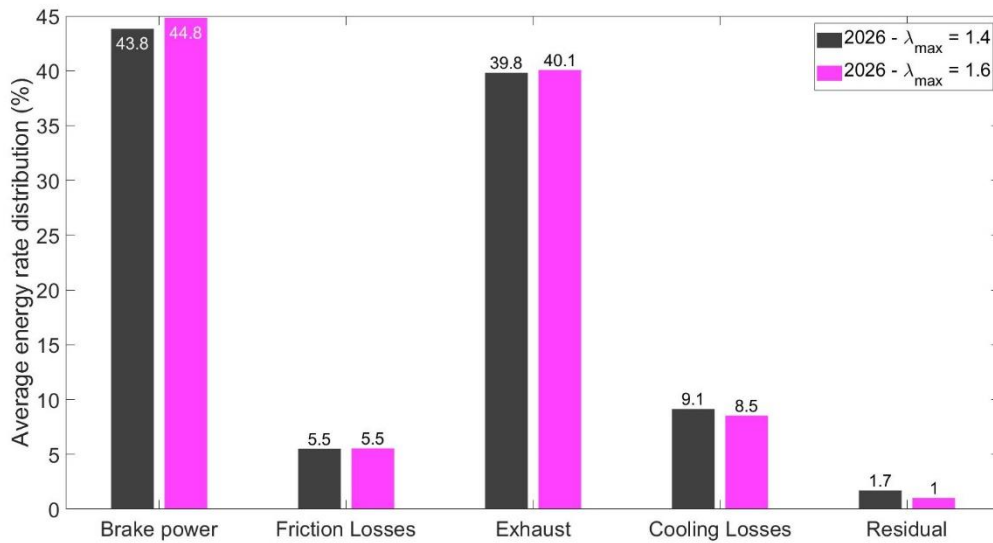


Figure 6-25 – Average energy distribution – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

The energy distribution is a summary of what has been observed previously in this section. The engine running at $\lambda_{max} = 1.6$ is more efficient, i.e. a higher fraction of the fuel energy is converted into brake power. Friction losses does not evolve as in-cylinder pressures of both models are different but not significantly, the increase in air/fuel ratio results in the lowest pressure increase for the same efficiency improvement as suggested by the “exchange rate” analysis in Section 6.2.

The lower in-cylinder temperatures lead to a reduction in losses to the cooling systems but thanks to the higher engine mass flow, the exhaust enthalpy is slightly increased in average.

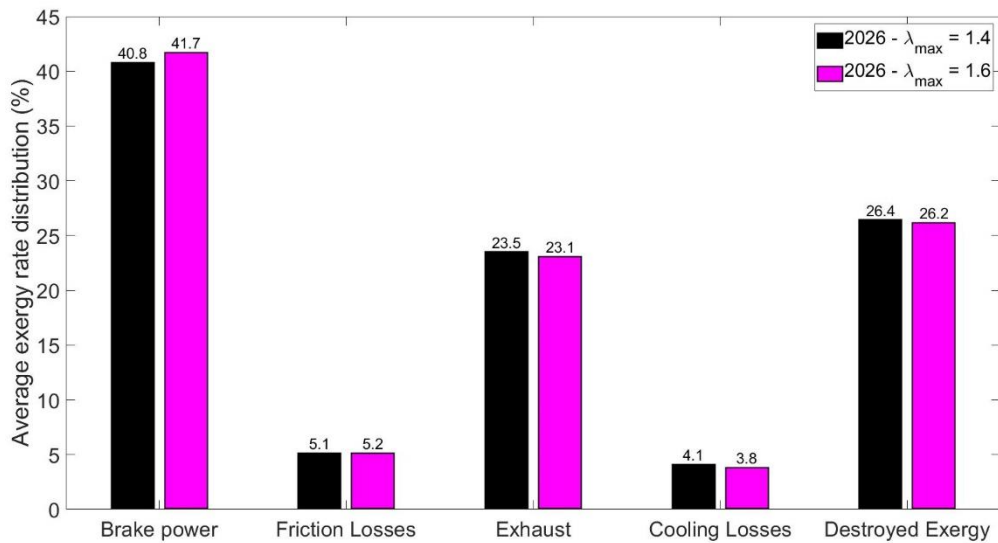


Figure 6-26 - Average exergy distribution – $\lambda_{max} = 1.4$ vs $\lambda_{max} = 1.6$ 2026 engine models

From an exergy point of view, the high dilution case is also the most efficient and friction losses are equivalent for both engine models. Interestingly, the available work of the exhaust gases for the $\lambda_{max} = 1.6$ case is lower than that of the $\lambda_{max} = 1.4$ which is the opposite of the energy point of view. This is due to the lower temperature of the exhaust gases when air/fuel ratios are increased, resulting in heat transfers occurring at lower temperature and thus being less efficient from the second law of thermodynamics perspective.

However, thanks to the lower in-cylinder temperatures, heat transfers to the cooling systems are limited, thus reducing the exergy lost there.

On average, both levels of dilution destroyed similar fraction of fuel exergy. By increasing air/fuel ratios, combustion irreversibility is raised but higher pressures are reached, a larger fraction of the fuel exergy content is converted into useful work and the lower temperature reduces exergy losses to the cooling systems.

Table 6-2 shows the relative differences between the 2026 Formula One™ engine model running at $\lambda_{max} = 1.6$ and the same engine running at $\lambda_{max} = 1.4$ but also the differences with the 2014+ Digital Twin operating in CWG mode.

Parameters	Average relative evolution between the 2026 engine model running at $\lambda_{max} = 1.4$ and $\lambda_{max} = 1.6$ (%)	Average relative evolution between the 2014+ (2014+ Digital Twin in CWG mode) and 2026 engine model running at $\lambda_{max} = 1.6$ (%)
Fuel mass flow	N/A	-26.8
Fuel energy flow	N/A	-31.8

Parameters	Average relative evolution between the 2026 engine model running at $\lambda_{max} = 1.4$ and $\lambda_{max} = 1.6$ (%)	Average relative evolution between the 2014+ (2014+ Digital Twin in CWG mode) and 2026 engine model running at $\lambda_{max} = 1.6$ (%)
Air mass flow	+14.7	-23.3
Engine power	+2.3	-29.8 (in CWG recharge mode) -34.4 (in CWG power mode)
Total PU peak power	+1.0	+4.8
BMEP	+2.3	-29.8 (in CWG recharge mode) -34.4 (in CWG power mode)
FMEP	+0.5	-8.3
Effective efficiency	+2.3	+3.0 (in CWG recharge mode) -3.8 (in CWG power mode)
BSFC	-2.2	+4.2
Engine cooling losses	-6.7	-32.5
Intercooler load	+39.1	-45.2
Exhaust enthalpy	+0.7	-34.5
Internal volumetric efficiency	+0.1	+8.7
Pressure ratio across the engine	+3.1	-17.0
Boost pressure	+15.2	-26.1
Compressor power	+37.1	-43.5
Peak in-cylinder pressure	+7.5	-28.1
Absolute NO _x production	-70.4	-84.0
Specific NO _x production	-71.1	-77.1
Average destroyed exergy fraction	-0.8	+5.2

Table 6-2 - Average relative evolution between the 2026 engine model running at $\lambda_{max} = 1.6$ compared to the same model running at $\lambda_{max} = 1.4$ and the 2014+ Digital Twin in CWG mode

6.4 Conclusions on the impact of the 2026 Formula One™ Power Unit regulations

This chapter investigated the impact on engine performance of the transition between the 2014+ and the 2026 Formula One™ Power Unit Regulations.

As both engines have the same geometry and are both fuel-limited with a similar speed dependence, their overall behaviours, using steady-state simulations, are very similar as shown by the energy and exergy analyses. The main difference between both sets of regulations comes from the fuel flow limit.

The fuel energy flow being reduced by approximately 30%, the engine performance output is also reduced by this amount as effective efficiency are relatively similar between both engines. However, when considering the removal of the MGU-H and its potential of providing energy to the MGU-K to increase the effective efficiency without requiring any energy from the ES, the relative reduction in power output is 35.9% and 2.81 percentage points reductions in terms of effective efficiency. By removing the MGU-H, the 2026 Formula One™ Power Unit will be less powerful than the 2014+ one and less efficient (although the efficiency assessment depends on the mode of operation of the 2014+ engine).

With a reduced fuel mass flow, less is demanded from the turbocharger and the intercooler leading to reduced cooling needs compared to the 2014+ Formula One™ engine even though the Energy Recovery System cooling needs must also be considered when designing the cooling system.

Despite the reduction in engine performance, the increase in MGU-K power makes the 2026 Formula One™ Power Unit are least as powerful as the current one if not more according to the engine models used in this research work. As for the 2014+ Digital Twin, the shape of the 2026 engine power curve follows that of the fuel flow limit. However, thanks to the very high torque at low speed of electric motors, the 2026 Formula One™ Power Unit should have a significantly higher torque at low speed than its 2014+ counterpart.

The reduction in fuel flow results in the engine not being pressure limited any more compared to its 2014+ counterpart which brings opportunity to push the combustion process further.

According to the “exchange rate” analysis done in Section 6.1, increasing air/fuel ratio is the most effective way to take advantage of the reduction in in-cylinder pressures to increase the engine efficiency and thus the output performance.

By increasing lambda values by 0.2 over the whole engine speed range to reach a maximum value of $\lambda = 1.6$, on average, an additional 1 percentage point in effective efficiency can be found. This comes at the expense of higher octane number requirement, higher total cooling requirements (engine + intercooler) and higher compressor power.

Overall, the 2026 Formula One™ Power Unit regulations are in the continuity of the 2014+ ones with still a fuel-limited electric hybrid Power Unit but with a change in thermal / electric power balance to reach close to equality. By reducing the stress / BMEP of the engine, the focus can be put on increasing mixture dilution, and the work presented in this thesis leads to believe that the engine manufacturer who will be able to operate the closest to the limit of passive pre-chamber capacities with reliable lean combustion will have the upper hand on its competitor when it comes to engine performance.

The collaboration between engine manufacturers and fuel suppliers will be of first importance to ensure that these advanced sustainable engineered fuels make leaner combustion possible by having extended ignitability limits and increased laminar flame speed.

As for the 2014+ Formula One™ engine, the behaviour of the 2026 Formula One™ engine will be strongly influenced by the shape of the fuel flow limit. Similarly, the engine will not be operated at high engine speed because of the power limitation imposed by the fuel flow limit and the increasing friction losses. Besides, the 15,000 RPM engine speed limit that was included in the first regulations of the second turbocharged era (2014-2021) has not been re-introduced in the 2026 regulations showing that there is no need for the regulators to implement such a limit.

With the 2014+ Formula One™ regulations, the engine is used within a narrow engine speed range which reflects on the turbocharger operation. For the 2026 regulations, if the cars are allowed the same number of gears as currently, the observation should be the same. This is why the vehicle model used in the transient 2026 AVL Boost™ engine model and presented in Chapter 4 has 8 gears and the engine speeds at which gear changes happen have been kept the same as the ones observed for the current Formula One™ Power Units.

Considering that the MGU-K nearly represents 50% of the PU power and that all the electric energy comes from the fuel (via regeneration), the energy management will be crucial to ensure the ES is kept at high enough SOC to power the MGU-K when required.

Turbo lag management will also be key to minimise the use of the ERS. The more the turbo lag is minimised, the faster high engine power are reached, reducing the dependency on the MGU-K to accelerate the car.

The steady-state simulations discussed in this chapter and Chapter 5 cannot capture the impact of turbo lag on engine performance during changes in load demand. This is why the next chapter will introduce transient AVL Boost™ engine simulations which will enable the investigation of turbo lag and anti-lag Power Unit modes.

7 Investigations of different ICE/MGU-K interaction modes to minimise turbo lag in the 2026 Formula One™ Power Unit transient operation

As mentioned in Chapter 6, managing turbo lag will be an important part of Power Unit suppliers responding to the 2026 Formula One™ regulations. Because of the MGU-H removal, the 2026 engine will feature a conventional turbocharger only powered by the energy recovered by the turbine. During part-load operation, as less fuel is burned and the engine mass flow is reduced, the exhaust enthalpy reduces and so does the amount of energy the turbine can recover and finally the turbocharger rotor speed is reduced as a result.

Chapter 2 briefly mentioned a range of anti-lag solutions, namely minimising the turbocharged moment of inertia, modifying the turbine aspect ratio, increasing exhaust enthalpy and externally assisting the turbocharger. These influence the behaviour of the turbocharger by directly intervening either during the design stage or while the engine is operating. There is no doubt that these solutions, and potentially others, will be investigated by the Power Unit manufacturers to improve the transient behaviour of their turbocharger within the 2026 Formula One™ Power Unit regulations framework. For instance, variable geometry turbines are not allowed for the 2026 engine but without doubt detailed inspection of the flow dynamics within a fixed geometry exhaust manifold and turbine housing will get design attention.

The aim of this chapter is not to investigate the usefulness of such solutions on the 2026 Formula One™ engine but to assess the other significant change introduced by the regulations which is how the internal combustion engine could interact with the much more powerful (350 kW) MGU-K to improve the efficiency of the whole Power Unit.

The MGU-K contributes to nearly half the total power of the 2026 PU so will be of great importance for on-track performance especially in the first stages of acceleration event (albeit up to the traction limit imposed by the tyres). Nevertheless, due to its high power, it consumes large amount of energy which comes from regeneration. It is helpful to remember that this comes indirectly from the fuel combustion required to accelerate the vehicle to high speed in the first place and is not totally 'free' energy, but an amount of energy reclaimed that otherwise would be lost. The energy management of the 2026 Formula One™ Power Unit will have less variables compared to the 2014+ PU due to the MGU-H removal but the notion of 'modes' of MGU-K and ICE operation will continue and must be optimised.

The state of charge (SOC) of the Energy Store (ES) needs to be managed dynamically to supply the MGU-K with enough energy when required and stay within the SOC variation allowed per lap by the regulations [15].

By minimising turbo lag, the engine power can be maximised during change in load demand and thus minimise the dependency on the MGU-K to accelerate the car within the tyre traction limit.

Experience from the past suggested that the regulations for the 2026 Formula One™ cars will also evolve to create a competitive package with the Power Unit. The 2026 cars will most probably have active aerodynamics [5] to minimise drag in straight lines for example and be consistent with the MGU-K power decrease from 300 km/h, thus minimising the MGU-K use to make the reduced output from the 2026 Power Unit acceptable.

To simulate situations in which turbo lag can have an impact on the engine performance, the steady-state simulations used so far in this research work are developed further. This requires an engine model not operating at constant speed and equipped with a full turbocharger model so that the engine reaction to modification in load demand can be investigated with its operating point evolving freely, i.e. AVL Boost™ transient engine models are required.

The engine models that will be used in this chapter are or are based on the AVL Boost™ transient model presented in Chapter 4.

7.1 Comparison of the transient performance of a motorsport ICE first fitted with an e-turbocharger and then with a conventional turbocharger with added P1-hybrid electric machine

This section is based on the research work presented in a paper that was presented at the 15th International Conference on Turbochargers and Turbocharging organised by the Institution of Mechanical Engineers (IMechE) [177].

The objective of this section is to discuss ways of deploying electrical machines (EM) on a turbocharged internal combustion engine and to assess their energy effectiveness. Inspired by the 2026 Formula One™ regulations [15], the use of an EM attached to the turbocharger rotor (eTC) and that of a powerful EM at the engine crankshaft (MGU-K) are investigated using AVL Boost™ transient engine simulations.

7.1.1 Motivation for work

For the past fourteen years, electric motors have progressively been introduced into Formula One™ Power Units. Early ‘push-to-pass’ hybrid motorsport powertrains, such as KERS [117]; [161], have been replaced with ‘torque fill’ architectures where the power output of the electric motor (EM) is sufficient to significantly enhance the acceleration of the vehicle at a range of road and internal combustion engine (ICE) speeds.

Whilst the current Formula One™ Power Unit, introduced in 2014, features two EMs, one linked to the crankshaft of the engine (MGU-K – P1 position) and the second connected to the turbocharger (MGU-H), with the introduction of the 2026 Formula One™ Power Unit regulations, the MGU-K will remain and will be more powerful, but the MGU-H will be removed, exposing the engine to turbo lag. Therefore, the consequence of the 2026 Formula One™ regulations is to focus on the interaction between a single EM and a turbocharged ICE.

The purpose of this section is to investigate how the position of a single electric motor on the engine can influence its behaviour in situation where the engine is exposed to turbo lag and how to best couple a single electric motor with an internal combustion engine to maximise the car acceleration performance and fuel conversion efficiency.

7.1.2 Presentation of the models used for this study

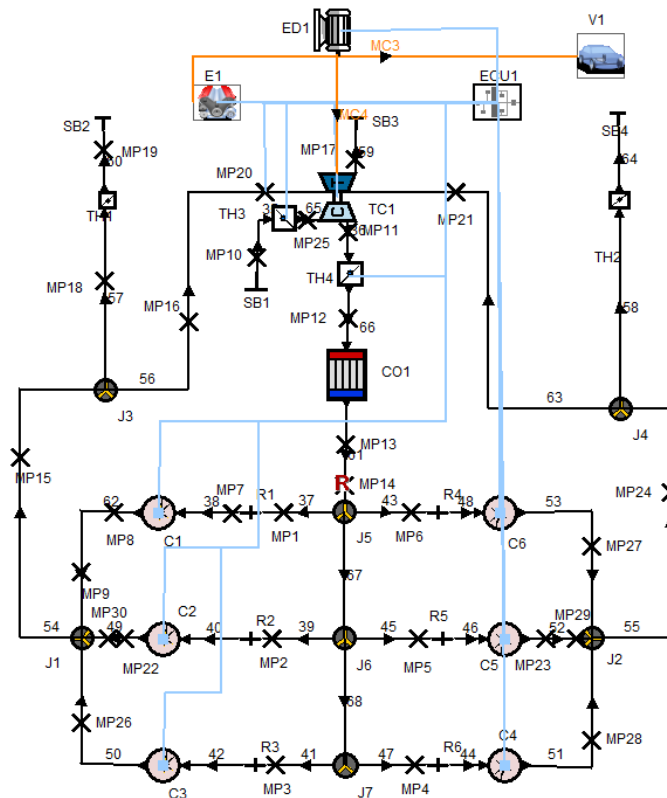


Figure 7-1 - AVL Boost layout of the engine model used for investigating the coupling of an electric motor with the ICE (here with an eTC)

The AVL Boost™ engine models used for this study are based on the 2026 transient model presented in Chapter 4 but are simplified versions as they do not require particularly complex control. Therefore, only the ECU model proposed in AVL Boost™ is used to control the parameters varying with engine operation, not the Formula Interpreter. The ECU controls fuelling, waste gate opening, engine load, throttle position and electric motor power when applicable. Combustion

timing control is maintained in the general parameter set for convenience. Figure 7-1 shows the layout of one of the engine models used for this study, namely the engine model with an electrically assisted turbocharger (eTC).

Three engine models are used for this study. A reference model which is a conventional turbocharged internal combustion engine without eTC or MGU-K. As a sub-model of the model presented in Chapter 4, this engine is also based on the 2026 Formula One™ regulations. The vehicle and turbocharger models are unchanged.

The second model has the same engine but features an eTC which is powered by a 30 kW MGU (shown in Figure 7-1 as 'ED1'). The objective of this MGU is to keep the turbocharger rotor always spinning at constant speed (125,000 RPM), even during part-load operation. The settings of this MGU are given in Table 7-1 and are derived from the Helix CTSM242 electric motor properties [32] which was used to define the characteristics of the MGU-K used for the 2026 transient model.

Parameter	Unit	Value
Maximum power	kW	30
Peak electrical efficiency	-	0.98
Moment of inertia	kg.m ²	0.005
Mechanical connection efficiency	-	0.97
Mechanical connection gear ratio	-	1:1

Table 7-1 - eTC electric motor parameters

The third model is the 2026 Formula One™ Power Unit. It has the same ICE as the two other models and a 350 kW MGU-K, the characteristics for which are given in Chapter 4 and shown as a reminder in Table 7-2.

Parameter	Unit	Value
Maximum power	kW	350
Maximum torque	Nm	500
Peak electrical efficiency	-	0.98
Moment of inertia	kg.m ²	0.042
Mechanical connection efficiency	-	0.97
Mechanical connection gear ratio	-	4:1

Table 7-2 - MGU-K parameters

For these models, the fuelling strategy is simple and only based on the engine load in first approximation. At 100% load, the fuel flow delivered to the engine is equal to the fuel flow limit defined by the 2026 Formula One™ Power Unit regulations. When no load is demanded to the engine, fuel flow is kept at 10 kg/h for all speeds so that the engine is kept sufficiently hot. For load between 0 and 100%, the ECU model linearly interpolates the required fuel flow value.

For the transient simulations carried out with the 2026 Formula One™ regulations, it was chosen to keep peak lambda values around $\lambda = 1.4$, this is with the calculations made for this level of dilution that the compressor map was chosen. This choice was made knowing that, despite the 2026 fuels being different to the 2014+, it is possible to operate the current generation of Formula One™ engine at $\lambda = 1.4$. As discussed in Chapter 6, it is expected that the engine manufacturers will push to increase dilution and thus efficiency, but the achievable level is not exactly known yet.

In addition, the studies conducted in this section and the next do not aim at predicting the precise behaviour of the 2026 Formula One™ PU but to investigate engineering solutions, to demonstrate the potential usefulness of ICE/EM interaction modes for which the precise dilution level makes a difference on the values presented but does not impact on the conclusions.

The results presented in this section are slightly different to the numbers given in the paper presented at the IMechE conference due to difference in the turbine swallowing curve. After the submission of the conference paper, it was judged, while conducting other studies, that the low-speed part of the turbine swallowing curve should be defined for even lower rotor speed to better represent the behaviour of the turbocharger at very low speeds.

By completing the definition of the turbine swallowing curve to reach the curve presented in Chapter 4 (Figure 4-10), the behaviour of the turbocharger at low speed is modified, hence the difference between the results presented here and in the conference paper. Nevertheless, the conclusions of the study are not impacted.

To place the engine in a situation where it is exposed to turbo lag, the simulation starts with the engine running at 10,000 RPM at 128 km/h in third gear with 10% engine load being applied for 0.9 s, then for 0.1 s, the load is linearly increased until reaching 100% and is kept there until the end of the simulation as shown in Figure 7-2. The simulations are run for 250 engine cycles. The drop seen around $t = 2.2$ s corresponds to a simulated gear change made by the model.

The study presented here focuses on the behaviour of the engine between the moment it “tips-in”, i.e. $t = 0.9$ s, and the first gear change (this point varies between simulations but can be seen at $t \approx 2.2$ s in Figure 7-2).

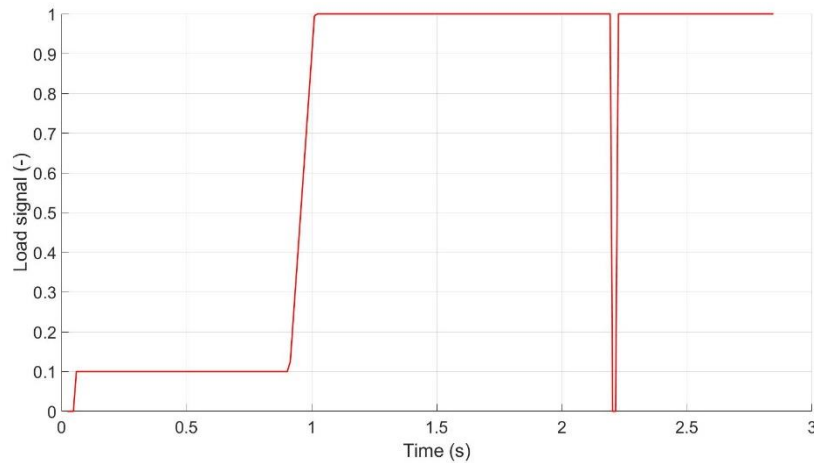


Figure 7-2 - Engine load evolution

7.1.3 Simulation results – How does an eTC improve engine efficiency?

As seen in the results of transient simulations (Figure 7-3), for the first 0.9 s of the simulation, the engine speed decreases because of the total engine friction losses (including pumping losses) being higher than the torque produced by the engine at 10% load. It can be noted that the engine speed drops less for the eTC than for the reference. This is one of the effects of the eTC being kept spinning at high rotor speed as the compressor thus provides the engine with more mass flow, leading to higher in-cylinder pressure and thus more power produced by the engine. This, of course, may need compensation from the driver using brakes or MGU-K to regulate vehicle speed.

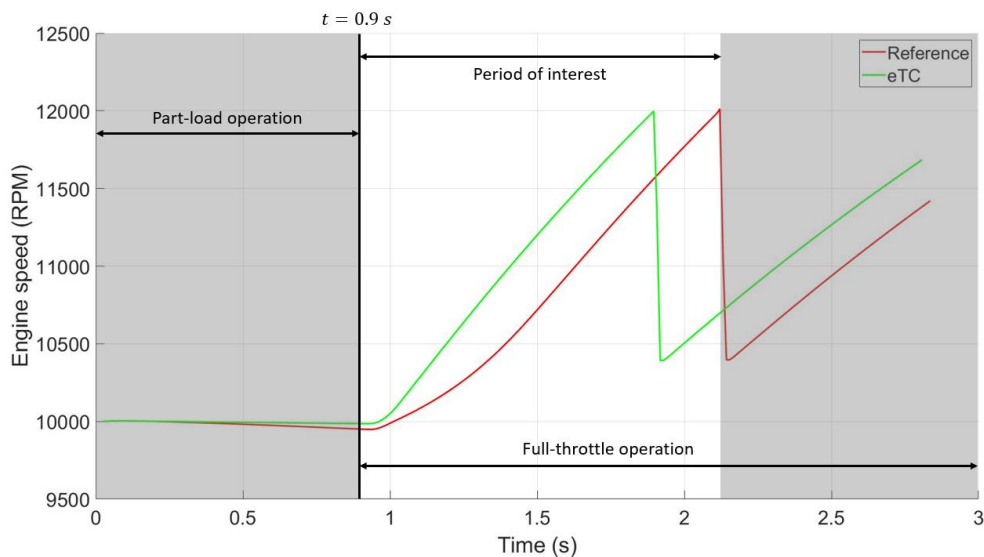


Figure 7-3 - Engine speed evolution for the reference and eTC cases

After $t = 0.9\text{ s}$, the engine speed starts increasing in response to the load change from 0 to 100% until reaching 12,000 RPM where the model simulates a gear change. The overall zone of interest for this study is shown by the clear part of Figure 7-3. For the eTC case, the period of interest stops at the first green curve peak. Both engines are operating with the same fuelling strategy and only depends on fuel to accelerate the vehicle so have the same power output potential.

As previously mentioned, and as seen in Figure 7-4, the electric motor in the eTC case is controlled to keep the turbocharger rotor speed at the 125,000 RPM target (top graph - green line).

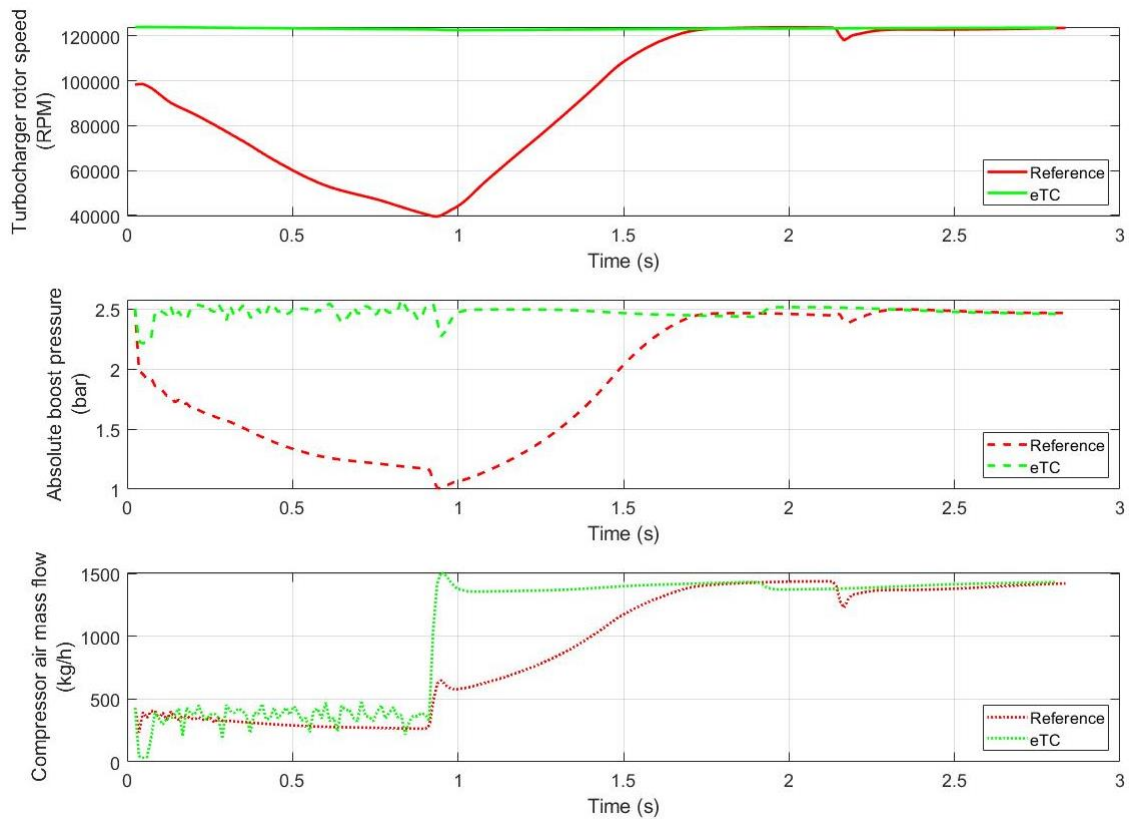


Figure 7-4 - Turbocharger rotor speed (top), absolute boost pressure (middle) and compressor air mass flow (bottom) evolutions for the reference and eTC cases

It can be noticed in Figure 7-4 that the compressor mass flow and boost pressure for the eTC case are very erratic at the beginning of the simulation suggesting that the compressor is operating around the surge line. This is due to the compressor spinning at high speed while the engine mass flow is limited by the throttle. To avoid this, the throttle angle could be increased to let more air flow through the compressor and the excess air could be recirculated using a dedicated valve.

On the other hand, looking at the red solid line (top graph in Figure 7-4), it can be noted that turbo lag is being modelled realistically. In this case, turbo lag is defined as the time necessary for the conventional turbocharger to reach the rotor speed of the eTC from the start of the 'tip-in' manoeuvre (here $t = 0.9\text{ s}$).

Otherwise, if this measure is not available, the evolution of inlet boost pressure can be used to measure turbo lag. Given the specific narrow operating window of the studied fuel-limited Formula One™ engine, in its full-load behaviour, the compressor provides the engine with a nearly constant boost pressure which can serve as a reference to estimate turbo lag.

In the reference case of Figure 7-4, all values are dropping for the first part of the simulation as the turbine is supplied with low exhaust enthalpy. Then, when full load is demanded, it takes time to build up rotor speed and thus boost pressure and air mass flow.

As the turbocharger is kept spinning at high speed during part-throttle operation in the eTC case, when the throttle is fully opened and 100% load is demanded, Figure 7-4 shows that the turbocharger is nearly immediately capable of supplying the engine with the requisite mass flow at the required pressure. As a result, the engine can maintain a higher air/fuel ratio than with a conventional turbocharger and reach higher efficiency as shown in Figure 7-5.

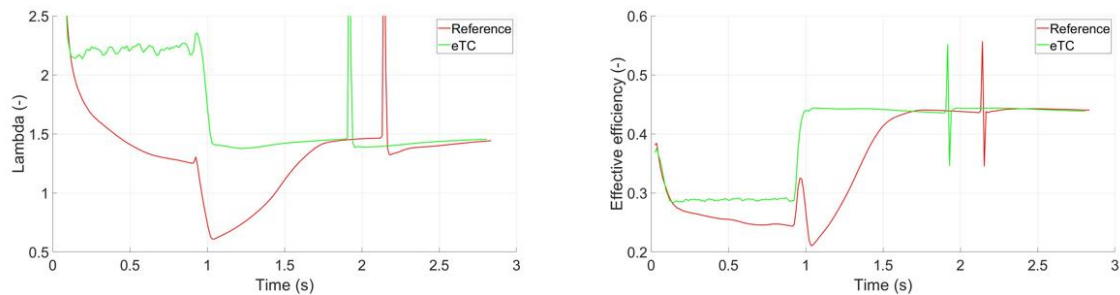


Figure 7-5 - Lambda and effective efficiency evolutions for the reference and eTC cases

In both the reference and the eTC cases, when 100% load is demanded, the model uses the maximum amount of fuel it is allowed to use under the published 2026 regulations (3000MJ/h above 10,500 RPM [15]), but because the eTC engine operation yields higher efficiency, the engine can accelerate faster to the gear change than in the reference case by 225 ms.

In addition to being useful when the engine ‘tips in’ to full load, the eTC also brings advantages during gear shifts. Fuel injection is cut during gear shift in these models meaning that the exhaust enthalpy briefly drops as there is no combustion. The turbine is thus not able to recover sufficient energy and the turbocharger rotor slows down. Because of the higher moment of inertia of the eTC compared to a regular turbocharger, a smaller reduction in rotor speed is observed, i.e. less turbo lag during gear shift. Plus, the electric motor is capable of momentarily modify its power supply to ensure constant speed operation, as can be seen in Figure 7-4.

Nevertheless, it is important to highlight that current Formula One™ shifting time are very short, taken here to be 12 ms, compared to more conventional manual gearboxes meaning that shifting lag is less of a concern as the exhaust enthalpy is not reduced over a long period.

7.1.4 Simulation results – Impact of the MGU-K on transient behaviour of the engine and turbocharger

As seen in Figure 7-6, the MGU-K case offers the shortest manoeuvre time despite a higher drop in engine speed than the eTC case for the opening segment of part throttle operation.

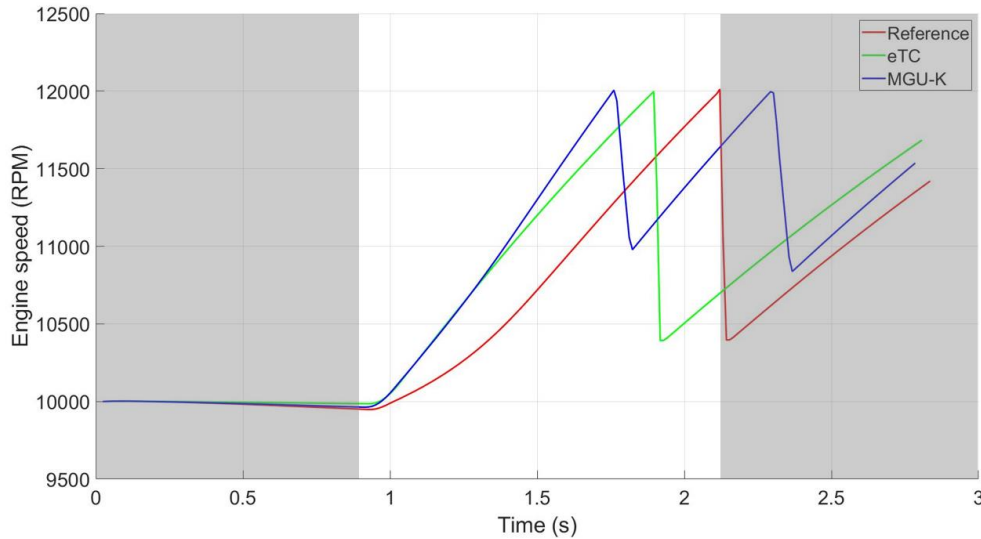


Figure 7-6 - Engine speed evolution for the reference, eTC and MGU-K cases

It is, of course, the most powerful PU as it combines the power of the ICE and of the MGU-K. In the MGU-K case, as the turbocharger is not electrically assisted, its speed will decrease during part-throttle operation due to low exhaust enthalpy (Figure 7-7) leading to lower air/fuel ratios and hence lower fuel conversion efficiency (Figure 7-8) and thus lower torque produced to counter total engine friction and pumping losses. As evident in Figure 7-7, the evolution of the turbocharger behaviour is very similar for the reference and MGU-K cases.

However, the engine speed drop in the MGU-K case is lower than that of the reference one because of the higher moment of inertia of the engine / electric motor assembly.

The higher inertia could also be a disadvantage for acceleration performance but thanks to the high MGU-K power (350 kW [15]), this configuration yields the shorter manoeuvre time. The other big impact of the MGU-K added moment of inertia, as observable in Figure 7-6, is that the engine speed drop during gear shift is reduced compared to the other cases.

Inspecting Figure 7-7, it is interesting to note that the reference (red lines) and MGU-K (blue lines) cases are nearly identical at the beginning of the acceleration phase before diverging.

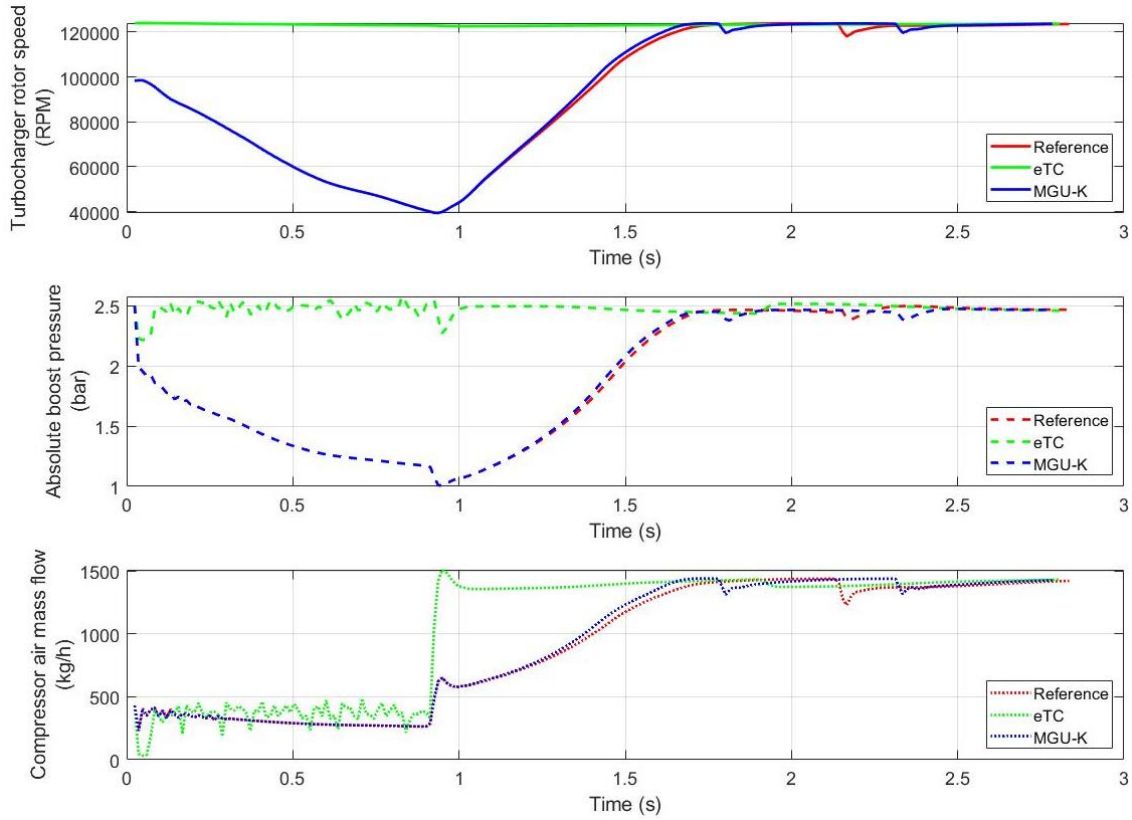


Figure 7-7 - Turbocharger rotor speed (top), absolute boost pressure (middle) and compressor air mass flow (bottom) evolutions for the reference, eTC and MGU-K cases

This can be explained by looking at Figure 7-8, on the left-hand side graph, the MGU-K case has the slightly lower excess air ratio values between $t = 1$ and $t = 1.3$ s. As seen in Figure 7-7, in this period, the turbocharger is spinning at the same speed for both the reference and MGU-K cases meaning that the boost pressures and compressor mass flow are also similar. However, as the engine is spinning faster in the MGU-K case due to the additional torque brought by the EM and because of the chosen fuelling strategy which matches the fuel flow limit given by the regulations and which is a function of engine speed, the fuel mass flow increases at a faster rate in the MGU-K case than in the reference one.

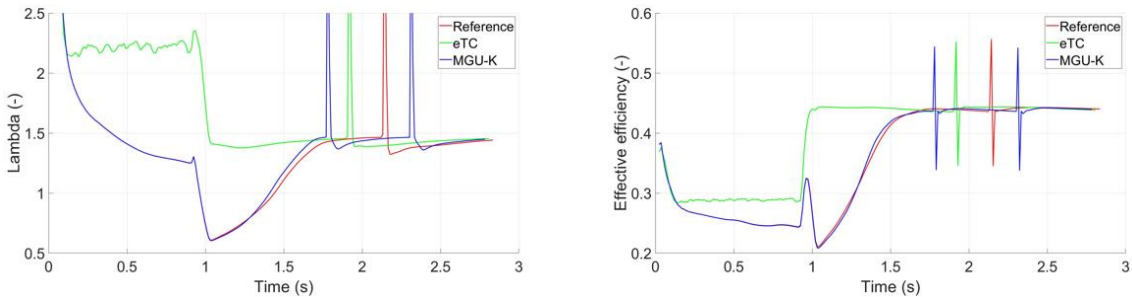


Figure 7-8 - Lambda and effective efficiency evolutions for the reference, eTC and MGU-K cases

Thus, with a higher fuel mass flow but a similar air mass flow, momentarily lower air/fuel ratio and efficiency values are achieved (Figure 7-8).

However, because of this faster increase in fuel flow rate, the exhaust enthalpy is higher in the MGU-K case than that in the reference. In consequence of the higher exhaust enthalpy, the turbine can achieve more work and thus spin the turbocharger rotor faster with the drawback of operating momentarily the engine at lower efficiency (Figure 7-8). This explains the difference in turbocharger behaviour between both the reference and MGU-K observable in Figure 7-7.

In the fuelling strategy chosen in this study, the fuel enrichment discussed above is not deliberate, it is a consequence of determining the fuel mass flow only based on the load demand and not considering compressor air mass flow supply. When the ECU instructs full load from the engine, the fuel supply is thus automatically set to the maximum even though it takes longer for the air mass flow to correct to the new engine speed and load condition. This strategy works very well with the eTC which is capable of immediately supplying the engine with the right air mass flow and thus allows to operate at high air/fuel ratios and maximum efficiency most of the time as shown in Figure 7-8.

Even though it results in lower fuel conversion efficiency at the beginning of the acceleration, this strategy in the MGU-K case leads to shorter acceleration time for the turbocharger rotor, enabled by the higher exhaust enthalpy, which then allows for higher fuel conversion efficiencies to be subsequently achieved faster thanks to the turbocharger delivering high boost pressure and mass flow earlier (Figure 7-7). As shown in Figure 7-8, because of the difference in engine speed evolution, the MGU-K case achieves 40% effective efficiency faster than the reference case by 25 ms ($t = 1.44\text{ s}$ vs $t = 1.465\text{ s}$).

Despite this behaviour, the MGU-K case remains less efficient than the eTC during this specific manoeuvre as seen in Figure 7-8. As the turbocharger rotor speed is kept constant, when 100% load is demanded, the eTC immediately provides the engine with the right air mass flow leading to lean operation and an efficiency advantage over more than 0.6 s compared to the MGU-K case.

7.1.5 Assessment of the difference in engine performance depending on the position of the electric motor on the engine

As discussed, the MGU-K option is the fastest route to increased engine speed compared to the other cases (Figure 7-6) and this is obviously also the case when considering vehicle speed, see Figure 7-9, as gear ratios are the same for all cases. The vehicle model used for these simulations assumes that the tyres can transmit all torque produced by the Power Unit without breaking traction. To test this assumption, the average acceleration for the MGU-K case from $t = 0.9\text{ s}$ to the end of the simulation is of 10.68 m/s^2 (1.09 g) which is commonplace for current Formula One™ cars and therefore the assumption is taken as sufficiently robust.

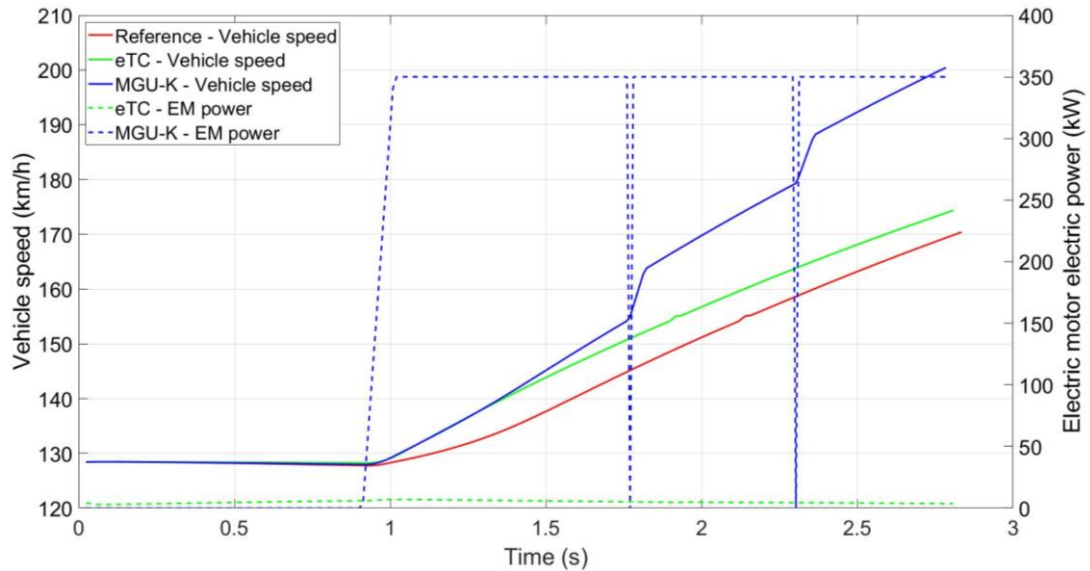


Figure 7-9 – Vehicle speed (solid lines) and electric motor electric power (dashed lines) evolutions for the reference, eTC and MGU-K cases

Note in Figure 7-9 the very sudden increase in vehicle speed when the clutch is reengaging after a gear change for the MGU-K case. This behaviour is unrealistic and due to limitations in the way the clutch is modelled in AVL Boost™. The higher the moment of inertia, the bigger the step, explaining why this phenomenon is particularly visible for the MGU-K case and not the two others. But, because the period of interest for this study stops right before the gear change, this phenomenon does not influence the results presented here.

Regarding powertrain performance, because of the difference in power between the electric motor of the eTC and the MGU-K, the vehicle performance comparison can be biased. As visible in Figure 7-9 (dashed lines), the difference between the MGU-K and the eTC's EM power is very large with respective peak power of 350 kW and 7.1 kW. Table 7-3 shows how much energy is consumed in each case and the manoeuvre time.

Case	Manoeuvre time (s)	Fuel energy (kJ)	Electric energy (kJ)	Total energy consumption (kJ)
Reference	1.221	961.8	N/A	961.8
eTC	0.996	780.2	10.0	790.2
MGU-K	0.861	667.1	279.5	946.6

Table 7-3 - Energy consumption during the studied manoeuvre for the reference, eTC and MGU-K cases

For the eTC case, keeping the turbocharger rotor speed constant during the part throttle operation is a fundamental control goal so, in Table 7-3, the electric energy used in this period is also accounted whereas for the MGU-K and for fuel energy, only the period between full load demand ($t = 0.9$ s) and the first gear shift is considered.

Importantly, even though it is the fastest solution, the MGU-K case also consumes more energy than the eTC (+156.4 kJ, +19.8% relative increase) which is a consequence of the very high-power output of the MGU-K.

For the eTC case, the fuel energy is the only source of propulsion for the vehicle with the electric energy being solely used to control the turbocharger speed. However, thanks to the constant speed control regime, the fuel energy is more efficiently converted into work as demonstrated in Figure 7-8, leading to lower manoeuvre time compared to the reference case. The 10 kJ of electric energy used to power the eTC is an investment to maximise fuel conversion efficiency. Over the investigated manoeuvre, the average effective efficiency of the eTC case is 6 and 8.9 percentage points higher than that of the reference and MGU-K cases respectively leading to lower total energy consumption (-171.2 and -156.4 kJ respectively)

However, even though the eTC case requires more fuel energy than the MGU-K case, the huge difference in electrical energy consumption must be highlighted, the MGU-K requiring nearly thirty times more electrical energy than the eTC for a 0.135 s gain in manoeuvre time.

In addition, the 2026 Formula One™ Power Unit Regulations states that a maximum of 9 MJ can be recovered each lap and that the difference between the maximum and the minimum state of charge of the battery cannot be greater than 4 MJ at any time the car is on track meaning that the vehicle can deploy a maximum of 13 MJ for a given lap. In the general and simplified case where the acceleration manoeuvre studied in this paper is repeated on subsequent segments of the track, the MGU-K can only operate for circa 40 s within the 13 MJ limit against 1294.8 s for the eTC before consuming all the energy available for a lap (considering the data in Table 7-3). Or put another way, the MGU-K manoeuvre described above can be undertaken around a dozen times.

The reference case is only relying on fuel to accelerate the engine and because of the initial drop in air/fuel ratio and efficiency, it requires the most time and total energy to reach gear change at 12,000 RPM. It must also be highlighted that at $t = 0.9$ s, this is the case for which the engine speed is the lowest, so the change in engine speed is greater than for the other cases. The MGU-K case significantly reduces the fuel consumption compared to the reference but requires 279.5 kJ of electric energy to achieve that. This results in the total energy consumption being reduced by only 1.5% when using the MGU-K, despite the 29.5% shorter manoeuvre time.

Traditionally, the impact of turbo lag on engine performance is assessed by measuring the time necessary to complete a given manoeuvre, for example, the time required for torque or engine speed to rise from and to user-defined values [90]. Considering the fuel-limited nature of the 2026 Formula One™ engine, efficiently using the limited available energy is key, the fastest solution might not allow the car to reach the end of the race if it consumes too much energy. This is why the amount of energy consumed by the Power Unit (fuel and electric energy) to accelerate the car must be integrated in the assessment of the different configurations, in addition to manoeuvre time.

For this study, a new performance indicator based on the ratio of vehicle acceleration to energy input is offered. This integrates the acceleration capacity of the vehicle between the ‘tip-in’ moment ($t = 0.9 \text{ s}$) and the first gear shift and the energy consumption. The performance indicator is defined by the following equation:

$$I_{performaces} (g/MJ) = \frac{\left(\frac{V_{gearshift} - V_{t=0.9}}{t_{gearshift} - 0.9}\right) \times \frac{1}{g}}{\int_{t=0.9s}^{t_{gearshift}} \left(\frac{dE_{electric} + Q_{lwh} \times dm_{fuel}}{dt}\right) \times dt} \quad (7-1)$$

Using this equation, the vehicle acceleration effectiveness indicators have been calculated for each case and are shown in Figure 7-10. The performance of the reference case is much lower than the two others showing how impacting turbo lag is on vehicle performance.

The performance of the eTC and MGU-K cases are very close. However, according to the vehicle acceleration effectiveness indicator, the eTC solution makes better use of the energy. Even when accounting for the energy required to keep the turbocharger spinning during the part-throttle operation, the fact that the engine efficiency is kept high any time during full load demand makes a big difference. Put another way, investing a small amount of electric energy to maintain fuel conversion efficiency high makes a big difference.

The MGU-K cases can be close to the eTC performance indicator value thanks to its much faster manoeuvre time but is penalised by its higher energy consumption. If the MGU-K power was even higher, both cases could have the same performance indicator value but that would be at the expense of an even higher electric consumption.

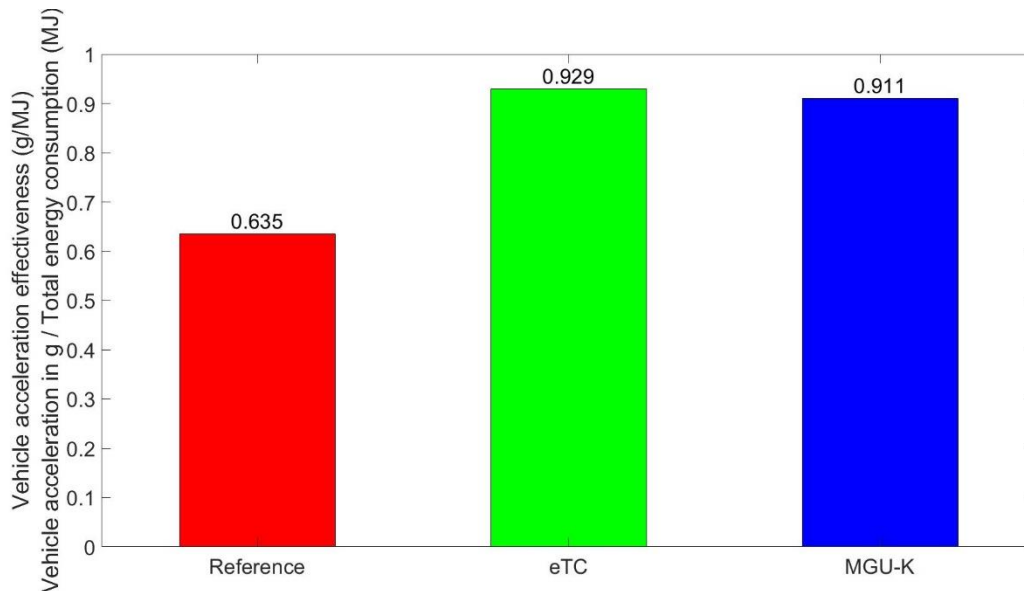


Figure 7-10 - Performance indicators for the reference, eTC and MGU-K cases

Nevertheless, it must be highlighted that the eTC case is only able to compete with the MGU-K configuration at the beginning of the acceleration event when the

ICE is exposed to turbo lag. As soon as the turbocharger speed has risen to its optimal operating value, the total power output of the MGU-K case Power Unit is significantly higher than that of the eTC which only has the ICE to accelerate the vehicle. The difference in acceleration capacity over a longer period of time of the different configurations is shown in Figure 7-9.

7.1.6 Conclusions on the impact of the electric motor on the ICE

The simulations reported here demonstrate that coupling a single electric motor to a turbocharged internal combustion engine, either by connecting it to the crankshaft or to the turbocharger rotor, helps to improve the transient behaviour of the Power Unit and vehicle and more importantly for motorsport applications, enhances vehicle acceleration performance. The reference case, which does not feature any electric motor, is subject to turbo lag without any compensation and yields the highest time and total energy consumption for the sudden acceleration manoeuvre studied.

The electrically assisted turbocharger (eTC) enables the direct control of the rotating speed of the turbocharger rotor independently of the conditions at the turbine and thus to effectively eliminate turbo lag. As demonstrated in this study, this method is very effective as it maintains air/fuel ratio and engine fuel conversion efficiency high, allowing to make the most of the limited amount of fuel available and that only by using a small amount of electric energy. This device is without doubt a contributor to the 50% efficiency reportedly achieved by the current Formula One™ Power Units [13]; [14] and observed in this research work with the 2014+ Digital Twin.

On the other hand, the 350 kW MGU-K proved to be a most useful device to accelerate the vehicle. Even though the engine simulations showed that the engine suffers from turbo lag during the first phase of acceleration, the 'torque-to-fill' capacity of the MGU-K allows rapid acceleration of the vehicle despite delay in the turbocharger response, but this is at the expense of low fuel conversion efficiency at the beginning of the manoeuvre. However, the MGU-K can also be used during braking events to recover kinetic energy, which is not investigated and considered in this study.

In addition, it was demonstrated that the non-deliberate fuel enrichment allowed by the applied fuel strategy and the engine speed increase due to the MGU-K leads to higher exhaust enthalpy and thus a more responsive behaviour of the turbocharger. However, to operate, the MGU-K requires a significant amount of electric energy meaning that given the electric consumption dictated by the regulations, it can only be used for circa 40 s per lap for the specific manoeuvre studied here. Furthermore, when comparing the acceleration performance normalised by the total amount of energy used, the MGU-K is less efficient than the eTC.

Even though, the MGU-K will be of high-importance for vehicle performances in 2026, the Formula One™ engine manufacturers will have to implement additional

solutions to mitigate the effect of turbo lag considering the quantum of energy that must be drawn out of the battery to power the MGU-K.

The fuel strategy implemented in this study was interesting to compare the eTC and MGU-K cases but will not be viable for 2026 because of the significant drop in air/fuel ratio and efficiency at the beginning of the manoeuvre. However, if the fuel strategy is modified, for example, not to ever drop below $\lambda = 1$, the effective efficiency is increased but less fuel is burnt so the vehicle acceleration is slower, and it takes more time for the turbocharger speed to rise as seen in Figure 7-11:

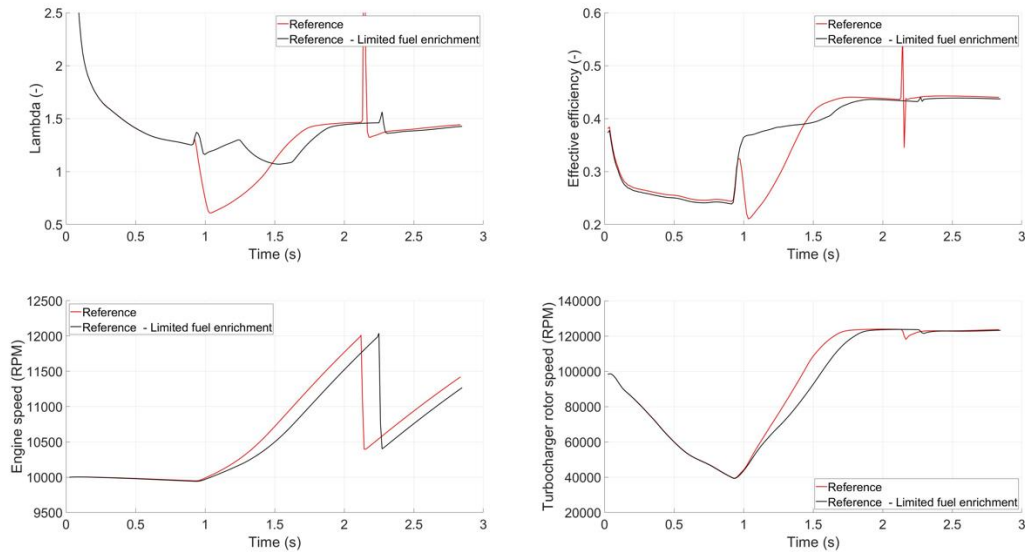


Figure 7-11 - Lambda (top left), effective efficiency (top right), engine speed (bottom left) and turbocharger rotor speed (bottom right) evolutions for the reference case and the reference case with a fuel enrichment limiting strategy

Engine manufacturers will need to find a balance between maintaining high fuel conversion efficiency and rapidly accelerating the turbocharger rotor. For instance, it can be useful to briefly sacrifice fuel conversion efficiency, albeit less than observed in this study, to speed up the turbocharger and achieve high efficiencies faster. It must be kept in mind that additional anti-lag solutions will also be implemented by the engine manufacturers.

The situation simulated here also results in a very severe turbocharger rotor speed drop during the first part of the simulation, this is an extreme case. On a less severe turbo lag cases, the results for the reference and MGU-K cases would be closer to that of the eTC thanks to higher compressor air mass flow and thus higher air/fuel ratios and fuel conversion efficiency.

In a more general context, coupling the ICE with an electric motor helps to improve performance. If the aim of the Power Unit is to maximise transient efficiency, the electric assisted turbocharger is the solution to select, and it only requires modest electric energy supply. If the objective is highest vehicle acceleration performance, the high-power MGU-K is preferred, but at the cost of high electric energy consumption and lower average fuel conversion efficiency than with an eTC. Obviously, if allowed, the best solution for efficiency and vehicle performance is to combine both solutions (as done with the 2014+ Formula

One™ Power Unit). Any of the electrical hybrid solutions outperform the powertrain that has no electrical assistance.

Energy management strategies are not discussed in this paper but due to their hybrid nature, the Formula One™ Power Units must be developed as a system to maximise the efficiency of the overall system to make the best use of the limited energy. For example, the MGU-K can be used to ‘torque fill’ as investigated in this paper but also to ‘torque consume’ during low torque demand from the driver as investigated in the following section.

7.2 Assessment of the MGU-K as a torque consuming device for the 2026 Formula One™ Power Unit to minimise turbo lag

7.2.1 Presentation of the “torque consuming” concept

As seen in Section 7.1, the 350 kW MGU-K featured in the 2026 Formula One™ Power Unit consumes large amount of electric energy when utilised at full torque capacity. To make sure it is always available for use in race-critical moments, for example at the beginning of a straight or defence against an opponent trying to overtake, energy regeneration and storage must be optimised, and the least power is required from the MGU-K to accelerate the car, the better and this is linked to reducing turbo lag to maximise engine output power.

To do so, the interaction between the internal combustion engine and the MGU-K can be assessed further than the torque filling only as investigated in Section 7.1.

To reduce turbo lag, exhaust enthalpy must be kept high so that the turbine can recover enough energy to maintain the turbocharger rotor speed. Exhaust enthalpy depends on mass flow but also on temperature meaning that having a high exhaust enthalpy is linked to fuel combustion.

As the 2026 Formula One™ engine is fuel-limited, and therefore the imperative is that fuel energy should only be used to generate work output, the combustion timing cannot be excessively retarded so that most of the combustion happens in the exhaust system to generate high exhaust temperature as an anti-lag solution.

The idea behind the ‘torque consuming’ concept discussed here, is to operate the engine at higher load than demanded by the driver or even acceptable at the tyre contact patch. By doing so, the throttle is open at higher angle, reducing throttle and pumping losses, the engine mass flow increases, and fuel is injected to be burned, meaning that exhaust enthalpy is kept high. In consequence, the turbine can recover more energy than if the engine was running at lower load, the turbocharger speed and fuel conversion efficiency of the ICE are kept high.

The excess energy, that is not required by the driver to accelerate the vehicle according to their demand for torque, is recovered using the MGU-K.

In this operating mode, the MGU-K is used 'against' or 'in resistance to' the ICE to recharge the battery when the vehicle is traction limited preventing more torque from the engine to be used for propulsion and/or when the MGU-K is not already recovering energy at full power during regenerative braking. Considering that the MGU-K power is high and close to that of the ICE, it makes possible the recovery of large amount of energy to add to that from conventional regenerative braking. However, there is a circa 10% energy conversion penalty to pay each time energy is transferred to the ERS and vice versa.

Nevertheless, if this mode allows the engine to run at significantly higher fuel conversion efficiency, for the same amount of fuel burnt and the same amount of energy directly sent to the wheel, despite the 90% efficiency of the electric system, a higher fraction of the fuel will be transformed into useful work.

7.2.2 Presentation of the torque consuming modes investigated in this case study

The idea of this case study is to investigate torque consuming as an anti-lag solution, so in situations where the engine is exposed to turbo lag. It can also be used in other situations, for example at the end of a straight if low power is required to power the car as currently done with the 2014+ Formula One™ Power Unit but that is not investigated here.

Thus, this case study will focus on the usefulness of torque consuming during braking event and the first stages of an acceleration event.

Torque consuming during braking:

During braking events, the MGU-K is partly used in place of the rear mechanical brakes to slow down the car and thus recovers kinetic energy which otherwise is lost in form of heat because of the friction between brake discs and pads.

Thanks to the higher power of the 2026 MGU-K, more energy can be recovered during braking than with the 2014+ Formula One™ regulations. However, as the car slows down, tyre grip typically reduces with reducing downforce meaning that less and less braking torque can be applied on the rear tyres. Thus, there is a moment where the MGU-K cannot operate at maximum power anymore, and this is when torque consuming during braking (TCB) can intervene if the SOC of the ES needs to be raised further.

When the MGU-K cannot operate at maximum braking power, the ECU can compute the difference between the MGU-K maximum power and the power at which it currently operates. By knowing the behaviour of the engine from engine dyno testing, the ECU can determine at which load to operate the ICE so that the power produced can be recovered by the MGU-K to maximise its recovering capacity but without compromising the car deceleration.

As the braking needs will progressively decrease, engine load can be increased accordingly so that the MGU-K recovering power is kept high for as long as possible or required.

By using this mode, the idea is that the throttle is more open, engine mass flow is high, and fuel is burned so exhaust temperature is also high which results in significant amount of exhaust enthalpy and thus of energy for the turbine to recover which minimises the turbocharger rotor drop during a braking event. When the driver starts accelerating again, the turbocharger is already spinning at high speed as it is provided with high exhaust enthalpy, so the engine is theoretically more responsive to further change in load demand from the driver (i.e. turbo lag is reduced).

Torque consuming during acceleration:

After a braking event, the driver progressively demands torque from the engine in response to tyre grip increasing as vehicle speed and thus downforce build up.

As the driver demand progressively increases, the throttle opening increases accordingly meaning that throttle and pumping losses are still impacting the engine performance at the beginning of the acceleration event.

In the torque consuming during acceleration (TCA) simulation case considered here, as soon as the driver starts accelerating, the engine load signal is set to 100%. Based on the throttle pedal position and the power produced by the engine, the ECU computes the difference with the power demand. If the ICE power output is higher than demanded, then the MGU-K is used to absorb it. Otherwise, the MGU-K torque filling capacity is used.

By doing so, the throttle switches directly to fully open, thus minimising the throttle and pumping losses, more fuel is burned so exhaust enthalpy is increased leading to a higher rate of change in turbocharger rotor speed and additional energy is recovered and stored in the ES if the MGU-K torque consumes.

It must be highlighted that this mode can be combined with the previously presented one so that the MGU-K is used for torque consuming both during braking and acceleration. This is the TCAB mode. These three modes are investigated in this case study using AVL Boost™ transient engine simulations.

Figure 7-12 shows a schematic summarising the different possible uses of the MGU-K within the 2026 Formula One™ Power Unit.

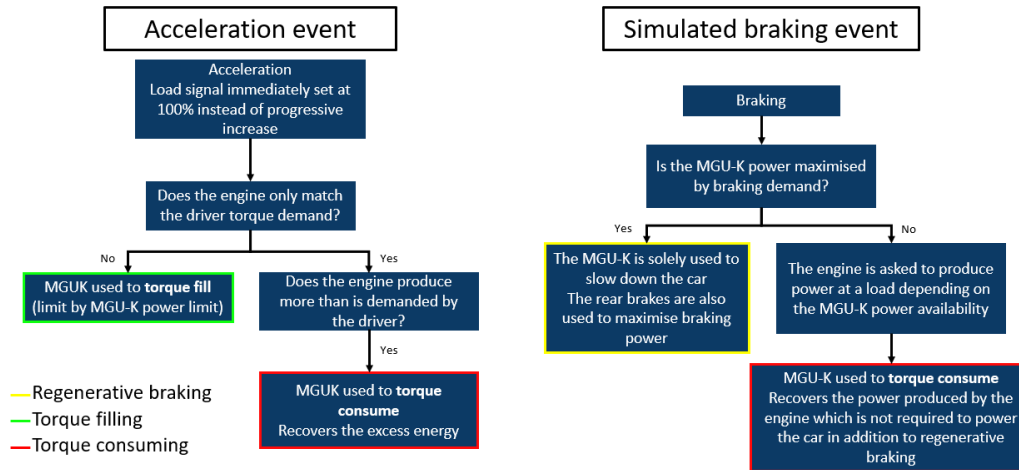


Figure 7-12 - Different possible MGU-K usages for the 2026 Formula One™ Power Unit

7.2.3 Modelling of the torque consuming modes using AVL Boost™

AVL Boost™ transient engine model

Torque consuming is investigated using the AVL Boost™ transient model of the 2026 Formula One™ engine presented in Chapter 4 and shown in Figure 7-13 as a reminder. The modelling of the different ICE/MGU-K interaction modes requires more complex algorithms than the controls allowed with the ECU model of AVL Boost™, this is why the Formula Interpreter (FI1 in Figure 7-13) is used.

Compared to the models presented in Section 7.1, the fuelling strategy has been modified to consider, in addition to engine speed, inlet air mass flow measured at the inlet reference measuring point (MP14).

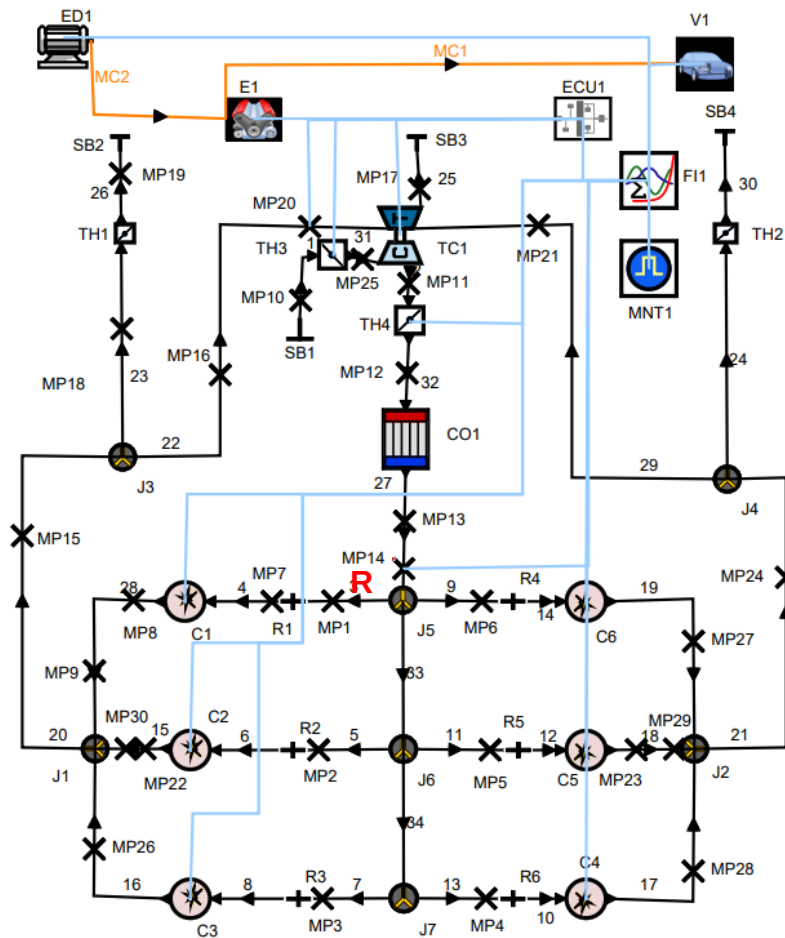


Figure 7-13 - AVL Boost™ transient engine model used to investigate the ICE/MGU-K interaction modes

For each engine speed, a target air/fuel ratio is defined. The evolution of target air/fuel ratio follows the same curve applied to the steady-state engine models and is reminded in Figure 7-14.

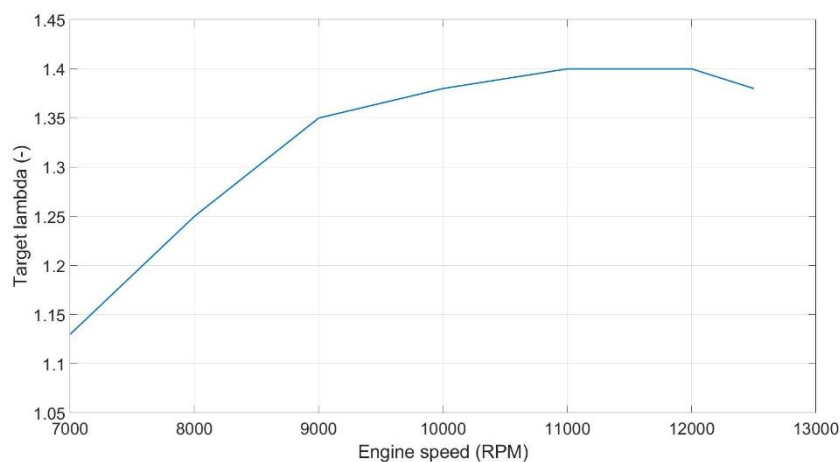


Figure 7-14 - Target lambda for torque consuming transient simulations

If engine speed is below 7,000 RPM, the target lambda is set at 1.1 and at 1.3 if engine speed is above 12,500 RPM.

The ECU in the model tries to maximise the quantity of fuel at any time up to the energy limit imposed by regulations. However, on the contrary to the previous transient model, if the inlet air mass flow is not sufficient to burn the quantity of fuel dictated by the fuel energy flow limit at the desired air/fuel ratio, less fuel is injected.

It is considered that if air mass flow is not sufficient, it means that the turbocharger rotor speed is not optimal (turbo lag), so the target lambda is voluntarily reduced to slightly increase exhaust enthalpy and the rate of change of turbocharger rotor speed. This is an example of fuelling strategy to balance fuel conversion efficiency and turbo lag management as discussed at the end of Section 7.1.

The minimum lambda λ_{min} is set at 1 to ensure that all the fuel energy is still released by complete combustion, and that fuel is not wasted in the exhaust. Based on the actual air mass flow $\dot{m}_{air_measured}$, the target lambda λ_{target} , the air mass flow required to meet the target lambda with full fuel flow $\dot{m}_{air_required}$ and the minimum possible lambda λ_{min} , the updated lambda target $\lambda_{updated_target}$ is calculated (Equation 7-2), and the injected fuel mass flow is computed using this value and the air mass flow.

$$\lambda_{updated_target} = \lambda_{min} + (\lambda_{target} - \lambda_{min}) \frac{\dot{m}_{air_measured}}{\dot{m}_{air_required}} \quad (7-2)$$

The fuelling strategy is summarised in Figure 7-15:

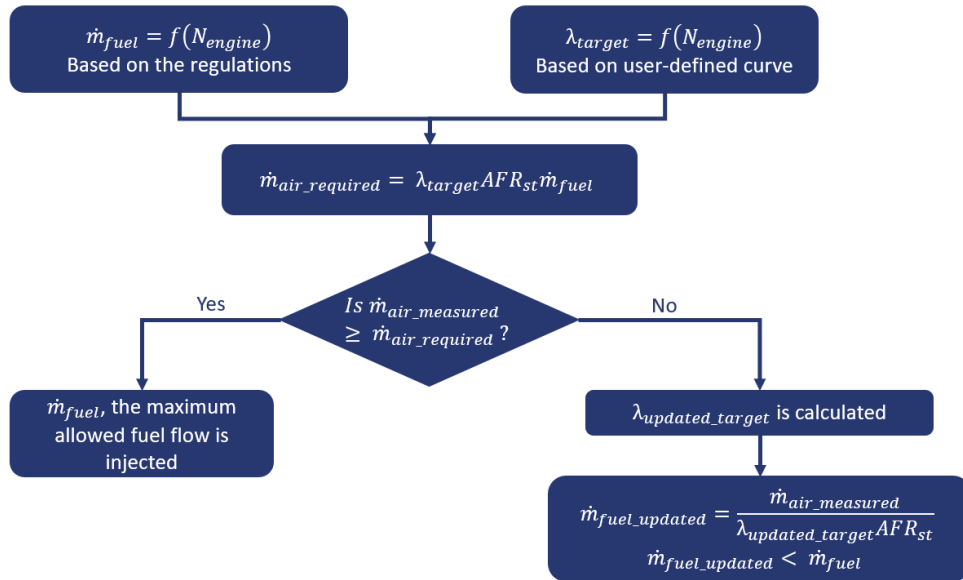


Figure 7-15 - Fuelling strategy for torque consuming transient simulations

Fuelling and MGU-K supply are cut during gear changes and as explained in Chapter 4, during braking events fuelling is cut but not MGU-K recovery.

The Energy Storage (ES) is not modelled in this simulation as its exact behaviour is not of interest for this case study. It is assumed that the MGU-K can be supplied

at any time to the required electric power and that the ES can store all the energy recovered by the MGU-K.

As presented in Chapter 4, for transient simulations to run in AVL Boost™, a resistive load must be applied on the engine. In this case, the vehicle model was chosen however, this is not a complex vehicle model as for a vehicle dynamic simulation. This model is just used to give a simple estimation of the load that a vehicle would apply on the engine. This load depends on a constant rolling resistance and a user-defined drag coefficient as presented in Chapter 4. This modelling approach is similar to the assumptions used when operating an ICE on a transient dynamometer.

In the next sections, 'vehicle speed' is mentioned. This speed is the theoretical / idealised speed that would be observed at the tyre based on the speed the engine is running at, the theoretical current gear ratio and a non-changing tyre rolling radius with zero slip. This is not an actual vehicle speed but an approximate equivalent or notional vehicle speed.

As for the vehicle, no gearbox is modelled in the engine model but the effect of changing gear (drop in engine speed, fuelling cut, etc.) is modelled in a simplified way using the vehicle model provided by AVL Boost™.

For the simulated speed events which are later presented, there is the need to assess how the MGU-K would behave during braking events for regenerative braking, therefore an estimation of the maximum power that can be applied by the MGU-K on the rear tyres is required.

As the aim of this study is not to investigate how the vehicle behaviour would affect the braking event but just to investigate how the MGU-K could be used to torque consume during braking, an estimation of the force that can be applied for braking is calculated in a very simplified way that only considers vehicle weight, downforce and a constant friction coefficient as presented in Chapter 4 (Equation 4-9). The equation used for this estimation is provided as a reminder here:

$$\text{Braking force (N)} = \mu \times \left(\frac{1}{2} \times \rho \times C_L \times A_{car} \times V^2 + m_{car} \times g \right) \quad (7-3)$$

As only an estimation of the braking force is required for this case study, factors such as steering angle, weight transfer, individual car corner weight are not considered. This approach leads to a maximum deceleration of 5 g when braking from 320 km/h which is the value reported for actual Formula One™ vehicles [178]. The 60/40 brake bias balance is then used to estimate the braking force applicable to the rear tyres.

Estimation of the applicable force at the tyre is only done for braking. For acceleration, another approach, presented later in this section, is adopted.

Four ICE/MGU-K interaction modes are simulated and compared in this case study:

- **TFO:** This is the reference mode where the use of the MGU-K is what would obviously be done with an electric motor in P1 position. The MGU-K is used for regenerative braking, to increase the SOC of the ES, and for torque filling, i.e. bring additional torque to what the engine can provide to meet the torque demand.
- **TCB:** This mode does everything that the TFO mode does but in addition, the MGU-K is used to torque consume during braking following the concept presented in Section 7.2.2.

To do torque consuming, the engine must produce power, i.e. burn fuel. This fuel is not used to power the vehicle but only for regeneration in an already fuel-limited ICE. This mode thus must be used only in situation where the opportunities for fuel-to-electric energy conversion are high. If the MGU-K is only capable of recovering say 20 kW from the engine in addition to 330 kW of regenerative braking, it means that the engine must be operated at very low load to produce so little power, which corresponds to low fuel conversion efficiency conditions and high pumping losses and therefore is unattractive.

This is why torque consuming during braking is activated in this simulated case study only if the ECU detects that the MGU-K can recover at least 200 kW from the ICE (arbitrary value which correspond to approximately 60% of the engine peak power thus enabling it to run at high load when TCB is activated). The load at which the engine is instructed to operate at is based on the torque consuming capacity of the MGU-K and the maximum achievable power by the engine. When the MGU-K is capable of torque consuming 200 kW from the engine in addition to regenerative braking, the engine starts producing power and the MGU-K regenerative power is increased to be closer to its 350 kW limit.

- **TCA:** Similarly, this mode operates the MGU-K for torque filling and regenerative braking but in addition, it is used for torque consuming during acceleration following the concept described in Section 7.2.2.

During simulated braking events, the MGU-K only does regenerative braking, no torque consuming at all.

In this case study, from the moment the torque demand becomes non-zero, the engine load is switched to 100% and based on the difference between the power demand and the engine output, the ECU computes at

which power the MGU-K must be operated to consume the excess power not to exceed the power demand.

- **TCAB:** This mode is the combination of the three other modes. The MGU-K is used to torque fill and regenerative braking (TFO mode) but also to torque consume during braking from the moment the MGU-K can absorb an additional 200 kW from the engine (TCB mode) and to torque consume during acceleration by immediately switching engine load to 100% and absorbing the excess power compared to the demand by using the MGU-K (TCA mode).

For each of the four modes, the algorithm implemented in the Formula Interpreter is adapted so that engine load and MGU-K behaviour are relevant to the mode simulated. The MGU-K is always operated in the torque and power limits dictated by the 2026 Formula One™ Power Unit regulations [15].

Specific virtual dyno test – Presentation of the simulated speed events

Compared to the engine models presented in Section 7.1, the simulated situation has also evolved. Instead of a simple part-load / full-load case, the different Power Unit modes are here tested on specific speed events representative of common Formula One™ car operation [5].

When strictly looking at the code implemented in the Formula Interpreter of the AVL Boost™ transient engine model, engine speed evolution is dictated by engine load which is linearly linked to power demand.

However, despite the difference in ICE/MGU-K interaction modes, all simulated cases can meet the power demand but with different contributions from the ICE and the MGU-K. This means that for a given power demand, as all modes meet it, the engine speed evolution, and thus the equivalent vehicle speed, is the same for all modes. So, even if engine speed is not directly controlled, the simulations presented here are equivalent to a simulation controlled using equivalent vehicle speed vs time profiles.

Depending on the minimum speed for the simulated event, the amount of torque that the driver can apply to the tyres during the first phase of the acceleration varies because of difference in downforce and thus tyre grip. Throttle position / engine load evolution must thus evolve in consequence.

To replicate this phenomenon without a tyre model (as it is outside of the scope of this study), generic throttle application evolutions have been extracted from a real Formula One™ qualification lap data [5] for the different speed events studied here. The curves are shown in Figure 7-16 and for this case study, it is assumed that engine load / power demand is linearly dictated by these throttle position evolutions.

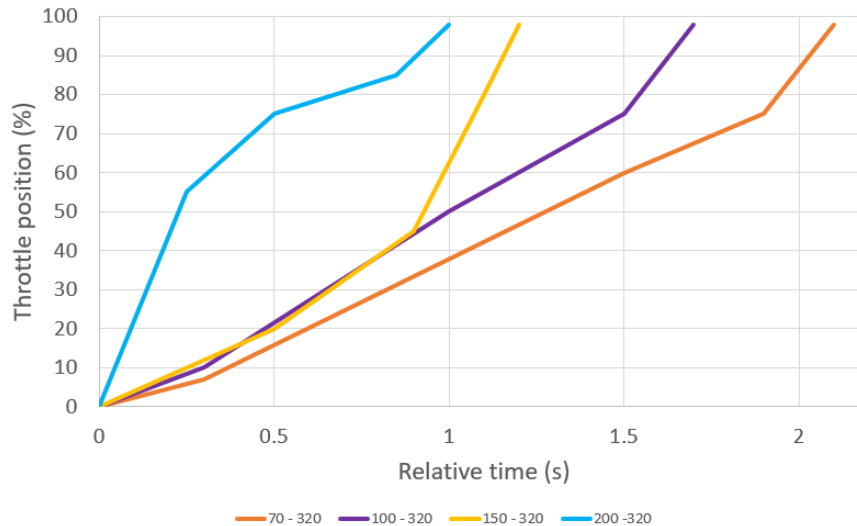


Figure 7-16 - Throttle position evolution depending on the speed event

For these specific ‘virtual dyno tests’, the simulation starts with the engine at 11,511 RPM in 5th gear, corresponding to 200 km/h in equivalent (notional) vehicle speed. The engine is instructed by a load signal to accelerate at full load, i.e. 100% power demand, until it reaches an equivalent vehicle speed of 225 km/h from where the power demand is switched to 0% and the estimated vehicle load is adapted to model a braking event until reaching a user-defined minimum equivalent vehicle speed, selected each time from these options: 70, 100, 150 or 200 km/h depending on the speed event investigated.

Then the engine load signal is progressively increased, at a rate depending on the manoeuvre minimum equivalent vehicle speed and based on the throttle application evolution shown in Figure 7-16, until reaching 100% power demand. Power demand is then kept at 100 % until 320 km/h in equivalent vehicle speed is reached. For the engine, what changes between the speed events is the simulated braking event duration, thus the turbo lag magnitude.

In this case study, the period of interest starts from the beginning of the simulated braking event until the moment the ICE reaches its nominal peak load behaviour and there is no longer any difference between ICE/MGU-K interaction modes. The speed equivalent vehicle speed at which this happens depends on the speed event investigated (150, 170, 200 and 240 km/h respectively). One representative example of the period of interest for the 70-320 km/h case is shown in Figure 7-17.

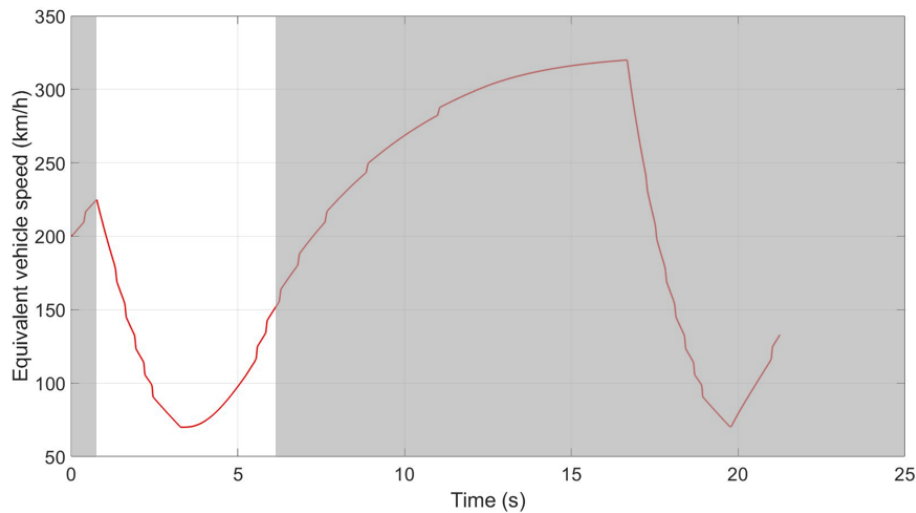


Figure 7-17 - Example of equivalent vehicle speed evolution for the 70-320 km/h speed event

As only a fraction of the speed event is of interest for this case study, only a reduced part of the simulation results will be shown in the following figures.

It can be noted in Figure 7-17 that the sudden increases in equivalent vehicle speed observed in the MGU-K configuration in Section 7.1 are also observable here. Nevertheless, despite this behaviour due to the clutch modelling in AVL Boost™ being only simple, it affects all the ICE/MGU-K interaction modes in the same way, thus does not influence the conclusions drawn in this case study.

There are thus 4 ICE/MGU-K interaction modes, each assessed on 4 different speed events for a total of 16 simulations.

In the following section, only the results observed for the first speed event (70-320 km/h) are discussed for clarity of presentation. In Section 7.2.5, the other speed events will be considered. By using this order of presentation, the behaviour of each individual ICE/MGU-K interaction mode can be understood before analysing how the impact of torque consuming on the Power Unit behaviour evolves as the manoeuvre minimum speed increases.

7.2.4 Analysis of the 2026 Formula One™ Power Unit behaviour depending on the ICE/MGU-K interaction modes for the 70-320 km/h speed event

In this section, the behaviour of the Power Unit for the four ICE/MGU-K interaction modes is investigated for the 70-320 km/h case only, i.e. the case for which the minimum speed in the investigated manoeuvre is 70 km/h. The other speed events are not investigated in such detail as the 70-320 case, but their results are presented and discussed in the next section.

It is in the 70-320 case that the engine faces the most severe turbo lag as the simulated braking event lasts longer and thus the turbine is provided with low exhaust enthalpy for the longest period. In consequence, this is the case that

shows the most the difference between the different ICE/MGU-K interaction modes investigated in this case study.

These modes are investigated with two objectives:

- Determine which ICE/MGU-K interaction mode is the most effective to mitigate turbo lag,
- Determine which ICE/MGU-K interaction mode is the most useful to maintain a high ES's SOC and reduce the dependency on the MGU-K to match the power demand.

The behaviour of each ICE/MGU-K interaction mode is observed individually to appreciate their difference before their energy consumption and anti-lag usefulness are compared.

70-320 km/h speed event – Torque Filling Only mode (TFO)

Starting with the Torque Filling Only (TFO) case, Figure 7-18 shows the evolution of equivalent vehicle speed (the simulated speed-time profile), torque (hence power) demand and the estimated available grip at the rear tyres for braking purpose (considers as negative as used to slow down the engine, see Equation 7-3).

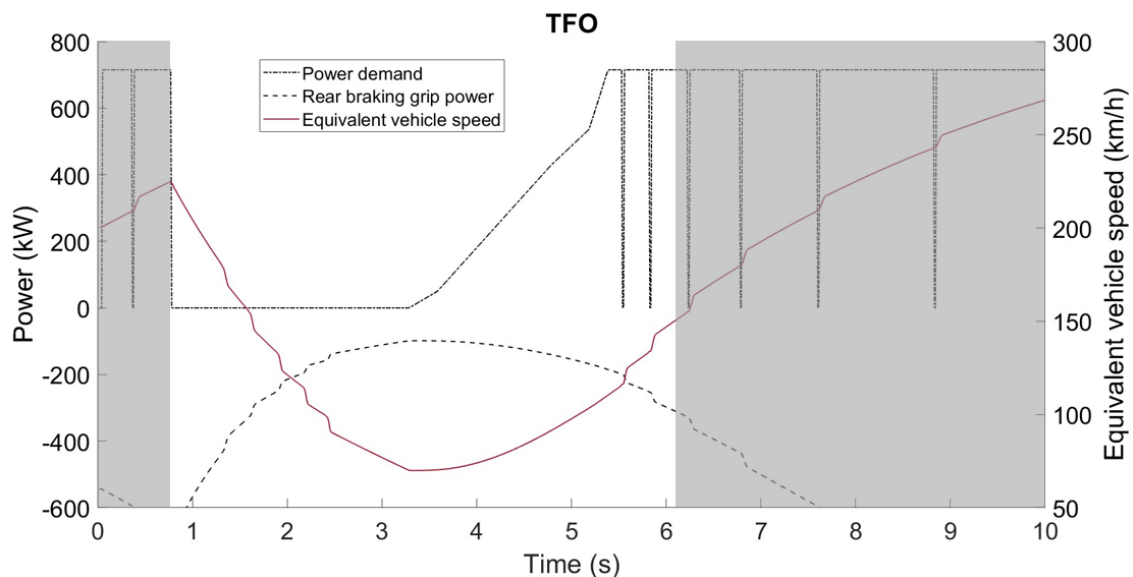


Figure 7-18 – 70-320 km/h speed event – TFO power and vehicle speed evolutions (1/3)

Based on the engine speed and simulated gear ratios, the equivalent vehicle speed at the start of the simulation is 200 km/h and accelerates up to 225 km/h. From here, a braking event is simulated until 70 km/h is reached. Then the vehicle speed raises again, up to 320 km/h where a second braking event is simulated but because at this point, the behaviours of each ICE/MGU-K are the same, Figure 7-18 only focuses on the period around the interesting part of the simulation and the second braking event is not shown.

In Figure 7-18 and for the 70-320 km/h case, the period of interest, which is shown between the grey areas, starts from the beginning of the first braking event to $t \approx 6$ s, when the equivalent vehicle speed is of 150 km/h and the engine behaviour in all ICE/MGU-K interaction modes becomes the same.

The dashed black line at the top of Figure 7-18 represents the power demanded based on the maximum power of the power unit and the throttle position evolution shown in Figure 7-16. The sudden drops represent gear change. Even though in a real case, the driver would keep the throttle open during gear shifts with the current gearbox technology, the drops were necessary to simulate fuel cuts. It can be noticed that the power demand is 0 during simulated braking events and that for the acceleration event included in the zone of interest, power demand increases gradually according to the throttle position evolution shown in Figure 7-16.

The rear braking grip power curve shows the estimation of the maximum available tyre grip for braking based on Equation 7-3. As the equivalent vehicle speed decreases, the estimated downforce does too so less braking force can be applied.

Figure 7-19 is the same graphic as Figure 7-18 but with ICE power being added (blue curve). One can observe that ICE power is highly correlated to the driver power demand which reflects throttle pedal position and thus engine load signal evolution.

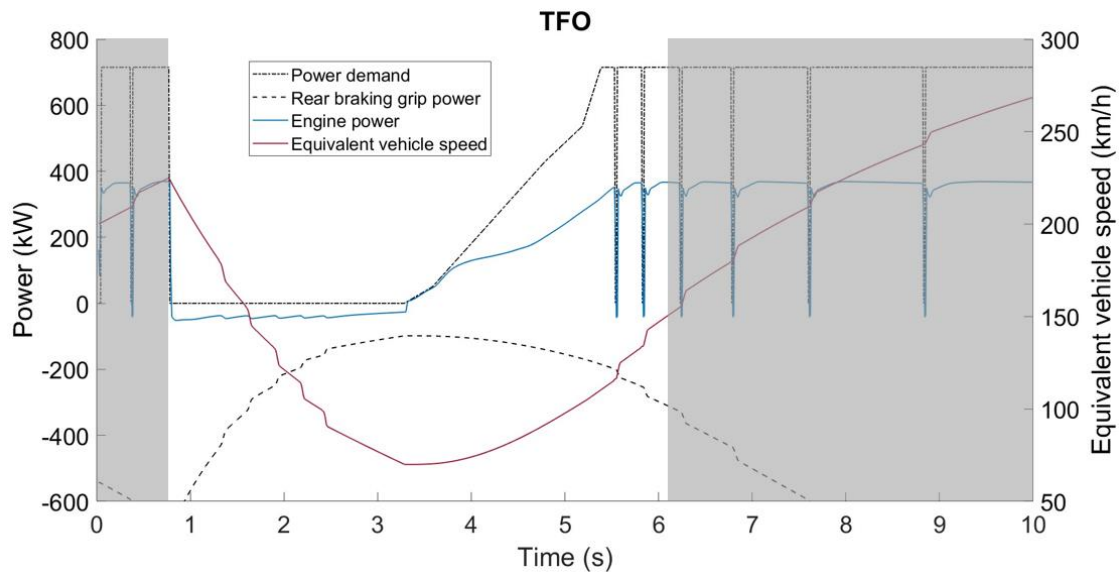


Figure 7-19 - 70-320 km/h speed event – TFO power and vehicle speed evolutions (2/3)

Engine power drops during gear changes as fuelling is cut for these events. It can also be highlighted that during the simulated braking events, engine power goes negative as the fuelling is also cut but the engine keeps breathing air, i.e. the engine acts as a pump. This represents engine braking.

Figure 7-20 is an evolution of Figure 7-19 with MGU-K power at the crankshaft (orange curve) and total PU power at the engine crankshaft (yellow curve) being

added. The MGU-K operates both at positive (deploys energy) and negative (recovers energy) power within its 350 kW limit.

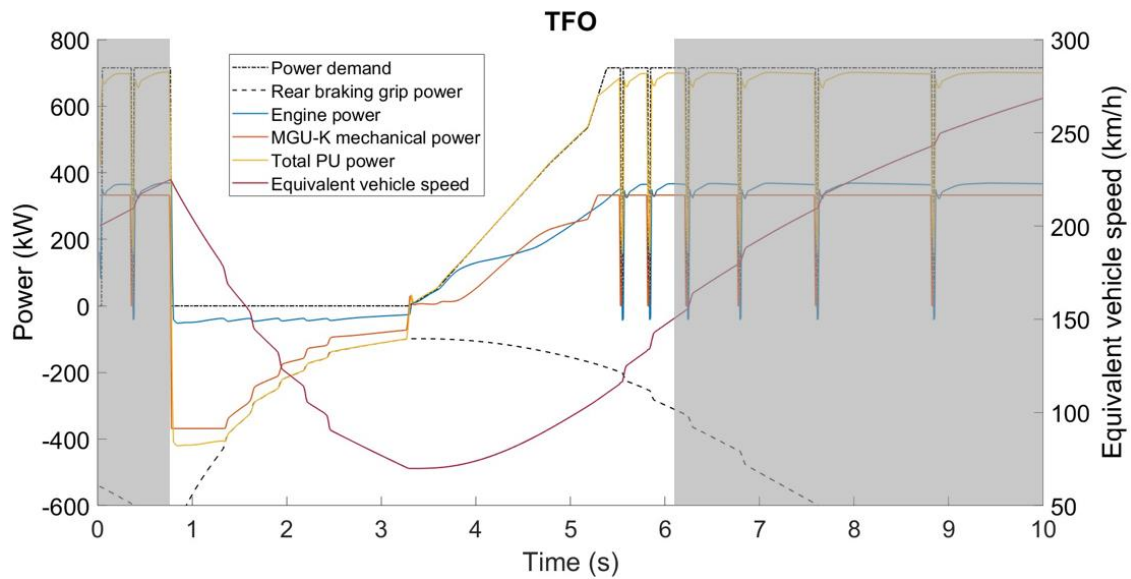


Figure 7-20 - 70-320 km/h speed event – TFO power and vehicle speed evolutions (3/3)

In the TFO mode, the MGU-K is only used for kinetic energy recovery (regenerative braking) and torque filling. During the simulated braking event (from the beginning of the zone of interest to $t \approx 3.2$ s where the minimum speed is reached), it can be observed in Figure 7-20 that the total PU power is either higher or equal to the available braking rear grip. In the first part of the braking event, the estimated vehicle downforce is still very high and so is the grip. The MGU-K regenerative power is maximised, and the rest of the available braking grip is modelled as being covered by the mechanical brakes (there computed as an increased load from the vehicle on the engine). As the available rear braking grip reduces with speed and downforce, the MGU-K power is reduced so that the estimated grip limit is not exceeded.

The torque filling capacity of the MGU-K is also shown in Figure 7-20, at the end of the investigated acceleration event. The engine power increases but not as fast as the demand.

The MGU-K thus switches into deployment (torque filling) mode and helps the engine to meet the demand.

During the full-load part of the simulation, the maximum power demand is higher than the maximum PU power by a small margin. This was implemented in the Formula Interpreter code to ensure that the PU was always operating at full load when it had too. If the demand was too close to the achievable power, it was possible for the PU output power to be momentarily higher than the demand causing the PU to operate at lower load than it should, only for control reasons.

70-320 km/h speed event – Torque Consuming during Braking mode (TCB)

Figure 7-21 shows the same as for Figure 7-20 but for the Torque Consuming during Braking (TCB) mode. In this mode, the MGU-K torque fills and does kinetic regenerative braking but is also used during simulated braking events to torque consume. The difference in behaviour compared to TFO mode is highlighted by the red ellipse in Figure 7-21.

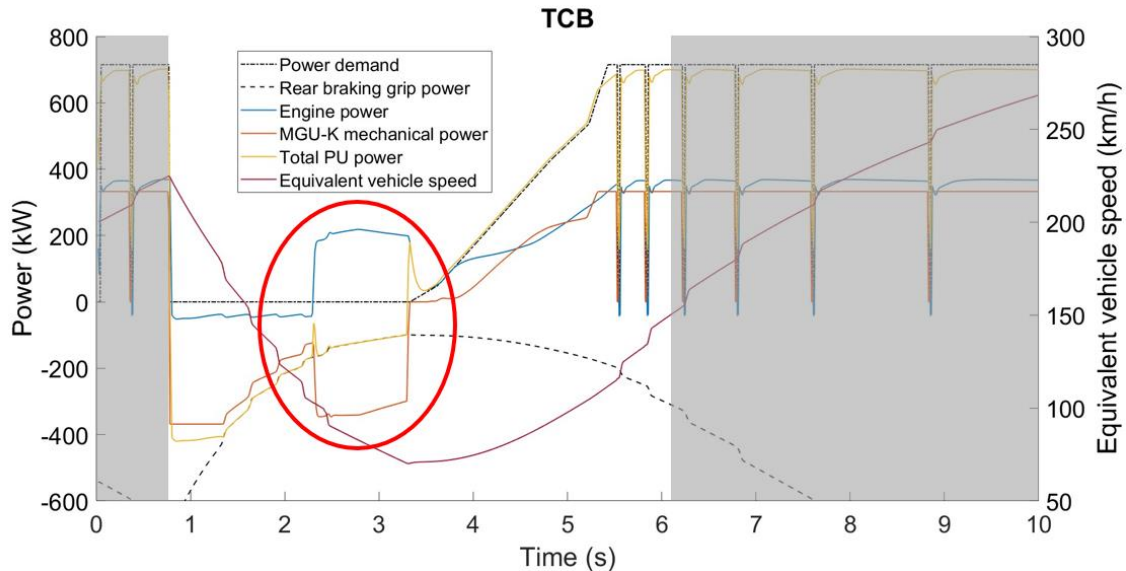


Figure 7-21 - 70-320 km/h speed event – TCB power and vehicle speed evolutions

The first phase of the braking event is the same as for the TFO mode. The MGU-K regenerative power is maximised and fully dedicated to braking. Then as the estimated available braking grip reduces, the MGU-K power also decreases but torque consuming is not immediately used as the difference between the MGU-K regenerative power and its 350 kW limit does not allow it yet to absorb 200 kW from the engine.

When the MGU-K is capable of torque consuming 200 kW, Figure 7-21 shows that the ICE starts producing power (from $t \approx 2.3$ s) and the MGU-K regenerative power is increased to be closer to its 350 kW limit meaning that more energy is recovered by the MGU-K than in TFO mode. However, for torque consuming to occur, additional fuel must be burned.

The total PU power (yellow curve) still matches the estimated braking grip limit (lower dashed black curve) showing that all the power produced by the engine is absorbed by the MGU-K and does not result in a different simulated braking event than the TFO mode.

Once the equivalent vehicle speed has reached the target minimum speed, the engine behaves in the same way as in TFO mode and only produces the power necessary not to exceed the demand at the beginning of the acceleration phase, meaning that the engine power must drop between the end of the braking event and the beginning of the acceleration. The engine powers the vehicle, and the MGU-K is used to torque fill if necessary.

70-320 km/h speed event – Torque Consuming during Acceleration mode (TCA)

Figure 7-22 shows the behaviour of the Power Unit in Torque Consuming during Acceleration (TCA) mode. During the simulated braking event, as no torque consuming is done, the behaviour is the same as in TFO mode. It is during the first part of the acceleration that the behaviour changes.

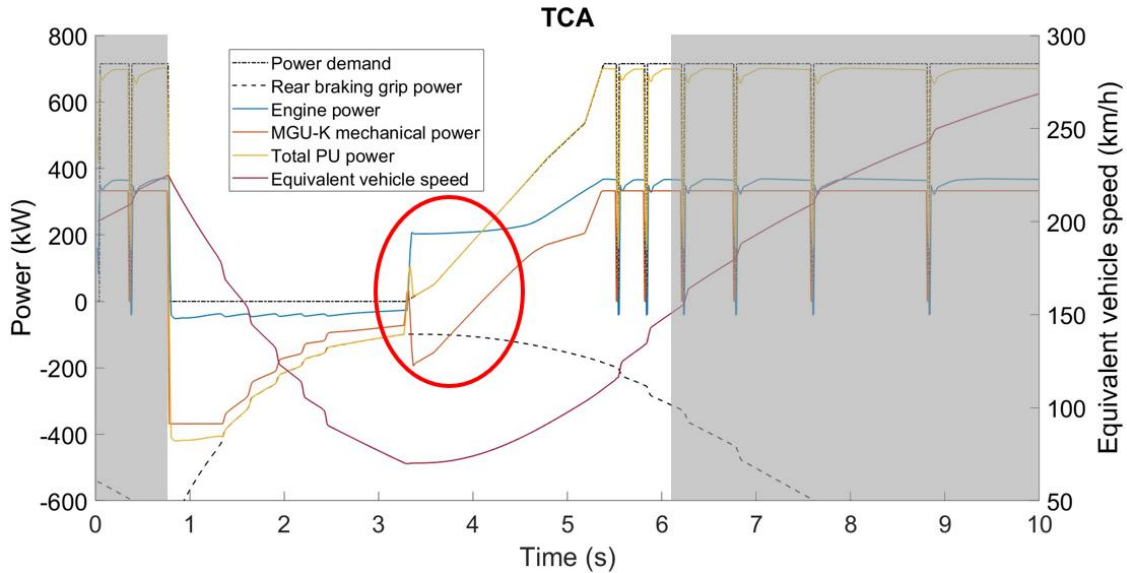


Figure 7-22 - 70-320 km/h speed event – TCA power and vehicle speed evolutions

As soon as the power demand is different than 0 after reaching the speed event minimum speed ($t \approx 3.2$ s), the engine load signal does not evolve following the increasing power demand but is immediately switched to 100%, the engine is instructed to operate at full load, thus nearly eliminating throttle losses.

As highlighted by the red ellipse in Figure 7-22, on the contrary to the TFO and TCB modes, the ICE produces more power than demanded (blue curve higher than the upper black dashed one). The difference is absorbed by the MGU-K to recharge the ES (negative values for the orange curve), resulting in the PU total power output (yellow curve) to match the demand.

This lasts until the power produced by the ICE exactly matches the power demand, after which the MGU-K switches into torque fill mode to meet the power demand. For the entire acceleration phase, the engine is operated at full load, there is no part-load operation.

70-320 km/h speed event – Torque Consuming during Acceleration and Braking mode (TCAB)

Figure 7-23 shows the behaviour of the Power Unit in Torque Consuming during Acceleration and Braking (TCAB) mode which is a combination of the TCA and TCB mode.

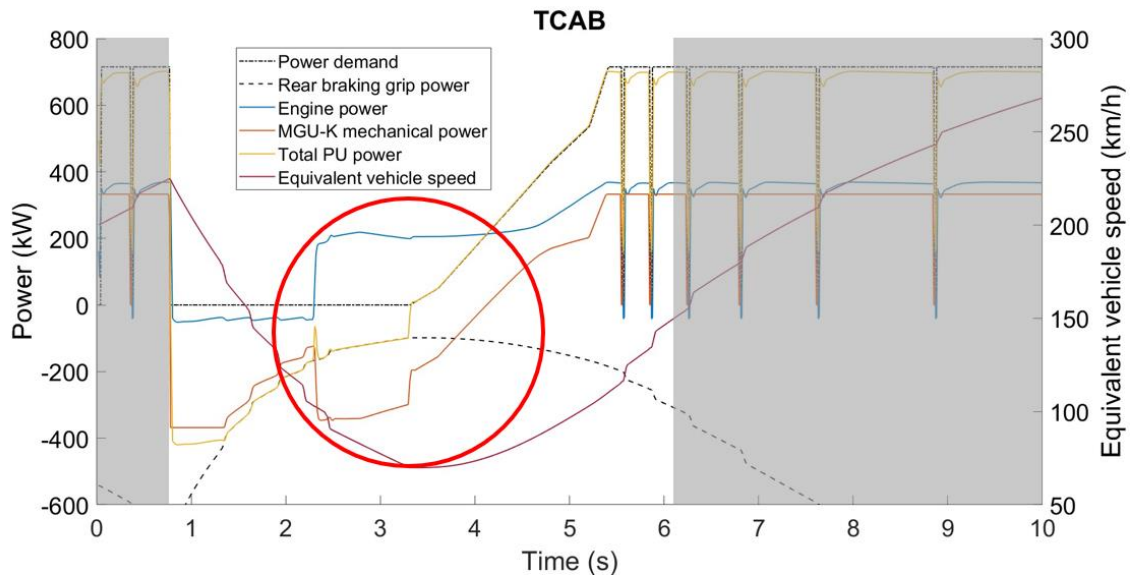


Figure 7-23 - 70-320 km/h speed event – TCAB power and vehicle speed evolutions

As a combination of both TCA and TCB modes, the TCAB behaviour is also a combination of their behaviour. The MGU-K starts torque consuming as soon as it can recover 200 kW from the engine ($t \approx 2.3$ s) in addition to regenerative braking. When the minimum equivalent vehicle speed is reached, the PU starts torque consuming during acceleration, so the ICE load signal is set at 100% and the MGU-K recovers the excess energy. When the ICE power output matches the demand, the MGU-K goes from consuming to deploying mode to torque fill.

On the contrary to the TCB mode, the engine power does not need to suddenly drop when the acceleration phase begins as the excess power is absorbed by the MGU-K. The engine produces power and the fraction that is transmitted to the wheel to accelerate the car is controlled by the MGU-K regenerative power.

As visible in Figure 7-23, the period during which the engine is operated at 0% load is significantly reduced compared to the other modes.

70-320 km/h speed event – Turbocharger rotor speed evolution for the different ICE/MGU-K interaction modes

Figure 7-24 shows the evolution of turbocharger rotor speed for the different ICE/MGU-K interaction modes. When the simulated braking event starts, the throttle body closes to its minimum angle value, suddenly reducing mass flow and thus compressor load which results in the spike observed in Figure 7-24 at this moment ($t \approx 0.9$ s).

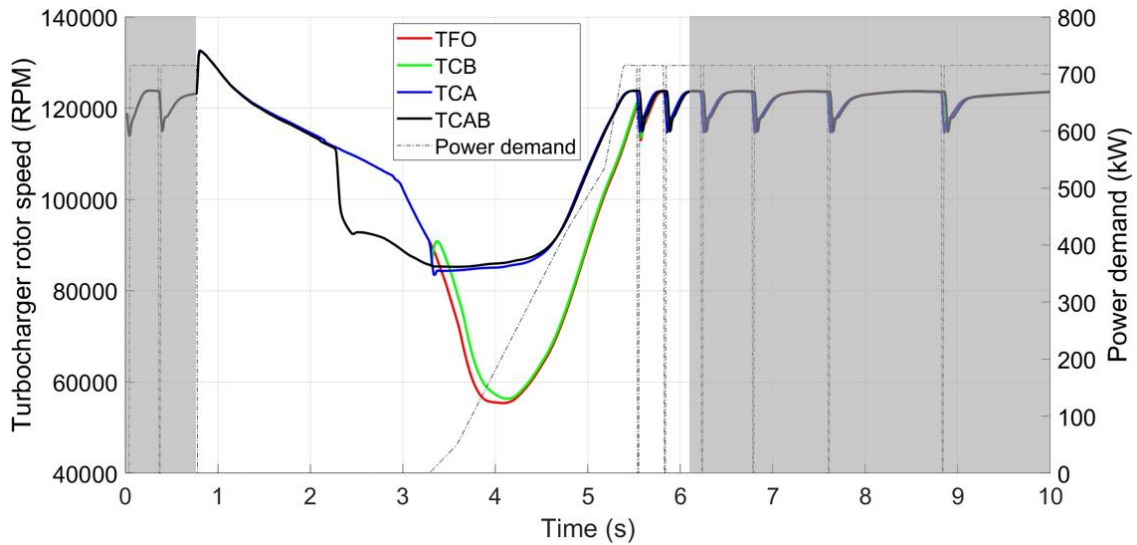


Figure 7-24 - 70-320 km/h speed event - Turbocharger rotor speed evolutions of the different ICE/MGU-K interaction modes

As the fuelling is cut during the simulated braking event, exhaust enthalpy is low, the turbine does not recover enough energy to keep the rotor spinning at optimal speed.

Out of the four modes, TCB (green curve) and TCAB (black curve) are the first to re-open their throttle to torque consume during braking. As torque consuming is only started when the engine can operate at high load, the throttle suddenly open to a high angle value which depends on engine load, leading to a sudden increase in compressor air mass flow and aerodynamic load in the compressor. As exhaust enthalpy does not increase immediately, there is an imbalance between the compressor and turbine power resulting in a turbocharger rotor speed drop until exhaust enthalpy builds up.

Once the speed event minimum speed is reached ($t \approx 3.2\text{ s}$), all modes start producing power to accelerate the vehicle. For the TFO mode (red curve), engine load, fuel flow and thus exhaust enthalpy increases progressively, and it takes time for the turbine power to match the compressor need, causing a significant power drop. For the TCB mode (green curve), the observation is the same. Despite the turbine being provided with exhaust enthalpy by the torque consuming during braking, the fact that the engine load is suddenly reduced not to exceed the power demand results in no anti-lag effect.

Regarding the TCA mode (blue curve), despite a drop in rotor speed before turbine power builds up, the drop is significantly reduced compared to the TFO and TCB modes as the engine operates immediately at full load and thus produces large amount of exhaust enthalpy to feed the turbine. The observation is the same for the TCAB mode (black curve), making them effective turbo lag mitigation solutions.

70-320 km/h speed event – Effective efficiency evolution for the different ICE/MGU-K interaction modes

Figure 7-25 shows the evolution in effective efficiency (brake fuel conversion efficiency) for the different ICE/MGU-K interaction modes. During the simulated braking events, when the fuel injection is cut, AVL Boost™ is not capable of calculating realistic efficiency, hence the absence of values in these zones.

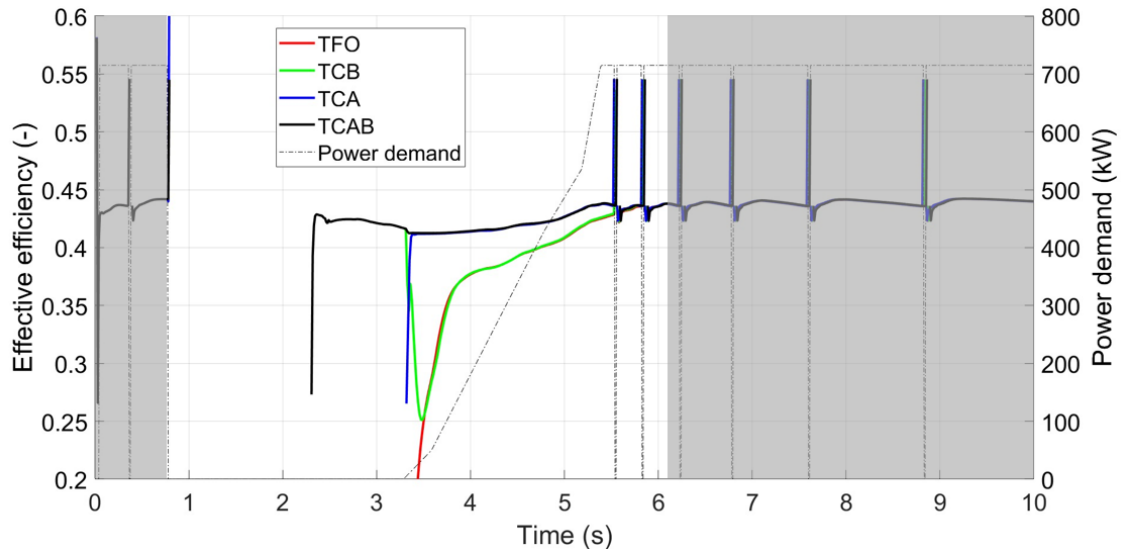


Figure 7-25 - 70-320 km/h speed event – Effective efficiency evolutions of the different ICE/MGU-K interaction modes

For the TCB (green curve) and TCAB (black curve) modes, it can be highlighted that thanks to the torque consuming during braking being only activated when the MGU-K is capable of consuming significant values of engine power, the engine operates at high load and thus at high effective efficiency, between 0.4 and 0.45, i.e. close to peak efficiency.

For the TCA (blue curve) and TCAB (black curve) modes, because the engine is operating at full load immediately from the beginning of the acceleration, with the implemented fuelling strategy, fuel flow and air/fuel ratios are higher, leading to high effective efficiency whereas the engine load only progressively increases in TFO and TCB modes, with lower fuel flow and air/fuel ratios leading to low fuel conversion efficiency in the first stages of the acceleration.

70-320 km/h speed event – Assessment of turbo lag characteristics of the different ICE/MGU-K interaction modes

Figure 7-26 shows a summary of the turbo lag situation for each of the four ICE/MGU-K interaction modes using three normalised performance values. The values are normalised by the highest values of the four studied modes for visual clarity, but the real values are displayed at the top of each bar.

The turbocharger rotor speed drop is calculated as the difference in rotor speed between the beginning of the zone of interest ($t \approx 0.9 \text{ s}$), and the minimum value observed during the studied speed event.

Turbo lag duration is here calculated as the time necessary for the turbocharger rotor speed to reach 120,000 RPM from the acceleration start (equivalent vehicle speed of 70 km/h, $t \approx 3.2 \text{ s}$). 120,000 RPM is considered, following analysis of the simulation results, as the threshold turbocharger rotor speed for which in these operating conditions, nominal turbocharger operation is reached. Considering Section 7.1 definition of turbo lag duration, 120,000 RPM would be the target rotor speed given to an electrically assisted turbocharger if it were to be featured in this 2026 Formula One™ transient engine model.

The mean effective efficiency is calculated from the acceleration start to the moment 150 km/h is reached.

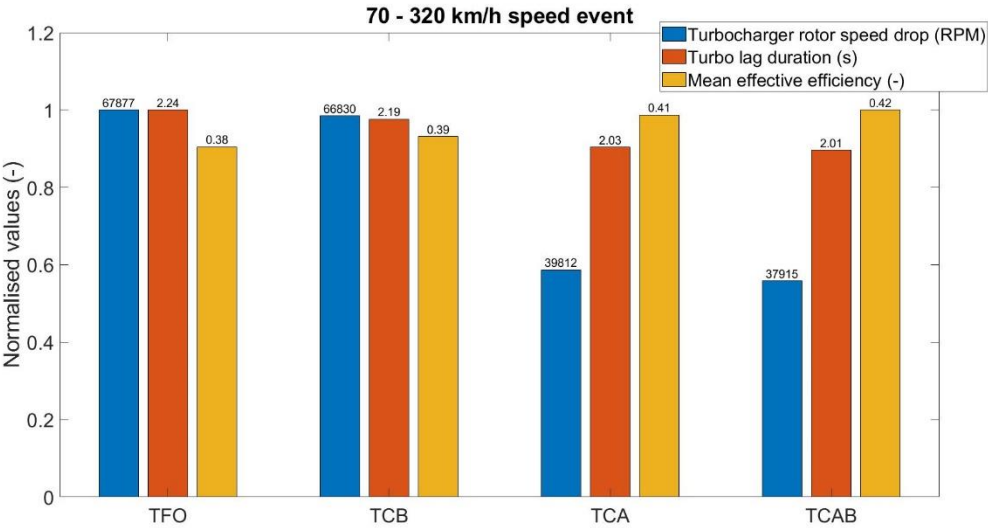


Figure 7-26 - 70-320 km/h speed event – Turbocharger and effective efficiency characteristics of the different ICE/MGU-K interaction modes

The TCB mode yields very similar results to that of the TFO mode. Turbo lag is only slightly reduced as the engine operates at low load at the beginning of the acceleration event causing an imbalance between the compressor need and the turbine supply, which results in a significant turbocharger rotor speed drop. As the engine faces turbo lag, the air mass flow provided by the turbocharger is not optimal, the engine operates at lower air/fuel ratios than nominal values which results in lower fuel conversion efficiency. As the engine is affected by turbo lag, the MGU-K is required to torque fill sooner and thus to consume more electric energy.

On the contrary, the TCA and TCAB modes, by operating the engine at full load for the whole acceleration phase effectively reduce turbo lag, operate at higher air/fuel ratios and yields higher fuel conversion efficiency.

70-320 km/h speed event – Analysis of the energy of the different ICE/MGU-K interaction modes

Figure 7-27 presents a summary of the energy usage of each mode. The total energy consumed (i.e. the change in energy within the Power Unit between the beginning and the end of the period of interest) is shown and broken down into total electric energy difference and fuel energy consumed. The two latter are respectively broken down into the energy used for regeneration (green outline) and for propulsion (red outline).

When analysing Figure 7-27, it must be kept in mind that all the simulated ICE/MGU-K interaction modes meet the power demand and thus achieve the same speed profile. Nevertheless, the contributions of the internal combustion engine and of the ERS are different depending on the mode and this is the balance between use of fuel and electric energy which is investigated in Figure 7-27.

Table 7-4 provides a guide on how to analyse the values of each bar presented in Figure 7-27 and what would be the ideal value / shape for each. However, obtaining the ideal value for each bar is not realistically possible as there are compromises to be made as explained in the following paragraphs.

Meaning Bar colour (Bar position)	Ideal bar value	Explanation
Total energy consumption Grey (1st bar)	Low value	The lower the value, the lower the amount of total energy used to accomplish the desired manoeuvre in the given ICE/MGU-K interaction mode
Total fuel energy consumption Yellow (2nd bar)	Low value	The lower the value, the less fuel has been used to accomplish the desired manoeuvre in the given ICE/MGU-K interaction mode
Electric energy delta Blue (2nd bar)	High negative value	A positive value means that more electric energy has been used than recovered and inversely The higher the negative value, the more the ES has been recharged
Electric energy consumed Blue + red outline (3rd bar)	Low value	The lower the value, the less the MGU-K has been used to torque fill and match the power demand. i.e. the less electric energy has been drawn out of the ES

Meaning Bar colour (Bar position)	Ideal bar value	Explanation
Electric energy recovered Blue + green outline (3 rd bar)	High negative value	The higher the value, the more electric energy has been recovered via regenerative braking and torque consuming and the more electric energy has been sent to the ES
Fuel energy used for propulsion Yellow + red outline (4 th bar)	Low value	The lower the value, the less fuel has been used to match the power demand
Fuel energy used for recovery Yellow + green outline (4 th bar)	Low value	Represents the additional fuel invested to make torque consuming possible, so a low value means that less additional fuel has been burned However, this value needs to be considered in regard to the increase in electric energy regeneration (blue bar with green outline) to assess the usefulness of torque consuming allowed by this additionally burned fuel energy but also in regard to the evolution of turbo lag

Table 7-4 - Presentation of the ideal energy usage distribution

TFO mode

For the TFO mode, between the beginning and the end of the manoeuvre, the energy difference is 1,430 kJ (grey bar) with 1,383 kJ coming from fuel energy and only 47 kJ from electric energy (second bar).

The electric energy difference is so low because when looking at the details, the MGU-K has recovered 487 kJ from regenerative braking (no torque consuming done in this mode) and has consumed 534 kJ to torque fill.

Regarding fuel energy, as this mode does not torque consume, fuel energy is only invested into propulsion, not regeneration. Figure 7-27 states that 15 kJ are used for torque consuming but this is due to the imperfect control of the MGU-K power allowed by AVL Boost™ and its Formula Interpreter which causes spikes in MGU-K power and to the energy consumption calculation algorithm. 15 kJ is just an offset and must be considered as the zero for this case study.

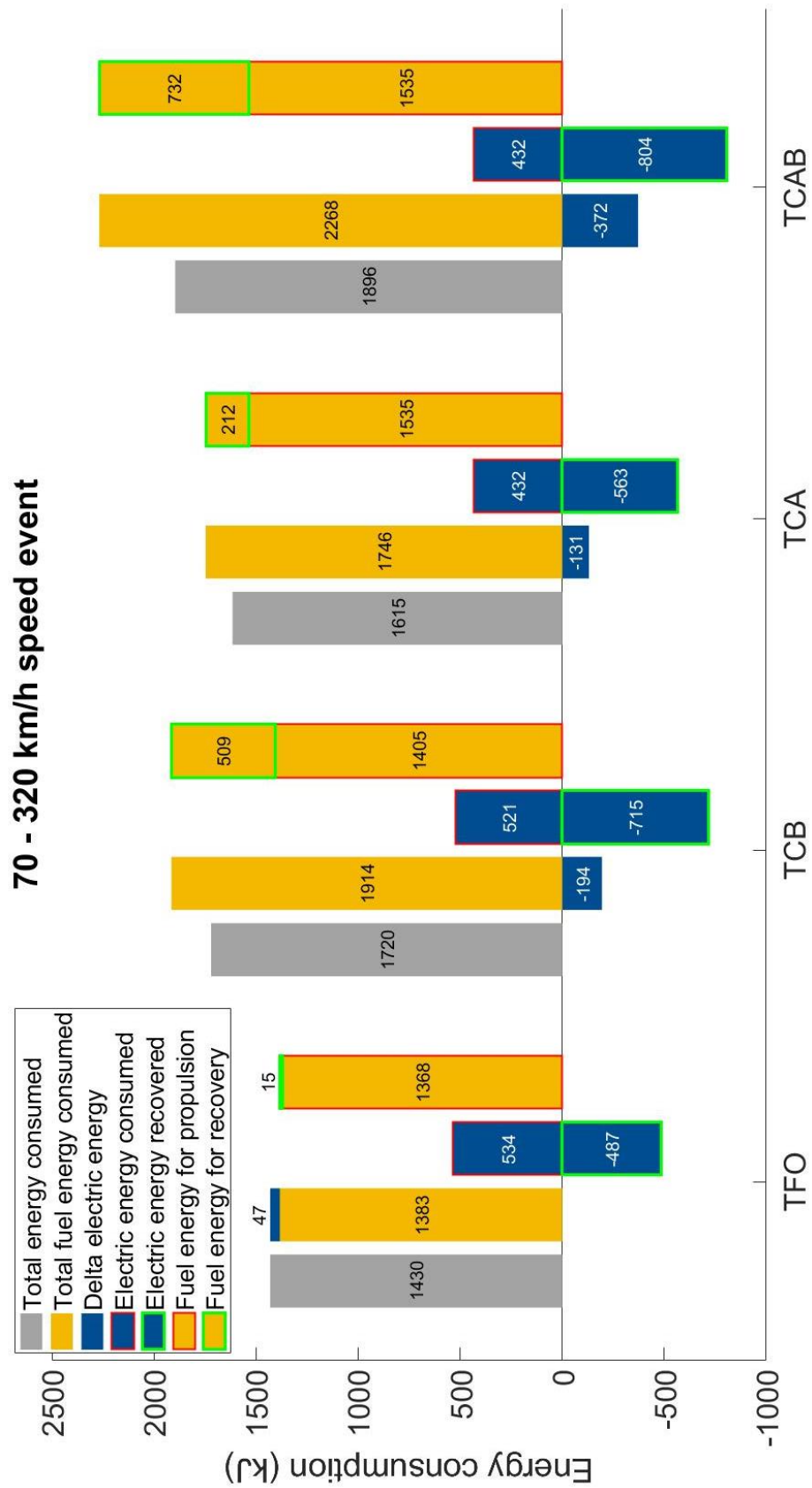


Figure 7-27 - 70-320 km/h speed event – Fuel and electric energy use of the different ICE/MGU-K interaction modes

TCB mode

For the TCB case, the overall energy difference is higher (grey bar). The consumed fuel energy (1914 kJ) is higher than the overall energy difference (1720 kJ) as there is an excess in recovered electric energy (715 kJ of electric energy recovered).

With the torque consuming during braking done in this mode, an additional 228 kJ of electric energy is recovered by the MGU-K compared to the TFO mode.

26.6% of the total consumed fuel energy is dedicated to producing engine power for torque consuming. This would mean that the 228 kJ of additional electric energy comes from 509 kJ of fuel energy used for torque consuming which represents an efficiency of 44.8%. This value is higher than the average effective efficiency of the engine observed in Figure 7-25 for the TCB mode when the MGU-K torque consumes, and here the mechanical and electric efficiencies of the MGU-K are not considered.

When looking in more depth at the results, only 197 kJ of the 228 kJ recovered energy is through torque consuming, thus the efficiency of torque consuming is of 38.7% which is more relevant. The fact that the additional electric energy recovered is not entirely due to torque consuming is caused by engine braking.

In TFO mode, no fuel is burned during the simulated braking event, so the engine works as an air pump, effectively acting as a brake. A fraction of the braking effort is taken care of by using the engine whereas in TCB mode, there is no engine braking when torque consuming is active as the engine produces power. The MGU-K thus must compensate for this lack of engine braking, meaning that more energy is recovered from kinetic energy regeneration in TCB mode than in TFO.

TCA mode

In TCA mode, the overall energy difference (grey bar) is higher than in TFO mode but lower than in TCB mode and more electric energy is recovered than consumed (blue part of the second bar for the TCA mode is negative).

The behaviour of the TFO and TCA modes during simulated braking events are the same as no torque consuming during braking is done, the difference thus comes from the acceleration part of the studied period during which the TCA mode torque consumes. As the engine in TCA mode is always operating at full load and as fuel flow is dependent on engine load, more engine power is produced but at the cost of a higher fuel consumption.

In addition, as the engine is less affected by turbo lag, it can produce more power and meet the power demand for longer without the support of the MGU-K, further explain the higher fuel consumption.

Compared to the TFO mode, the total fuel consumption is increased by 26.2% (1383 kJ vs 1746 kJ) but if only considering the energy invested in propulsion and not torque consuming, the increase is only of 11% (1383 kJ vs 1535 kJ).

The 212 kJ of fuel energy invested into torque consuming allows the MGU-K to recover an additional 76 kJ compared to the TFO mode (35.8% efficiency) but indirectly to also reduce the dependency on the MGU-K to meet the power demand resulting in a 102 kJ reduction in electric energy deployment (534 kJ vs 432 kJ).

As a result, over the studied manoeuvre, the TFO mode sees its electric storage reduced (positive electric consumption, 47 kJ) whereas the TCA mode recharges its ES for a total gain of 178 kJ (102 kJ not deployed compared to TFO and an additional 76 kJ recovery).

TCAB mode

In TCAB mode, it is interesting to highlight that the same amount of energy (fuel and electrical) is consumed to power the car than in TCA mode (see boxes with red outlines). Maintaining significantly higher turbocharger rotor speed helps to reduce the need of MGU-K torque fill in this mode in the same way it occurs for the TCA mode.

However, much more fuel is invested in torque consuming. Relatively to the TCA mode, 245% more fuel is used for torque consuming (212 kJ vs 732 kJ) with a return of only 42.8% in recovered electric energy (563 kJ vs 804 kJ) and this considering that the MGU-K also compensates for the lack of engine braking when torque consuming during simulated braking events as it does in TCB mode.

The TCAB mode results in the greatest recovered electric energy, the highest electric energy difference (highest negative value) but also the highest fuel consumption. Compared to the TFO mode, an additional 885 kJ of fuel is used (1383 kJ vs 2268 kJ), and the electric consumption reduces by 419 kJ (47 kJ vs -372 kJ) resulting overall in a 466 kJ increase in energy consumption (1430 kJ vs 1896 kJ).

Analysis of the impact of the ICE/MGU-K interaction modes based on the results of the 70-320 km/h speed event

As explained at the beginning of the section, the ICE/MGU-K interaction modes are investigated with two objectives in mind, reducing turbo lag and reducing the dependency on the MGU-K to meet the power demand which is linked to maintaining a high ES SOC level.

As seen previously, all torque consuming modes result in an increased fuel consumption, it is the necessary cost for the improvement brought by torque consuming as the energy must come from somewhere.

Being fuel-limited, it is important to consider the benefits of torque consuming on the 2026 Formula One™ Power Unit operating with regards to the fuel consumption cost and this is what the following figures are representing, a cost-benefit analysis of torque consuming.

For all these figures, the ideal mode is in the bottom left corner where the fuel penalty is the lowest and the other parameter under study is improved.

Figure 7-28 shows the difference between the torque consuming modes (TCB, TCA and TCAB) with the TFO mode as reference for fuel consumption and turbo lag duration.

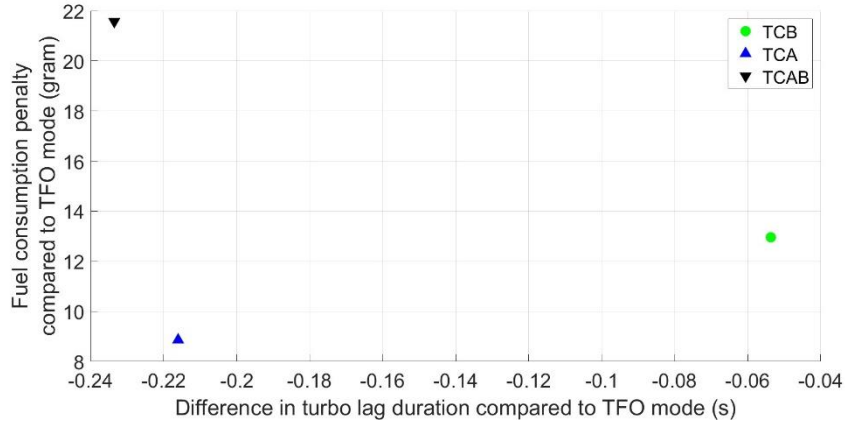


Figure 7-28 - 70-320 km/h speed event - Fuel penalty vs turbo lag duration of torque consuming modes

As previously observed, the TCB mode barely reduces turbo lag duration on the contrary to the ICE/MGU-K interaction modes that torque consume during acceleration (TCA and TCAB). However, for these two latter modes, the fuel penalty is very different as part of the fuel consumption increase in TCAB mode is for torque consuming during braking which does not have any anti-lag effect.

Considering turbo lag and this 70-320 km/h speed event, TCA is the most useful ICE/MGU-K interaction mode.

Concerning the dependency on the MGU-K to meet the power demand, it has been observed that the lower the turbo lag, the higher the engine output power and thus the lower the need of the MGU-K thereby maintaining SOC.

By always operating the ICE at full load, the torque consuming during acceleration modes (TCA and TCAB) reduce turbo lag duration (Figure 7-28) but also reduce the dependency on the MGU-K to maximise PU output power (Figure 7-29).

Using Figure 7-29, the TCA mode is identified as the most efficient ICE/MGU-K interaction mode when it comes to reducing the dependency on the MGU-K for acceleration as the torque consuming during braking of the TCAB mode results in a higher fuel consumption that does not help with engine power output.

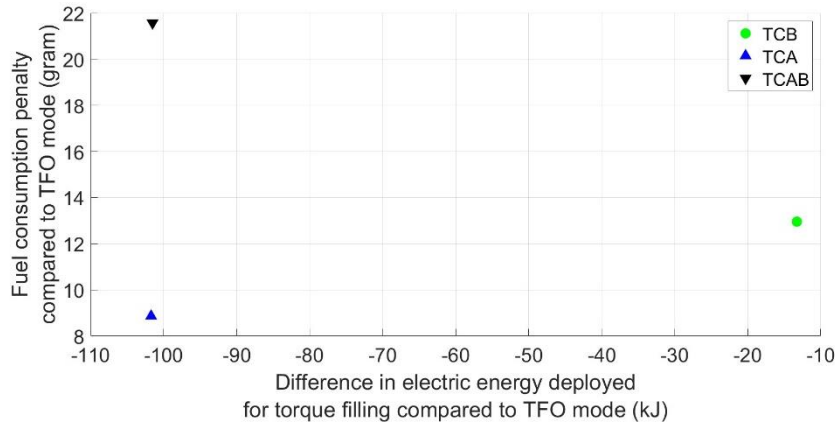


Figure 7-29 - 70-320 km/h speed event - Fuel penalty vs electric energy deployment for torque filling of torque consuming modes

Considering the higher power of the 2026 MGU-K, it has been explained that there is a need to keep the ES at high SOC level so that the MGU-K is available for use at race-critical moments.

As shown by Figure 7-30, the more the ES energy level (another way of interpreting SOC) is increased, the higher the fuel penalty as the improvement in energy level is due to torque consuming which requires fuel to be burned. And as seen previously, this is torque consuming during braking modes that allow for the most energy recovery.

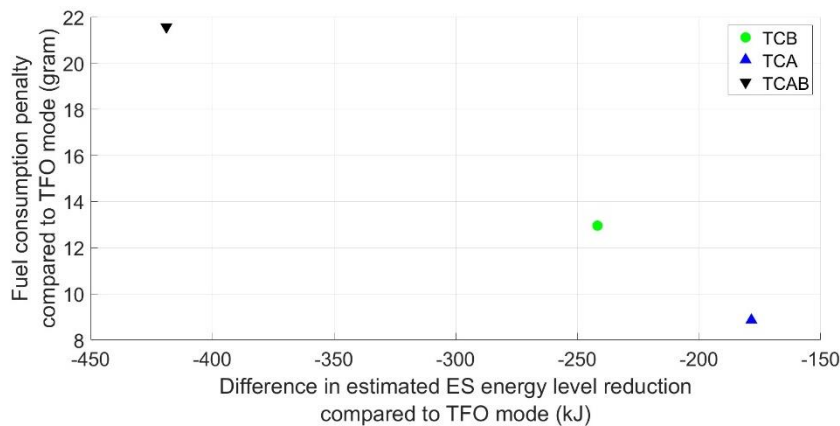


Figure 7-30 - 70-320 km/h speed event - Fuel penalty vs estimated ES energy level reduction of torque consuming modes

The TCA mode which results in the lowest fuel penalty for the best anti-lag characteristic and highest MGU-K dependency reduction yields the lowest ES energy level improvement of the three torque consuming modes.

As for the case study presented in Section 7.1, acceleration effectiveness (estimated vehicle acceleration / energy consumption) can be calculated. As previously discussed, all ICE/MGU-K interaction modes achieve the same speed-time profile so yield the same acceleration. However, because of the

difference between torque consuming modes, the acceleration effectiveness of all modes is not the same as shown in Figure 7-31.

This performance indicator only considers the energy transfer during the acceleration phase, including energy recovery during torque consuming during acceleration.

It can be observed that the torque consuming during acceleration modes (TCA and TCAB) yield the lowest acceleration efficiency. As previously described, the TCA and TCAB modes rely more on the ICE to power the car than the TFO and TCB modes. However, because of the relatively low effective efficiency of an internal combustion engine compared to that of an electric motor, a significant fraction of the fuel energy is “wasted” in the conversion process which negatively impact the acceleration effectiveness even when accounting for the energy recovery thanks to torque consuming during acceleration.

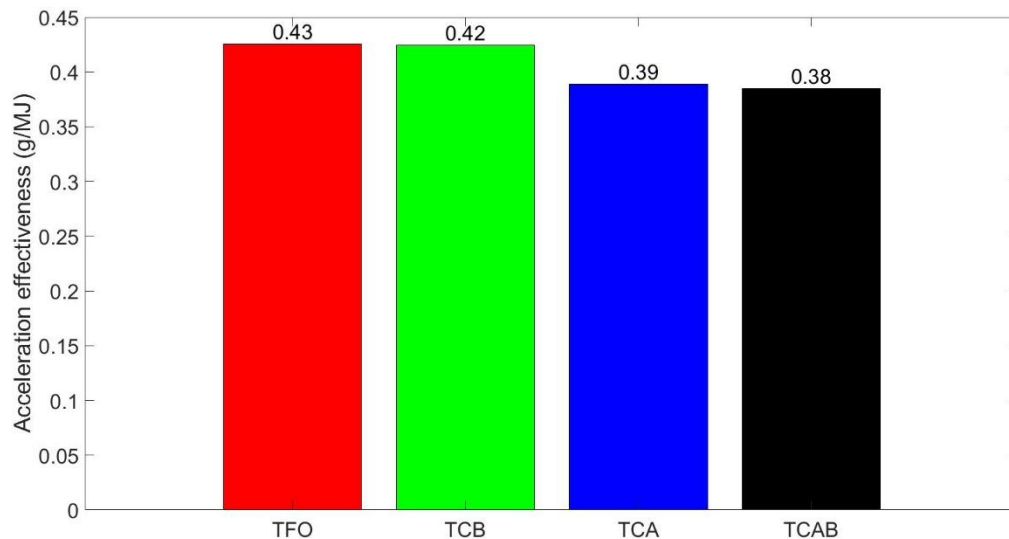


Figure 7-31 - 70-320 km/h speed event - Acceleration effectiveness of the different ICE/MGU-K interaction modes

Despite enabling the ICE to operate at higher effective efficiencies when in TCA and TCAB modes, the fact that the TFO and TCB modes use more the MGU-K, which has a significantly higher energy conversion efficiency, leads to improved apparent acceleration efficiency. However, as already mentioned several times, the electrical energy used by the MGU-K comes from energy recovery, i.e. indirectly from fuel combustion, which is not accounted for in this performance indicator.

The next section will investigate the evolution of the impact of the different modes on the 2026 Formula One™ Power Unit behaviour for all speed events, after which conclusions on the usefulness of these ICE/MGU-K interaction modes will be drawn.

7.2.5 Evolution of the impact of the different ICE/MGU-K interaction modes on the 2026 Formula One™ Power Unit behaviour with different speed events

For the other investigated speed events, 100-320, 150-320 and 200-320 km/h, the period of interests also lasts until the PU nominal behaviour is reached which corresponds to equivalent vehicle speeds of 170, 200 and 240 km/h respectively.

Figure 7-32 shows a summary of the evolution of the fraction of the fuel consumption dedicated to torque consuming as the minimum equivalent vehicle speed of the manoeuvre increases. Comparing the values between the different speed events is not very useful as not achieved for the same acceleration event but it is interesting to observe how the fraction of fuel consumption dedicated to torque consuming varies between modes and between speed events.

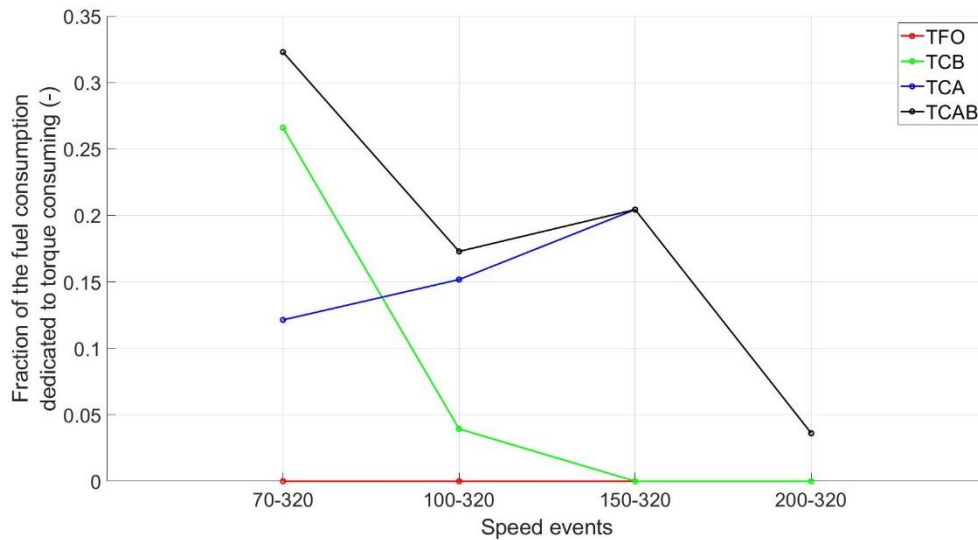


Figure 7-32 - Evolution of the fraction of fuel consumption dedicated to torque consuming with different speed events

For the TFO mode, the values are always zero as this mode does not involve any torque consuming.

Going from the 70-320 to the 100-320 km/h case, significant trend can be observed in Figure 7-32. The TCA mode (blue curve) has not changed as much as the TCB (green curve) and TCAB (black curve) modes for which the fraction of fuel dedicated to torque consuming has significantly reduced meaning that torque consuming during braking has been modified.

As explained in Section 7.2.4, torque consuming during braking is activated when the MGU-K can recover at least 200 kW from the engine in addition to regenerative braking. As the minimum speed of the speed event increases, the period during which the MGU-K can torque consume reduces as the available rear braking grip is high enough for most of the braking event to allow the MGU-K to be used in regenerative braking mode at high power, thus not leaving room for torque consuming.

In the 70-320 km/h case, a significant fraction of fuel is dedicated to torque consuming during braking for the TCB and TCAB modes but as the minimum speed increases, there is less and less room for this energy recovery mode.

As shown in Figure 7-32, for the 150-320 and 200-320 km/h in TCB mode, the values drop to zero indicating that no torque consuming during braking occurs. The mode behaves as TFO by using the MGU-K only to torque fill.

Similarly, for these speed events, the TCAB mode cannot do any torque consuming during braking, thus behaves like the TCA mode. Nevertheless, because the power demand never switches immediately from 0 to 100% (see Figure 7-16) on the contrary to engine load signal, there are, for the four speed events studied, opportunities to torque consume during the first part of the acceleration. This is why the amount of fuel dedicated to torque consuming in TCA and TCAB is always higher than zero.

Figure 7-33, which shows the evolution of turbo lag duration for the different modes and speed events, also shows the convergence between the TCA and TCAB and between the TFO and TCB modes as the minimum speed of the manoeuvre increases.

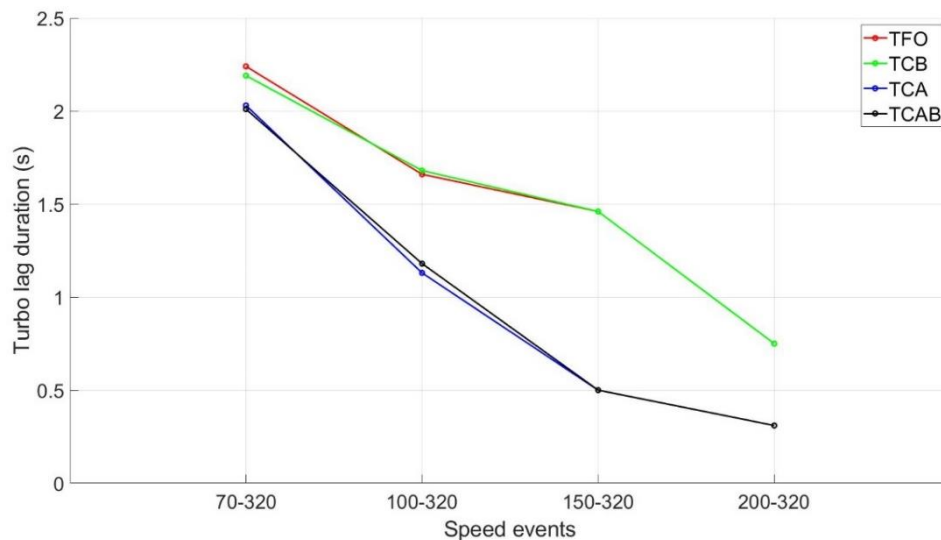


Figure 7-33 - Evolution of turbo lag time with different speed events

Because the simulated braking event is shorter and shorter, the exhaust enthalpy is reduced for shorter periods meaning that turbocharger rotor speed drop and turbo lag duration (duration for the turbocharger speed to reach 120,000 RPM from the moment power demand is different than zero) are minimised as shown in Figure 7-33.

Nevertheless, for all speed events, the TCA and TCAB modes have an advantage in terms of turbo lag over the TFO and TCB modes. Even if torque consuming during acceleration is more and more limited, the fact that the engine load is directly switched to 100% with the MGU-K recovering the excess power, is a significant help to increase exhaust enthalpy, reducing turbo lag and the dependency on the MGU-H to power the vehicle.

As previously seen, with the controls implemented in the models used for this case study, the reduction in turbo lag does not result in faster manoeuvre time for any speed event or ICE/MGU-K interaction mode, as the total power demand is always met but leads to differences in ICE/MGU-K power balance to meet this demand. What this case study shows is that by torque consuming during acceleration, the ICE can be provided with higher air mass flow to burn more fuel and at higher air/fuel ratio which results in higher fuel conversion efficiency in addition to higher power output thus reducing the need for the MGU-K to torque fill. Here the advantage of turbo lag reduction is not shown as a time gain but as a reduction in MGU-K contribution to the power demand and a more efficient use of the limited fuel flow.

The biggest gap in turbo lag duration is observed for the 150-320 km/h speed event. By torque consuming during acceleration, turbo lag duration is reduced from 1.49 s to 0.5 s. As a result, the average effective efficiency of the studied part of the acceleration goes from 39 to 43% (10.25% relative increase), the MGU-K contribution to acceleration is reduced from 480 kJ to 434 kJ (9.6% relative decrease) and ES SOC is kept constant (as much energy recovered than deployed), however the cost is a 27% relative increase in fuel consumption (34.9 gram vs 44.4 gram).

7.2.6 Conclusions on the usefulness of torque consuming modes

By torque consuming in addition to regenerative braking, the investigated ICE/MGU-K modes allow the 2026 Formula One™ Power Unit to recover more electric energy but at the expense of an unavoidable greater fuel consumption. The overall energy consumptions of the torque consuming modes are greater than that of the reference TFO mode even with the additional energy recovery partly because of the inherent lower efficiency of an internal combustion engine.

When it comes to torque consuming as an anti-lag solution, the previous sections showed that only the TCA and TCAB modes effectively reduces turbo lag and the dependency on the MGU-K to accelerate the car. With the TCB mode, the fact that the engine is operated at very low engine load at beginning of the acceleration event overcomes the effort made when torque consuming during braking to keep exhaust enthalpy high and results in no turbo lag improvement.

Only torque consuming during acceleration is effective in minimising turbo lag. Overall, it consumes more fuel energy because the engine is operated at higher load and thus at higher fuel mass flow but by reducing turbo lag, the average fuel conversion efficiency is significantly increased. More fuel is consumed but at a much higher efficiency. This results in higher electric energy recovery compared to the TFO mode and a reduced need for the MGU-K to torque fill for the PU to match the power demand.

Torque consuming during braking can be useful if the SOC of the ES is low and significant energy recovery is necessary as it enables to recover more energy

than torque consuming during acceleration but is less and less available as the minimum speed of the braking event increases. As TCB alone does not help for turbo lag mitigation, torque consuming during braking is more likely to be used in TCAB mode than in TCB to combine both the advantages of torque consuming during braking and acceleration.

However, the most promising mode is TCA as it brings the most advantages (turbo lag duration reduction and MGU-K dependency reduction) with the smallest fuel consumption penalty. It leads to higher engine efficiency, lower dependency on the MGU-K torque filling capacity, lower fuel consumption increase, lower turbo lag, and additional energy recovery (even though additional energy conversion steps and losses occur).

As for the traditional anti-lag solution in which the fuel is burnt in the exhaust, torque consuming leads to additional fuel consumption which must be seen as an investment to later improve the performance of the ICE and efficiency. However, on the contrary to older techniques, with torque consuming, a fraction of the additional fuel energy consumed is recovered as electric energy which can then be used to recharge the ES / maintain the ES at a reasonable SOC and thus power the MGU-K when necessary.

Nevertheless, with more classic anti-lag solutions that do not require additional fuel consumption and that will be investigated by the Power Unit manufacturers, the TCA mode is less of interest as the turbo lag should be less significant. But this mode can prove useful if there is a need to recharge the ES in addition to mitigate turbo lag.

Despite being the most promising mode, TCA consumes more fuel than if no torque consuming occurs. As the 2026 Formula One™ Power Unit is fuel limited, any additional fuel consumption must be investigated in detail to assess its worth. As for the Open Waste Gate mode of the 2014 Formula One™ Power Unit (Chapter 5), torque consuming cannot be seen as a sustainable mode but as a useful mode when additional energy recovery is necessary to maintain high ES SOC level.

To fully appreciate the usefulness of this mode, it would be necessary to couple the engine simulation model with a lap time simulation to analyse the overall energy consumption around a lap, the lap time gain and if such mode would allow the car to finish the race with the limited fuel onboard and if it has its place on the car.

7.3 Conclusions on the impact and opportunities related to coupling an internal combustion engine with an electric motor

By using AVL Boost™ transient engine simulations based on the 2026 Formula One™ Power Unit regulations, this chapter investigated the usefulness of coupling electric motor(s) to the internal combustion engine.

Coupling the internal combustion engine to an electric motor at the turbocharger (eTC) or at the crankshaft (MGU-K) helps to improve PU (and hence vehicle) acceleration performances but in different ways. With an eTC, turbo lag is nearly eliminated, and it becomes possible to precisely control the turbocharger rotor speed and air mass flow to significantly improve the engine effective efficiency during transient manoeuvres, and this by only investing small amount of electric energy. The eTC solution can only compete with an ICE coupled with an MGU-K when this latter is compromised by turbo lag, after which the combined power of the ICE and of the MGU-K results in much higher acceleration capacity. However, the MGU-K configuration consumes much more electric power and fuel conversion efficiency is affected by turbo lag which leads to non-optimal air/fuel ratios.

The 2026 Formula One™ Power Unit will not feature an eTC meaning that the engine will be exposed to turbo lag. PU manufacturers will need to find anti-lag solutions to maximise ICE fuel conversion efficiency and improve on-track performance without relying too much on the MGU-K which consumes large amount of electric energy that comes from energy regenerative, i.e. indirectly from fuel combustion.

However, as the MGU-K represents nearly 50% of the 2026 Formula One™ Power Unit output power, its use is necessary to achieve competitive lap times. Thus, it must be ensured that the ES SOC is maintained at acceptable level to use the MGU-K when necessary.

As explained in Section 7.1, the 2026 Formula One™ PU will feature anti-lag solutions which are not investigated here but that will already improve the efficiency and performance of the ICE. One example is a pre-compressor valve, which was already used during the first turbocharger era, which acts as a variable pre-swirl valve to maximise the efficiency of the compressor but also shuts when braking to remove the load on the compressor while exhaust enthalpy is still high, causing the rotor speed to spike thus minimising the speed drop and the turbo lag.

These solutions do not lead to additional fuel consumption but do not recharge the ES either. By using the high power of the MGU-K, this latter can be used in torque consuming mode, i.e. absorb the power produced by the ICE to recharge the ES, to operate the ICE at higher loads than it would do without torque consuming. By doing so, the exhaust enthalpy is kept high, turbo lag is mitigated, and engine effective efficiency is increased.

Torque consuming during acceleration (TCA) proved to be the most promising ICE/MGU-K interaction mode according to the simulations run for the case study as it effectively reduces turbo lag, reduces throttle and pumping losses, recharge the ES, minimises the use of the MGU-K to torque fill and results in a smaller fuel consumption penalty compared to torque consuming during braking which does not serve as an anti-lag solution.

To be able to draw definitive conclusions on the usefulness of torque consuming, these modes need to be investigated alongside the other anti-lag solutions that

will be featured on the 2026 Formula One™ fuel-limited engine and with a more complete vehicle simulation to assess the benefits in terms of on-track performance. But this 350 kW MGU-K makes torque consuming an interesting solution to consider as a special PU mode for momentarily reducing turbo lag, recharging the ES, improving fuel conversion efficiency, and reducing the need for the MGU-K torque filling capacity.

8 Conclusions and future work

8.1 Conclusions

8.1.1 Contribution to the body of knowledge

This thesis presents a critical assessment of the thermodynamic behaviour of a fuel-limited internal combustion engine featuring an electrically assisted turbocharger and running with a lean mixture, a pre-chamber ignition system, direct injection, a high geometric compression ratio and waste gates used as pressure-relief valves. The assessment has been investigated using advanced computer simulation techniques drawing on both energy and exergy analyses. The literature review set out in Chapter 2 concludes that such an assessment has not been published previously and therefore represents a contribution to the understanding of highest performing internal combustion engines.

This engine is the 2014+ Formula One™ engine for which only little information is available in the literature. By using a steady-state 0D/1D model for the thermodynamic and fluid dynamic performance of the engine within unavoidable constraints on component-level-detail data (2014+ Digital Twin), it has been possible to highlight and discuss the main performance insights of this engine and to understand how each enabling technology allows the ICE to reach the 50+% brake thermal efficiency claimed by the Formula One™ engine manufacturers and to also appreciate how the evolution of the fuel flow limit with engine speed imposed by the regulations influences the behaviour of the whole engine.

The investigation conducted on the significance of the uncertainty related to the input dataset of the 2014+ Digital Twin on its performance predictions demonstrated that despite the sparse available data on the 2014+ Formula One engine, the 2014+ Digital Twin could be confidently used to assess the behaviour of this engine but with a 0.7 IMEP uncertainty.

In 2026, Formula One™ will introduce new Power Unit regulations which will share a similar fuel-limited philosophy as the 2014+ one but with a thermal / electric power balance brought closer to 50/50 and the electrically assisted turbocharger replaced by a conventional turbocharger only powered by exhaust gas enthalpy.

This new engine is currently under development so no data about design or even performance is available. By also using a steady-state 0D/1D engine model based on the 2026 Formula One™ regulations published by the FIA, it has been possible to make a prediction on the level of performance that this engine could reached but also to highlight the differences with its 2014+ counterpart, specifically the reduction in in-cylinder pressure which will enable the combustion process to be more rapid than previously and increasing air/fuel ratio to best exploit the advantages of this situation.

By expanding the assessment to include transient 0D/1D engine models of the 2026 Formula One™ engine, the possible interaction between the 350 kW MGU-K and the internal combustion engine to improve the transient behaviour of the Power Unit has been investigated. It was demonstrated for the first time that the MGU-K can be set to absorb a fraction of the power produced by the engine, enabling this latter to operate at higher load than otherwise required, thus increasing exhaust enthalpy, reducing turbo lag and reducing dependency on the electric system of the car for propulsion but at the expense of a higher fuel consumption.

In addition, the work carried out during this Doctoral Thesis demonstrated the robustness of the currently available 0D/1D engine simulation software which successfully modelled the behaviour of high-performance Formula One™ engines despite the sparse input dataset enabling to investigate the thermodynamic characteristics of these engines and to draw conclusions about the way they reach very high fuel conversion efficiency.

8.1.2 Results achieved relating to the objectives of the research

As previously mentioned, there is little data available in the literature on the 2014+ Formula One™ engine and even though the specialised press explains periodically the technologies used in this engine, there is no in-depth investigation of the engine from a fundamental physics point of view with simulation-backed data and even the Power Unit manufacturer may not have the time to carry out this type of investigation, nor necessarily the motivation as long as the engine performance is improved and the Power Unit remains competitive. There is some, limited, un-verified knowledge about this engine in the public domain but no clear explanation of how it works.

By using the knowledge acquired during the Cranfield F1™ Design Sprint and creating a 2014+ Digital Twin of the 2014+ Formula One™ engine thanks to the advice from the sponsor of this Doctoral Thesis, insights regarding the engine and turbocharger behaviours have been exposed, making possible to scientifically investigate from the points of view of energy and exergy how this engine can achieve 50+% brake fuel conversion efficiency and the significance of the MGU-H in achieving peak efficiency and thus confirm the claims made by Power Unit manufacturers using results from 0D/1D engine simulations. With the analysis of the 2014+ Formula One™ Power Unit presented in this thesis, a unique structured scientific exposition of its enabling technologies is proposed.

The investigations also demonstrated how much the shape of the fuel flow limit imposed by the regulations influenced the behaviour of the engine and of the turbocharger leading to them being operated in a narrow window of high efficiency.

The observations made on the 2014+ Digital Twin behaviour can then be considered in the development of future motorsport but also road engines to

maximise fuel conversion efficiency and help the transition towards a more efficient and low-carbon future.

By running similar settings with an engine following the 2026 Formula One™ regulations, it was observed that the 2026 Power Unit will have similar peak performance than its 2014+ counterpart but with a much higher contribution of the electric system and lower engine effective efficiency (2.81 percentage points) because of the removal of the MGU-H and the impossibility of transferring exhaust energy to the crankshaft via both MGUs. Due to the high torque production capacity at low speed of an electric motor, the low-speed acceleration capacity of the 2026 Power Unit will be higher than that of the 2014+ one but less energy per lap is available because of the reduction fuel flow.

When investigating acceleration, the 2026 Power Unit will appear more energy efficient than the 2014+ Power Unit as a higher fraction of the power comes from the electric system which has a higher efficiency than an internal combustion engine (95% vs 50%) so less energy is lost because of inefficiencies. Nevertheless, it must be kept in mind at any time when studying these Formula One™ fuel-limited hybrid Power Unit that all the energy onboard the car during a race originates directly or indirectly from fuel meaning that fuel must be burned for the MGU-K to be used and for the ES SOC to be kept high.

With the reduction in fuel flow limit and the unchanged swept volume, the in-cylinder pressure of the 2026 Formula One™ engine will significantly be reduced making it no longer pressure-limited. There is thus room for the engine manufacturer to improve their combustion process in ways that were not possible under the 2014+ regulations because of mechanical limits. The investigation conducted in this thesis highlighted that increasing air/fuel ratio is the most promising solution to increase fuel conversion efficiency and this could be allowed by the advanced sustainable fuel which will be used in 2026 as their composition will be optimised to result in higher lean combustion limit and higher laminar flame speed.

As shown with the AVL Boost™ transient engine simulations, the electrification of the internal combustion engine is very effective at increasing fuel conversion efficiency and/or improving acceleration performance. An electrically assisted turbocharged is identified as the best solution to precisely control air/fuel ratio and thus fuel conversion efficiency during changes in load demand whereas an electric motor connected to the crankshaft of the ICE leads to better acceleration capacities because of the higher total output power.

With the MGU-H being removed from the 2026 Formula One™ Power Unit regulations, the engine with its conventional turbocharger will be exposed to turbo lag which negatively impacts engine performance and thus requires more intervention of the electric system to torque fill to match the power demand which in turn requires more energy recovery.

Nevertheless, by optimising the interaction between the internal combustion engine and the MGU-K, this latter can be used to 'torque consume' (deliberately absorb energy produced through combustion of fuel in the engine) which results

in higher energy recovery (and thus a higher ERS SOC), lower turbo lag and a reduced dependency on the MGU-K to accelerate the car, but these benefits come from a higher fuel consumption in an already fuel-limited engine. However, considering the significance of the MGU-K power relatively to the total PU output power, it will be essential to keep the ERS SOC high so that the MGU-K can be used to achieve relevant lap time meaning that over a whole race, torque consuming could prove useful despite the higher fuel consuming and thus the higher mass of fuel onboard.

As shown in this thesis, torque consuming during acceleration (TCA), i.e. always operating the engine at 100% load and using the torque absorption capacity of the MGU-K to control the amount of power transmitted to the wheels, is the most interesting mode as it significantly reduces turbo lag, improves engine efficiency and reduces the dependency on the MGU-K on the contrary to torque consuming during braking which is more effective for energy recovery.

8.2 Future work

If the trend observed with the former Formula One™ engine regulations continues, papers about the 2014+ Formula One™ Power Unit are expected to be published by Power Unit manufacturers once the regulations come to an end after the 2025 season. At that moment, it will become possible to verify some of the assumptions made for the creation of the 2014+ Digital Twin and to judge on the accuracy of the model predictions. By doing so, it will also be possible to improve the 2014+ Digital Twin and enhance the understanding of this very specific fuel-limited engine.

Nevertheless, because the philosophy of the 2026 Formula One™ Power Unit regulations will share similarities with the current one (2014+), it may take more time for information to be published without engine manufacturers compromising their competitive advantage.

With the engine model created for this thesis, predictions were made regarding the performance of the 2026 Formula One™ Power Unit but actual data will only become available when the Power Units will hit the track in 2026.

If the Cranfield F1™ Design Sprint continues, by using the research presented here and the feedback from the industry, a Digital Twin of the 2026 Formula One™ engine could be created and analysed in the same way it was done with the 2014+ Digital Twin to confirm or not the assumptions made in this thesis but also enhance the understanding of this new engine and the potential impact of fuel composition on its behaviour.

Torque consuming has been proposed to use the high power of the MGU-K to keep the ERS SOC high, reduce turbo lag and the dependency on the electric system to accelerate the car but it is also only in 2026 that it will be possible to assess if such an ICE/MGU-K interaction mode will be used by the different Power Unit manufacturers.

In the meantime, the usefulness of this Power Unit mode could be further investigated by integrating it to a lap time simulation to assess if the extra fuel weight is compensated by the possibility to use the MGU-K more often.

As previously mentioned, the composition of the advanced sustainable fuels that will be used in the 2026 Formula One™ engine will have the possibility to be optimised for enhanced combustion properties as engineered fuels and can be the key to these future regulations. This was not possible within this thesis but understanding how the composition of the fuel can influence the performance of this future engine could prove to be of primary importance and require improved work using 2-zone rate-kinetics simulations.

In 2026, thanks to the reduction in fuel flow and the associated reduction in peak in-cylinder pressure, it will be possible to run at higher air/fuel ratios than that of the 2014+ Formula One™ engine. However, the 2026 regulations will still only allow one injector per cylinder making the use of active pre-chamber impossible. Combined with the sole spark event allowed by cycle, the fuel preparation optimisation to ensure the ignition of the passive pre-chamber of the 2026 Formula One™ engine will be crucial. This will require detailed CFD simulation to understand the interaction between the design of the combustion chamber, the shape of the piston, the geometry of the pre-chamber and the fuel spray dynamics.

With the limited operation window of a passive pre-chamber, there must be a point of diminishing return where even with the distributed ignition points and high ignition energy of the pre-chamber, the benefits of higher air/fuel ratios are outweighed by suboptimal combustion characteristics (difficult pre-chamber filling, increased combustion duration because of lower laminar flame speed, unreliable ignition, partial burning). The position of this point of diminishing return and the influence that engineered fuel composition can have on it could be investigated using rate-kinetics simulation, potentially associated with CFD models.

To develop an in-depth understanding of these fuel-limited Formula One™ engines and of component-level requirements, experimental work will be necessary. By doing so, the experimental results can be fed into detailed CFD models to optimise them and thus expanding the research possibilities as the basic mechanisms will be better understood and represented.

As shown by the increase in number of pages in Formula One™ Technical regulations describing the requirements of the Power Unit, the number of components and variables that can be optimised to deliver excellent performance is also increased. The incorporation of Artificial Intelligence and Machine Learning is a promising solution to support engineers in the development of their designs and in the optimisation of the design process especially considering the more and more stringent limitations on engine testing and CFD simulations. Considering these limitations, software in the loop testing is also a solution that should be explored.

Interestingly, in the latest issue of the 2026 Formula One™ Power Unit regulations (3rd issue) published on the 20th of June 2023 by the FIA [15], the article defining the turbocharger (article 5.5.1) has been modified. The part of the article stating that the exhaust turbine is the only component permitted to transfer energy to the rotating part of the turbocharger has been removed. At first, this modification could open the door for electrically assisted turbocharger which, as described in this thesis in Chapter 7, is the most useful solution to maintain high fuel conversion efficiency during transient operation. This modification could drastically change the development of the 2026 Formula One™ engine which could benefit from the transient capacity of the MGU-H featured in the 2014+ Formula One™ engine but without the associated very complex energy management strategies. Nevertheless, the energy flow diagram provided in the 2026 Formula One™ regulations shows that no energy transfer is possible between the ES and the turbocharger which prevents the implementation of such a system.

REFERENCES

- [1] Sakurahara, K. (2009) 'Summary of Honda Formula One Engine in Third-Era Activities', *Honda R&D Technical Review F1 Special (The Third Era Activities)*, pp. 14–18.
- [2] Theissen, M., Duesmann, M., Hartmann, J., Kietz, M. and Schulz, U. (2010) '10 Years of BMW F1 Engines', in *31. Internationales Wiener Motorensymposium*. Vienna, 29-30 April 2010.
- [3] Hassan, W. T. F. (1966) 'The Coventry Climax Racing Engine 1961-1965', *SAE Technical Papers*, 660742. doi: 10.4271/660742.
- [4] Henry, A. (1998) *The Formula 1 Turbo Era*. Hazleton Publishing.
- [5] 'Private communication' (2023).
- [6] Otake, Y., Goto, O., Miyano, H., Kawamoto, M., Aoki, A. and Ogawa, T. (1989) 'Honda Formula One Turbo-Charged V-6 1.5L engine', *SAE Technical Papers*, (890877). doi: 10.4271/890877.
- [7] Kondo, K., Motohashi, Y., Ogiyama, K., Endo, H., Nakamura, S. and Yanagisawa, N. (2009) 'Explanation of Honda's Third Era Formula One Engine Development', *Honda R&D Technical Review F1 Special (The Third Era Activities)*.
- [8] Fédération Internationale de l'Automobile (FIA) (2012) *2014 Formula One Technical Regulations*, *FIA Technical Regulations*.
- [9] Fédération Internationale de l'Automobile (FIA) (2014) '2014 Formula One Sporting Regulations', *FIA Sporting Regulations*.
- [10] Fédération Internationale de l'Automobile (FIA) (2021) *2021 Formula One Sporting Regulations*, *FIA Sporting Regulations*.
- [11] Shell Global (2020) *Fuelling Scuderia Ferrari into a new era for F1*. Available at: <https://www.shell.com/inside-energy/fuelling-scuderia-ferrari-into-a-new-era-for-f1.html> (Accessed: 26 June 2023).
- [12] Fédération Internationale de l'Automobile (FIA) (2023) *2024 Formula One Technical Regulations*, *FIA Technical Regulations*.
- [13] Mercedes-AMG Petronas Formula One Engine (2017) *F1 Explained: The Most Powerful Mercedes F1 Engine Ever Made!* Available at: <https://www.youtube.com/watch?v=rGDJqTDXgtg> (Accessed: 26 June 2023).
- [14] Mercedes-AMG Petronas Formula One Engine (2022) *Power Unit 101 with Petronas: Internal Combustion Engine, EXPLAINED!* Available at: <https://www.youtube.com/watch?v=l2b4edxCClc> (Accessed: 26 June 2023).
- [15] Fédération Internationale de l'Automobile (FIA) (2023) *2026 Formula One Power Unit Technical Regulations*, *FIA Technical Regulations*.

- [16] Chapman, F. A. (2022) *Automated optimisation of engine design parameters using 1D performance simulations*. MSc Thesis. Cranfield University.
- [17] Imperial College London (2023) *Turbine Dynamometer*. Available at: <https://www.imperial.ac.uk/turbochargers/facilities/dyno/> (Accessed: 23 March 2023).
- [18] European Federation for Transport and Environment (2018) 'Roadmap to decarbonising European cars by 2050'. Available at: https://www.transportenvironment.org/publications/roadmap-decarbonising-european-cars%0Ahttps://www.transportenvironment.org/sites/te/files/publications/2050_strategy_cars_FINAL.pdf.
- [19] United States Environmental Protection Agency (EPA) (2023) 'Inventory of U.S. Greenhouse Gas Emissions and Sinks: 1991-2021'. Available at: <https://www.epa.gov/ghgemissions/inventory-us-greenhouse-gas-emissions-and-sinks-1990-2021>.
- [20] Diaz, S., Tietge, U. and Mock, P. (2016) 'CO₂ Emissions From New Passenger Cars in Europe: Car Manufacturers' Performance in 2015'. The International Council on Clean Transportation. Available at: <https://theicct.org/publication/co2-new-passenger-cars-europe-aug22/>.
- [21] European Parliament (2022) *EU ban on the sale of new petrol and diesel cars from 2035 explained*. Available at: <https://www.europarl.europa.eu/news/en/headlines/economy/20221019STO44572/eu-ban-on-sale-of-new-petrol-and-diesel-cars-from-2035-explained> (Accessed: 6 June 2023).
- [22] Reitz, R. D., Ogawa, H., Payri, R., Fansler, T., Kokjohn, S., Moriyoshi, Y., Agarwal, A. K., Arcoumanis, D., Assanis, D., Bae, C., Boulouchos, K., Canakci, M., Curran, S., Denbratt, I., Gavaises, M., Guenther, M., Hasse, C., Huang, Z., Ishiyama, T., Johansson, B., Johnson, T. V., Kalghatgi, G., Koike, M., Kong, S. C., Leipertz, A., Miles, P., Novella, R., Onorati, A., Richter, M., Shuai, S., Siebers, D., Su, W., Trujillo, M., Uchida, N., Vaglieco, B. M., Wagner, R. M. and Zhao, H. (2020) 'IJER editorial: The future of the internal combustion engine', *International Journal of Engine Research*, 21(1), pp. 3–10. doi: 10.1177/1468087419877990.
- [23] Zhao, F., Chen, K., Hao, H. and Liu, Z. (2020) 'Challenges, Potential and Opportunities for Internal Combustion Engines in China', *Sustainability*, 12(12):495. doi: 10.3390/su12124955.
- [24] Thøgersen, J., Vatn, A., Aasen, M., Dunlap, R. E., Fisher, D. R., Hellevik, O. and Stern, P. (2021) 'Why do people continue driving conventional cars despite climate change? Social-psychological and institutional insights from a survey of Norwegian commuters', *Energy Research and Social Science*, 79. doi: 10.1016/j.erss.2021.102168.
- [25] Yang, Z., Wang, B. and Jiao, K. (2020) 'Life cycle assessment of fuel cell, electric and internal combustion engine vehicles under different fuel scenarios

and driving mileages in China', *Energy*. Elsevier Ltd, 198, p. 117365. doi: 10.1016/j.energy.2020.117365.

[26] Fédération Internationale de l'Automobile (FIA) (2023) *FIA Environmental Report 2023*. Available at: <https://www.fia.com/news/committed-doing-our-part-fia-releases-its-2023-environmental-report> (Accessed: 26 June 2023).

[27] Formula One (2019) *Sustainability Strategy*. Available at: <https://corp.formula1.com/wp-content/uploads/2019/11/Environmental-sustainability-Corp-website-vFINAL.pdf> (Accessed: 26 June 2023).

[28] Fédération Internationale de l'Automobile (FIA) (2020) *FIA Environmental Strategy 2020-2030*. Available at: <https://www.fia.com/multimedia/publication/fia-environmental-strategy-2020-2030> (Accessed: 26 June 2023).

[29] Motorsport UK (2021) *Sustainability Strategy*. Available at: <https://www.motorsportuk.org/wp-content/uploads/2021/11/Motorsport-UK-Sustainability-Strategy-2021-2030.pdf> (Accessed: 26 June 2023).

[30] Fédération Internationale de l'Automobile (FIA) (2023) *All Formula 1 Teams Achieve FIA Three-Star Environmental Accreditation*. Available at: <https://www.fia.com/news/all-formula-1-teams-achieve-fia-three-star-environmental-accreditation> (Accessed: 22 June 2023).

[31] Caton, J. A. (2018) 'Maximum efficiencies for internal combustion engines: Thermodynamic limitations', *International Journal of Engine Research*, 19(10), pp. 1005–1023. doi: 10.1177/1468087417737700.

[32] Integral Powertrain Limited (2022) *CTSM242 Platform*. Available at: <https://ehelix.com/products/ctsm242-electric-motor/> (Accessed: 22 November 2022).

[33] Wang, Z., Shuai, S., Li, Z. and Yu, W. (2021) 'A review of energy loss reduction technologies for internal combustion engines to improve brake thermal efficiency', *Energies*, 14(20). doi: 10.3390/en14206656.

[34] Heywood, J. (2018) *Internal combustion engine fundamentals*. 2nd Edn. New York: McGraw-Hill Education.

[35] Caton, J. A. (2000) 'A review of investigations using the second law of thermodynamics to study internal-combustion engines', *SAE Technical Papers*, 2000-01–10. doi: 10.4271/2000-01-1081.

[36] Automotive Council UK (2021) *Thermal Propulsion Systems Roadmap 2020*. Available at: https://www.apcuk.co.uk/wp-content/uploads/2021/09/https___www.apcuk_.co_.uk_app_uploads_2021_02_Exec-summary-Technology-Roadmap-Thermal-Propulsion-Systems-final.pdf.

[37] Rakopoulos, C. D. and Giakoumis, E. G. (2006) 'Second-law analyses applied to internal combustion engines operation', *Progress in Energy and Combustion Science*, 32(1), pp. 2–47. doi: 10.1016/j.pecs.2005.10.001.

- [38] Farrell, J. T., Stevens, J. G. and Weissman, W. (2006) 'A Second Law Analysis of High Efficiency Low Emissions Gasoline Engine Concepts', *SAE Technical Papers*, (2006-01-0491). doi: <https://doi.org/10.4271/2006-01-0491>.
- [39] Razmara, M., Bidarvatan, M., Shahbakhti, M. and Robinett, R. D. (2016) 'Optimal exergy-based control of internal combustion engines', *Applied Energy*. Elsevier Ltd, 183, pp. 1389–1403. doi: 10.1016/j.apenergy.2016.09.058.
- [40] Rakopoulos, C. D., Michos, C. N. and Giakoumis, E. G. (2008) 'Availability analysis of a syngas fueled spark ignition engine using a multi-zone combustion model', *Energy*, 33(9), pp. 1378–1398. doi: 10.1016/j.energy.2008.05.007.
- [41] Sayin Kul, B. and Ciniviz, M. (2021) 'An evaluation based on energy and exergy analyses in SI engine fueled with waste bread bioethanol-gasoline blends', *Fuel*. Elsevier Ltd, 286(2). doi: 10.1016/j.fuel.2020.119375.
- [42] Yao, Z. M., Qian, Z. Q., Li, R. and Hu, E. (2019) 'Energy efficiency analysis of marine high-powered medium-speed diesel engine base on energy balance and exergy', *Energy*. Elsevier Ltd, 176, pp. 991–1006. doi: 10.1016/j.energy.2019.04.027.
- [43] Dunbar, W. R. and Lior, N. (1994) 'Sources of combustion irreversibility', *Combustion Science and Technology*, 103(1–6), pp. 41–61. doi: <https://doi.org/10.1080/00102209408907687>.
- [44] Liu, D., Wang, H., Liu, H., Zheng, Z., Zhang, Y. and Yao, M. (2020) 'Identification of factors affecting exergy destruction and engine efficiency of various classes of fuel', *Energy*. Elsevier Ltd, 211. doi: 10.1016/j.energy.2020.118897.
- [45] Caton, J. A. (2010) 'Implications of fuel selection for an SI engine: Results from the first and second laws of thermodynamics', *Fuel*. Elsevier Ltd, 89(11), pp. 3157–3166. doi: 10.1016/j.fuel.2010.05.005.
- [46] Sonntag, R. E., Borgnakke, C. and van Wylen, G. J. (2003) *Fundamentals of Thermodynamics*. 6th Edn. New York: John Wiley & Sons.
- [47] Toulson, E., Schock, H. J. and Attard, W. P. (2010) 'A review of pre-chamber initiated jet ignition combustion systems', *SAE Technical Papers*, 2010-01-22. doi: 10.4271/2010-01-2263.
- [48] Zhu, S., Akehurst, S., Lewis, A. and Yuan, H. (2022) 'A review of the pre-chamber ignition system applied on future low-carbon spark ignition engines', *Renewable and Sustainable Energy Reviews*. Elsevier Ltd, 154. doi: 10.1016/j.rser.2021.111872.
- [49] Almarzooq, Y. M., Schoegl, I. and Petersen, E. L. (2023) 'Laminar flame speed measurements of a gasoline surrogate and its mixtures with ethanol at elevated pressure and temperature', *Fuel*. Elsevier Ltd, 343(February), p. 128003. doi: 10.1016/j.fuel.2023.128003.

- [50] Li, G., Long, Y., Zhang, Z., Liang, J., Zhang, Xiaowu, Zhang, Xintang and Wang, Z. (2019) 'Performance and emissions characteristics of a lean-burn marine natural gas engine with the addition of hydrogen-rich reformat', *International Journal of Hydrogen Energy*. Elsevier Ltd, 44(59), pp. 31544–31556. doi: 10.1016/j.ijhydene.2019.10.007.
- [51] Onofrio, G., Li, C., Garcia Valladolid, P., De La Morena, J., Garcia, A., Tunestal, P. and Beatrice, C. (2020) 'Experimental and Numerical Assessment of Active Pre-chamber Ignition in Heavy Duty Natural Gas Stationary Engine', *SAE Technical Papers*, 2020-01–08. doi: 10.4271/2020-01-0819.
- [52] Jamrozik, A. (2015) 'Lean combustion by a pre-chamber charge stratification in a stationary spark ignited engine', *Journal of Mechanical Science and Technology*, 29(5), pp. 2269–2278. doi: 10.1007/s12206-015-0145-7.
- [53] Peters, N., Pothuraju Subramanyam, S. K., Bunce, M., Blaxill, H., Beilke, N., Tew, D. and Vishwanathan, G. (2021) 'The Effect of Homogeneous Lean Combustion on Efficiency and Emissions Trends in Natural Gas-Fueled Small Engines', *SAE Technical Papers*, 2021-01–06. doi: 10.4271/2021-01-0652.
- [54] Beér, J. M. (1994) 'Minimizing NO_x emissions from stationary combustion; reaction engineering methodology', *Chemical Engineering Science*, 49(24, Part A), pp. 4067–4083. doi: 10.1016/S0009-2509(05)80006-5.
- [55] Shudo, T. and Mizuide, T. (2002) 'NO_x emission characteristics in rich-lean combustion of hydrogen', *JSAE Review*, 23(1), pp. 9–14. doi: 10.1016/S0389-4304(01)00163-1.
- [56] De Oliveira, W. P., Coelho Baeta, J. G., Silva Mendonça, M., Teixeira Malaquias, A. C., Rodrigues Filho, F. A. and Resende Soares, E. P. (2023) 'A review of passive pre-chamber in turbulent jet ignition system applied on low emissions SI engines', *SAE Technical Papers*, 2022-36–00. doi: 10.4271/2022-36-0066.
- [57] Palakunnummal, M. F., Sahu, P., Ellis, M. and Nazha, M. (2020) 'Simulation-Aided Development of Prechamber Ignition System for a Lean-Burn Gasoline Direct Injection Motor-Sport Engine', *Journal of Engineering for Gas Turbines and Power*, 142(8), pp. 1–10. doi: 10.1115/1.4047767.
- [58] Serrano, D., Zaccardi, J. M., Müller, C., Libert, C. and Habermann, K. (2019) 'Ultra-Lean Pre-Chamber Gasoline Engine for Future Hybrid Powertrains', *SAE Technical Papers*, 2019-24–01. doi: 10.4271/2019-24-0104.
- [59] Benajes, J., Novella, R., Gomez-Soriano, J., Barbery, I., Libert, C., Rampanarivo, F. and Dabiri, M. (2020) 'Computational assessment towards understanding the energy conversion and combustion process of lean mixtures in passive pre-chamber ignited engines', *Applied Thermal Engineering*. Elsevier, 178(May), p. 115501. doi: 10.1016/j.applthermaleng.2020.115501.
- [60] Benajes, J., Novella, R., Gomez-Soriano, J., Martinez-Hernandez, P. J., Libert, C. and Dabiri, M. (2019) 'Evaluation of the passive pre-chamber ignition concept for future high compression ratio turbocharged spark-ignition engines',

Applied Energy. Elsevier, 248, pp. 576–588. doi: 10.1016/j.apenergy.2019.04.131.

[61] Almatrafi, F., Uddeen, K., Houidi, M. Ben, Cenker, E. and Turner, J. (2022) 'Direct Injection Strategy to Extend the Lean Limit of a Passive Pre-Chamber', in *Proceedings of ASME 2022 ICE Forward Conference, ICEF 2022*. Indianapolis, Indiana, USA. 16-19 October. doi: 10.1115/ICEF2022-89021.

[62] Hua, J., Zhou, L., Gao, Q., Feng, Z. and Wei, H. (2021) 'Influence of pre-chamber structure and injection parameters on engine performance and combustion characteristics in a turbulent jet ignition (TJI) engine', *Fuel*. Elsevier Ltd, 283. doi: 10.1016/j.fuel.2020.119236.

[63] Novella, R., Gomez-Soriano, J., Martinez-Hernandez, P. J., Libert, C. and Rampanarivo, F. (2021) 'Improving the performance of the passive pre-chamber ignition concept for spark-ignition engines fueled with natural gas', *Fuel*. Elsevier Ltd, 290. doi: 10.1016/j.fuel.2020.119971.

[64] Atis, C., Chowdhury, S. S., Ayele, Y., Stuecken, T., Schock, H. and Voice, A. K. (2020) 'Ultra-Lean and High EGR Operation of Dual Mode, Turbulent Jet Ignition (DM-TJI) Engine with Active Pre-chamber Scavenging', *SAE Technical Papers*, 2020-01–11. doi: 10.4271/2020-01-1117.

[65] Zhao, F., Harrington, D. L. and Lai, M.-C. (2002) *Automotive Gasoline Direct-Injection Engines*. Warrendale, USA: SAE.

[66] Aleiferis, P. G., Van Romunde, Z. R., Larson, G., Lawes, M. and Sheppard, C. G. W. (2015) 'On the Effect of Ambient Turbulence and Thermodynamic Conditions on Fuel Spray Development for Direct-Injection Spark-Ignition Engines', *Flow, Turbulence and Combustion*, 95, pp. 29–60. doi: 10.1007/s10494-015-9605-5.

[67] Williams, P. A., O'Donoghue, S., Anderson, R. W. and Richardson, S. H. (2001) 'An experimental study of the spray characteristics of pressure-swirl atomizers for DISI combustion systems', *SAE Technical Papers*, 2001-01–19. doi: 10.4271/2001-01-1974.

[68] Liu, J., Chen, H., Lin, S., Li, Y., Ye, L. and Zhang, S. (2017) 'Test study on characteristics of multi-hole injector for gasoline direct injection engine', in SAE China (ed.) *Lecture Notes in Electrical Engineering*. Singapore, Singapore: Springer Nature Singapore Pte Ltd, pp. 161–170. doi: 10.1007/978-981-10-3527-2_15.

[69] Kaminaga, T., Yamaguchi, K., Ratnak, S., Kusaka, J., Youso, T., Fujikawa, T. and Yamakawa, M. (2019) 'A Study on Combustion Characteristics of a High Compression Ratio SI Engine with High Pressure Gasoline Injection', *SAE Technical Papers*, 2019-04–01. doi: 10.4271/2019-24-0106.

[70] Lake, T., Stokes, J., Murphy, R., Osborne, R. and Schamel, A. (2004) 'Turbocharging Concepts for Downsized DI Gasoline Engines', *SAE Technical Papers*, 2004-01–00. doi: 10.4271/2004-01-0036.

- [71] Hu, B., Turner, J. W. G., Akehurst, S., Brace, C. and Copeland, C. (2017) 'Observations on and potential trends for mechanically supercharging a downsized passenger car engine: a review', *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 231(4), pp. 435–456. doi: 10.1177/0954407016636971.
- [72] Alexander, J. and Porpatham, E. (2021) 'Thermodynamic and experimental analysis of turbocharger for a downsized LPG fuelled automotive SI engine', *International Journal of Energy and Environmental Engineering*. Springer Berlin Heidelberg, 12, pp. 383–399. doi: 10.1007/s40095-020-00373-x.
- [73] Huang, Z., Wang, L., Wang, T. and Shen, K. (2022) 'Effects of Dilution Combustion and Miller Cycle on the Performance of Gasoline Engine', *International Journal of Automotive Technology*, 23(2), pp. 511–519. doi: 10.1007/s12239-022-0047-5.
- [74] Tavakoli, S., Jazayeri, S. A., Fathi, M. and Jahanian, O. (2016) 'Miller cycle application to improve lean burn gas engine performance', *Energy*. Elsevier Ltd, 109, pp. 190–200. doi: 10.1016/j.energy.2016.04.102.
- [75] Wei, H., Yu, J. and Zhou, L. (2019) 'Improvement of engine performance with high compression ratio based on knock suppression using Miller cycle with boost pressure and split injection', *Frontiers in Energy*, 13(4), pp. 691–706. doi: 10.1007/s11708-019-0621-3.
- [76] Shen, K., Xu, Z., Chen, H. and Zhang, Z. (2021) 'Investigation on the EGR effect to further improve fuel economy and emissions effect of Miller cycle turbocharged engine', *Energy*. Elsevier Ltd, 215. doi: 10.1016/j.energy.2020.119116.
- [77] Rosetti, A., Iott, C. and Cantore, G. (2019) 'Intake manifold primary trumpet tuning options for fuel flow limited high Performance I.C.E.', *SAE Technical Papers*, 2019-24-00. doi: 10.4271/2019-24-0005.
- [78] Benaitier, A., Krainer, F., Jakubek, S. and Hametner, C. (2023) 'Optimal energy management of hybrid electric vehicles considering pollutant emissions during transient operations', *Applied Energy*. Elsevier Ltd, 344. doi: 10.1016/j.apenergy.2023.121267.
- [79] Liu, T., Feng, L., Hellgren, M. and Wikander, J. (2019) 'A Binary Controller to Ensure Engine Peak Efficiency for a Parallel Hybrid Electric Car', in *2019 IEEE Intelligent Transportation Systems Conference, ITSC 2019*. Auckland, New Zealand, October 27-30: IEEE, pp. 726–732. doi: 10.1109/ITSC.2019.8917315.
- [80] Saju, C., Michael, P. A. and Jarin, T. (2022) 'Modeling and control of a hybrid electric vehicle to optimize system performance for fuel efficiency', *Sustainable Energy Technologies and Assessments*. Elsevier Ltd, 52. doi: 10.1016/j.seta.2022.102087.
- [81] Wang, P., Duan, X., Chen, C. and Liu, S. (2023) 'Experimental investigation on the exhaust emissions and performance of a hybrid electric motor coupled

with an energy recovery system', *Case Studies in Thermal Engineering*. Elsevier Ltd, 45. doi: 10.1016/j.csite.2023.102994.

[82] Park, S., Matsumoto, T. and Oda, N. (2010) 'Numerical analysis of turbocharger response delay mechanism', *SAE Technical Papers*, 2010-01-12. doi: 10.4271/2010-01-1226.

[83] Lee, W., Schubert, E., Li, Y., Li, S., Bobba, D. and Sarlioglu, B. (2017) 'Overview of Electric Turbocharger and Supercharger for Downsized Internal Combustion Engines', *IEEE Transactions on Transportation Electrification*. IEEE, 3(1), pp. 36–47. doi: 10.1109/TTE.2016.2620172.

[84] Kouhyar, F. and Nikzadfar, K. (2022) 'A Model-Based Investigation of Electrically Split Turbocharger Systems Capabilities to Overcome the Drawbacks of High-Boost Downsized Engines', *SAE Technical Papers*, 2022-01-50. doi: 10.4271/2022-01-5052.

[85] Barman, J., Patchappalam, K. and Gambhir, H. (2019) 'Compressed Air in Engine Exhaust Manifold to Improve Engine Performance and Fuel Economy', *SAE Technical Papers*, 2019-26-00. doi: 10.4271/2019-26-0043.

[86] Baietta, L., Alshammari, M., Pesyridis, A. and Gohil, D. (2020) 'Design of an Axial Turbine for Highly Downsized Internal Combustion Engines', *Applied Sciences*, 10(17). doi: <https://doi.org/10.3390/app10175935>.

[87] Zeng, T., Upadhyay, D., Sun, H., Curtis, E. and Zhu, G. G. (2018) 'Regenerative Hydraulic Assisted Turbocharger', *ASME. J. Eng. Gas Turbines Power*, 140(10). doi: 10.1115/1.4039937.

[88] Katrasnik, T., Rodman, S., Trenc, F., Hribernik, A. and Medica, V. (2003) 'Improvement of the Dynamic Characteristic of an Automotive Engine by a Turbocharger Assisted by an Electric Motor', *ASME. J. Eng. Gas Turbines Power*, 125(2), pp. 590–595. doi: 10.1115/1.1563246.

[89] Walkingshaw, J., Iosifidis, G., Scheuermann, T., Filsinger, D. and Ikeya, N. (2015) 'A Comparison of a Mono-, Twin-, and Double-Scroll Turbine for Automotive Applications', *ASME. J. Eng. Gas Turbines Power*, 138(5). doi: <https://doi.org/10.1115/1.4031449>.

[90] Spessa, E. and Baratta, M. (2009) 'Turbocharged CNG Engines for Urban Transportation: Evaluation of Turbolag Reduction Strategies by Means of Computational Analyses', in *Proceedings of the ASME Internal Combustion Engine Division 2009 Spring Technical Conference*. Milwaukee, Wisconsin, USA, 3-6 May: ASME, pp. 573–582.

[91] Stoffels, H., Dunstheimer, J. and Hofmann, C. (2017) 'Potential of Electric Energy Recuperation by Means of the Turbocharger on a Downsized Gasoline Engine', *SAE Technical Papers*, 2017-24-01. doi: 10.4271/2017-24-0162.

[92] Nemmour, A., Inayat, A., Janajreh, I. and Ghenai, C. (2023) 'Green hydrogen-based E-fuels (E-methane, E-methanol, E-ammonia) to support clean

energy transition: A literature review', *International Journal of Hydrogen Energy*. Hydrogen Energy Publications LLC. doi: 10.1016/j.ijhydene.2023.03.240.

[93] Brynolf, S., Hansson, J., Anderson, J. E., Skov, I. R., Wallington, T. J., Grahn, M., Korberg, A. D., Malmgren, E. and Taljegård, M. (2022) 'Review of electrofuel feasibility - Prospects for road, ocean, and air transport', *Progress in Energy*, 4(4). doi: 10.1088/2516-1083/ac8097.

[94] Singh, H., Li, C., Cheng, P., Wang, X. and Liu, Q. (2022) 'A critical review of technologies, costs, and projects for production of carbon-neutral liquid e-fuels from hydrogen and captured CO₂', *Energy Advances*. Royal Society of Chemistry, 1(9), pp. 580–605. doi: 10.1039/d2ya00173j.

[95] Taha, A. A., Abdel-Salam, T. and Vellakal, M. (2015) 'Hydrogen, Biodiesel and Ethanol for Internal Combustion Engines: A Review Paper', in *Proceedings of the ASME 2015 Internal Combustion Engine Division Fall Technical Conference*. Houston, Texas, USA, 8-11 November: ASME. doi: 10.1115/ICEF2015-1011.

[96] Splitter, D., Pawlowski, A. and Wagner, R. (2016) 'A historical analysis of the co-evolution of gasoline octane number and spark-ignition engines', *Frontiers of Mechanical Engineering*, 1. doi: 10.3389/fmech.2015.00016.

[97] Kalghatgi, G. T. (2014) 'The outlook for fuels for internal combustion engines', *International Journal of Engine Research*, 15(4), pp. 383–398. doi: 10.1177/1468087414526189.

[98] Rockstroh, T., Burger, V., Yates, A. and Smit, D. (2014) 'Laminar Flame Speed Characterization of Synthetic Gasoline Components', *SAE Technical Papers*, 2014-01–26. doi: 10.4271/2014-01-2616.

[99] Corrubia, J. A., Capece, J. M., Cernansky, N. P., Miller, D. L., Durrett, R. P. and Najt, P. M. (2020) 'RON and MON chemical kinetic modeling derived correlations with ignition delay time for gasoline and octane boosting additives', *Combustion and Flame*. Elsevier Inc., 219, pp. 359–372. doi: 10.1016/j.combustflame.2020.05.002.

[100] Rockstroh, T., Floweday, G. and Yates, A. (2016) 'Optimisation of synthetic gasoline blend recipes for use in modern charge boosted GDI engines', *Fuel*. Elsevier Ltd, 186, pp. 800–820. doi: 10.1016/j.fuel.2016.09.001.

[101] Rossi, E., Hummel, S., Cupo, F., Vacca, A., Chiodi, M., Bargende, M., Villforth, J., Kulzer, A. C. and Deeg, H. P. (2021) 'Experimental and Numerical Investigation for Improved Mixture Formation of an eFuel Compared to Standard Gasoline', *SAE Technical Papers*, 2021-24–00. doi: 10.4271/2021-24-0019.

[102] Naiki, T., Obata, K. and Watanabe, M. (2020) 'Research of Fuel Components to Expand Lean-limit in Super Lean-burn Condition (Part II)', *SAE Technical Papers*, 2020-001–2. doi: 10.4271/2020-01-2042.

[103] Jurić, F., Ban, M., Priesching, P., Schmalhorst, C., Duić, N. and Vujanović, M. (2022) 'Numerical modeling of laminar flame speed and autoignition delay

using general fuel-independent function', *Fuel*, 323. doi: 10.1016/j.fuel.2022.124432.

[104] Han, W., Sun, Z., Scholtissek, A. and Hasse, C. (2021) 'Machine Learning of ignition delay times under dual-fuel engine conditions', *Fuel*, 288. doi: 10.1016/j.fuel.2020.119650.

[105] Mahle Powertrain (2023) *MAHLE Powertrain Jet Ignition (MJI) - Passive*. Available at: <https://www.mahle-powertrain.com/en/experience/mahle-jet-ignition/> (Accessed: 26 June 2023).

[106] Mahle Powertrain (2023) *MAHLE Powertrain Jet Ignition (MJI) - Active*. Available at: <https://www.mahle-powertrain.com/en/experience/mahle-jet-ignition/> (Accessed: 26 June 2023).

[107] Maserati of Akron (2023) *Maserati MC20 Engine*. Available at: <https://www.akronmaserati.com/mc20-engine/> (Accessed: 26 June 2023).

[108] Mazda (2023) *Mazda Skyactiv-X technology*. Available at: <https://www.mazda.co.uk/why-mazda/skyactiv-x/> (Accessed: 26 June 2023).

[109] Garrett Motion (2023) *Electric Turbo Whitepaper | A key Technology for EU7*. Available at: https://www.garrettmotion.com/electric-hybrid/garrett-e-turbo/?utm_source=Google&utm_medium=SearchAd_BC_01&utm_campaign=Traffic-Electric-Hybrid_GLOBAL&gclid=EAlaIQobChMIsvnRqJ7S_wIVB-3tCh1pxQYyEAAYASAAEgIMEPD_BwE (Accessed: 26 June 2023).

[110] Trevas, I., Baeta, A. J., Pimenta, C., Fernandes, H., Carvalho, M. and Montemor, R. (2017) 'Combustion Analysis on a Variable Valve Actuation Spark Ignition Engine Operating with E22 and E100', *SAE Technical Papers*, 2017-01-10. doi: 10.4271/2017-01-1069.

[111] Stellantis (2010) *Fiat's 1.4 MultiAir Turbo is Best Engine of the Year*. Available at: <https://www.media.stellantis.com/uk-en/fiat/press/fiat-s-1-4-multi-air-turbo-is-best-new-engine-of-the-year> (Accessed: 26 June 2023).

[112] Fédération Internationale de l'Automobile (FIA) (2000) *2001 Formula One Technical Regulations, FIA Technical Regulations*.

[113] Fédération Internationale de l'Automobile (FIA) (2004) *2005 Formula One Technical Regulations, FIA Technical Regulations*.

[114] Fédération Internationale de l'Automobile (FIA) (2005) *2006 Formula One Technical Regulations, FIA Technical Regulations*.

[115] Fédération Internationale de l'Automobile (FIA) (2012) *2013 Formula One Technical Regulations, FIA Technical Regulations*.

[116] Montgomery, J. (2021) *A critical analysis of hybrid power unit regulations and their role in innovation*. MSc Thesis. Cranfield University.

[117] Fédération Internationale de l'Automobile (FIA) (2009) *2009 Formula One Technical Regulations, FIA Technical Regulations*.

- [118] Fédération Internationale de l'Automobile (FIA) (2017) *2017 Formula One Technical Regulations, FIA Technical Regulations*.
- [119] Fédération Internationale de l'Automobile (FIA) (2022) *2022 Formula One Technical Regulations, FIA Technical Regulations*.
- [120] Fédération Internationale de l'Automobile (FIA) (2003) *2004 Formula One Sporting Regulations, FIA Sporting Regulations*.
- [121] Fédération Internationale de l'Automobile (FIA) (2023) *2023 Formula One Sporting Regulations, FIA Sporting Regulations*.
- [122] Mohammadkhani, F. and Yari, M. (2019) 'A 0D model for diesel engine simulation and employing a transcritical dual loop Organic Rankine Cycle (ORC) for waste heat recovery from its exhaust and coolant: Thermodynamic and economic analysis', *Applied Thermal Engineering*. Elsevier, 150(July 2018), pp. 329–347. doi: 10.1016/j.applthermaleng.2018.12.158.
- [123] Bellér, G., Árpád, I., Kiss, J. T. and Kocsis, D. (2021) 'AVL Boost: A Powerful Tool for Research and Education', *Journal of Physics: Conference Series*, 1935(1). doi: 10.1088/1742-6596/1935/1/012015.
- [124] Badiger, R. (2022) *The experimental assessment of bubble formation caused by boiling in the cooling circuits of internal combustion engine*. MSc Thesis. Cranfield University.
- [125] Curran, H. J. (2019) 'Developing detailed chemical kinetic mechanisms for fuel combustion', *Proceedings of the Combustion Institute*. Elsevier Inc., 37, pp. 57–81. doi: 10.1016/j.proci.2018.06.054.
- [126] McBride, B., Gordon, S. and Reno, M. (1993) 'Coefficients for Calculating Thermodynamic and Transport Properties of Individual Species', *Nasa Technical Memorandum*, 4513. Available at: http://ntrs.nasa.gov/archive/nasa/casi.ntrs.nasa.gov/19940013151_1994013151.pdf.
- [127] Lawrence Livermore National Laboratory (2023) *Combustion*. Available at: <https://combustion.llnl.gov/> (Accessed: 26 June 2023).
- [128] Lawrence Livermore National Laboratory (2023) *Mechanisms*. Available at: <https://combustion.llnl.gov/mechanisms> (Accessed: 26 June 2023).
- [129] Pei, Y., Mehl, M., Liu, W., Lu, T., Pitz, W. J. and Som, S. (2015) 'A multicomponent blend as a diesel fuel surrogate for compression ignition engine applications', *Journal of Engineering for Gas Turbines and Power*, 137(11), pp. 1–9. doi: 10.1115/1.4030416.
- [130] Mehl, M., Pitz, W. J., Westbrook, C. K. and Curran, H. J. (2011) 'Kinetic modeling of gasoline surrogate components and mixtures under engine conditions', *Proceedings of the Combustion Institute*. Elsevier Inc., 33(1), pp. 193–200. doi: 10.1016/j.proci.2010.05.027.

- [131] Ansys (2023) *Ansys Chemkin-Pro*. Available at: <https://www.ansys.com/en-gb/products/fluids/ansys-chemkin-pro> (Accessed: 17 April 2023).
- [132] Zheng, L., Nygaard, Z., Figueroa-Labastida, M., Susa, A. J., Ferris, A. M. and Hanson, R. K. (2023) 'Atmospheric-pressure shock-tube measurements of high-temperature propane laminar flame speed across multiple equivalence ratios', *Combustion and Flame*. Elsevier Inc., 251. doi: 10.1016/j.combustflame.2023.112726.
- [133] Yang, S. (2023) 'An improved TRF mechanism for a new turbulent premixed combustion model with application to engine combustion CFD', *International Journal of Engine Research*, 24(7), pp. 3231–3249.
- [134] Lin, K. C., Dahiya, A., Tao, H. and Kao, F. H. (2022) 'Combustion mechanism and CFD investigation of methyl isobutanoate as a component of biodiesel surrogate', *Energy*. Elsevier Ltd, 249. doi: 10.1016/j.energy.2022.123589.
- [135] Ismail, H. M., Ng, H. K., Gan, S., Lucchini, T. and Onorati, A. (2013) 'Development of a reduced biodiesel combustion kinetics mechanism for CFD modelling of a light-duty diesel engine', *Fuel*, 106, pp. 388–400. doi: 10.1016/j.fuel.2012.10.015.
- [136] Honda R&D (2009) 'F1 Special (The Third Era Activities)', *Technical Review 2009*.
- [137] Royce, M. J. and Royce, K. J. (1996) 'An Analysis of the Output of the 1993 3.5L Chrysler-Lamborghini CL-01 Formula 1 Engine', *SAE Technical Papers*, 962543, pp. 201–208. doi: 10.4271/962543.
- [138] Lyon, K. (1996) 'The Chrysler/Lamborghini 3.5L Formula One Engine: Control System Development and Calibration Methodology', *SAE Technical Papers*, 962544, pp. 209–223. doi: 10.4271/962544.
- [139] Ravaglioli, V. and Bussi, C. (2019) 'Model-Based Pre-Ignition Diagnostics in Race Car Application', *Energies*, 12(12):227. doi: 10.3390/en12122277.
- [140] Neumann, M. P., Fieni, G., Balerna, C., Duhr, P., Cerofolini, A. and Onder, C. H. (2023) 'Low-level Online Control of the Formula 1 Power Unit with Feedforward Cylinder Deactivation', *IEEE Transactions on Vehicular Technology*. IEEE. doi: 10.1109/TVT.2023.3246130.
- [141] Bopaiah, K. and Samuel, S. (2020) 'Strategy for Optimizing an F1 Car's Performance Based on FIA Regulations', *SAE International Journal of Advances and Current Practices in Mobility*, 2(5), pp. 2516–2530. doi: 10.4271/2020-01-0545.
- [142] Boretto, A. (2019) 'Energy flow of a 2018 FIA F1 racing car and proposed changes to the powertrain rules', *Nonlinear Engineering*, 9, pp. 28–34. doi: 10.1515/nleng-2018-0171.

- [143] Dec, J. E., Yang, Y. and Dronniou, N. (2012) 'Improving Efficiency and Using E10 for Higher Loads in Boosted HCCI Engines', *SAE International Journal of Engines*, 5(3), pp. 1009–1032. doi: 10.4271/2012-01-1107.
- [144] Dernet, J., Dec, J. E. and Ji, C. (2015) 'Energy Distribution Analysis in Boosted HCCI-like / LTGC Engines - Understanding the Trade-Offs to Maximize the Thermal Efficiency', *SAE International Journal of Engines*, 8(3), pp. 956–980. doi: 10.4271/2015-01-0824.
- [145] Yang, R., Hariharan, D., Zilg, S., Lawler, B. and Mamalis, S. (2018) 'Efficiency and Emissions Characteristics of an HCCI Engine Fueled by Primary Reference Fuels', *SAE International Journal of Engines*, 11(6), pp. 993–1006. doi: 10.4271/2018-01-1255.
- [146] AVL (2023) 'Design a BOOST Calculation Model', in *BOOST User Manual*.
- [147] Harrison, M. F. (2019) 'Cooling and Lubrification', *Motorsport Powertrain Design Module Lecture Notes*.
- [148] Harrison, M. F. (2019) 'Introduction to Valve Flow', *Motorsport Powertrain Design Module Lecture Notes*.
- [149] NewGuy012 (2023) *spider_plot*. Available at: https://github.com/NewGuy012/spider_plot (Accessed: 26 June 2023).
- [150] AVL (2023) *Boost Theory*.
- [151] AVL (2023) *Boost Validation*.
- [152] Chen, Z., Dai, P. and Chen, S. (2012) 'A model for the laminar flame speed of binary fuel blends and its application to methane/hydrogen mixtures', *International Journal of Hydrogen Energy*. Elsevier Ltd, 37(13), pp. 10380–10389. doi: 10.1016/j.ijhydene.2012.04.015.
- [153] AVL (2023) *About AVL*. Available at: <https://www.avl.com/en/about-avl> (Accessed: 7 April 2023).
- [154] AVL (2023) *Boost Gas Properties Tool*.
- [155] Fédération Internationale de l'Automobile (FIA) (2023) *2023 Formula One Technical Regulations*, *FIA Technical Regulations*.
- [156] Pirelli Tyre S.p.A (2023) *F1 tyres*. Available at: <https://www.pirelli.com/tyres/en-ww/motorsport/f1/tyres> (Accessed: 12 April 2023).
- [157] Seward, D. (2014) *Race Car Design*. London: Palgrave.
- [158] Harrison, M. F. (2004) *Vehicle refinement: controlling noise and vibration in road vehicles*. Oxford: Elsevier.
- [159] Garrett Motion Inc. (2022) *G-Series G30-770 58mm*. Available at: <https://www.garrettmotion.com/racing-and-performance/performance-catalog/turbo/g-series-g30-770/> (Accessed: 26 June 2023).

- [160] Stepanov, V. S. (1995) 'Chemical energies and exergies of fuels', *Energy*, 20(3), pp. 235–242. doi: 10.1016/0360-5442(94)00067-D.
- [161] Yoshida, M., Kita, M. and Atarashi, H. (2009) 'Development of Hybrid System for Formula One', *Honda R&D Technical Review F1 Special (The Third Era Activities)*, pp. 225–238.
- [162] Waston, N. and Janota, M. S. (1982) *Turbocharging the internal combustion engine*. London: Macmillan.
- [163] Formula One World Championship Limited (2014) *A racing revolution? Understanding 2014's technical regulations*. Available at: <https://www.formula1.com/en/latest/features/2014/1/A-racing-revolution-Understanding-2014s-technical-regulations.html> (Accessed: 26 June 2023).
- [164] Scarborough, C. (2014) *Complete guide to F1's technical changes for 2014*. Available at: <https://www.autoweek.com/racing/formula-1/a1890431/complete-guide-f1s-technical-changes-2014/> (Accessed: 26 June 2023).
- [165] De Groote, S. (2013) *2014 regulations: F1's engine revolution*. Available at: <https://www.f1technical.net/features/19037> (Accessed: 26 June 2023).
- [166] Livengood, J. C. and Wu, P. C. (1955) 'Correlation of autoignition phenomena in internal combustion engines and rapid compression machines', *Symposium (International) on Combustion*, 5(1), pp. 347–355. doi: 10.1016/S0082-0784(55)80047-1.
- [167] Douaud, A. M. and Eyzat, P. (1978) 'Four-Octane-Number Method for Predicting the Anti-Knock Behavior of Fuels and Engines', *SAE Technical Papers*, 780080. doi: <https://doi.org/10.4271/780080>.
- [168] Sodré, J. R. (1998) 'A parametric model for spark ignition engine turbulent flame speed', *SAE Technical Papers*. doi: 10.4271/982920.
- [169] Wayne, W. S., Clark, N. N. and Atkinson, C. M. (1998) 'A parametric study of knock control strategies for a bi-fuel engine', *SAE Technical Papers*, (724). doi: 10.4271/980895.
- [170] Hoepke, B., Jannsen, S., Kasseris, E. and Cheng, W. K. (2012) 'EGR Effects on Boosted SI Engine Operation and Knock Integral Correlation', *SAE International Journal of Engines*, 5(2), pp. 547–559. doi: 10.4271/2012-01-0707.
- [171] Chen, L., Li, T., Yin, T. and Zheng, B. (2014) 'A predictive model for knock onset in spark-ignition engines with cooled EGR', *Energy Conversion and Management*. Elsevier Ltd, 87, pp. 946–955. doi: 10.1016/j.enconman.2014.08.002.
- [172] Soyulu, S. and Van Gerpen, J. (2003) 'Development of an autoignition submodel for natural gas engines', *Fuel*, 82(14), pp. 1699–1707. doi: 10.1016/S0016-2361(03)00147-9.

- [173] Garrett Motion Inc. (2023) *Garrett G40-1150 71mm*. Available at: <https://www.garrettmotion.com/racing-and-performance/performance-catalog/turbo/g-series-g40-1150-71mm/> (Accessed: 26 June 2023).
- [174] Honda the other side (2023) *Honda K20C1 Engine Specs and Performance*. Available at: <https://hondatheotherside.com/k20c1/> (Accessed: 26 June 2023).
- [175] Goodger, E. M. and Allen, J. E. (2000) *Transport Fuels Technology: Mobility for The Millenium*. Norwich: Landfall Press.
- [176] Wouters, C., Lehrheuer, B., Pischinger, S., Seifert, P., Raabe, T., Kolbeck, M., Rausch, B., Menger, L. and Casal Kulzer, A. (2022) 'Evaluation of Synthetic Gasoline Fuels and Alcohol Blends in a Spark-Ignition Engine', *SAE International Journal of Fuels and Lubricants*, 15(3), pp. 333–347. doi: 10.4271/04-15-03-0017.
- [177] Le Roux de Bretagne, O. and Harrison, M. F. (2023) 'Comparison of the transient performance of a motorsport ICE first fitted with an e-turbocharger and then with a conventional turbocharger with added P1-hybrid electric machine', in Institution of Mechanical Engineers (ed.) *Proceedings of the 15th International Conference on Turbochargers and Turbocharging*. Twickenham, London, 16-17 May.
- [178] Amazon Web Services (2023) *The Development of F1 Insights, Powered by AWS - Braking Performance*. Available at: <https://aws.amazon.com/fr/sports/f1/braking-performance/#:~:text=A key point of intrigue,that of the unaccelerated state.> (Accessed: 26 June 2023).

APPENDICES

Appendix A - Formula One™ Engine / Power Unit Technical regulations evolution analysis

Era Period of application	V10 2000-2005		V8 2006-2013		V6 2014-2025		V6 2026 - Unknown
Year	2001	2005	2006	2013	2014	2024	2026
power	Engine only [5.2.1]	Engine only [5.2.1]	Engine only [5.3.1]	Engine + KERS [5.2.1]	Engine + MGU-K [5.2.1]	Engine + MGU-K [5.3.1]	Engine + MGU-K [5.4.1]
Engine type	4-stroke [5.1.1]	4-stroke [5.1.1]	4-stroke [5.1.1]	4-stroke [5.1.1]	4-stroke [5.1.1]	4-stroke [5.2.1]	4-stroke [5.3.1]
Arrangement	10 cylinders [5.1.4]	10 cylinders [5.1.4]	90° V8 [5.1.4]	90° V8 [5.1.5]	90° V6 [5.1.7]	90° V6 [5.2.7.]	90° V6 [5.3.3]
Swept volume	3L [5.1.2.]	3L [5.1.2]	2.4L [5.1.2]	2.4L [5.1.2]	1.6L [5.1.2]	1.6L [5.2.2]	1.6L [5.3.2]
Forced induction	Forbidden [5.1.3.]	Forbidden [5.1.3]	Forbidden [5.1.3]	Forbidden [5.1.4]	Allowed (Single stage compressor, single stage turbine mandatory) [5.1.6]	Allowed (Single stage compressor, single stage turbine mandatory) [5.2.6]	Allowed (Single stage compressor, single stage turbine mandatory) [5.5.1]

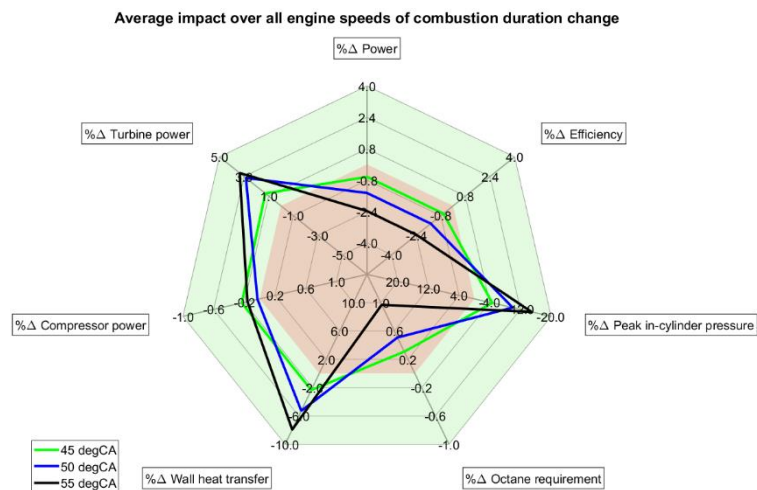
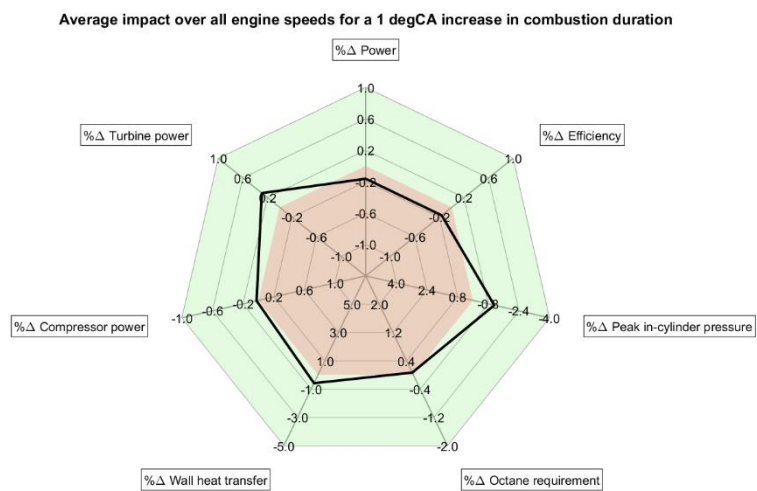
Era Period of application	V10 2000-2005		V8 2006-2013		V6 2014-2025		V6 2026 - Unknown
Year	2001	2005	2006	2013	2014	2024	2026
Inlet air pressure limit	N/A	N/A	N/A	N/A	No	No	4.8 barA [5.5.2]
Fuel flow restrictions	No	No	No	No	Yes (100 kg/h above 10,500 RPM) [5.1.4 and 5.1.5]	Yes (100 kg/h above 10,500 RPM) [5.2.3 and 5.2.4]	Yes (3,000MJ/h above 10,500 RPM) [5.4.3 and 5.4.4]
Engine speed limit	Not mentioned	Not mentioned	Not mentioned	18,000 RPM [5.1.3]	15,000 RPM [5.1.3]	Not mentioned	Not mentioned
Maximum engine allowance	No	Yes, 1 engine for 2 consecutive events [2005 Sporting regulations, Article 85] 19 Grand Prix	Yes, 1 engine for 2 consecutive events [2006 Sporting regulations, Article 87] 18 Grand Prix	Yes, 8 ICE [2013 Sporting regulations, 28.4] 19 Grand Prix	Yes, 5 ICE [2014 Sporting regulations, 28.4] 19 Grand Prix	Yes, 4 ICE [2023 Sporting regulations, 28.2 (2024 Sporting regulations not yet available)] 23 Grand Prix in 2023	Yes, 3 ICE [2026 Sporting regulations, 2.1] Number of Grand Prix to be announced
Compression ratio limit	Not mentioned	Not mentioned	Not mentioned	Not mentioned	Not mentioned	18:1 [5.4.6]	16:1 [5.6.3]
Bore limit	Not mentioned	Not mentioned	98 mm [5.4.1]	98 mm [5.3.1]	80 mm [5.3.1]	80 mm [5.4.1]	80 mm [5.6.1]

Era Period of application	V10 2000-2005		V8 2006-2013		V6 2014-2025		V6 2026 - Unknown
Year	2001	2005	2006	2013	2014	2024	2026
Number of injectors allowed per cylinder	Not mentioned	Not mentioned	1 [5.7.2]	1 [5.10.2]	1 [5.10.2]	1 [5.11.2]	1 [5.11.2]
Number of spark plug allowed per cylinder	Not mentioned	Not mentioned	1 ignition coil + 1 spark plug Plasma, laser of high frequency ignition systems banned [5.8.1]	1 ignition coil + 1 spark plug Plasma, laser of high frequency ignition systems banned [5.11.1]	1 ignition coil + 1 spark plug Plasma, laser of high frequency ignition systems banned [5.11.1]	1 ignition coil + 1 spark plug, max 5 spark events per engine cycle Plasma, laser of high frequency ignition systems banned [5.12.1]	1 ignition coil + 1 spark plug, max one spark per cycle, spark energy limited to 120 mJ [5.12.1] and [5.12.3]
Variable valve timing and variable valve lift systems	Not mentioned	Not mentioned	Banned [5.6.3]	Banned [5.9.3]	Banned [5.9.2]	Banned [5.10.2]	Banned [5.9.2]
Variable geometry inlet systems	Not mentioned	Not mentioned	Banned [5.6.1]	Banned [5.9.1]	Banned (in 2014 only) [5.9.3]	Not mentioned [5.10.1]	Banned [5.9.3]
Variable geometry exhaust systems	Banned [5.4]	Banned [5.4]	Banned [5.6.2]	Banned [5.9.2]	Banned [5.9.1]	Banned [5.10.1]	Banned [5.9.1]

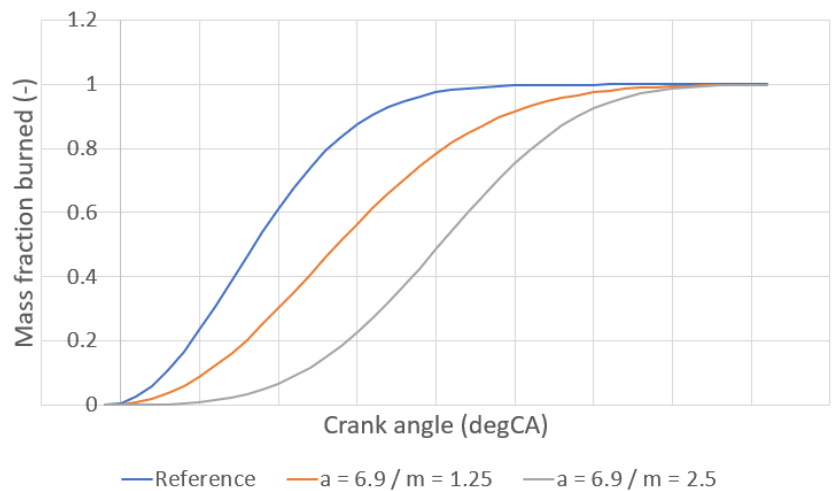
Era Period of application	V10 2000-2005		V8 2006-2013		V6 2014-2025		V6 2026 - Unknown
Year	2001	2005	2006	2013	2014	2024	2026
Fuel injection type	Not mentioned	Not mentioned	Port injection [5.7.2]	Port injection [5.10.2]	Direct injection [5.10.2]	Direct injection [5.11.2]	Direct injection [5.11.2]
Fuel injection pressure limit	Not mentioned	Not mentioned	100 bar [5.7.1]	100 bar [5.10.1]	500 bar [5.10.1]	500 bar [5.11.1]	350 barG [5.11.1]
Fuel composition	Regulated [19.3 and 19.4]	Regulated [19.3 and 19.4]	Regulated [19.3 and 19.4]	Regulated [19.3 and 19.4]	Regulated [19.3 and 19.4]	Regulated [16.3 and 16.4]	Regulated [16.3 and 16.4]

Appendix B - Spider diagrams summarising the sensitivity analysis conducted on the input dataset of the 2014+ Formula One™ engine Digital Twin

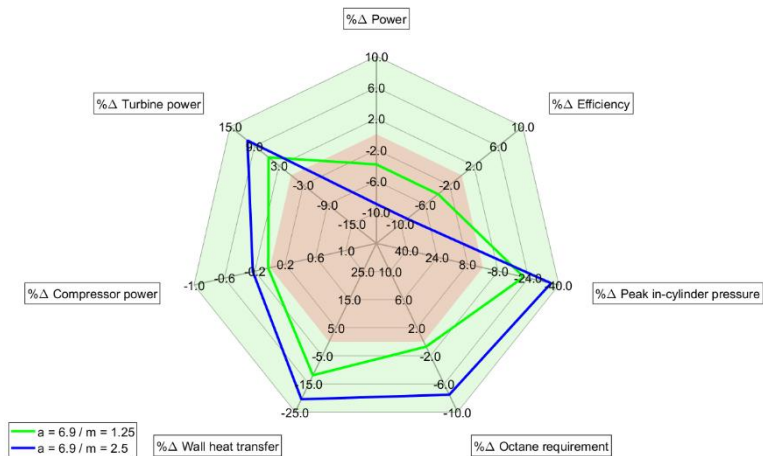
Combustion duration:



Weibe function curve:

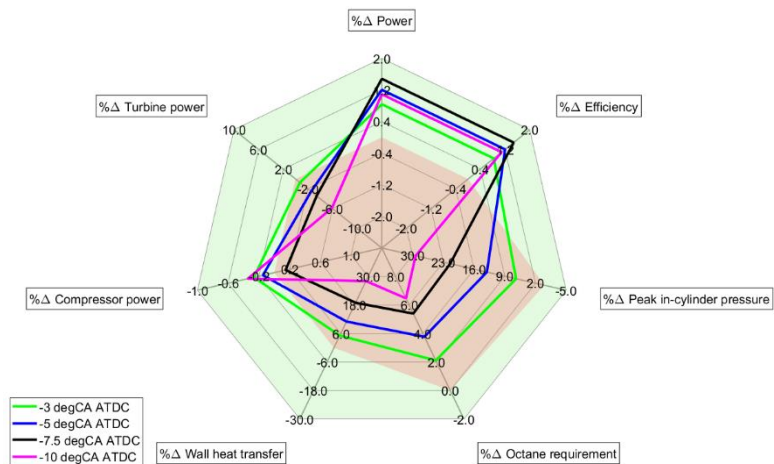


Average impact over all engine speeds of a Weibe curve shape modification

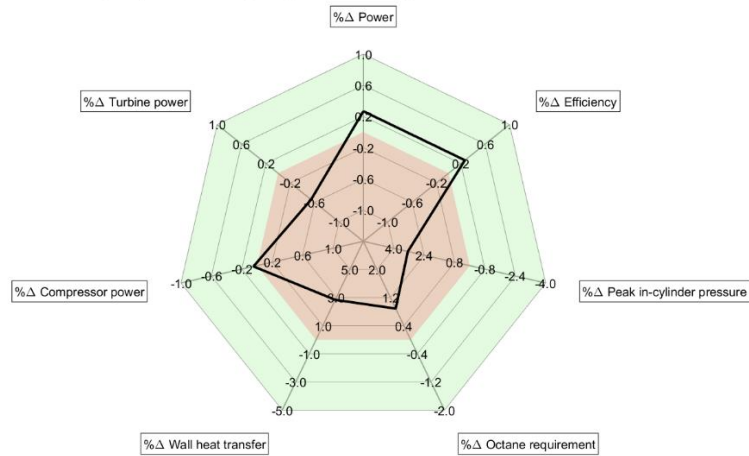


Start of combustion:

Average impact over all engine speeds of combustion advance change

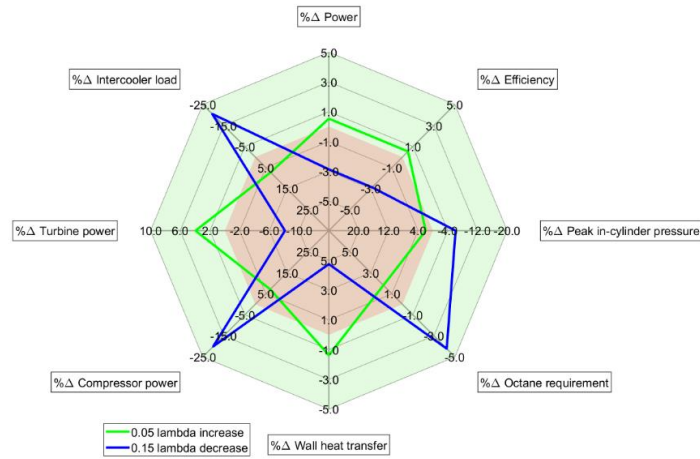


Average impact over all engine speeds for a 1 degCA increase in combustion advance

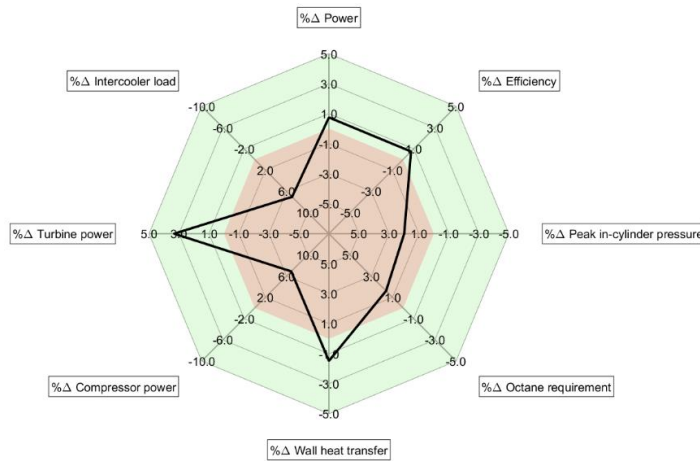


Air/fuel ratio:

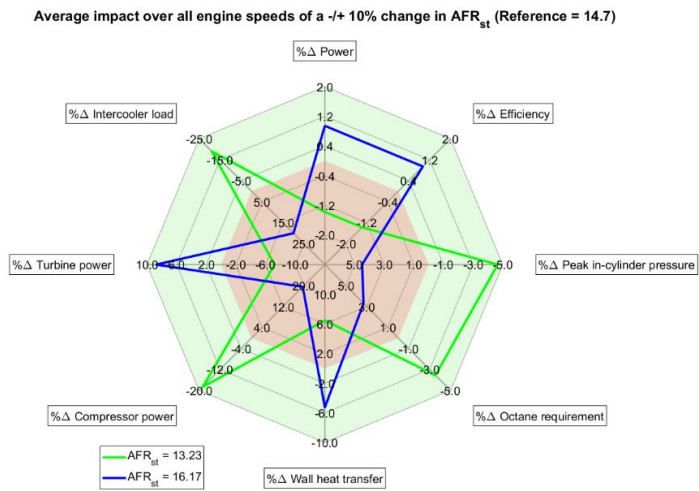
Average impact over all engine speeds of air/fuel ratio modifications



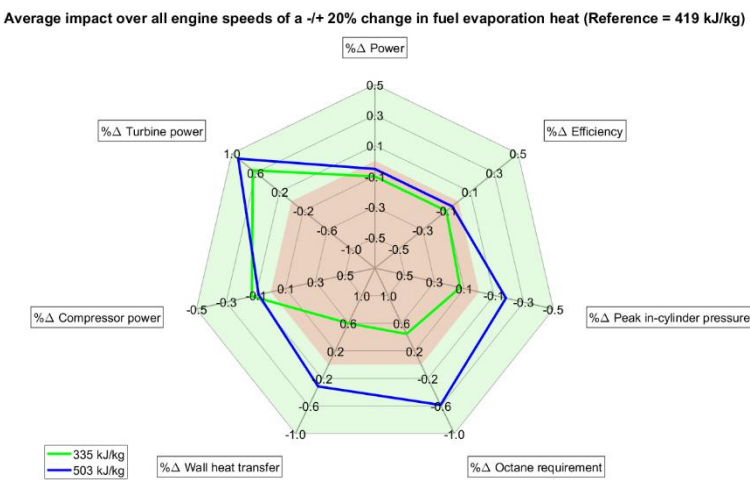
Average impact over all engine speeds of a 0.05 lambda increase



Stoichiometric air/fuel ratio:

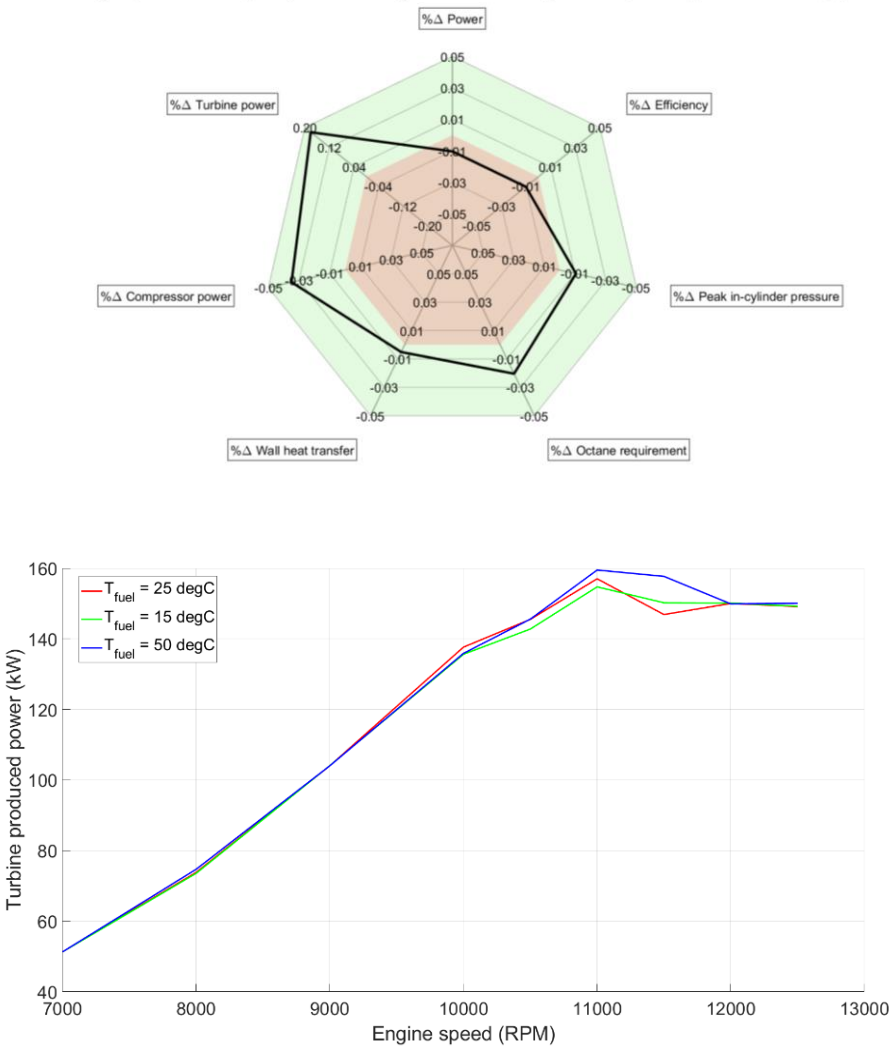


Fuel heat of evaporation:



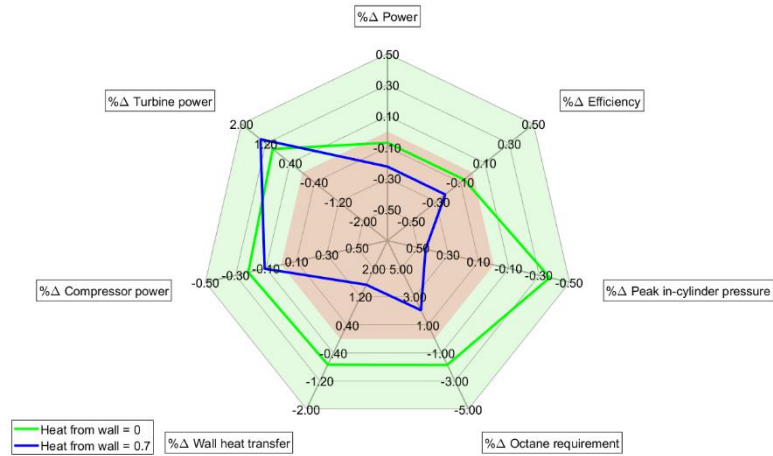
Fuel temperature:

Average impact over all engine speeds of a 5 degC increase in fuel injection temperature (Reference = 25 degC)

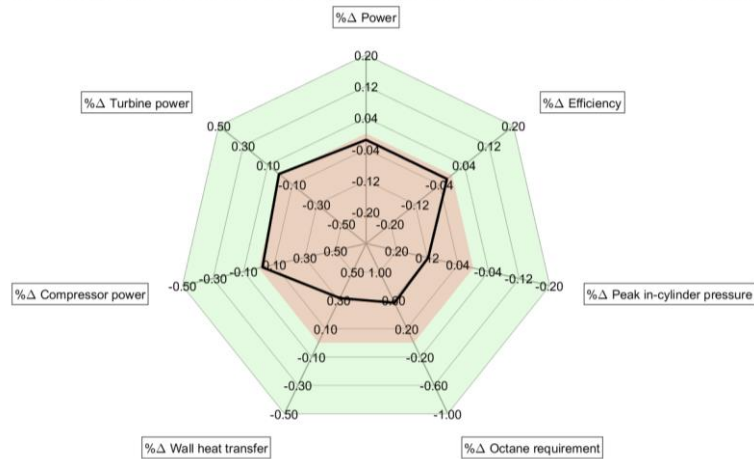


Heat from wall:

Average impact over all engine speed of changes in evaporation heat fraction coming from combustion walls (Reference = 0.3)

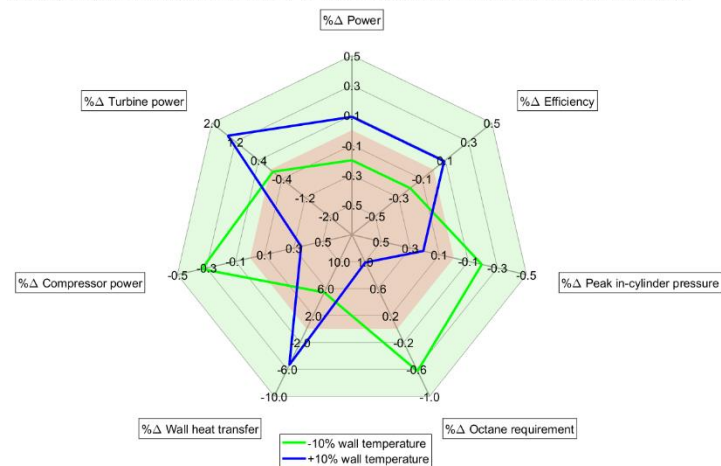


Average impact over all engine speed of a 0.1 increase in evaporation heat fraction coming from combustion walls (Reference = 0.3)

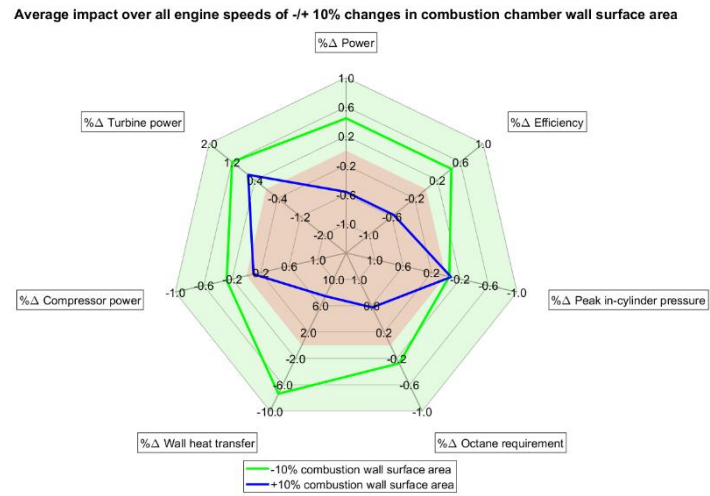


Combustion chamber wall temperature:

Average impact over all engine speeds of +/- 10% changes in combustion chamber wall temperature

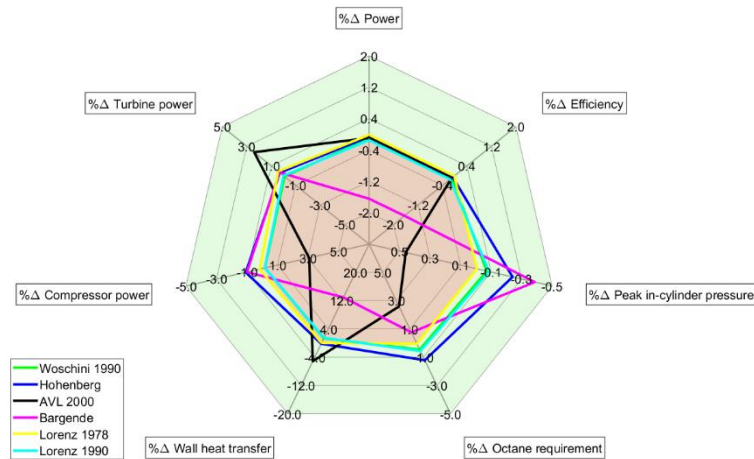


Combustion chamber surface area:

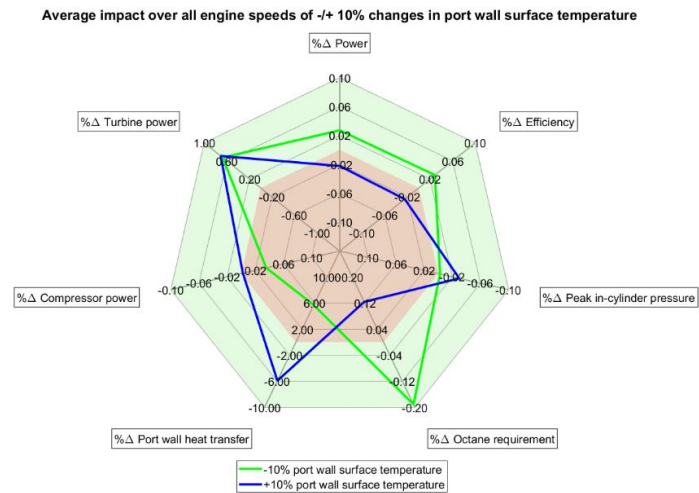


Heat transfer coefficient model:

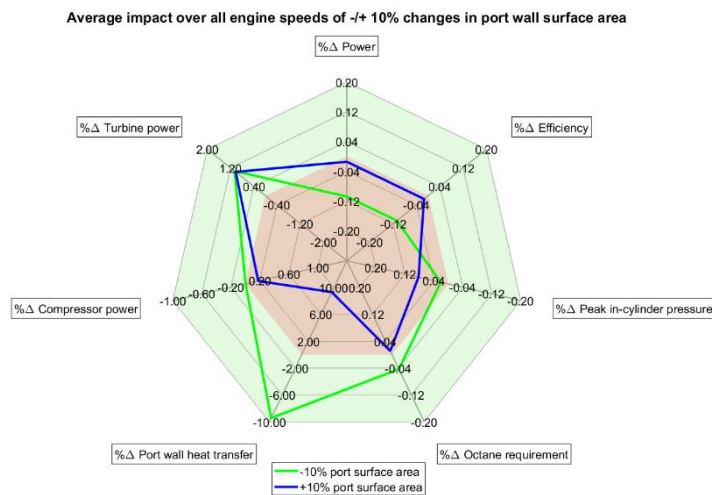
Average impact over all engine speeds of different heat transfer coefficient models (Reference: Woschini 1978)



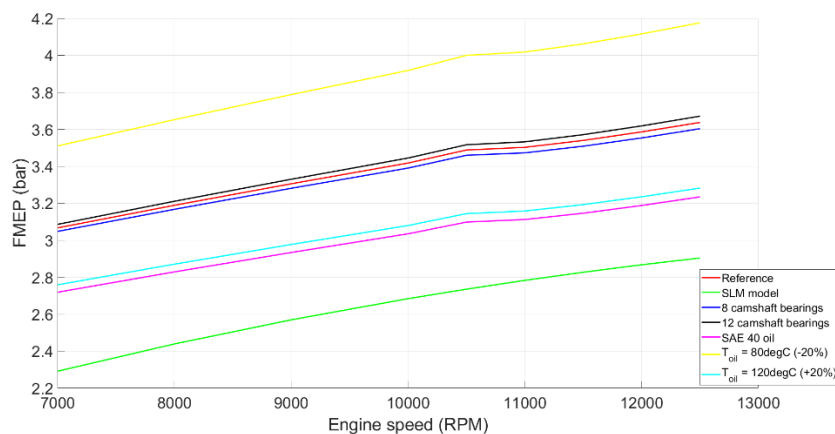
Port heat transfer – Wall temperature:

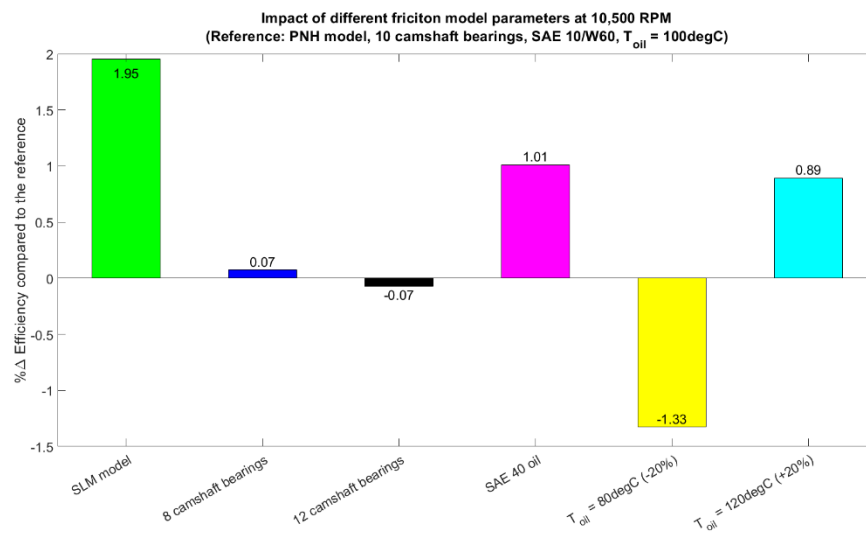


Port heat transfer – Surface areas:



Friction:





Appendix C - Detailed description of AVL Boost 2014+ Digital Twin and 2026 steady-state model input parameter dataset

Parameters	2014+	2026
Engine parameters		
Engine Type	4-stroke	4-stroke
Number of Cylinders	6	6
Cylinder Arrangement	V6	V6
Firing Order (Cylinder numbers shown on the model layout figures in Chapter 4)	1 / 6 / 3 / 4 / 2 / 5	1 / 6 / 3 / 4 / 2 / 5
Identical Cylinders	Yes	Yes
Engine Friction Model	Patton, Nitschke, Heywood	Patton, Nitschke, Heywood
Friction Multiplier	1	1
Type of Valve Train	DOHC-Direct Acting	DOHC-Direct Acting
Number of Camshafts Bearings	10	10
Type of Cam Follower	Flat Follower	Flat Follower
Oil Type	SAE 10/W60	SAE 10/W60
Oil Temperature	100 °C	100 °C
Injection Pump	No	No
Simulation control settings		
Species Transport	Classic	Classic
End of Simulation	40 cycles	40 cycles
Average Cell Size	15 mm	15 mm
Fuel properties		
Fuel Properties	Standard Fuel	User-defined fuel
Fuel	Gasoline	80% Gasoline / 20% Ethanol in volume

Parameters	2014+	2026
Lower Heating Value	44 MJ/kg	41 MJ/kg
Stoichiometric Air/Fuel Ratio	14.7	13.51
Reference Ambient Conditions		
Pressure	1 bar	1 bar
Temperature	25 °C	25 °C
Turbocharger model settings		
Turbocharger Model	Split	Combined
Calculation Type	Simplified	Simplified
Turbine Wall Heat Transfer	No	No
Calculation Mode	N/A	Turbine Layout Calculation & Waste Gate Calculation
Compressor Mode	Corrected Mass Flow	Corrected Mass Flow
Compressor Pressure Reference	1 bar	1 bar
Compressor Temperature Reference	25 °C	25 °C
Turbine Flow Type	Corrected Mass Flow (Swallowing curve)	Discharge Coefficient
Turbine Reference Area	N/A	6,300 mm ²
Turbine Reference Pressure Ratio	3	N/A
Turbine Reference Pressure	2.25 bar	N/A
Turbine Reference Temperature	1086 °C	N/A
Turbine Size Multiplier	N/A	1
Mechanical Efficiency	0.98	0.98
Turbine Isentropic Efficiency	0.7	0.7
Inlet Interference Flow Coefficient	0.5	0.5
Turbine to Total Mass Flow	1	1 in Turbine Layout Calculation Variable in Waste Gate Calculation

Parameters	2014+	2026
MGU-H settings		
Calculation Type	Simplified	N/A
AMEP Consideration of Mechanical Power	Yes	N/A
Electrical Efficiency	1	N/A
Intercooler settings		
Total Air Cooler Volume	5 L	2.5 L
Inter Collector Volume	1 L	0.5 L
Outlet Collector Volume	1 L	0.5 L
Length of Cooling Core	320 mm	320 mm
Hydraulic Setting of Cooling Core	No	No
Friction Specification	Target Pressure Drop	Target Pressure Drop
Heat Transfer Specification	Target Outlet Temperature	Target Outlet Temperature
Reference Mass Flow	0.7 kg/s	0.42 k/s
Reference Inlet Air Temperature	200 °C	150 °C
Reference Inlet Pressure	4.8 bar	2.5 bar
Target Pressure Drop	0.2 bar	0.2 bar
Coolant Temperature	35 °C	35 °C
Target Outlet Temperature	40 °C	40 °C
Flow Coefficients	1	1
Cylinder settings		
Bore	80 mm	80 mm
Stroke	53.052 mm	53.052 mm
Effective Blow-By Gap	0.0008 mm	0.0008 mm
Mean Crankcase Pressure	1 bar	1 bar
Scavenge Model	Perfect Mixing	Perfect Mixing
Initial Conditions at EO - Pressure	5 bar	5 bar

Parameters	2014+	2026
Initial Conditions at EO - Temperature	730 °C	730 °C
Initial Gas Composition – Ratio Type	Air/Fuel Ratio	Air/Fuel Ratio
Initial Gas Composition – Ratio Value	14.7	13.51
Initial Gas Composition – Fuel Vapour	0	0
Initial Gas Composition – Combustion Products	1	1
Heat Release	Vibe 2-Zone	Vibe 2-Zone
Fuelling	Fuel Mass / Cycle	Fuel Mass / Cycle
Mixture Preparation	External	External
Fuel Temperature	25 °C	25 °C
In-Cylinder Evaporation	Yes	Yes
Fuel Evaporation Heat	419 kJ/kg	532.5 kJ/kg
Heat from Wall	0.3	0.3
In-Cylinder Evaporation Curve	User-defined	User-defined
Pollutants	AVL Boost™ default settings	AVL Boost™ default settings
Knock	AVL Boost™ default settings except pressure exponent set at 1.485	AVL Boost™ default settings except pressure exponent set at 1.62
Cylinder Heat Transfer Model	Woschini 1978	Woschini 1978
Port Heat Transfer Model	Zapf	Zapf
Piston Surface Area	5036.55 mm ²	5036.55 mm ²
Piston Calibration Factor	1	1
Cylinder Head Surface Area	5529.2 mm ²	5529.2 mm ²
Cylinder Calibration Factor	1	1
Liner Layer Discretization	No	No
Liner Surface Area	889 mm ²	889 mm ²
Liner Calibration Factor	1	1
Combustion System	DI	DI

Parameters	2014+	2026
In-Cylinder Swirl Ratio	0	0
Variable Wall Temperature	No	No
Valve settings		
Intake Port Surface	8800 mm ²	5655 mm ²
Intake Port Temperature	150 °C	150 °C
Exhaust Port Surface	8400 mm ²	5030 mm ²
Exhaust Port Temperature	240 °C	240 °C
Intake Valve Inner Seat Diameter	30 mm	30 mm
Intake Valve Clearance	0 mm	0 mm
Intake Valve Scaling Factor for Effective Flow Area	1.6	1.6
Intake Valve Reference Lift for Valve Timing	0 mm	0 mm
Intake Valve Reference Lift Base	Effective	Effective
Exhaust Valve Inner Seat Diameter	27 mm	27 mm
Exhaust Valve Clearance	0.25 mm	0.25 mm
Exhaust Vave Scaling Factor for Effective Flow Area	1.6	1.6
Exhaust Valve Reference Lift for Valve Timing	0 mm	0 mm
Exhaust Valve Reference Lift Base	Effective	Effective
Pipe settings		
Compressor Inlet Diameter	120 mm	80 mm
Compressor Inlet Length	350 mm	450 mm
Compressor Outlet Diameter	95 mm	75 mm
Compressor Outlet Length	425 mm	300 mm
Intercooler Outlet Diameter	95 mm	79.5 mm
Intercooler Outlet Length	150 mm	100 mm

Parameters	2014+	2026
Plenum Diameter	90 mm	79.5 mm
Length Between Cylinders	100 mm	101 mm
Intake Runner Diameter	(Cylinders 1 & 6) 75 / 75 / 70 mm (Cylinders 2 & 5) 78 / 75 / 70 mm (Cylinders 3 & 4) 82 / 75 / 70 mm	45 mm
Intake Runner Length	(Cylinders 1 & 6) 0 / 60 / 150 mm (Cylinders 2 & 5) 0 / 60 / 140 mm (Cylinders 3 & 4) 0 / 60 / 125 mm	130 mm
Intake Port Diameter	70 mm	45 mm
Intake Port Length	40 mm	40 mm
Exhaust Port Diameter	67 mm	N/A
Exhaust Port Length	40 mm	N/A
Exhaust Primary Diameter	67 mm	40 / 45 / 50 mm
Exhaust Primary Length	150 mm	0 / 107 / 400 mm
Exhaust Secondary Diameter	80 mm	80 mm
Exhaust Secondary Length	200 mm	320 mm
Turbine Inlet Diameter	80 mm	80 mm
Turbine Inlet Length	130 mm	60 mm
Turbine Outlet Diameter	100 mm	90 mm
Turbine Outlet Length	450 mm	300 mm
Waste Gate Intake Diameter	40 mm	38.8 mm
Waste Gate Intake Length	100 mm	80 mm
Waste Gate Outlet Diameter	38.8 mm	38.8 mm
Waste Gate Outlet Length	100 mm	240 mm
Intake Wall Temperature (Except Compressor Outlet)	40 °C	40 °C

Parameters	2014+	2026
Compressor Outlet Wall Temperature	120 °C	120 °C
Exhaust Wall Temperature	600 °C	600 °C
Turbine Outlet Wall Temperature	300 °C	300 °C
Pipe Laminar Friction Coefficient	64 (AVL Boost™ Default Value)	64 (AVL Boost™ Default Value)
Pipe Turbulent Friction Coefficient	0.019 (AVL Boost™ Default Value)	0.019 (AVL Boost™ Default Value)
Pipe Gas/Wall Heat Transfer Model	Colburn	Colburn
Heat Transfer Factor	1	1
Hydraulic Settings	No	No
Bent Pipe	No	No
Absorptive Material	No	No
Variable Wall Temperature	No	No
Mechanical connection settings		
Calculation Type	Simplified	N/A
Efficiency	1	N/A
Waste Gate settings		
Reference Diameter	39 mm	38.8 mm
Flow Coefficients	Linear evolution from 0 to 1 when the throttle opens from 0 to 90°	Linear evolution from 0 to 1 when the throttle opens from 0 to 90°
Junction settings		
Junction Type	Constant Static Pressure (Constant Pressure for J3 and J4)	Constant Static Pressure
Flow Coefficients for Constant Pressure Junction	Inlet Pipe: Inlet = 1 / Outlet = 0 Outlet Pipe: Inlet = 0 / Outlet = 1	N/A
System boundary settings		
Pressure	1 bar	1 bar
Temperature	25 °C	25 °C

Parameters	2014+	2026
Fuel Vapour	0	0
Combustion Products	0	0
Air/Fuel Ratio	10,000	10,000

