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Agreement **DE-FC26-07NT43274**
Subject **Optimally Controlled Flexible Fuel Powertrain System**
Primary Author **Completion Report – Report Period Oct 1, 2007 to Oct 31, 2010**
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Executive Summary

Distribution

Original: GS/ENS-NA DSP
Exec. Summary: GS/NE, GS/RP-NA, GS/ENS, GS/ENA, GS/NE-MKT
Complete Report: Department of Energy – Michael Ursic, Kevin Stork
GS/ENS-NA – Yilmaz, Brdarski, Cavanaugh, Jiang, Mond, Vanier
RBNA/GOV – Caruso, Johnson
Ricardo – M. Christie
University of Michigan – A. Stefanopoulou, K. Ahn, D. Lee

Subject

Advanced FFV – Optimally Controlled Flexible Fuel Powertrain System Development Project
Completion Report October 1, 2007 – October 30, 2010.

Administrative Status

- The Advanced FFV project was completed on Oct 30 with all key milestones accomplished.
- Bosch organized a recognition event for collaboration with the U.S. Department of Energy in Farmington Hills on August 25, 2010 during which FFV project demo vehicle and key deliverables were presented.

Budget Status

The federal budget for project phase 4 was \$464,056, which was fully exhausted. The last and final invoice was issued on November 30th, 2010 and all necessary ARRA reporting has been completed and submitted as required.

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Technical Status

GOAL I: Improved Fuel Economy with E85

- Base Flex-Fuel Prototype Engine and Vehicle Calibration was completed for E0 and E85
- Flex-Fuel Prototype Engine achieved 420N•m Peak Torque / 295hp Peak Power with E85 and 360N•m Peak Torque / 241hp Peak Power with Gasoline at the engine dynamometer
- Optimized Flex-Fuel Powertrain approach resulted in 10 and 12% fuel economy improvement with E85 compared to baseline production-level engine in a city and highway drive cycle respectively as indicated by the simulation models correlated with prototype engine and vehicle test data

GOAL II: Reduced Emissions with E85

- LEVII-ULEV Emissions Standards were achieved with E0 and E85 during FTP75 drive cycle at the chassis dynamometer

GOAL III: Advanced Flex-Fuel Engine Management System

- Closed-Loop Spark Control and Combustion Phasing Optimization using Extremum Seeking via In-Cylinder Pressure Sensor were developed for efficient combustion with various ethanol fuel blends
- Ethanol Detection and Fuel System Diagnosis via Exhaust Gas Oxygen and In-Cylinder Pressure Sensor Fusion was developed
- Proposed control algorithms were validated at engine dynamometer using rapid prototyping, and integrated into the Bosch MED17.3 ECU

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Project Overview

The primary objective of this project was to develop a true Flex Fuel Vehicle capable of running on any blend of ethanol from 0 to 85% with reduced penalty in usable vehicle range. A research and development program, targeting 10% improvement in fuel economy using a direct injection (DI) turbocharged spark ignition engine was conducted. In this project a gasoline-optimized high-technology engine was considered and the hardware and configuration modifications were defined for the engine, fueling system, and air path. Combined with a novel engine control strategy, control software, and calibration this resulted in a highly efficient and clean FFV concept. It was also intended to develop robust detection schemes of the ethanol content in the fuel integrated with adaptive control algorithms for optimized turbocharged direct injection engine combustion. The approach relies heavily on software-based adaptation and optimization striving for minimal modifications to the gasoline-optimized engine hardware system. Our ultimate objective was to develop a compact control methodology that takes advantage of any ethanol-based fuel mixture and not compromise the engine performance under gasoline operation.

ADVANCED FFV PROJECT DEMONSTRATION VEHICLE

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BOSCH ADVANCED FFV PROJECT TEAM

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Administration

The overall project period is October 1, 2007 to October 31, 2010.

Phase 1 milestones include:

- Combustion characterization engine test data
- Control oriented model that predicts the trends and the dynamic behavior of the flex-fuel engine system. Simulink models, physical equations used in modeling, model simulation results and correlation with experimental data
- WAVE simulation results, Flex Fuel System Proposal for prototype engine
- Vehicle level fuel economy prediction as a function of fuel blend
- Demonstration of BOSCH MED 17 ECU with proposed engine platform and Labcar simulation environment
- A GO/NO GO decision based on the overall results from each of the previously identified milestones

Phase 2 milestones include:

- Specification of components optimized for ethanol combustion
- Engine hardware design for increased cylinder pressure and compatibility of ethanol fuels
- Control algorithms in Simulink/Ascet SD and block diagrams showing the clear concepts, and transmission shift strategy as application data
- Demonstration of prototype vehicle with MED 17 ECU and base level engine/vehicle platform
- A GO/NO GO decision based on the overall results from each of the previously identified milestones

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Phase 3 milestones include:

- Operational prototype engines and vehicle
- A multivariable feedback controller in Simulink, with block diagrams showing the clear concept, for turbocharging and variable valve timing actuation
- A GO/NO GO decision based on the overall results from each of the previously identified milestones

Phase 4 milestones include:

- Real-time code for control algorithms
- Verification data of engine and vehicle level performance with new algorithms and hardware
- Fully operational prototype vehicle ready for demonstration for all results
- A GO/NO GO decision based on the overall results from each of the identified milestones

Phase I

Phase I data gathering tasks related to dynamometer testing of the baseline Ecotec LNF engine with fuels E0, E24, E55, and E85 was completed. The 1D engine simulation model, vehicle simulation model, and control-oriented engine model were developed. Multiple component packages for the Bosch MED17 controller including in-cylinder pressure sensor interface were developed.

Phase II

Components in the air induction and fuel systems were specified for ethanol combustion. Engine hardware design was completed for a higher compression ratio and increased maximum cylinder peak pressure. An ethanol detection methodology for direct injection engines was developed, in which the charge cooling effects of ethanol fuels are extracted from the evolution of cylinder pressures. The model-based control architecture for a flex-fuel engine was proposed for the closed-loop control of ethanol combustion and the coordination of throttling, turbocharging, and variable valve-train. The vehicle-level development and calibration of the base ECU (Bosch MED 17) was completed. The rapid prototyping scheme for the control strategy development and evaluation was identified.

Phase III

After the completion of system specification and design of the engine components, procurement and adaptation of the hardware was initiated in earnest to build prototype optimized FFV engines and vehicle. A block diagram that illustrates the architecture of the multivariable feedback controller for turbocharging, variable valve timing, and spark ignition actuation was completed.

Phase IV

After two prototype optimized FFV engines were built, one of them was set up at the dynamometer for base engine parameterization and calibration. Multiple injection cold start and emission control functionalities were calibrated on the prototype vehicle. After all the proposed control algorithms were validated using rapid-prototyping techniques, they were integrated into the Bosch Motronic Engine Control Unit MED17.3 software.

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Project budget

FFV FINACIAL SUMMARY				
PHASE	NON-FEDERAL BUDGET	NON-FEDERAL ACTUAL	FEDERAL BUDGET	FEDERAL ACTUAL
1	\$ 607,293.00	\$ 563,238.84	\$ 530,952.00	\$ 530,952.00
2	\$ 496,537.00	\$ 561,313.36	\$ 438,676.00	\$ 438,676.00
3	\$ 473,712.00	\$ 555,998.57	\$ 415,851.00	\$ 415,851.00
4	\$ 521,917.00	\$ 281,134.18	\$ 464,056.00	\$ 464,056.32
TOTAL	\$ 2,099,459.00	\$ 1,961,684.95	\$ 1,849,535.00	\$ 1,849,535.32

	Non-Federal	Federal
Actuals to date	\$1,961,684.95	\$1,849,535
Budget	\$2,099,459.00	\$1,849,535
Remaining	\$ 137,774.05	\$0

Federal obligation

PHASE 4			
Phase 4	Invoices	DOE Avail.	Notes
10-Jan		\$464,056	DOE phase 4 budget
10-Feb			
10-Mar			
10-Apr			
10-May	\$158,539	\$305,517	1 st QTR Invoice submitted to DOE
10-Jun			
10-Jul	\$124,860	\$180,657	2 nd QTR Invoice submitted to DOE
10-Aug			
10-Sep			
10-Oct			
10-Nov	\$180,657	\$0	3rd QTR/Final Invoice submitted to DOE.

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GOAL I: Improved Fuel economy with E85

During the first phase of the project, a baseline investigation was carried out with a stock General Motors 2.0L DI Turbocharged LNF engine. The baseline investigation included gathering the necessary data to parameterize a 1-D engine model and investigate the effect of different valving strategies and compression ratios.

Once the thermodynamic 1D engine model was parameterized, fuel and induction systems were specified and the subsequent design of the modified engine components took place. Finally, the procurement and integration of modified components.

DATA GATHERING

A General Motors 2.0L DI Turbo LNF engine was set up for testing at Ricardo Inc in January 2008. Dynamometer testing of four fuel blends including E0 (gasoline), E24, E55 and E85 has been completed. The engine was stripped and machined for cylinder pressure transducers to enable the measurement of combustion parameters. Various locations were identified for the installation of thermocouples and pressure transducers, and the turbocharger was instrumented for the measurement of its rotational speed. Additional fittings were also added to measure engine emissions and smoke. As shown in Figure 1, the engine was re-assembled and set up in a test cell at Ricardo Inc.

Figure 1: Engine set-up on the test stand

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At eight engine speeds ranging from 800RPM up to 4000RPM, a set of operating points, as illustrated in Figure 2, was designed using the Design of Experiment (DoE) Tool, around which the engine was operated and data were recorded. In the current Optimally Controlled Flexible Fuel Powertrain study, all operating points were designed using five key engine operation variables including engine load, intake valve opening, exhaust valve closing, spark angle, and fuel rail pressure.

Figure 2: Design and tested points in the operation space for intake Mass Air Flow (MAF), Intake Valve Opening (IVO), Exhaust Valve Closing (EVC), spark angle, and fuel rail pressure

In order to select optimum actuator set-points in a Flexible Fuel Powertrain, additional dynamometer testing has been conducted to investigate the effects of boost pressure, variable valve timing, lambda control as well as the influence of different injection strategies on system performance under steady operation conditions. All steady-state tests for E0 are listed in Table 1 and those for fuel blend E24, E55 and E85 were operated at the same engine speeds and loads except for labeled cases.

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Fuel Blend	Speed [RPM]	Load [kg/hr]	Variable Valve Timing Control			Boost Control			Injection Mode		Lambda Control		
			Std	Inc	Red	Std	Inc	Red	INJ1	INJ2	Std	Lean	Rich
E0	1000	20	√			√			√	√	√	√	√
	1000	60	√			√			√	√	√	√	√
	2000	100 ¹	√	√	√	√			√	√	√	√	√
	3500	80 ²	√			√			√	√	√	√	√
	3500	300	√			√	√	√	√		√		√

¹ With the other operating conditions the same, tests are conducted at 50 kg/hr for E24, E55 and E85.

² With the other operating conditions the same, tests are conducted at 70 kg/hr for E24 and E55, and 60 kg/hr for E85.

Table 1: List of steady-state tests for fuel E0 at various actuator settings

Table 2 lists actuator setting values employed during the tests at speeds and loads indicated in Table 1. Please note that listed values for Intake Valve Opening (IVO) and Exhaust Valve Closing (EVC) were the setting values for tests operated at a speed of 1000RPM and a load of 60kg/hr. The standard IVO and EVC values employed in other tests refer to pre-calibrated maps stored in the Engine Control Unit (ECU). In addition, listed values for wastegate duty cycle through which desired boost pressure can be achieved were the setting values for the tests operated at a speed of 3500RPM and a load of 300kg/hr. Since desired boost pressure depends on engine speed and load, the wastegate duty cycle also varies across speed and load. For our current operation conditions listed in Table 1, the standard setting for the wastegate duty cycle is 0%.

Control Strategy		Setting	Comments
Variable Valve Timing	Standard (Std)	IVO = -27 [°ATDC], EVC = 28 [°ATDC]	@ 1000 RPM, 60 kg/hr
	Increased (Inc)	IVO = -32 [°ATDC], EVC = 33 [°ATDC]	
	Reduced (Red)	IVO = -22 [°ATDC], EVC = 22 [°ATDC]	
Boost Control	Standard (Std)	Wastegate duty cycle = 40%	@ 3500 RPM, 300 kg/hr
	Increased (Inc)	Wastegate duty cycle = 45%	
	Reduced (Red)	Wastegate duty cycle = 35%	
Injection Mode	Homogenous single injection mode (INJ1)	Injection angle = 275 [°BTDC]	
	Homogenous split injection mode (INJ2)	Begin angle of 1 st Injection = 275 [°BTDC] End angle of 2 nd Injection = 140 [°BTDC] Split factor = 50%	
Lambda Control	Standard (Std)	Desired lambda = 1	
	Lean	Desired lambda = 1.05	
	Rich	Desired lambda = 0.9	

Table 2: Settings for the various control strategies

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Although spark angle is not listed explicitly as a control variable, it was selected based on results from a spark sweep test under each set of operation conditions and control parameter settings. The spark sweep test was conducted by starting the spark at angles of (A1) knocking (if possible), (A2) Maximum Braking Torque (MBT), (A3) the median point between A1 and A2, (A4) 6degCA after A2, and (A5) 12degCA after A2. In addition, tests have been conducted for fuel blends E0, E24, E55 and E85. The engine was subjected to transient operation conditions, such as a step change in the throttle angle, under both injection modes and lambda control on/off conditions. The transient data will be used to calibrate and validate the control-oriented model transient prediction capability.

PERFORMANCE MODEL SIMULATION

A 1-D engine simulation model has been constructed in WAVE to represent the thermodynamic behavior of the LNF engine more accurately. This model has been correlated with the data collected from dynamometer tests for fuel E0 in Figure 3 and for fuel E85 in Figure 4. A compressor map has also been established in Figure 5 based on the data collected during Phase I. Compared with fuel E0, the turbo-charger, as shown in Figure 6, operates at a lower pressure ratio with fuel E85 due to its improved spark efficiency.

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Figure 3: Simulated engine variables using a 1-D WAVE model for E0

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Figure 4: Simulated engine variables using a 1-D WAVE model for E0

Figure 5: Compressor map established via WAVE with operating line of E0 marked

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Figure 6: Comparison of the operating line for E85 with that for E0 at the same torque level

The engine performance of E85 is limited by the current maximum in-cylinder pressure capability. In order to further investigate the potential of improving engine performance with fuel E85, simulation has been conducted at different maximum in-cylinder pressure capabilities in Figure 7 and Figure 8. According to the simulation results, the base engine is able to achieve a torque up to 500N·m and a Brake Mean Effective Pressure (BMEP) of 31bar if it is designed to withstand a maximum in-cylinder pressure of 150 bar.

In Figure 8, the operating lines on a compressor map for E85 at different maximum in-cylinder pressure capabilities are compared. When the maximum in-cylinder pressure capability reaches 140bar, the operating line of E85 starts to deviate from the desirable operation zone of the compressor. Therefore, in order to avoid a change of the turbo-charger, the engine should be designed with a maximum in-cylinder pressure capability of 140bar so that it can achieve a torque up to 470N·m and BMEP of 29.5bar. However, if the goal is to exploit the full potential of E85, a maximum in-cylinder pressure capability around 150bar will be suggested.

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Figure 7: Simulated engine variables using a 1-D WAVE model for E85 when the engine is equipped with different maximum in-cylinder pressure capability

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Figure 8: Operating lines of E85 on the compressor map when the engine is equipped with different maximum in-cylinder pressure capability

SYSTEM SPECIFICATION

A. Induction System Layout

In order to exploit the benefits of higher knock resistance in ethanol fuels, 1-D simulations have been conducted to evaluate the performance of an engine with increased compression ratio and maximum peak cylinder pressure. The candidate design increases the engine compression ratio from 9.25:1 in the base engine to 11.5:1 due to the various constraints. The design of the piston bowl should ensure desirable engine performance during cold start and enable a higher compression ratio while providing the necessary cam phaser authority.

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According to the simulation results from the engine performance analysis, no changes will be required for the intake air box, air filter and ducts, intake manifold, or throttle body.

B. Late Intake Valve Closing (LIVC) Strategy

Late Intake Valve Closing strategy is able to adjust the effective compression ratio for different ethanol fuels by optimizing the cam phasing and modifying the cam profiles to enable duration extension. Thus, LIVC strategy is able to minimize the performance penalty for gasoline or fuels with low ethanol content in an engine with the compression ratio of 11.5:1.

Figure 9 compares the simulated outputs for the Wide-Open-Throttle (WOT) engine performance of the base engine with that of the candidate flex-fuel engine that is equipped with a higher compression ratio of 11.5:1 and modified intake cam profiles. In this study, two intake cam profiles with 20degCA and 40degCA duration extension are evaluated. In addition, in order to evaluate the knock behaviors of such engine and cam profile modifications, the built-in knock model is correlated with cylinder pressure measurements for gasoline, from which the baseline knock index is established.

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Figure 9: Simulation-based comparison of engine performance for gasoline (E0) of the base engine with that of an engine with increased compression ratio and modified intake cam profiles

Note that performance simulations have only been conducted for gasoline (E0) here because it has the worst knocking behaviors. To achieve the same knock level, the timing for spark and intake valve closing are both advanced. As shown in Figure 9, there exist conflicting operation requirements at the low-speed wide-open-throttle regime. On one hand, it is essential to retard the intake cam phasing when running on gasoline, which lowers the effective compression ratio and hence reduces knocking. On the other hand, the late intake valve closing lowers the volumetric coefficient significantly. Furthermore, with the wastegate fully closed in this operation regime, it is not possible to overcome the loss of torque via boosting. In addition, it can be observed that the torque loss at low-speed operation regime increases as the intake cam duration is further extended. Thus, all these conflicting requirements will be taken into account in the design process.

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(a) base engine with regular intake cam profile

(b) engine with increased compression ratio and modified cam profile (20degCA duration extension)

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(c) engine with increased compression ratio and modified cam profile (40degCA duration extension)

Figure 10: Simulation-based comparison of compressor performance for gasoline (E0) of the base engine with that of an engine with increased compression ratio and modified intake cam profiles

Figure 10 compares the simulated compressor performance of base engine with that of an engine with increased compression ratio and modified intake cam profiles. As shown in the figures, the increase of engine compression ratio and the duration extension in the intake cam profile do not have significant effects on the compressor performances. In fact, as observed during the earlier dynamometer testing, fuels with higher ethanol content requires less boosting at the same torque level as a result of their higher spark efficiencies. Therefore, no changes will be required for the compressor and turbine in the candidate flex-fuel engine for ethanol combustion. However, a continuous wastegate actuator may be used to replace the current solenoid valve for improved controllability.

Here, three part-load conditions at (1) 1300RPM-3.0bar BMEP, (2) 2000RPM-2.0bar BMEP, and (3) 2100RPM-4.0bar BMEP, were selected to evaluate the fuel economy benefits of the proposed engine modification. The fuel economy with optimized phasing for intake valve open at these part load conditions is predicted in Table 3. Results indicate that there is a 6% fuel economy improvement with the increased compression ratio and the modified intake cam profile with a duration extension of 20degCA. However, the fuel economy improvement between the intake cam duration of 20degCA and 40degCA is not remarkable, which again confirms the extension of intake cam profile by 20degCA.

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In general, two-thirds of the listed fuel economy improvement can be expected during a regular driving cycle. Thus, the fuel economy of the candidate flex-fuel engine with the increased compression ratio of 11.5:1 and modified intake cam profile is expected to be improved by 4-5%, compared to the base engine, at part-load conditions.

Engine Configuration	BSFC [kg/kW/hr]	Improvement [%]	Residual [%]
1300 RPM / 3.0 bar			
Base LNF	0.396	-	18.33
LNF CR=11.5	0.382	4	18.13
LNF CR=11.5 w/20 Deg Dur Ext, Opt IVO	0.374	6	9.02
LNF CR=11.5 w/40 Deg Dur Ext, Opt IVO	0.369	7	14.09
2000 RPM / 2.0 bar			
Base LNF	0.552	-	20.48
LNF CR=11.5	0.524	5	36.61
LNF CR=11.5 w/20 Deg Dur Ext, Opt IVO	0.521	6	21.82
LNF CR=11.5 w/40 Deg Dur Ext, Opt IVO	0.520	6	12.78
2100 RPM / 4.0 bar			
Base LNF	0.396	-	18.33
LNF CR=11.5	0.382	4	18.13
LNF CR=11.5 w/20 Deg Dur Ext, Opt IVO	0.374	6	9.02
LNF CR=11.5 w/40 Deg Dur Ext, Opt IVO	0.369	7	14.09

Table 3: Predicted performance at part load conditions

The simulations in Figure 11-Figure 13 illustrate that a 20degCA extension on the intake cam duration enables an engine with the increased compression ratio to achieve the production torque level in the low-speed wide-open-throttle regime with gasoline. However, a longer intake cam duration extension of 40degCA does not improve the engine performance in that regime. The torque loss caused by the reduction in volumetric efficiency is generally compensated by advancing the intake cam phasing. However, such a measure for torque loss compensate is constrained by the current cam phaser authority of 50degCA. A review of the piston-to-valve clearance and the trade-off between the intake cam phasing advanced and increase of cam phaser authority is essential and will be discussed in the remainder of the report.

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Figure 11: WAVE simulation-based analysis of the effects of Late Intake Valve Closing (LIVC) strategy on engine knocking in terms of torque and fuel economy: intake cam duration of 40degCA

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Figure 12: WAVE simulation-based analysis of the effects of Late Intake Valve Closing (LIVC) strategy on engine knocking in terms of torque and fuel economy: intake cam duration of 20degCA

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Figure 13: WAVE simulation-based analysis of cam phaser authority

Table 4 illustrates the performance with different engine configurations running with E0 and E85. The engine configurations that were investigated include the two compression ratios (10.66:1 and 11.25:1) and the two intake cam designs (Ecotec LNF cam and FFV LIVC cam). Evaluated at four commonly visited operation points, it has been concluded that the configuration with 10.66:1 compression ratio and LIVC cam strategy yields the best performance.

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1200 rpm / 2.0 Bar

Engine Configuration (eng. / int. cam / CR / fuel)	Int/Exh Cam Pos. (°CA ATDC)	M/rI (Nm/%)	BSFC (g/Kwh)	COV of IMEP (%)	PMEP (Bar)	MAP (mBar)	THC (ppm)	NOx (ppm)	Smoke (FSN)
FFV / LVC / 11.25:1 / E0	12 / 34	1.71	306	1.70	-0.45	567	3088	379	0.01
FFV / Prod. / 11.25:1 / E0	-8 / 34	1.57	325	2.00	-0.50	481	4586	186	0.00
FFV / LVC / 10.66:1 / E0	12 / 34	1.58	330	1.21	-0.40	574	3034	392	0.00
Production LNF / E0	-4 / 3	1.40	360	0.86	-0.64	374	2393	1790	-
FFV / LVC / 10.66:1 / E85	12 / 34	1.68	457	1.22	-0.42	563	1780	195	0.00

1200 rpm / 4.0 Bar

Engine Configuration (eng. / int. cam / CR)	Int/Exh Cam Pos. (°CA ATDC)	M/rI (Nm/%)	BSFC (g/Kwh)	COV of IMEP (%)	PMEP (Bar)	MAP (mBar)	THC (ppm)	NOx (ppm)	Smoke (FSN)
FFV / LVC / 11.25:1 / E0	2 / 44	2.01	251	1.15	-0.27	795	2534	639	0.03
FFV / Prod. / 11.25:1 / E0	-18 / 34	1.92	259	1.07	-0.31	685	3435	452	0.04
FFV / LVC / 10.66:1 / E0	12 / 44	1.93	263	0.91	-0.16	842	2600	849	0.00
Production LNF / E0	-16 / 21	1.76	281	1.78	-0.40	618	2085	1251	-
FFV / LVC / 10.66:1 / E85	12 / 44	1.98	365	0.89	-0.19	817	1349	409	0.00

2000 rpm / 2.0 Bar

Engine Configuration (eng. / int. cam / CR)	Int/Exh Cam Pos. (°CA ATDC)	M/rI (Nm/%)	BSFC (g/Kwh)	COV of IMEP (%)	PMEP (Bar)	MAP (mBar)	THC (ppm)	NOx (ppm)	Smoke (FSN)
FFV / LVC / 11.25:1 / E0	12 / 32	1.58	306	1.19	-0.54	522	2544	757	0.07
FFV / Prod. / 11.25:1 / E0	-17 / 34	1.52	332	1.31	-0.57	454	3460	376	0.00
FFV / LVC / 10.66:1 / E0	2 / 44	1.50	348	1.15	-0.48	525	2466	863	0.00
Production LNF / E0	-17 / 19	1.39	352	1.37	-0.64	409	1664	700	-
FFV / LVC / 10.66:1 / E85	12 / 44	1.55	476	1.52	-0.48	544	1121	247	0.00

2000 rpm / 4.0 Bar

Engine Configuration (eng. / int. cam / CR)	Int/Exh Cam Pos. (°CA ATDC)	M/rI (Nm/%)	BSFC (g/Kwh)	COV of IMEP (%)	PMEP (Bar)	MAP (mBar)	THC (ppm)	NOx (ppm)	Smoke (FSN)
FFV / LVC / 11.25:1 / E0	2 / 44	1.92	256	0.99	-0.42	708	2155	1133	0.02
FFV / Prod. / 11.25:1 / E0	-27 / 24	1.86	263	0.92	-0.43	597	2674	1212	0.04
FFV / LVC / 10.66:1 / E0	12 / 44	1.87	274	0.94	-0.38	769	1938	1436	0.02
Production LNF / E0	-27 / 27	1.78	270	1.05	-0.48	638	1477	983	-
FFV / LVC / 10.66:1 / E85	2 / 44	1.93	379	0.88	-0.33	737	1002	761	0.00

Table 4: Performance comparison with different engine configurations

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C. Fuel System Layout

In a direct injection engine, the fuel system is composed of the low-pressure and high-pressure paths. The low-pressure fuel path is integrated into an in-tank fuel pump module. The high-pressure fuel path, as illustrated in Figure 14, is composed of a fuel pump, a fuel rail, a pressure sensor, and a certain number of injectors.

As listed in Table 5, the operation pressure of an in-production 2.0L Ecotec LNF engine is 150bar, while the one planned for the optimized flex-fuel vehicle is targeted to be 200bar. The high-pressure fuel path in an in-production LNF engine includes a 3-lobe cam driven fuel pump with a nominal flow rate of 0.9cc/rev, a fuel rail with a volume of 135cc and an operation pressure up to 150bar, a pressure sensor with a measurable range of 0-200bar, and injectors with nominal flow rate of 22.5cc/s at 100bar. To accommodate the flex-fuel application, a 4-lobe cam driven fuel pump with a nominal flow rate of 1.1cc/rev will be used to introduce a higher fuel rail pressure. Correspondingly, the operation pressure of the fuel rail will be increased up to 200bar, and the measurable range of the pressure sensor will be extended to 0-260bar. In addition, the volume to the fuel rail will be decreased to 66cc to shorten the start time and reduce pulsation during catalyst heating and idle operations. Based on results from the on-going testing, injectors with the current nominal flow rate of 22.5cc/s will be adequate for the flex-fuel applications. However, spray targeting of the injectors will be optimized for the modified piston bowl design with ethanol fuels.

Fuel System Components	Base Engine: 2.0L Ecotec LNF	
	In Production (150bar)	Optimized for Flex-Fuel Vehicle (200bar)
High Pressure Pump	3-lobe cam driven, 0.9cc/rev	4-lobe cam driven, 1.1cc/rev
High Pressure Injector	22.5cc/s	22.5cc/s, optimized spray targeting
High Pressure Fuel Rail	135cc, 150bar	66cc, 200bar
High Pressure Sensor	200bar	260bar
In-Tank Fuel Pump Module	gasoline only	flex-fuel capable

Table 5: Specifications of fuel system components

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Figure 14: Bosch High Pressure Fuel System Layout

As addressed in [1], a novel strategy for SULEV emission requirements was developed after the completion of the advanced system engineering project between Bosch and Ricardo on DI Boost. The proposed strategy will be implemented in this project to achieve ULEV emission level for fuels with ethanol content up to E85 and investigate the potentials for SULEV standards. The proposed methodology includes the implementation of a multi-injection strategy, the modification of piston bowl design and exhaust system layout, and the optimization of spray targeting. One multi-injection strategy is proposed for a side-mounted direct injection engine is shown in Figure 15, and two schemes for spray targeting optimization are illustrated in Figure 16. As for piston bowl design for HC reductions, modifications include (i) increased bowl width for improved air-fuel mixture, (ii) optimized fuel spray deflection angles, (iii) smoother surface transitions, and (iv) decreased crevice volumes.

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Figure 15: Injection strategies for emission reduction [1]

Figure 16: Optimization of spray targeting

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DESIGN OF MODIFIED ENGINE COMPONENTS

A. Design Analysis

The performance analysis based on dynamometer testing has highlighted the need to increase the maximum cylinder pressure capability in the candidate FFV engine. Since the candidate design is not targeted for the production requirement of 120,000 miles, only key engine components will be analyzed. As listed in Figure 17 (in black font), the components and factors to be investigated include (i) connecting rod, piston pin, piston pin boss, (ii) main and connecting rod bearings, (iii) crankshaft, (iv) cylinder head bolt, and (v) oil film thickness. In addition, the cam profiles for the intake valve cam and the fuel pump will be optimized to achieve Ricardo's kinematics and dynamics design guidelines.

Figure 17: Key engine components

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A.1. Connecting Rod, Piston Pin, Piston Pin Boss

Design analysis presented in this section focuses on the assessment of the impacts introduced by the increased maximum cylinder peak pressure. Figure 18 shows the geometry of the connecting rod, piston pin and boss.

Figure 18: Schematic geometry of the piston pin

Figure 19 illustrated the target maximum cylinder peak pressure for the candidate flex-fuel engine. Point 1 on the curve indicates the worst-case loading for small-end bearing (129.8bar at 2000RPM), while Point 2 indicates the worst-case loading for boss (140bar at 5500RPM). According to the design constraints, specifications of the target design are listed in Table 6. A brief summary of the design analysis is included as follows.

- Base engine: GM Ecotec LNF
 - Stresses and deflections are within Ricardo guidelines
- Optimized design for the increased maximum cylinder peak pressure ($P_{\max} = 140\text{bar}$) with the same geometry in the base engine
 - Stresses and deflections are outside Ricardo guidelines
- Upgraded “Optimized II” design for the increased maximum cylinder peak pressure ($P_{\max} = 140\text{bar}$) with optimized mass
 - Pin mass is increased by 55g (48.2%)
 - Pin diameter is increased by 1mm (4.3%)
 - Pin thickness is increased by 1.5mm (30%)
 - Pin length is increased by 9mm (17.3%)

Karl Schmidt, the current supplier, can provide the piston boss with required stress of 85MPa.

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Figure 19: Target maximum cylinder peak pressure for the candidate flex-fuel engine

BASELINE 92.2 bar		OPTIMIZED II 140 bar Pin OD, t, v Baseline MPa		CONSTRAINTS	
INPUTS					
Pin OD [mm]	23.000	Pin OD [mm]	24.000	Gross Boss Stress [MPa]	85
Pin ID [mm]	13.000	Pin ID [mm]	11.000	Small End Stress [MPa]	190
t [mm]	5.000	t [mm]	6.500	Ovalization [mm]	0.030
a [mm]	2.647	a [mm]	2.647	Bending Deflection [mm]	0.120
v [mm]	13.311	v [mm]	17.811		
w [mm]	18.050	w [mm]	18.050		
x [mm]	1.910	x [mm]	1.910		
Speed for SE [rpm]	1754	Speed for SE [rpm]	2000		
Pmax for SE [bar]	92.2	Pmax for SE [bar]	129.8		
Length [mm]	52	Length [mm]	61.000		
OUTPUTS		OUTPUTS			
Mass [kg]	0.114	Mass [kg]	0.169		
Gross Boss Stress [MPa]	75.652	Gross Boss Stress [MPa]	85.180		
Gross Pin Stress [MPa]	110.620	Gross Pin Stress [MPa]	149.243		
Boss Stress [MPa]	74.621	Boss Stress [MPa]	77.980		
Small End Stress [MPa]	108.670	Small End Stress [MPa]	146.548		
Ovalization [mm]	0.025	Ovalization [mm]	0.017		
Bending Deflection [mm]	0.061	Bending Deflection [mm]	0.119		

Table 6: Design constraints and optimized specifications

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A.2. Crankshaft

Crankshaft analysis in Figure 20 and Figure 21 investigated the stress levels and minimum oil film thickness for both mains and big end bearings in the current crankshaft design. According to the oil film analysis, the minimum oil-film thickness for mains and big-end bearings, predicted for higher maximum cylinder pressures, are both above the Ricardo minimum acceptable value. However, the specific load for the big-end bearings cannot meet the minimum requirements. Therefore, the big-end bearings will be manufactured using high-grade materials such as FM G188 to improve its capability of withstanding higher loading.

A complete stress analysis, based on the finite element of the crankshaft in Figure 23, is being performed to assess the impact of higher loading on the crankshaft. As shown in Figure 22, the lumped-mass model of the timing drive and valvetrain is used to evaluate the effects of the extra loading from the new 4-lobe fuel pump on their durability. In addition, design analysis is conducted to investigate the impacts of the higher loading on the crank due to the increased cylinder peak pressure on the torsional damper matching. Figure 24 shows the initial calculations of the displacement of the baseline damper at current and target maximum cylinder peak pressures. The available analysis suggests no changes in the current design.

Figure 20: Minimum oil film thickness for mains and big end bearings with increased maximum cylinder peak pressure

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Figure 21: Specific loads for mains and big end bearings with increased maximum cylinder peak pressure

Figure 22: Lumped-mass model of the timing drive and valvetrain

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Figure 23: Finite element model for the crankshaft

Figure 24: Displacement of torsional damper at production and modified ratings

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A.3. Valvetrain Analysis

As illustrated in Figure 25, analysis has been conducted to evaluate the dynamic and static behaviors of the valvetrain for the modified cam profiles. Based on the kinematic models that are developed for the camshafts in the base engine, dynamic analysis and design for the cam profile with an extended duration of 37degCA and 20degCA is completed. In addition, the effects of the new 4-lobe fuel pump on the valvetrain are also evaluated.

Figure 25: Modified intake cam profile and the corresponding seating velocities

A.4. Piston-to-Valve Clearance

Figure 26 shows the valve position relative to the piston cut-out. Figure 27 illustrates the differences between the minimum piston-to-valve clearance and the phaser angle for the baseline cam profile and the modified ones with extended durations of 20degCA and 37degCA. The analysis indicates the feasibility of compensating the torque loss in low-speed wide-open-throttle engine operation regime by using cam profiles with extended durations. But it negates the option of retarding the intake cam phasing to compensate such torque loss. In addition, it also confirms the possibility of maintaining a phaser authority of 60degCA.

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Figure 26: Schematic layout of the valve and piston

Figure 27: Minimum piston-to-valve clearance

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B. Component Design

Based on the results from design analysis and performance simulation, CAD modeling of the revised components will be performed. Components to be re-designed include

- cylinder head: modified to enable the use of cylinder pressure sensors and a 4-lobe fuel pump
- piston bowl: optimized geometry to achieve a higher compression ratio and lower emissions
- connecting rod and pin: modified geometry to support higher pressure loading
- crankshaft: modified to compensate the effects of the change of piston mass and higher stress due to increased maximum cylinder peak pressure on balancing
- pin, connecting rod, main bearings, head gasket, crank damper: modified, if needed, to support a higher maximum cylinder peak pressure
- intake camshaft: modified to accommodate the new 4-lobe fuel pump cam and enable LIVC
- intake cam phaser: modified to increase phaser authority

Some of the component changes are illustrated in the following figures.

B.1. Fuel Pump Retrofit

As discussed above, a 4-lobe fuel pump will be used in the candidate flex-fuel engine to enable a higher fuel rail pressure and thus satisfy the injection requirements for ethanol fuels. Associated with the use of the 4-lobe fuel pump, its cam has been integrated with the camshaft model in Figure 28. Figure 29 illustrates the cross-section layout through the new 4-lobe fuel pump and its adaptor. The design of the fuel pump adaptor, such as the thickness of the adaptor flange, will be determined by the definition of spring pre-loading details.

Figure 28: Fuel pump cam model

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Figure 29: Cross-section layout through the new 4-lobe fuel pump and its adaptor

B.2. Cylinder Head

The cylinder head needs to be modified to accommodate the new 4-lobe fuel pump as well as the cylinder pressure sensors.

Figure 30: Schematic layout of the cylinder head

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A revised cylinder head made by NEMAC, rated to a maximum cylinder peak pressure of 130bar, will be manufactured for this program from General Motors. As illustrated in Figure 30, modification of the fuel pump bore depends on the definition of the cam follower bore and the keyway details. In addition, the flame deck of the cylinder head will be milled by 0.5mm for the increased compression ratio.

B.3. Connecting Rod

A drawing of the modified connecting rod is shown Figure 31. The connecting rod is modified to support higher pressure loading.

Figure 31: Modified design for the connecting rod

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B.4. Piston Bowl

Figure 32 compares the current and proposed design for the piston bowl. The major modifications are listed as follows.

1. The bowl width is increased with optimized fuel spray deflection angles and smooth surface transitions for improved air/fuel mixture.
2. Materials are added to increase the compression ratio from 9.25:1 to 10.7:1
3. The top, middle, and lower rings are moved up by 1mm, 1.5mm, and 1.75mm, respectively.
4. The bore is broadened with a large diameter and strengthened with additional materials to support a larger pin.

(a) Current Base Engine

(b) Proposed Design

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(c) Design details

Figure 32: Piston bowl design

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C. Piston Bowl

Hardware procurement and adaptation for the flex-fuel optimized engine is completed. One prototype engine for dynamometer parameterization has been built and instrumented for Design-of-Experiment (DoE) testing. Another prototype engine for vehicle-level calibration will be built after analyzing the DoE testing with E0 and E85. Details of the modified engine components are described as follows. A summary of the procured engine components is given in Appendix I.

C.1. Head Bolts

The head bolts have been changed into high-strength studs for a high pressure load capacity. ARP is selected as the supplier for these studs.

C.2. Cylinder Head and Block

Supplied by General Motors, the cylinder heads cast using the Roto technique can withstand higher cylinder pressures. They have also been machined to be equipped with cylinder pressure sensors. The cylinder blocks for Ecotec LNF MY2010 with higher strengths, also supplied by General Motors, will be used for the prototype engine.

Figure 33: GM 2.0L 135 bar peak pressure cylinder head

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C.3. Head Gasket

The head gaskets, supplied by Elring Klinger, are designed to withstand higher cylinder pressures.

C.4. Piston Bowl and Pin

The piston bowl design has been optimized to achieve a higher compression ratio while maintaining low emissions. Supplied by Kolbenschmidt, the piston bowl and pin have been designed for higher cylinder pressures.

C.5. Intake Cam Phaser and Camshaft

The intake cam phaser has been modified to obtain an increased phasing authority of 60 degCA. The intake cam profile is modified to achieve an extended valve open duration for Late Intake Valve Closing (LIVC) strategies. The intake camshaft has also been evaluated to accommodate the four-lobe fuel pump.

C.6. Rod Bearings and Crank Damper

The rod bearings, supplied by Mahle, have been modified to achieve a higher load capacity of 75MPa. Although no changes are required for the crank damper, additional diamond-coated washers will be used to increase the bolt cover factor.

C.7. Connecting Rod

The connecting rod, supplied by Corrillo, has been designed to withstand a higher pressure loading.

C.8. Fuel System

The fuel system components, supplied by Bosch, have been optimized to satisfy the requirements of direct injection with ethanol fuels. The modified fuel system components include the four-lobe fuel pump with an adaptor for the cylinder head, the injectors with optimized spray pattern, the fuel rail with an optimized volume, and the pressure sensor with a higher measurement range.

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VEHICLE SIMULATION STUDY

An initial vehicle model has been completed via vehicle simulation software EASY5 as illustrated in Figure 34. Its performance and fuel economy is currently being correlated with experimental data. An example of simulated outputs is also shown in Figure 35 based on the EASY5 vehicle simulation model for a small to mid-size SUV available in the US market.

Figure 34: Schematic layout for the vehicle simulation software EASY5

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Figure 35: Simulated outputs using the vehicle simulation software EASY5

Figure 36: Full load torque curves 3.6L V6 and 2.0L I4

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The initial vehicle model was built using a conventional naturally aspirated GM 3.6L HFV6 engine in a Saturn Vue vehicle using dyno data recorded by Ricardo at part load and wide open throttle. Once correlated on E0 to published EPA data (it should be noted that the gear shift schedule is not an exact replicated as Ricardo did not have access to that data), the model was converted to run with the GM LNF 2.0I Turbo engine using dyno data collected during phase 1. Table 7 shows a comparison of the predicted vehicle performance and vehicle fuel economy. Previously published data would suggest that for similar power and torque levels (Figure 36), the expected benefit of downsizing would be between 10 to 15%. The results show 13.9% on the FTP with an overall improvement of 10.5%. Vehicle 0 to 60 mph acceleration is comparable. The energy usage plot is shown for the FTP cycle in Figure 37 and for highway in Figure 38.

Saturn Vue FWD 3.6L / 6 spd AT (4250# ETW)		20.5	32.3	24.53				
RUN		FTP75	Hwy	M-H	70 mph	US06	0-10	0-60
1	3.6L / 6spd AT / 2.77 FDR - baseline	21.8	32.5	25.6	27.3	21.9	1.16	6.74
2	2.0L / 6spd AT / 2.77 FDR - baseline	24.9	34.1	28.3	27.5	23.3	1.10	6.30
	Change	3.0	1.6	2.7	0.2	1.4	-0.1	-0.4
	Improvement	13.9%	4.9%	10.5%	0.9%	6.4%	5.2%	6.5%

Table 7: Predicted performance for fuel economy

Figure 37: Most visited points FTP cycle

Figure 38: Most visited points highway cycle

Further simulations were performed to establish the predicted performance on varying blends of alcohol using data taken during phase 1. Table 8 shows a summary of the results. Overall there is a predicted increase in fuel economy of 26% between E0 and E85 assuming identical engine performance.

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Saturn Vue FWD 3.6L / 6 spd AT (4250# ETW) RUN		City FTP75	Highway Hwy	Mixed M-H
1	3.6L / 6spd AT / 2.77 FDR - baseline	21.8	32.5	25.6
2	2.0L / 6spd AT / 2.77 FDR - baseline	24.9	34.1	28.3
	Improvement over 3.6L V6	13.9%	4.9%	10.5%
3	E24	21.1	29.0	24.1
	Difference from 3.6L V6	-3.2%	-10.7%	-6.0%
	Difference from 2.0L on E0	-15.0%	-14.9%	-15.0%
4	E55	19.2	26.5	21.9
	Difference from 3.6L V6	-11.9%	-18.6%	-14.4%
	Difference from 2.0L on E0	-22.6%	-22.5%	-22.6%
5	E85	18.7	24.4	20.9
	Difference from 3.6L V6	-14.2%	-24.8%	-18.3%
	Difference from 2.0L on E0	-24.6%	-28.4%	-26.1%

Table 8: Fuel economy benefit comparison on different engine platforms

Based on the data collected during Phase I and from background information, it has been noted that the optimum fuel consumption points in the fuel economy map changes as a function of alcohol blend. To investigate the potential of achieving further improvements in fuel consumption by varying the operating point of the engine to take advantage of this, a study was performed to establish the ideal ratios required to drive along the optimum BSFC line for each fuel blend. Figure 39 shows the operating line when running on E0. The simulation tools use a Continuous Variable Transmission (CVT) model to enable the vehicle to be matched to the BSFC curve.

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Figure 39: E0 operating points with CVT

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Each blend of alcohol fuel was evaluated using this manner and the results are shown in Table 9. The results show that as expected there is approximately 2.7% improvement in on cycle fuel economy with a CVT. Interestingly as the alcohol content increases it shows bigger improvements, up to 4.7% could be realized. This suggests that there are worthwhile improvements in cycle fuel economy if the driveline match can be varied as a function of alcohol blend.

Saturn Vue FWD 3.6L / 6 spd AT (4250# ETW)			
RUN	FTP75	Hwy	M-H
1 3.6L / 6spd AT / 2.77 FDR - baseline	20.5	32.3	24.53316
	21.8	32.5	25.6
Saturn Vue FWD 2.0L / 6spd AT (4250# ETW)			
RUN	FTP75	Hwy	M-H
1 2.0L / 6spd AT / 2.77 FDR - baseline	24.9	34.1	28.3
2 E24	21.1	29.0	24.1
Change vs 2.0L w/E0	-15.0%	-14.9%	-15.0%
3 E55	19.5	26.8	22.2
Change vs 2.0L w/E0	-21.6%	-21.5%	-21.6%
4 E85	18.7	24.4	20.9
Change vs 2.0L w/E0	-24.6%	-28.4%	-26.1%

Saturn Vue FWD 2.0L / Ideal Drive Ratios			
RUN	FTP75	Hwy	M-H
1 E00- Baseline	26.1	33.8	29.1
Improvement over 3.6L V6	19.5%	4.1%	13.5%
Improvement over 2.0L w/6 Speed AT	5.0%	-0.8%	2.7%
2 E24	22.3	29.4	25.0
Improvement over 2.0L w/E24	5.4%	1.3%	3.8%
3 E55	20.6	26.9	23.0
Improvement over 2.0L w/E55	5.7%	0.5%	3.7%
4 E85	20.0	24.8	21.9
Improvement over 2.0L w/E85	7.0%	1.4%	4.7%
Change vs 2.0L w/E0	-19.4%	-27.4%	-22.6%

Table 9: List of steady-state tests for fuel E0 at various actuator settings

With the ability to optimize the part load calibration as a function of alcohol content (VVT schedule or external EGR), the fuel consumption can be estimated through the vehicle simulation model.

Table 10 below shows the result of this back-to-back comparison of the two calibration strategies. The results suggest that a 2.3% would be realized on the FTP cycle.

	1	2	3	2 plus 3	FTP75	Hwy	M-H
Fully Optimized E85	15.76	18.48	19.69	19.05	17.83	23.46	19.99
Spark Optimized E85	15.46	17.91	19.33	18.57	17.43	23.37	19.68
					2.31%	0.41%	1.58%

Table 10: Calibration optimization results

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During the program, it became apparent that an SUV as the original target vehicle was not going to be available. A decision was made to use a Chevrolet HHR vehicle which is already equipped with the LNF engine. As a result the vehicle analysis was repeated with the geometry of the HHR vehicle. The baseline performance was correlated to published data.

Reasonable correlation was achieved from a vehicle performance perspective. Table 12 shows the fuel consumption correlation compared to published data.

		2008 HHR SS (LNF) Published Data	LNF
		E0	E85/E0
Final Drive Ratio	--	4.05	4.05
0-30 mph	s		2.90
0-60 mph	s	6.2	6.59
0-100 mph	s	15.1	15.63
1/4 mile	s	14.8	15.15
1/4 mile	mph	99	98.50
Max. Power	HP	260	256
Max. Torque	ft-lbs	260	260

Table 11: Comparison of the Chevy HHR simulation model and published data

		2008 EPA Data	Simulation Model	Percent Difference
N/V ratio	--	37.9	38.2	0.79%
FTP City Cycle Fuel Economy	mpg	26.8	27.5	2.55%
FTP Highway Cycle Fuel Economy	mpg	40.5	40.2	0.74%

Table 12: Fuel Consumption correlation

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Correlation over the drive cycles was very good and well within the acceptable tolerance as shown in Figure 40.

Figure 40: Correlation between measured data and simulated data

The next step of the study was to look at ways of achieving a 10% improvement in fuel consumption. The results taken in the initial study that we did not achieve the expected improvement in thermodynamic efficiency so as a result a study was made to look at utilizing the extra torque capability of the flex fuel engine concept. Previous work has suggested that if the final drive ratio or the transmission shift pattern is re-optimized some improvements in fuel consumption can be made. The following data shows the potential improvement in cycle fuel consumption when the final drive ratio is varied:

	Fuel	FTP Fuel Economy [mpg]		% Difference	Highway Cycle Fuel Economy [mpg]		% Difference
		2.9 FDR	4.05 FDR		2.9 FDR	4.05 FDR	
FFV	E85	21.59	19.37	10.3%	32.36	28.47	12.0%
FFV	E0	29.83	27.02	9.4%	44.62	39.62	11.2%

Table 13: Final Drive Ratio potential improvement in drive cycle fuel consumption

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Figure 41: Final drive ratio sweeps with E0 and E85

The current final drive ratio of the HHR is 4.05:1, the study shows that if the drive ratio could be lowered to less than 3:1, it would enable an approximate 10% improvement in cycle fuel consumption. It was estimated that 2.9:1 would be the lowest final drive ratio that could be achieved in a production setting.

The downside of this reduction in final drive ratio is that there is some loss of vehicle performance. To counteract this, the flex fuel vehicle concept would also use the wide open throttle performance which the enhanced base engine is capable of achieving running on E85. Figure 42 shows a summary of the torque curves created on the engine dyno.

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Figure 42: Full Load Torque Curves with different engine platforms

		2008 HHR SS (LNF) Published Data E0	LNF E85/E0	LNF E85/E0	FFV E85	FFV E0
Final Drive Ratio	--	4.05	4.05	2.90	2.90	2.90
0-30 mph	s		2.90	3.07	2.96	3.38
0-60 mph	s	6.2	6.59	6.75	6.13	7.16
0-100 mph	s	15.1	15.63	16.00	13.87	16.82
1/4 mile	s	14.8	15.15	15.18	14.63	15.61
1/4 mile	mph	99	98.50	97.65	102.74	96.50
Max. Power	HP	260	256	256	300	234
Max. Torque	ft-lbs	260	260	260	307	264

Table 14: Chevy HHR Simulation Performance Results

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Since the maximum torque of E85 on a production-level Ecotec LNF engine is limited by the peak cylinder pressure capability, identical torque curves are used for E0 and E85 in vehicle performance simulations. On the other hand, the optimized flex-fuel engine exploits the high performance characteristics of E85 fuel, which results in a potential optimization of the final drive ratio. The conclusion that can be drawn from this study was that a 10% improvement in fuel economy can be realized without detriment to the vehicle performance. E85 provides an enhanced torque curve which provides the capability to use the lower final drive ratio.

A hypothetical study was also performed on the original SUV (A Saturn Vue). The performance results for this vehicle using the same philosophy of changed final drive ratio and extended torque curve are shown on Table 15.

		2008 Saturn Vue E0	LNF E85/E0	LNF E85/E0	FFV E85	FFV E85	FFV E0	FFV E0
Final Drive Ratio	--	2.77	3.00	2.50	3.00	2.50	3.00	2.50
0-30 mph	s	2.88	3.04	3.12	3.01	3.03	3.10	3.03
0-60 mph	s	6.69	6.91	6.96	6.42	6.43	7.14	7.09
0-100 mph	s	19.25	18.04	18.32	15.61	15.87	18.85	19.01
1/4 mile	s	15.49	15.43	15.57	14.96	15.06	15.65	15.68
1/4 mile	mph	91.45	93.91	93.36	98.62	97.70	92.61	91.80
Max. Power	HP	274	256	256	259	300	234	234
Max. Torque	ft-lbs	251	260	260	295	307	264	264

Table 15: Saturn VUE Simulation Performance Results

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GOAL II: Reduced Emissions with E85

To achieve the California Air Resources Board (CARB) regulations of Ultra Low Emissions Vehicle (ULEV) levels, a comprehensive vehicle level calibration was performed on the modified engine to achieve the targets shown in Table 16:

	NMHC	CO	NOX
	[g/mile]	[g/mile]	[g/mile]
Ca LEV2 ULEV Standard	0.055	2.10	0.070

Table 16: California ULEV standards

CALIBRATION WITH E85

The demonstration vehicle was updated with the E85 optimized engine which included the hardware modifications (Procurement and Adaptation of Hardware) to increase the efficiency on E85. The hardware changes to this engine required a new base engine calibration for pressure model, torque model, fuel path, camshaft positions, etc. which were then included in the vehicle calibration. To successfully run FTP tests, many other calibration areas needed checked and possibly changed. Cold start testing was used to calibrate the start performance, catalyst heating, and idle speed control. On-road driving was performed to check warmed-up operation of closed loop, rear trim, and drivability functionalities.

An updated catalyst was installed on the vehicle. This catalyst replaced only the first can on the exhaust system of this vehicle. The volume is larger and is specified by the supplier of the catalyst (GM) that it is SULEV capable and aged to full useful life (150k miles). Initial FTP tests showed emissions levels better than ULEV.

Further cold start testing was conducted with the use of fast HC analyzers to assess the effect of calibration changes on the emissions. Further debugging of the system was necessary to realize start performance improvement. The results of this testing were used to guide the setting of the start and after-start fueling factors, the desired lambda during the after-start period, and other parameters. The open-loop fueling factor is the main influence on engine torque and emissions performance. The start fueling factor was reduced until lambda showed no rich operation during the start transient. A lean start with a strong run-up is attributed to the use of a multiple injection strategy for all start and after-start injections. The start injection strategy implemented (HP3s) comprises one injection on the suction stroke, then two injections on the compression stroke. The injection timing and fuel mass split between the injection pulses can be calibrated.

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Figure 43: Diagram of start injection types available showing the HP3s type

The engine hardware changes included a different shape on the piston crown, which is important for the wall-guided homogeneous stratified injection (HSP) mode used for fast catalyst heating. The base parameters for HSP mode are set for specific hardware configuration, so this change warranted a review and possible change of these parameters. The engine dynamometer was used to select the camshaft phasing positions, injection angles, and fuel mass split for the HSP combustion mode. The final parameters provided good engine stability and lowest engine-out HC emissions. Before this work on the engine dynamometer, there was some chance of misfire during HSP, which was eliminated with the new data setting.

In an effort to further reduce the CO₂ emissions on the FTP cycle, the engine dynamometer was used to enable, calibrate and evaluate a split injection type during warmed-up homogenous combustion mode. The homogeneous combustion mode supports injection on the suction stroke with either one, two or three separate injections. This was enabled in a speed and load range which is used often on the FTP drive cycle. All testing was done with target lambda equal to one, and different fuel mass split, time pause between injections, and injection angles were explored to achieve a lower BSFC.

Functional areas changed for starts include: high pressure start injection type set to HP3s, start fueling factor, HP3s injection angles, HP3s fuel mass split, start fuel pressure, start throttle angle, start spark timing, and pressure model at start.

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Functional areas changed after the start include: after-start fueling factor, desired lambda during catalyst heating, catalyst heating enable range and heat rate, HSP injection angles, switchover timing to after-start fueling and torque model and camshaft phasing positions during catalyst heating.

Functional areas changed during warm operation include: lambda sensor heating for quick closed loop, transient fueling, rear trim target and gains, idle speed control, and accessory load control.

Functional areas disabled on the FFV engine: canister purge, knock control, catalyst diagnosis, fuel adaptation, lambda sensor dynamics diagnosis, and misfire detection.

Figure 44: Example measurement of the start on E77 fuel showing engine speed and measured exhaust lambda. Note the strong run-up of engine speed even though lambda remains greater than one until later in the after-start

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Figure 45 Entire FTP cycle measurement showing vehicle speed and tailpipe HC emissions with E85

Figure 46: Cold start and first hill of the FTP cycle showing vehicle speed, engine out and catalyst out emissions with E85

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EMISSIONS TESTING RESULTS

Chassis dynamometer testing was used to measure the regulated and CO2 emissions with California Phase 2 gasoline, E85, and E77 ethanol blends. The results of the FTP tests are shown in Table 17. The base LNF engine was run with gasoline using the MED9.6 production controller and calibration. The E85 test on the LNF engine used the project specific MED17.5 controller with custom software and calibration. This custom software was needed to enable operation of this controller in the vehicle system. The majority of the calibration was identical to the production engine, with exceptions where necessary for the new controller, and the start calibration to enable starting on E85.

Cat	Test - Fuel	NMHC [g/mile]	CO [g/mile]	NOX [g/mile]	CO2 [g/mile]
Green	20C FTP - CaPh2	0.033	0.650	0.001	378
Street	20C FTP - E85	0.016	0.198	0.046	341
SULEV	20C FTP - E0	0.034	0.489	0.028	345
SULEV	20C FTP - E85	0.029	0.246	0.029	349
SULEV	20C FTP - E77	0.021	0.281	0.010	333

Table 17: FTP emissions results from the LNF engine and the FFV engine

The catalyst system on this vehicle comprises a front catalyst, then the secondary oxygen sensor, then a rear catalyst. The system used on the gasoline LNF test was new, with less than 1000 miles of use. Extremely low NOX emissions and somewhat low NMHC emissions show the good conversion, with the majority of the NMHC emitting prior to the front catalyst reaching operating temperature. The system used on the E85 LNF test had been on a vehicle for some mileage, however, less than full useful life (10k-20k miles). The catalyst system used on the FFV engine tests was a “SULEV” capable front catalyst aged to full useful life (150k mile equivalent), with a production rear catalyst.

Catalyst heating performance of each engine is shown in Figure 47 below. The LNF engine uses a higher heat rate during HSP catalyst heating than the FFV engine, and achieved a temperature at 20 seconds after the start about 15 Celsius degrees higher. The heat rate on the FFV engine with E85 was set to a lower value in an effort to minimize the HC mass flow during the afterstart period. There is potential improvement of catalyst light-off with E77 fuel over E85. Because the LNF and FFV engines are turbocharged, with the first catalyst after the turbocharger, higher temperatures for light-off are not achievable without further system development.

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Figure 47: Catalyst temperature in LNF system higher at drive-away of FTP

The FFV engine had different components in the fuel system. A smaller volume fuel rail was fitted and a 4 lobe camshaft profile was used to drive the high pressure pump, where the production engine had only a 3 lobe profile. Both of these changes were intended to enable a quick pressure build during engine cranking for reliable high-pressure starts, a necessary feature for low emission E85 start. Figure 48 shows pressure measurements during engine cranking for both the base LNF engine and the FFV engine using E85 fuel. The time from feed-pump pressure level (0.5 MPa) to 16 MPa is about half with the FFV engine.

FFV = 0.95 seconds (6-7 deliveries, 1.5 rotations of camshaft)
LNF = 1.90 seconds (10 deliveries, 3+ rotations of camshaft)

Figure 48: Comparison of fuel system pressure rise at start of LNF and FFV engines

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IMPROVING CATALYST LIGHT-OFF THROUGH WASTEGATE POSITION VARIATION

In order to evaluate the potential for improving catalyst light-off by means of bypassing the exhaust turbine, a variation of the wastegate position was tested during several starts. Wastegate position was controlled by feeding the wastegate actuator regulated, compressed air, providing a means of simulating the flexibility of a turbo featuring an electronically-controlled wastegate. The test results are shown below in Figure 49 and Figure 50. Fully opening the wastegate, as seen in Figure 50, reduced the exhaust gas temperature by 30C at twenty seconds after start due to the resultantly lower intake manifold pressure and total air charge. Despite this, the catalyst temperature, shown in Figure 49, was roughly unchanged at twenty seconds after start from the conventional, closed wastegate case at the same 19kW heat request rate (i.e. baseline case).

Figure 49: Catalyst temperature with variations in wastegate position and exhaust heat request rate

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Figure 50: Exhaust gas temperature profile with variations in wastegate position and exhaust heat request rate

By adjusting the feed pressure to the wastegate actuator, the wastegate could be set to intermediate positions, thereby allowing control over the balance between exhaust energy sunk in the catalyst with that used to increase the engine's air charge. With the wastegate reduced from fully open to approximately 25% open, the catalyst light-off performance exceeded the baseline case by approximately 15C at twenty seconds after start.

Further increases in the heat request rate with the wastegate opened were ineffective due to the limited air charge available. With the wastegate operating conventionally, however, the heat request rate could be further increased. As observed, the best performing catalyst light-off of all tested configurations was achieved by increasing the heat request rate to 22kW on the conventional system.

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GOAL III: Advanced Flex-Fuel Engine Management System

A GM 2.0L Ecotec I4 engine has been selected as the base engine for our investigation. Cutting-edge automotive technologies such as turbocharging, direct fuel injection, and variable valve timing (VVT) incorporated into this engine are integral to the study of how to best to characterize and exploit the advantages of ethanol fuel.

Figure 51: Schematic layout of a turbocharged spark ignition direct injection engine

As illustrated in Figure 51, ambient air enters the system and a hot-film mass flow sensor measures the intake air flow rate. Next, the air passes through the compressor side of the turbocharger and then the intercooler, which results in a boost pressure that is higher than ambient pressure. After the air passes through the throttle, it accumulates in the intake manifold and enters the cylinder when the intake valve opens. Depending on intake and exhaust valve timing, a fraction of the exhaust gas from the current cycle remains in the cylinders and influences combustion behavior during the next cycle. In the mean time, the injector sprays a demanded amount of fuel directly into the combustion chamber by controlling fuel rail pressure and injection duration.

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Due to differing vaporization and combustion properties of ethanol, as listed in Table 18, the mixture of air and fuel in the combustion chamber is ignited at an optimized angle according to the ethanol content in fuel along with the injection timing and the amount of residual gas. On the exhaust side, part of the exhaust gas exits via the turbine and generates the power to drive the compressor, while the rest of it passes through the wastegate. The exhaust gas eventually returns back to the environment after being properly treated in the catalyst.

		Gasoline (E0)	Ethanol (E100)
RON		92	111
Stoichiometric air/fuel ratio		14.3	9.0
Density	g/cm ³	0.74	0.79
Boiling Temperature	°C	20-300	78.5
Heat of Combustion	MJ/kg	42.4	26.8
Enthalpy of Vaporization	kJ/kg	420	845

Table 18: Fuel properties of gasoline and ethanol

Figure 52 illustrates the block diagram of the proposed control architecture, with the nomenclature listed in Appendix I, to optimize the engine performance for ethanol fuels ranging from E0 up to E85. The proposed control scheme consists of the close-loop combustion control using cylinder pressure sensing as well as the model-based control of the air and fuel path. When a command from the driver is received, the controller first determines the desired engine torque, T^* . Taking into account the actuator dynamics, the feedforward controller then determines the operation set-points based on the desired engine torque, the engine speed, N , and the estimated ethanol content in the fuel, $E_{toh\%}$. Here, set-points are specified for the following engine variables: intake mass air flow rate, W_a , boost pressure, p_b , intake manifold pressure, p_{im} , throttle angle, θ_{th} , angle of the continuous wastegate, θ_{wg} , variable valve timing, ϕ_{VVT} , amount of fuel enrichment, λ , spark timing, ϕ_s , injection timing, ϕ_i , and combustion states such as angle at 50% mass fraction burn, CA50, and indicated mean effective pressure, IMEP.

Based on the combustion states estimated from the cylinder pressure measurements, the feedback combustion controller adjusts the spark timing to achieve desirable combustion behavior and account for variations in the fuel composition as well as changes in the mass of internally recirculated residual gas or boosting levels. In the air path, based on the measurements from the intake mass air flow sensor and the pressure transducers in the induction and intake manifolds, the feedback air path controllers adjusts the angle set-points of the throttle valve and wastegate to ensure desirable close-loop engine transient behavior and account for the potential variability of the variable valve timing. This controller may also consider the combustion behavior, such as the 10-90% mass fraction burn, CA₁₀₋₉₀, and the maximum rate of cylinder pressure rise, dp_{max} , for adjusting the variable valve timing.

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In the fuel path, the adaptation of fuel composition determines the ethanol content of the fuel that is combusted in the cylinders using the cylinder pressure measurements. The details of the ethanol detection methodology are presented in the remainder of this report. The fuel system adaptation then determines the amount of fuel to be injected based on the measurements from the Exhaust Gas Oxygen (EGO) sensor as well as the estimated ethanol content.

Figure 52: Block diagram for the proposed control architecture

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Figure 54: Integration of Robert Bosch standard Simulation Libraries (RBSL) for the development of the control-oriented model

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Inputs		Outputs	
θ_{th}	Throttle valve angle	p_b	Boost pressure
θ_{IVO}	Angle at intake valve open	p_{im}	Intake manifold pressure
θ_{EVC}	Angle at exhaust valve closing	p_{em}	Exhaust manifold pressure
N_{eng}	Engine rotational speed	W_{air}	Mass flow rate of intake air
p_{fuel}	Fuel rail pressure	W_{fuel}	Mass flow rate of injected fuel
t_{inj}	Fuel injection duration	x_r	Fraction of residual gas
		N_{turbo}	Turbocharger rotational speed
		λ	In-cylinder normalized air-to-fuel ratio

Table 19: List of input and output signals in proposed engine model

As illustrated in Figure 54, the intercooler was modeled as a heat exchanger with varying efficiency, while the throttle is modeled as a standard nozzle with its cross-section area varying as a function of throttle angle. The turbocharger model is composed of the modeling of a centrifugal compressor, a turbine with an integrated wastegate, and a connecting shaft. Based on the performance map for the twin-scroll turbocharger from BorgWarner, the efficiencies of the compressor and turbine are modeled as functions of the upstream/downstream pressure ratio, and the corrected speed and flow parameters. The mass air flow rate through the compressor is modeled using the Euler's equation for turbomachinery [1], while the turbine is modeled as a standard nozzle with a varying cross-section area. Due to the nonlinear effects of the boost pressure, pressure drop across the turbine, and control valve duty cycle on the wastegate lift, the effective cross-section area of the wastegate is then modeled using a small-scale radial-basis neural network.

The model was developed using the first and second law of thermodynamics, in which the conservation of mass and enthalpy flow is captured. In this model, all the volume blocks (induction volume, intake manifold, exhaust manifold, catalyst) use the states of mass and pressure, and output the temperature. The dynamic behavior of the turbocharger is captured via the state of the rotational speed of the shaft that connects the compressor and turbine. Based on the experimental data with fuel E0, the model has been calibrated based on the experimental data.

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B. Sub-system and Component Models

B.1. Intercooler

The intercooler is modeled as a heat exchanger that cools the inlet air temperature towards a given coolant temperature at a certain efficiency.

Figure 55¹: Modeling of the downstream temperature of the intercooler (T_{IC}^{up} and T_{IC}^{dn} denote the upstream and downstream air temperature at the intercooler)

¹ Note that the relative absolute error in the remainder of this section was calculated as

$$e = \frac{1}{N} \sum_{i=1}^N \frac{|y_e(i) - y(i)|}{y(i)} \times 100\%,$$

where y and y_e denote the measured and modeled values, and N is the number of samples

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B.2. Throttle

The throttle is modeled as nozzle with varying cross-section area A_{th} that is proportional to the throttle angle θ_{th} .

Figure 56: Modeling of mass air flow through the throttle

B.3. Turbocharger

The turbocharger includes the compressor, the turbine, and the connecting shaft. Based on the map provided by the manufacturer, the compressor and turbine efficiency are modeled in following equation via regression.

$$\eta_C = f_{\eta_C}(\pi_C, T_C^{up}, N_{turbo}) \text{ and } \eta_T = f_{\eta_T}(\pi_T, T_T^{up}, N_{turbo})$$

where $\pi_C = \frac{p_C^{dn}}{p_C^{up}}$ and $\pi_T = \frac{p_T^{dn}}{p_T^{up}}$. Here p_C^{dn} and p_C^{up} denote the downstream and upstream pressures of the compressor, while p_T^{dn} and p_T^{up} are those from the turbine.

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Figure 57: Modeling of compressor and turbine efficiency

On the other hand, the mass air flow through the compressor and turbine are modeled based on the first law of thermodynamics [1][4]. The principle of the compressor is equivalent to a throttle. Due to the fast rotation of the rotor, the centrifugal-power imparts a high velocity to the air as well as an increase to the downstream pressure. Similarly, the turbine is modeled as nozzle with varying cross-section that is proportional to the ratio between its upstream and downstream pressures.

With the experimental data collected when the engine was operated under steady conditions, the power of compressor and turbine in Figure 60 calculated based on their efficiency and mass air flow model described above matches with each other as expected.

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Figure 58: Modeling of the mass air flow through the compressor

Figure 59: Modeling of the cross-section area and mass air flow through the turbine (W_{EG} denotes the mass air flow rate on the exhaust side)

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Figure 60: Compressor and turbine power calculated based on the efficiency and mass air flow model described above

B.4. Dual-Cam Variable Valvetrain

The cam phasing block in Figure 54 models the fraction of residual gas that is caused by the valve overlap, also known as the internal Exhaust Gas Recirculation (iEGR). Since the fraction of residual gas was not measured directly during the dynamometer testing, Model B is first developed based on the available measurements, and then Model A is parameterized based on the outputs from Model B.

Model B [5] is derived based on the energy balance at the intake valve closing, in which the fraction of residual gas can be obtained in an iterative manner. Model A [6] captures the contribution due to the backflow of the burned gas from the exhaust valve overlap period, and the trapped gas in the cylinder just before the start of valve-overlap. Model A is established as a function of valve overlap period and the ratio between intake and exhaust manifold.

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Figure 61: Fraction of residual gas modeled using Model A and Model B

B.5. Engine

The mean-value engine model describes the average engine performance in terms of intake, compression, combustion/combustion, and exhaust strokes. The compression and expansion strokes are considered as isentropic with different polytropic exponent. The charge temperature and pressure caused by the instantaneous release of fuel heat are captured in the model of the combustion stroke. The non-adiabatic blow-down process is captured in the model of exhaust stroke.

Since Model B estimates the fraction of residual gas in an iterative manner, Model A was used in the cam phasing block in the proposed model for a close-loop simulation. With the input of the fraction of residual gas, the engine block calculates the amount of trapped gas as well as the temperature at intake valve closing. The exhaust blow-down temperature can then be modeled along with the discrete-event based model. As shown in Figure 62 and Figure 63, the modeled values of the inlet mass air flow and the exhaust blow-down temperature can both achieve desirable modeling accuracy.

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Figure 62: Comparison of inlet air mass flow between the measurements and that is modeled based on the fraction of residual gas using Model A

Figure 63: Modeling of exhaust blow-down temperature

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B.6. Fuel Injector

The principle of the fuel injector is equivalent to a nozzle. Therefore, the mass of injected fuel is a function of the injection time and the injection rate that is actually a function of the fuel rail pressure.

Figure 64: Modeling of injected fuel mass via a fuel injector

C. Engine Simulation

The models presented above are parameterized using the experimental data that are collected from 2.0L four-cylinder turbocharged SIDI VVT engine. Various manifold locations, as indicated in Figure 54, were identified for the installation of pressure and thermocouples transducers. In addition, the engine is equipped with Kistler in-cylinder pressure transducers to enable the tracking of cylinder conditions, such as the in-cylinder pressure at IVC, and the polytropic exponents during the compression and expansion strokes.

The data used for model parameterization were collected when the engine was operated under steady operation conditions. During the dynamometer testing, the engine was operated from 800RPM up to 6500RPM with various control set-points for throttle angle, wastegate duty cycle, fuel injection and spark angle, variable valve timing, and fuel rail pressure. Moreover, transient engine data were also collected when the engine was subjected to step changes in the throttle angle at different speeds and loads. These data have been used for model validation.

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After the parameterized sub-models are implemented in Simulink using S-functions, a simulation model for the engine system, as illustrated in Figure 54, can then be established by linking them following the principles of mass and energy conservation. Figure 65-Figure 67 show the measured and simulated model outputs with gasoline and E85, which validates the transient behavior of the parameterized engine model. The engine was operated around the engine speed of 2000RPM and intake air mass flow rate of 100kg/h. The engine model captures the transient dynamics in the intake air and injected fuel flow rate, the boost and intake manifold pressure, and the turbocharger rotational speeds when a step change in the throttle angle occurs. Compared with the experimental measurements, the presented model is able to estimate these outputs with desirable accuracy.

Figure 65: Engine operation conditions during simulation

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Figure 66: Simulated and measured outputs under operation conditions illustrated in Figure 65 with E0

Figure 67: Simulated and measured outputs under operation conditions illustrated in Figure 65 with E85

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CLOSED-LOOP SPARK CONTROL

Spark advance, one of the key parameters for engine performance, is normally controlled in an open-loop manner using the map values stored in the ECU as a function of engine speed and load. However, these values determined during the calibration process do not take into consideration all the influencing factors such as air humidity, engine wear, and fuel properties [7]. In order to achieve a cost-effective calibration process for optimal spark advance, many approaches have been proposed in literature for spark advance control using cylinder pressure feedback. Most of the approaches employ a combustion phase indicator, derived from cylinder pressures on a cycle-to-cycle basis. As reviewed in [8], the most commonly used combustion phase indicators include the location of peak pressure (LPP), the location of 50% mass fraction burned (CA50), the location of maximum pressure rise, the location of maximum heat release rate, and the value of relative pressure ratio 10 CA° aTDC. In this study, CA50 is selected as the combustion phase indicator for maximum brake torque timing due to its comparatively lower sensitivity to cycle-by-cycle combustion variations and weaker dependence of engine speed and load [8].

Figure 68 illustrates the proposed cylinder-individual combustion control of spark timing using cylinder pressure feedback. The conventional map stores the spark timing set-points as a function of engine speed and loading conditions. The knock control is activated once knocking is detected using the knock sensor and then adjusts the spark timings depending on the knock intensity and the fuel ethanol content. Such feedforward control scheme is first improved by adding cylinder-individual adaptive feedback PI controllers in terms of CA50. Due to the presence of measurement noise and cycle-to-cycle combustion variation, the cylinder pressure measurements are first conditioned and then CA50 values are computed. To improve the transient response of the controller, the stored set-point values of spark timing are then adapted online based on the engine average spark advance adjustment commanded from these cylinder-individual PI controllers.

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Figure 68: Block diagram illustrating the concept for combustion control

Simulations in Figure 69 illustrate the performance of the PI feedback controller with different gains to track the commanded CA50*. The simulations are conducted at the engine speed of 2000RPM and the throttle angle of 30%. Due to the existence of high cycle-to-cycle variations in SI combustion and the resulting CA50 values, a small proportional gain, $k_p = 0.15$, is selected for the PI controller during simulations. The selection of the integral gain, k_i , depends on the desired response time and the level of combustion variations. The PI controller with gains of $k_p = 0.15$ and $k_i = 5$ achieves a rise time of approximately 5 engine cycles at the engine speed of 2000RPM, while one with gains of $k_p = 0.15$ and $k_i = 50$ achieves a shorter rise time of 1-2 engine cycles. However, a larger integral gain in the PI controller, in the mean time, also introduces more combustion variations into the following cycles via feedback. It can also be observed that the IMEP values increase after the selected combustion phasing indicator, CA50, reaches the desired value around 6degCA aTDC. Moreover, cycle-to-cycle variations in IMEP also significantly reduce.

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Figure 69: Spark advance control for optimal combustion phasing using PI feedback controller

The proposed algorithm has been implemented into the Bosch MED software and validated during engine dynamometer testing.

COMBUSTION PHASING OPTIMIZATION

A. Introduction

An Extremum Seeking (ES) control algorithm for both spark timing and variable valve timing (VVT) has been proposed in the previous chapter and tested with an engine mean value model (MVM) to better understand the closed-loop engine system response. Based on the engine MVM characteristic, the parameters in the ES algorithm have been designed and tested. To prove the possibility for the ES algorithm to be implemented in a real vehicle, we better implement the ES algorithm in a dynamometer engine, which allows us to have more controllable test circumstance than a real vehicle.

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This chapter focuses on the implementation and test of the ES control algorithm in a dynamometer engine. First of all, net mean effective pressure (NMEP) and net specific fuel consumption (NSFC) for various spark ignition timings and variable valve timings are introduced based on the dynamometer engine test. We then present a modified ES control algorithm to be implemented in the dynamometer engine. The ES algorithm test is conducted for different engine conditions, i.e. different engine speeds and loads. In addition to the steady-state load condition, transient load inputs with a sinusoidal form are applied to show the ES algorithm capability under a real driving profile.

B. Approach

The turbo charged (TC) spark ignition (SI) direct injection (DI) engine with VVT has been modified primarily to have a higher compression ratio and different variable valve cam profile. The cylinder compression ratio has been changed from 9.25 to 10.6 while intake valve open duration was increased from 230 to 264 crank angle degrees (CAD). Therefore, in this section, we present spark ignition timing and variable valve timing effects on net specific fuel consumption (NSFC). The ES control algorithm proposed initially in the previous report is slightly modified to be implemented into a dynamometer engine.

B.1. Spark Timing and Variable Valve Timing Effects on NSFC

Two different tests were conducted to understand how different spark timing and variable valve timing affect engine performance:

- various spark timings have been applied with other engine inputs fixed and
- intake valve cam timing has been changed with other engine inputs fixed

Spark sweep from -27 to -11 CAD aTDC has been conducted at the relative load (RL) of 50% and engine speed of 2000 RPM as shown in Figure 1. Top two subplots show NMEP and NSFC for different spark timings while two in the bottom present NMEP and NSFC for corresponding combustion phase θ_{CA50} . At the same engine operating condition, intake valve open timing sweep from -8 to -32 CAD aTDC has been also applied as shown in Figure 71. Notice that only intake valve cam timing has been changed with fixed exhaust cam position (exhaust valve close at 20 CAD aTDC) since it has more dominant effect on internal residual gas recirculation (iEGR) due to longer valve open duration. Two plots in the top show NMEP and NSFC for various intake valve openings (IVO) and two in the bottom are combustion phase θ_{CA50} and load changes for same IVO. Error bars in these two figures indicate standard deviation of each variable.

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Figure 70: NMEP and NSFC for various spark timing and corresponding combustion phase (θ_{CA50}) at N = 2000 RPM and Relative Load = 50%

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Figure 71: NMEP, NSFC, combustion phase (θ_{CA50}) and load for various intake valve open timing at N = 2000 RPM and Relative Load = 50%

For different spark timing sweep, NMEP and NSFC curves are similar to parabolic shapes with the maximum fuel efficiency at θ_{CA50} between 5 and 10 CAD aTDC, which, in turn, allows us to apply an on-line optimization scheme such as extremum seeking control. For instance, we can increase approximately 3% fuel efficiency if spark timing θ_{CA50} is advanced by 5 CAD from -17.5 to -22.5 CAD aTDC (i.e. θ_{CA50} from -15 to -10 CAD aTDC). Fuel efficiency improvement rate increases exponentially as the corrected spark timing angle increases. There exists relatively flat area in both NMEP and NSFC curves around maximum fuel efficiency area at spark timing between -27 and -22.5 CAD aTDC (θ_{CA50} between 5 and 10 CAD aTDC).

Change of intake valve open timing affects combustion behavior. Advanced IVO introduces longer valve overlap (the portion of the operating cycle of engines when both the intake and exhaust valves are open) and higher iEGR. In Figure 71, for instance, once IVO is advanced from -8 to -32 CAD aTDC, combustion duration increases due to increased iEGR and, hence, θ_{CA50} is retarded from 9 to 14 CAD aTDC. Spark timing during this IVO sweep was fixed such that associated combustion for different IVO is not maximum brake torque (MBT) condition.

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If we assume that the optimal combustion phase θ_{CA50} is around 6-7 CAD aTDC for each IVO position, NSFC of 241 g/KWh at IVO = -32 CAD aTDC can be improved approximately 7 g/kWh based on the relationship between θ_{CA50} and NSFC in Figure 70 (the second subplot in the bottom). Unlike our expected benefit from higher iEGR on fuel economy, there is no significant engine combustion efficiency improvement through IVO in this engine configuration. There might be a couple of possibilities for the new engine with a higher compression ratio and longer intake valve open duration to have minor combustion efficiency benefit by VVT. Internal residual gas recirculation (iEGR) changes of the new engine through VVT seems to be less sensitive for most of engine operating conditions except very low load close to idle. Therefore, in the following section, we implement and conduct ES algorithm only for optimal spark timing.

In the current algorithm, pumping mean effective pressure (PMEP) is estimated using (5.4), which estimation error might propagate into NMEP and NSFC calculation. It is necessary to have more complicated test for detailed analysis.

C. Results

C.1. Experiment Setup with Rapid Prototyping

The current experimental setup at the engine dynamometer, as illustrated in Figure 72, consists of the latest Bosch MED17.3.2 ECU for engine controls, ETAS ES1000 VME for rapid prototyping, and A&D Redline CAS II System for real-time combustion analysis. The ETAS ES1000 VME rapid prototyping system communicates with the ECU and computer via CAN and Ethernet respectively.

With the cylinder pressure sensing capability integrated, the Bosch MED17.3.2. ECU is able to acquire the cylinder pressure measurements and conduct real-time calculation of various key combustion features such as the crank angle at 50% mass fraction burn and indicated mean effective pressure. In this study, Kistler 6125B cylinder pressure sensors are installed in each cylinder, whose outputs are converted into voltage signals via an external charge amplifier for the ECU and CAS system. Figure 74 illustrates the cylinder pressure measurements over 100degCA bTDC to 80degCA aTDC, recorded in ECU for 1sec, during vehicle-level testing under the operation conditions shown in Figure 73.

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Figure 72: Experimental setup at the engine dynamometer

Figure 73: Measurement collected during the vehicle-level testing

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Figure 74: Cylinder pressure measurements collected for 1sec

The rapid prototyping environment is set up to verify and validate new control and diagnostic functions under target operation conditions on a PC or experimentation system. The proposed virtual prototyping environment includes the following software and hardware components.

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- INTERCRIO: an open environment that enables the integration of AUTOSAR software components as well as MATLAB®/ Simulink® and ASCET models plus C code modules
- ASCET software components: ASCET-MD for modeling and design, ASCET-SE for software engineering, ASCET-RP for rapid prototyping
- MATLAB®/ Simulink® with Real-Time Workshop® and Embedded Coder
- ES1000 VME boards and housing: scalable rapid prototyping module that combines real-time calculation power with all relevant ECU bus interfaces, real-time analog and digital I/O to connect sensors and actuators, as well as customizable rapid prototyping boards

Figure 75: Proposed rapid prototyping scheme with ES1000 VME [9]

C.2. Steady-State Load Condition

The Extremum Seeking control for spark timing has been tested for two different engine speeds, $N = 1500$ and 2000 RPM. One and two load conditions were tested for $N = 1500$ and 2000 RPM, respectively. For each engine operation, three different spark timing offsets (i.e. $+6$, $+9$ and $+12$ CAD) from the ECU map values were applied for various initial spark timing values. Table 20 shows the engine operating conditions for the spark timing ES control test with a steady-state load condition.

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Engine speed N [RPM]	Relative Load [%]	Spark timing offset $\Delta\theta_{\sigma,0}$ [CAD]	Fuel [-]
1500	50	+6, +9, +12	E70
2000	50	+6, +9, +12	E85
	70	+6, +9, +12	E70

Table 20: Engine operating conditions for load steady-state Extremum Seeking (ES) control test

ES control algorithm test has been conducted continuously until five times of convergence detection event occur for each test. When an engine visits ES target condition during test for a certain period of time, ES control block is triggered to start spark timing excitation to search its optimal position. On the other hand, ES control block holds its state variables as soon as an engine departs from ES target region until it revisits the same target region.

Some dynamometer engine test results with steady-state load conditions are shown in Figure 76 through Figure 78. First column of each figure shows relative load, spark timing, crank angle degree of 50% fuel burned θ_{CH50} , and net specific fuel consumption (NSFC) from top to bottom. Second column represent ES control trigger signal, compensated spark timing by ES control block, accumulative moving average of optimal spark timing, and the number of convergence detection flag events.

First, an initial spark timing was retarded by 6 CAD from ECU value at $N = 2000$ RPM and Relative Load = 70% as shown in Figure 76. Note that ES control target engine speed and relative load for this test was set to be [1900 RPM, 2100 RPM] and [65%, 75%], respectively. Convergence detection event occurs five times with corrected spark timings of $\Delta\theta_{\sigma} = -6, -3, -2.25, -8.25$ and -4.5 CAD at $t = 40, 93, 142, 179$ and 210 seconds, respectively. Accumulated moving average of these corrected spark timings is -4.8 CAD aTDC (i.e. 10.8 CAD advanced from initial spark timing $\theta_{\sigma,0}$).

Different values of $\Delta\theta_{\sigma}$ for different convergence events are due to small NSFC variation near optimal spark timing as shown in Figure 70.

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Figure 76: Extremum seeking control test result: $N = 2000$ RPM, Relative load = 70%, $\Delta\theta_{g,0} = 6$ CAD, fuel = E70

In Figure 77 ES control test with the same engine operating condition, $N = 2000$ RPM and Relative Load = 70%, was performed with larger initial spark timing shift of $\Delta\theta_{g,0} = 9$ CAD aTDC. Corrected spark timings for five time optimizations are -4.5, -3.75, -2.25, -6 and -6 CAD at $t = 50, 80, 105, 140, 165$ seconds, respectively. Accumulative moving average of these five $\Delta\theta_g$ is -4.5 CAD, which is very similar to the case with $\Delta\theta_{g,0} = 6$ CAD aTDC. Note that corresponding combustion phase θ_{C450} for this optimal spark timing is 5.5 CAD aTDC, which achieves NSFC with 195 g/kWh.

Finally, spark timing has been retarded by 12 CAD from ECU, $\Delta\theta_{g,0} = 12$ CAD aTDC, at the same engine operating condition as shown in Figure 78. We have conducted 5 times of spark timing optimization and optimal spark timings are -4.5, -3.75, -4.5, -2.25 and -3.75 CAD determined at $t = 38, 75, 100, 135$ and 165 seconds. Accumulative moving average is -3.75 CAD, which locates combustion phase θ_{C450} at 6 CAD aTDC. Equivalent NSFC value is 196 g/kWh.

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ES control algorithm was test for three different initial spark timings, $\Delta\theta_{\sigma,0} = 6, 9$ and 12 CAD, at the engine speed of 2000 RPM and relative load of 70%. Although optimal spark timing has variations between -8.25 (the most advanced optimal spark timing) and -2.25 (the most retarded optimal spark timing), accumulative moving averages after fifth iteration of spark timing optimization demonstrate negligible variation with $-4.8, -4.5$ and -3.75 , respectively. It takes approximately less than 20 seconds to converge to optimal spark position for most of iterations. Notice that similar ES control performances have been observed from other dynamometer engine test data. Parameters in the implemented ES control algorithm were fixed for all different tests.

Figure 77: Extremum seeking control test result: $N = 2000$ RPM, Relative Load = 70%, $\Delta\theta_{\sigma,0} = 9$ CAD, fuel = E70

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Figure 78: Extremum seeking control test result: $N = 2000$ RPM, Relative Load = 70%, $\Delta\theta_{\sigma,0} = 12$ CAD, fuel = E70

C.3. Transient Load Conditions

Strict steady-state driving profiles in a real vehicle are rarely accomplished even in a cruise condition due to unexpected torque disturbances such as curves and altitude changes in the road. To be implemented in a real vehicle, therefore, ES control algorithm needs to be feasible against small variances in both engine load and speed even though ES control test with steady-state engine load and speed conditions were successful as shown in the previous section.

Since the optimal spark timing is more sensitive to engine load than engine speed, ES control test has been conducted under various transient load conditions. At the engine speed of 2000 RPM and mean relative load of 50%, for instance, sinusoidal load inputs with two different amplitudes ΔRL were commanded to produce a load variation. For each sinusoidal amplitude, the frequency f_{RL} has been increased from 0.01 to 0.15 as shown in Table 21.

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Note that initial spark timing was shifted by +12 CAD from the value stored in an ECU map. In this section, we only present the case with the relative load amplitude of $A_{RL} = 3\%$, which has qualitatively comparable results with the other case, $A_{RL} = 5\%$.

Engine speed N [RPM]	Relative Load [%]	Spark timing offset $\Delta\theta_{\sigma,0}$ [CAD]	A_{RL} [%]	f_{RL} [-]	Fuel [-]
2000	50	+12	3	0.01, 0.05, 0.10, 0.15	E85
	50	+12	5	0.01, 0.05, 0.10	E85

Table 21: Engine operating conditions for load transient extremum seeking (ES) control test

For transient load conditions, the ES control algorithm was tested on the dynamometer engine in a similar manner as the steady-state load cases. Test results at the engine speed of 2000 RPM and relative load of 50% are presented in Figure 79 through Figure 82. Contents of these figures are identical with ones in the previous section. The blue dashed line in the first top subplot is demanded relative load signal while the red solid line indicates achieved relative load by the dynamometer engine. We measure ISFC instead of NSFC as shown in the last subplot of first column.

We first applied the sinusoidal load input with the amplitude of 3% and the frequency of 0.01 as shown in Figure 78Figure 79. Five times of spark timing convergence take place at $t = 86, 109, 162, 189$ and 227 seconds with corrected spark timing $\Delta\theta_{\sigma} = 3.75, -0.75, -3, -7.5$ and 0.75 CAD. Its accumulative moving average is -1.35 CAD. Corresponding combustion phase θ_{CA50} is located from 2 to 10 CAD aTDC. Range of associated ISFC mean values is from 185 to 188 g/kWh.

Next, a higher frequency of sinusoidal load input, $f_{RL} = 0.05$, was commanded during ES control test and its result is shown in Figure 80. Initial combustion phase with the spark timing shifted by 12 CAD from ECU value was 22 CAD aTDC, which moves to 5 CAD aTDC in 15 seconds after the first ES iteration. ISFC decreased from 210 to 185 g/kWh with approximately 12% fuel efficiency improvement. Corrected spark timings $\Delta\theta_{\sigma}$ of five ES iterations are $-4.5, -2.25, -1.5, 0.75$ and -5.25 CAD and the accumulative moving average, in turn, is -2.55 CAD. Equivalent combustion phase θ_{CA50} ranges from 3 to 8 CAD aTDC and ISFC varies from 185 to 186 g/kWh.

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Figure 79: Extremum seeking control test result: $N = 2000$ RPM, Relative Load = 50%, $\Delta\theta_{\sigma,0} = 12$ CAD, $A_{RL} = 3\%$, $f_{RL} = 0.01$, fuel = E85

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Figure 80: Extremum seeking control test result : $N = 2000$ RPM, Relative Load = 50%, $\Delta\theta_{\sigma,0} = 12$ CAD, $A_{RL} = 3\%$, $f_{RL} = 0.05$, fuel = E85

In Figure 81 the sinusoidal load frequency f_{RL} was increased to 0.1 for the next ES algorithm feasibility test. Corrected spark timing $\Delta\theta_{\sigma}$ has been achieved with 0.75, -3, -4.5, 4.5 and -0.75 CAD for five ES iterations at $t = 31, 90, 124, 154$ and 194 seconds. The accumulative moving average is -0.28 CAD. Combustion phase θ_{cas0} from 5 to 12 CAD aTDC and ISFC from 185 to 189 g/kWh are associated with these optimal spark timings. Note that, during first and fourth ES iterations, the optimization process had not been completed even though the convergence detection was set to be zero. Due to the slow change in the NSFC, the ES algorithm was stopped before the state was truly converged. Such a problem can be resolved with a better calibrated convergence criterion.

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Figure 81: Extremum seeking control test results: $N = 2000$ RPM, Relative Load = 50%, $\Delta\theta_{\sigma,0} = 12$ CAD, $A_{RL} = 3\%$, $f_{RL} = 0.1$, fuel = E85

We also conducted the ES control algorithm test with a sinusoidal load frequency f_{RL} of 0.15 and its result is shown in Figure 82. The current ES control algorithm was not capable to determine the optimal spark timing for this frequency level since the state in the algorithm does not satisfy the convergence detection criteria. The ES algorithm, however, regulates combustion phase θ_{cas0} in the range from 4 to 8 CAD aTDC, and also achieves ISFC in the range from 185 to 189 g/kWh.

The ES algorithm has been tested on the dynamometer engine and was shown to accomplish the optimal spark timing under various transient load conditions. The accumulative moving average of $\Delta\theta_{\sigma}$ after five ES iterations are -1.35, -2.55 and -0.28 CAD for $f_{RL} = 0.01$, 0.05 and 0.1, respectively. Similar level of ISFC around 185 g/kWh can be achieved. Even though the ES test with $f_{RL} = 0.15$ does not show any satisfied converging event of spark timing, its result demonstrates convincingly that the ES algorithm controls spark timing near optimum to achieve best fuel efficiency.

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Figure 82: Extremum seeking control test result: $N = 2000$ RPM, Relative Load = 50%, $\Delta\theta_{\sigma,0} = 12$ CAD, $A_{RL} = 3\%$, $f_{RL} = 0.15$, fuel = E85

D. Conclusion

This chapter presents an ES control algorithm which has been tested in a dynamometer engine. NMEP and NSFC characteristics have been analyzed based on different spark timing and variable valve timing. Due to small and negligible NSFC changes for IVO changes, the ES control has been tested only for spark timing. Implemented ES control algorithm has been modified with two additional function blocks, saturation and dead-zone, which guarantee stabilized ES controller dynamics. Also, convergence detection criteria were introduced to determine whether the current spark timing is the optimum or not.

Two different types of load input have been applied during ES algorithm test in the dynamometer engine, steady-state and transient load inputs. The transient load input has been realized with sinusoidal load commands with various amplitudes and frequencies. It takes approximately 20 seconds to search the optimal spark timing under the steady-state load condition.

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For the transient load inputs, the current ES algorithm is capable to determine the optimum up to the frequency of $f_{RL} = 0.1$, while the current convergence detection strategy is not capable to determine the optimum with a higher frequency, $f_{RL} = 0.15$. Therefore, it might be necessary to modify the logic to be implemented in a real vehicle if a normal driving profile often changes with a high frequency in a steady-state maneuver.

ETHANOL DETECTION AND FUEL SYSTEM DIAGNOSIS

A. Introduction

It is a common practice that the ethanol content is estimated based on the stoichiometric air-to-fuel ratio (SAFR) identified through the closed-loop air/fuel regulation using the exhaust gas oxygen (EGO) sensor. However, the SAFR-based ethanol estimation is sensitive to fuel injector drifts [1], occurring as a common deterioration over the life-time of the fuel system.

Due to the large ethanol estimation error associated with injector drift, the existing strategy in FFVs compensates for the drift by switching between the ethanol estimation and the fuel drift estimation. Specifically, the ethanol content estimation is generally triggered by the detection of a refueling event. Once the ethanol estimation reaches a steady-state value (does not change appreciably) or after a pre-determined period, the estimation for fuel injector drifts starts. However, such a switching strategy of estimating two unknown variables, ethanol content and fuel injector drifts, using information derived from the common exhaust gas oxygen (EGO) sensor is prone to accumulation of estimation errors [11].

An independent ethanol content estimation of the SAFR-based estimation by using the in-cylinder pressure measurements in direct injection (DI) engines was shown to be insensitive to the fuel injector drifts [12]. Since the use of cylinder pressure sensors for closed-loop combustion control might be more widely spread in the future, its application for ethanol estimation and fuel system on-board diagnosis could demonstrate additional benefits of in-cylinder pressure sensing. This estimation scheme exploits the principle of the charge cooling when fuel is directly injected inside the cylinder when the valves are closed. As the ethanol content increases in the fuel, the observed charge cooling via lower cylinder pressure should also increase due to the fact that ethanol has higher latent heat of vaporization (LHV) than gasoline.

More specifically, the LHV-based detection feature (γ) that quantifies the ethanol content in the fuel [13] is calculated from the in-cylinder pressure measurements of two consecutive cycles with different fuel injection modes:

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- Single injection (Si) mode: All the commanded fuel is injected during the intake stroke, and hence even if the fuel is rich in ethanol, the charge cooling observed by cylinder pressure should be minimal.
- Split injection (Sp) mode: A fraction of the commanded fuel is injected during the intake stroke, and the rest is injected during the compression stroke after intake valve is closed. The fuel that is injected during the compression stroke contributes most of cylinder charge cooling proportional to the ethanol content, and an associated pressure drop could be observed.

By comparing the cylinder pressure from the two modes, differentiation of the charge cooling effect from the other thermo-physical phenomena and their effects on cylinder pressure evolution is possible, and the LHV detection feature is the quantity of this comparison.

The LHV-based detection feature provides an independent indication of the fuel ethanol content in addition to the SAFR-based detection feature under fuel injector drifts, and thus improves the robustness of SAFR-based ethanol detection strategy. However, as presented in [12], the correlation between the LHV-based detection feature and the fuel ethanol content cannot be perfectly modeled and it exhibits various errors at various operating conditions. Therefore, the methodology proposed in this report employing the LHV-based detection feature is used to correct the SAFR ethanol content estimation in operation region where certain confidence levels and accuracies are satisfied [14]. Due to the enhanced ethanol content estimation, a more reliable injector fault detection is realized as estimation of injector drift relies on accurate estimation of fuel ethanol content. We demonstrate the applicability of the methodology via a simulation with slightly exaggerated injector drift happening exactly after a refueling event that changes the ethanol content of the fuel. The simulation uses a realistic cycle-to-cycle variability and modeling error for the LHV-estimation hence it is a good demonstration of the feasibility and potential benefits of the proposed algorithm. The integrated estimation algorithm is also validated with engine dynamometer test.

B. Approach

B.1. LHV-Based Ethanol Detection Feature

This section provides supplementary information on calculation of the LHV-based ethanol detection feature, α , introduced in the previous section.

Figure 83 shows the cylinder pressure evolutions during the compression stroke under Si and Sp injection modes. The third plot in Figure 83 shows the total amount of injected fuel, where it increases at around 40 CADeg under the Sp mode as half of the stoichiometric amount is injected after all the valves are closed.

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By the charge cooling effect, the cylinder pressures under the Sp mode are lower than those under the Si mode for the same engine operating conditions. This pressure drop is quantified by the following residue at an instance:

$$Res(k) = \ln(p_0^{Si}(k)/p_0^{Sp}(k)), \quad (E1)$$

where p_0^{Si} , p_0^{Sp} are corrected cylinder pressures [13] under Si and Sp injection modes, and k denotes a step in time or crank angle. Even though the cylinder pressures under different modes are obtained in different cycles, we regard them as synchronized quantities in time. The second plot in Figure 83 shows the evolution of the residue, Res , which quantifies the pressure gap under different injection modes appeared in the first plot. Finally, the LHV-based detection feature, r , is obtained by averaging Res over the interval of several steps near the end of compression where the instantaneous residue is usually maximized corresponding to the rectangle in the second plot in Figure 83.

$$r = \frac{\sum_{k=k_1}^{k_2} Res(k)}{k_2 - k_1 + 1} \quad (E2)$$

Figure 83: Cylinder pressure and residue evolutions during the compression stroke. (2 consecutive engine cycles regarded as 1 synchronized cycle)

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B.2. SAFR-Based Ethanol Content Estimation

The ethanol content or the volume fraction of ethanol e in a fuel blend is related to the mass fraction e_m as

$$e_m = \frac{e \rho_{eth}}{e \rho_{eth} + (1-e) \rho_{gas}}, \quad (E3)$$

where ρ_{eth} and ρ_{gas} denote the densities of ethanol and gasoline, respectively. Since the densities of ethanol (789 kg/m³) and gasoline (747 kg/m³) are similar, e is almost equal to e_m . The stoichiometric air-to-fuel ratio (SAFR) AFR_s is then expressed in as:

$$AFR_s(e_m) = 9 \times e_m + 14.6 \times (1 - e_m) = 14.6 - 5.6 \times e_m. \quad (E4)$$

Let us denote the fresh mass air flow (MAF) into the cylinder as $W_{a,cyl}$ and the actually injected fuel flow into the cylinder W_f so that the actual air-to-fuel ratio (AFR) is expressed as $AFR = W_{a,cyl}/W_f$. The availability of exhaust gas oxygen (EGO) sensing allows the regulation of λ so that $AFR = AFR_s$ at steady state, where λ denotes the relative air-to-fuel ratio $\lambda \triangleq AFR/AFR_s$. Hence $AFR_s = W_{a,cyl}/W_f$ at steady state and hence measuring air flow and knowing the injected fuel provides the numerical value of AFR_s which can then be used for ethanol estimation. In fact a quick way for estimating ethanol content would be to filter a direct calculation of

$$e_m = AFR_s^{-1}(W_{a,cyl}/W_f) = (14.6 - W_{a,cyl}/W_f)/5.6, \quad (E5)$$

based on the interpretation of the air flow measurements and the fuel flow commands.

Clearly, this ethanol estimation directly depends on mass air flow measurement errors and fuel injector shifts:

$$\frac{\partial e_m}{\partial W_{a,cyl}/W_{a,cyl}} = -(2.6 - e_m), \quad \frac{\partial e_m}{\partial W_f/W_f} = 2.6 - e_m. \quad (E6)$$

The MAF drift has been accounted in [11] and here we develop an ethanol estimation methodology that is tolerant to injector drifts using an additional independent ethanol estimation based on in-cylinder pressure measurements and the charge cooling effects of ethanol [12][13][14][15].

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B.3. LHV-Based Ethanol Content Estimation

Fault-tolerant ethanol estimation needs to account for drifts in the mass air flow sensor and the injectors. As the MAF sensor drifts can be corrected through [11], we only address the estimation of ethanol content and fuel injector drifts. Therefore, we need an extra equation which is independent from the λ regulation to account for both unknown ethanol content e and fuel amount W_f . The LHV-based ethanol detection feature in [13] calculated by the in-cylinder pressure measurements is utilized to provide this additional information in this report. The feature at the stoichiometric condition, λ_s , should be then monotonic at a fixed known $W_{f,sto}$ with respect to the change of the ethanol content e to solve e by inversion. The experimental data do not show the desirable monotonicity, introducing inevitable ethanol calculation errors. The monotone detection feature model is obtained by monotone regression of experimental data, where the ethanol content calculation error should be minimized.

Data have been collected from the experiments at near-stoichiometry on a 2.0L turbo-charged spark ignition direct injection (SIDI) variable valve timing (VVT) engine equipped with Kistler 6125B in-cylinder pressure sensors as in [12]. Table 22 illustrates an example of the nominal conditions for the experiments.

Description	Value
Relative AFR (λ)	1
Split injection ratio (Sp mode)	50/50 (%)
Start of 1 st injection	105 CAdeg bBDC
End of 2 nd injection (Sp mode)	40 CAdeg aBDC
Spark timing	MBT map for E0

Table 22: Nominal experiment conditions

At each nominal set point with different engine speed, load and ethanol content, the ethanol content and detection feature measurement data are averaged. The averaged data are regarded as the steady state data, as we are only interested in the steady state ethanol estimation error.

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Figure 84: Experimental data obtained at the engine speed of 2500 RPM

Figure 84 shows the LHV-based detection feature data at the engine speed of 2500 RPM from the experiments. The detection features in Figure 84(a) obtained at mass air flow of 127.1 kg/hr are raw data corresponding to the calculation (E2) which shows high cycle-to-cycle variability, while those in Figure 84(b) are averaged data which are regarded as steady-state values and used for the regression (E7) in the next subsection.

B.3.1. Affine Regression

At each engine speed and load, we want to build a suitable monotone regression model which maps the ethanol content e to the detection feature r_2 and find the coefficients in the regression model by using the averaged data. The affine regression is utilized in this report. A higher order monotone function can be derived using [16] if more data with more set points are available. The following regression model is considered here:

$$r_2 = r_2^{reg}(N, W_a, e) = a_1(N, W_a) + a_2(N, W_a)e, \quad (E7)$$

where N and W_a denote the engine speed and load, respectively. At each speed and load corresponding to an experimental set point, the coefficients a_1 and a_2 are resolved by the least squares fit using the averaged ethanol content and detection feature data. Since minimization of the ethanol calculation error is desirable, the standard least squares regression is actually applied to the inverse regression model of (E7):

$$e^{reg}(N, W_a, r_2) = b_1(N, W_a) + b_2(N, W_a)r_2, \quad (E8)$$

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where $b_1 \triangleq -a_1/a_2$ and $b_2 \triangleq 1/a_2$. Once the coefficients a_1 and a_2 corresponding to several experimental set points are identified, those coefficient values provide a lookup table so that a_1 and a_2 at arbitrary speed and load can be determined by interpolation. On the other hand, the averaged detection feature at arbitrary operating condition, i.e., arbitrary speed, load and ethanol content can be calculated by multi-dimensional interpolation using the experimental data. Let us denote this interpolated feature by $r_s^{intp}(N, W_e, \phi)$. The steady state ethanol content estimate $\hat{\phi}_{ss,LHV}$ is then calculated by inverting (E7):

$$\hat{\phi}_{ss,LHV}(N, W_e, \phi) = \frac{r_s^{intp}(N, W_e, \phi) - a_1(N, W_e)}{a_2(N, W_e)} \quad (E9)$$

Therefore, the steady state ethanol content estimation error under the LHV-based estimation $\hat{\phi}_{ss,LHV} \triangleq \phi - \hat{\phi}_{ss,LHV}$ is expressed as:

$$\hat{\phi}_{ss,LHV}(N, W_e, \phi) = \phi - \frac{r_s^{intp}(N, W_e, \phi) - a_1(N, W_e)}{a_2(N, W_e)} \quad (E10)$$

Figure 85 shows the steady state ethanol content estimation $\hat{\phi}_{ss,LHV}$ at several loads for two different engine speeds. From the figure it is clear that the steady state LHV estimation performance could provide satisfactory accuracy at certain limited speed and load conditions. At these operating points the LHV estimation could be used judiciously to reduce large ethanol estimation errors due to unaccounted injector drifts.

Figure 85: The steady state ethanol content estimate $\hat{\phi}_{ss,LHV}$ vs. the actual ethanol content

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B.3.2. Availability of LHV-Based Estimation

When the ethanol content estimation using the LHV-based detection feature is available, the fuel amount can also be estimated due to the stoichiometry constraint even under injector drifts. This simultaneous LHV-based estimation of both ethanol content and fuel injector drifts then depends on reliability of LHV-based ethanol estimation. However, there are certain operating conditions that such reliability is relatively degraded as shown in Figure 85 and more reliable estimation depends on longer duration of nearly fixed operating conditions to avoid undesirable transient responses. It is also desirable that the period of activated alternative injection modes (Si and Sp) to obtain the LHV-based detection feature is minimized as the split injection mode sacrifices chance to enhance volumetric efficiency by charge cooling while the valves are closed at the second injection, although the activation/deactivation of the alternative injection mode is omitted in the following sequel. Therefore, the simultaneous estimation of ethanol content and fuel injector drifts is not achieved directly from the LHV-based estimation considering the occasional availability of the LHV-based estimation. We show below a design of the integrated SAFR and LHV-based estimations under fuel injector drifts that takes into account the LHV modeling error identified in this section.

B.4. Integrated Estimation Algorithm

The integration of the LHV and SAFR estimation relies on the λ feedback control loop regulating the fuel amount to match the stoichiometric level of the inducted cylinder air. Hence both estimations and hence their integration depends and affects the λ controller which is discussed below.

B.4.1. Relative Air-to-Fuel Ratio Controller

The relative air-to-fuel ratio (RAFR) controller is composed of feedforward and feedback:

$$W_{inj} = W_{ff} + W_{fb}, \quad (E11)$$

where W_{inj} is the injector command, i.e., issued fuel flow, W_{ff} the feedforward command and W_{fb} the feedback command. The feedforward command is calculated to realize stoichiometric condition based on the estimated cylinder air flow and the fuel type considering the fuel injector shift as well:

$$W_{ff} = \frac{\dot{W}_{a,est}}{\lambda_{sto} \cdot \rho_f} \cdot \frac{1}{1 + f_{inj}}, \quad (E12)$$

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where $\dot{W}_{a,cyl}$ denotes the estimated fresh air flow into the cylinder, $\bar{AFR}_s$ the estimated stoichiometric air-to-fuel ratio (SAFR) depending on the estimated fuel ethanol content $\hat{e}$, i.e., $\bar{AFR}_s = AFR_s(\hat{e})$, f_{ef} the estimated fuel injector shift, where the actual injector shift f_{ef} expresses how much mass fraction of issued fuel is additionally injected into the cylinder:

$$W_f = (1 + f_{ef})W_{inj}, \quad (E13)$$

where W_f denotes the actually injected fuel flow. The feedback controller is a proportional and integral (PI) controller:

$$W_{fb} = W_{fb,P} + W_{fb,I}, \quad (E14)$$

where

$$W_{fb,P} = k_P(\bar{\lambda} - 1), \quad (E15)$$

$$W_{fb,I} = k_I(\bar{\lambda} - 1), \quad (E16)$$

where k_P and k_I are the proportional and integral control gains, respectively and, $\bar{\lambda}$ is the measured λ .

B.4.2. Fuel Ethanol Content and Injector Drift Estimation

The fuel ethanol content e and the fuel injector drift f_{ef} are estimated by integrating the RAFR feedback control signal W_{fb} . However, both estimations cannot be simultaneously performed by commonly using the feedback signal W_{fb} , or the RAFR measurements $\bar{\lambda}$ [11]. Since the ethanol content change only happens during a certain period after refueling event, the estimation periods for ethanol content update and injector shift update could be divided using a switching signal which detects the ethanol content change period. The switching signal is denoted by α_e in this report. Enabling of α_e , i.e., setting $\alpha_e = 1$ happens at the refueling event or a bit later. Disabling of α_e , i.e., setting $\alpha_e = 0$ should happen after the ethanol content in the injected fuel has settled to a new value. An algorithm for enabling/disabling of α_e could be formulated along with the algorithms in this report. However, it is assumed that a robust algorithm for the switching signal α_e is provided *a priori* independent of the algorithms in this report. The ethanol content and fuel injector shift estimators are then expressed as:

$$\dot{\hat{e}} = \alpha_e L_e W_{fb}, \quad (E17)$$

$$\dot{\hat{f}}_{ef} = (1 - \alpha_e) L_{inj} W_{fb}, \quad (E18)$$

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where L_e and L_{inj} are ethanol content and fuel injector shift estimator gains, respectively. Note that $\hat{e}$ and $\hat{f}_{if}$ depends on each other. Therefore, as refueling events happen again and again, errors may accumulate in $\hat{e}$ and $\hat{f}_{if}$ with respect to their actual values by any transient responses at switching, i.e., at the instant of changing α_e , and by the fuel injector drifts [11]. In order to prevent from too much accumulation of errors, the estimation should be reset with a more reliable value. In the following sections, the algorithms for the resetting will be proposed.

B.4.3. LHV-Based Ethanol Estimation

The steady state estimation of ethanol content using the LHV-based detection feature at the stoichiometric condition is insensitive to the fuel injection error as the injection command is not used in detection feature calculation in section 4, while that of the SAFR-based estimation is very sensitive to the fuel injection error as explained in section 3. Hence, if the model (E7) is sufficiently accurate in a region of operating conditions, a reliable estimation of fuel ethanol content is possible in that limited operation region by the LHV-based estimation. Therefore, the LHV-based ethanol content estimation could be used as a resetting value of the SAFR-based ethanol content estimation. The LHV-based ethanol estimator is given by:

$$\hat{e}_{LHV} = L_{LHV}(\bar{r} - \hat{r}_e), \quad (E19)$$

where L_{LHV} is the estimator gain, $\bar{r}$ the measured feature calculated by the in-cylinder pressure measurements, $\hat{r}_e$ the estimated detection feature from the regression model (E7):

$$\hat{r}_e = r_e^{*est}(\bar{N}, \hat{W}_{a,cyl} \hat{e}_{LHV}), \quad (E20)$$

where $\bar{N}$ denotes the measured engine speed and $\hat{W}_{a,cyl}$ the estimated load or the estimated fresh air flow into the cylinder [11]. The high cycle-to-cycle variability in Figure 84(a) is filtered online via the estimator (E20) and delays reaching the steady state in Figure 84(b).

B.4.4. LHV-Based Estimation Reliability Examiner

Since the steady state LHV-based ethanol content estimate $\hat{e}_{ss,LHV}$ is accurate in limited operating conditions, the assessment of reliability on the LHV-based ethanol content estimate $\hat{e}_{LHV}$ should be performed by checking the history of operating conditions. First, let $a_{op}(\hat{e})$ denote a flag indicating whether the operating conditions satisfy a certain level of reliability of the steady state estimation $\hat{e}_{ss,LHV}$ or not. For example, a_{op} can be designed as:

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$$\alpha_{op} = \begin{cases} 1 & \text{if } \bar{N} \in [1900, 2100] \text{ RPM and } \bar{W}_{a, opt} \in [170, 190] \text{ kg/hr,} \\ & \text{or if } \bar{N} \in [2400, 2600] \text{ RPM and } \bar{W}_{a, opt} \in [70, 140] \text{ kg/hr,} \\ 0 & \text{otherwise,} \end{cases} \quad (E21)$$

based on the estimation accuracy established offline in section 6. Second, let us define the following flag at time t , which tracks the recent history of α_{op} :

$$\alpha_{op, hist}(t) = \begin{cases} 1 & \text{if } \alpha_{op}(\tau) = 1, \forall \tau \in [t - t_{LHV}, t], \\ 0 & \text{otherwise,} \end{cases} \quad (E22)$$

where t_{LHV} is a tuning parameter. Third, let $\alpha_{LHV}(t)$ denote a flag indicating whether the LHV-based ethanol content estimate $\hat{\theta}_{LHV}$ at time t is reliable or not. Since the LHV-based ethanol estimate $\hat{\theta}_{LHV}$ exhibits a dynamic response as it is filtered by (E19), the flag α_{LHV} depends on the history of α_{op} and is designed as:

$$\alpha_{LHV}(t) = (1 - \bar{\alpha}_e(t))\alpha_{op, hist}(t), \quad (E23)$$

$$\bar{\alpha}_e(t) = \begin{cases} 1 & \text{if } \exists \tau \text{ such that } t \in [\tau, \tau + t_e] \text{ and } \alpha_e(\tau) = 1, \\ 0 & \text{otherwise} \end{cases}, \quad (E24)$$

where $\alpha_e(t)$ is the flag associated with the switching between the ethanol estimation (E17) and the fuel injector drift estimation (E18) evaluated at time t and t_e is another tuning parameter. Larger value of t_{LHV} sets longer time span to allow longer transient period in $\hat{\theta}_{LHV}$ before enabling α_{LHV} , i.e., $\alpha_{LHV} = 1$. The parameter t_e is tuned so that the flag α_{LHV} is disabled ($\alpha_{LHV} = 0$) until the transient response of the LHV-based ethanol content estimation induced by the actual ethanol content change but not by the operating condition changes is removed. Indeed, the operating condition history examiner (E22) takes account for the delay from the filtering (E19) and the tuning of t_{LHV} in (E22) relies on the estimator gain tuning of L_{LHV} in (E22).

B.4.5. Fault Detection

In normal steady state, the ethanol content estimate $\hat{\theta}$ and the LHV-based ethanol content estimate $\hat{\theta}_{LHV}$ should be sufficiently close. Therefore, the following ethanol content fault detection residual is defined:

$$s_e = \alpha_{LHV} |\hat{\theta} - \hat{\theta}_{LHV}|, \quad (E25)$$

where $|\cdot|$ denotes the absolute value, and α_{LHV} is multiplied in order to take the transient responses into account. If the residual s_e is greater than a certain threshold, a fault should be triggered. Therefore, the fault trigger signal β_{FT} is designed as:

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$$\beta_{FT} = \begin{cases} 1 & \text{if } x_e > \theta_{thid} \\ 0 & \text{otherwise} \end{cases} \quad (E26)$$

where θ_{thid} denotes a pre-determined threshold value. Once the fault trigger is enabled, the ethanol estimate $\hat{e}$ is gradually reset to θ_{LHV} , and then the fault trigger β_{FT} would eventually be set to zero again. This fault can be actually better understood and quantified as the fault flag α_{FT} which explicitly inspects the estimated fuel injector drift:

$$\alpha_{FT} = \begin{cases} 0 & \text{if } \hat{f}_{ef} \in [f_{efn}, f_{efh}] \\ 1 & \text{otherwise} \end{cases} \quad (E27)$$

where $[f_{efn}, f_{efh}]$ is the pre-determined range of allowable fuel injector drift. The fault flag α_{FT} could be displayed as a warning signal on the driver's front panel to inform that the maintenance service is required. Note that α_{FT} could be enabled with the normal function of the estimator (E17) and (E18) without ever triggering the resetting event.

B.4.6. Resetting

Immediately after the fault trigger β_{FT} is enabled, the LHV-based ethanol content estimate θ_{LHV} is registered at a memory buffer and resetting of the ethanol content estimation $\hat{e}$ to θ_{LHV} in the buffer is executed. If the estimated ethanol content is instantly reset to a new value calculated by the LHV-based estimation (E19), the feedforward control signal (E12) is abruptly reset as well, hence causing discontinuous RAFR control signal and undesirable transient response in λ . Therefore, the instant resetting of the ethanol content estimate requires resetting of the RAFR controller state so that the discontinuity in control signal is prevented. There could be still other control logic parts, however, which depend on the ethanol content estimate and could possibly show undesirable responses by the instant resetting. By this reason, we use gradual resetting of $\hat{e}$ to θ_{LHV} in the buffer, which takes some time period starting at the moment of rising edge of β_{FT} , allowing all the control algorithms using the ethanol content estimate unchanged.

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Figure 86: Block diagram of the integrated estimation algorithm

Figure 86 shows the block diagram of the integrated algorithm proposed in this section.

B.5. Fault Detection Residual Threshold

Let $|\hat{e}|_{ss,LHV}^{\max}(N, W_a)$ denote the maximum absolute steady state ethanol content estimation error at the engine speed N and the load W_a under the LHV-based estimation:

$$|\hat{e}|_{ss,LHV}^{\max}(N, W_a) = \max_{e \in [0, 0.02]} |\hat{e}_{ss,LHV}(N, W_a, e)|, \quad (E28)$$

where $\hat{e}_{ss} = \hat{e} - e_{ss}$ denotes the ethanol content error in the steady state, with e_{ss} denoting the estimate of the ethanol content e in the steady state.

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Lemma 1 *If the steady state fault detection residual $s_{e,ss}$ is greater than $2|\theta_{ss,LHV}^{max}|$, the steady state ethanol estimation performance is worse than that of the LHV-based estimation, i.e.,*

$$s_{e,ss} > 2|\theta_{ss,LHV}^{max}| \Rightarrow |\theta_{ss}| > |\theta_{ss,LHV}|. \quad (E29)$$

Proof: By the definition of the fault detection residual s_e in (E25), the following inequality holds in the steady state:

$$s_{e,ss} \leq |\theta_{ss} - \theta_{ss,LHV}|. \quad (E30)$$

Applying the triangle inequality to the right side of (E30),

$$\begin{aligned} |\theta_{ss} - \theta_{ss,LHV}| &= |\theta_{ss} - \theta_{ss,LHV}| \\ &\leq |\theta_{ss}| + |\theta_{ss,LHV}| \\ &= |\theta_{ss}| - |\theta_{ss,LHV}| + 2|\theta_{ss,LHV}| \\ &\leq |\theta_{ss}| - |\theta_{ss,LHV}| + 2|\theta_{ss,LHV}^{max}|. \end{aligned} \quad (E31)$$

From (E30) and (E31),

$$|\theta_{ss}| - |\theta_{ss,LHV}| \geq s_{e,ss} - 2|\theta_{ss,LHV}^{max}|. \quad (E32)$$

Therefore, if $s_{e,ss} > 2|\theta_{ss,LHV}^{max}|$, $|\theta_{ss}| - |\theta_{ss,LHV}| > 0$. ■

According to Lemma 1, the fault detection residual threshold θ_{thid} should be specified within the upper bound $2|\theta_{ss,LHV}^{max}|$ as reaching the condition such that $s_{e,ss} > 2|\theta_{ss,LHV}^{max}|$ always results in worse steady state estimation performance than resetting the ethanol estimate by the LHV-based estimate.

If $\alpha_{LHV} = 1$ in the steady state, i.e., the LHV-based estimation is reliable in the steady state, since $|\theta_{ss} - \theta_{ss,LHV}| \leq \theta_{thid}$, the following inequality holds:

$$|\theta_{ss}| \leq |\theta_{ss,LHV}| + \theta_{thid}. \quad (E33)$$

There are special considerations for choosing the fault detection residual threshold θ_{thid} . On one hand a large value of fault detection residual threshold θ_{thid} may increase the steady state estimation error $|\theta_{ss}|$ by delaying resetting even if the LHV estimation is deemed accurate enough and within reliable steady-state conditions when $\alpha_{LHV} = 1$ from (E17)-(E23).

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On the other hand, a small value of σ_{ethid} may frequently cause additional estimation error by resetting. The additional error occurs due to the occasional large LHV estimation error. Hence, we need to allow resetting only if the two ethanol estimates (the current and the LHV) are alarmingly apart. As an extreme case, if $\sigma_{ethid} = 0$, then we will get $\hat{\theta}_{ss} = \hat{\theta}_{ss,LHV}$ when $\alpha_{LHV} = 1$. In this case, if the estimation error without resetting was actually zero, there is loss of estimation accuracy as $\hat{\theta}_{ss,LHV}$. By the inequality on the steady state estimation error (E33), the following bound regardless of the actual ethanol content e holds if $\alpha_{LHV} = 1$:

$$|\hat{\theta}_{ss}| \leq |\hat{\theta}_{ss,LHV}^{max}(N, W_a)| + \sigma_{ethid}(N, W_a). \quad (E34)$$

Let us impose a universal bound on the inequality (E34), as we want the steady state estimation error to be within certain bound regardless of the operating conditions:

$$|\hat{\theta}_{ss}| \leq |\hat{\theta}_{ss,LHV}^{max}(N, W_a)| + \sigma_{ethid}(N, W_a) \leq |\hat{\theta}_{ss}^{bnd}|, \quad (E35)$$

where the universal bound $|\hat{\theta}_{ss}^{bnd}|$ is a tuning parameter. From (E35) and the upper bound of σ_{ethid} derived from Lemma 1, $\sigma_{ethid} \geq 0$ is constrained with the upper bound:

$$\sigma_{ethid} \leq \min(|\hat{\theta}_{ss}^{bnd}| - |\hat{\theta}_{ss,LHV}^{max}(N, W_a)|, 2|\hat{\theta}_{ss,LHV}^{max}(N, W_a)|). \quad (E36)$$

B.5.1. Probabilistic Reliability Examiner

The ethanol content estimation (E17) provides the SAFR-based estimation if switching to the fuel injector shift estimation is always off, i.e., $\alpha_e \equiv 1$, which solely depends on the λ measurements without resetting by the LHV-based estimation. Let $\hat{\theta}_{SAFR}$ denote this SAFR-based estimation and $\hat{\theta}_{ss,SAFR}$ its steady state estimate. It is desirable that if $|\hat{\theta}_{ss,SAFR} - \hat{\theta}_{ss,LHV}|$ is bounded by the residual threshold σ_{ethid} , a better steady state estimation is obtained by the SAFR-based estimation than by the LHV-based estimation more often. In other words, we want to find σ_{ethid} bounded by (E36) such that the following conditional probability is maximized at each load and speed:

$$P(|\hat{\theta}_{ss,SAFR}| \leq |\hat{\theta}_{ss,LHV}| | (|\hat{\theta}_{ss,SAFR} - \hat{\theta}_{ss,LHV}| \leq \sigma_{ethid})) \\ = \frac{P(|\hat{\theta}_{ss,SAFR}| \leq |\hat{\theta}_{ss,LHV}| | |\hat{\theta}_{ss,SAFR} - \hat{\theta}_{ss,LHV}| \leq \sigma_{ethid})}{P(|\hat{\theta}_{ss,SAFR} - \hat{\theta}_{ss,LHV}| \leq \sigma_{ethid})}. \quad (E37)$$

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It is difficult to perform the double integration over the domain $\{(\phi, \phi_{ss,DAFR}) | \phi \in [0, 0.85], \phi_{ss,DAFR} \in (-\infty, \infty)\}$ which is required to evaluate the conditional probability (E37). Instead, a geometric approximation is applied to evaluate (E37) using discrete data points utilized in the detection feature regression. Although the details of approximation are not shown here, the approximated conditional probability is finally expressed as a function of the residual threshold ϕ_{thld} at each speed and load, denoted as $l_p(N, W_a, \phi_{thld})$. By maximizing l_p at the given speed and load, within the threshold bound (E36), the optimal threshold ϕ_{thld}^* is then obtained. The optimal threshold is actually solely determined by $\phi_{thld}^* = \min(|\tilde{e}_{ss}^{bnd} - |\tilde{e}_{ss,LHV}^{max}(N, W_a)|, 2|\tilde{e}_{ss,LHV}^{max}(N, W_a)|)$ since l_p monotonically increases up to $\phi_{thld}^* = 2|\tilde{e}_{ss,LHV}^{max}$ as shown in Figure 87.

(a) engine speed 2000RPM and MAF 176.2kg/hr (b) engine speed 2500RPM and MAF 127.1 kg/hr

Figure 87: The conditional probability l_p vs. the residual threshold ϕ_{thld}

However, the conditional probability evaluated at ϕ_{thld}^* , l_p^* , is utilized in designing the LHV-based estimation reliability examiner, as larger value of l_p^* is required for more confidence on the LHV-based estimation reliability. Given a specific operating condition, the flag α_{op} should satisfy:

$$\alpha_{op} = \begin{cases} 1 & \text{if } l_p \geq l_{p,thld} \\ 0 & \text{otherwise} \end{cases} \quad (E38)$$

where the probability threshold $l_{p,thld}$ is a tuning parameter defining the confidence level necessary for designing (E21).

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Figure 87 shows the conditional probability l_p versus the residual threshold θ_{chid} at two different operating conditions. With the universal bound $|\theta|_{\text{end}}^{\text{end}} = 0.20$, the vertical dash dot line corresponds to $\theta_{chid} = |\theta|_{\text{end}}^{\text{end}} - |\theta|_{\text{end}}^{\text{max}}$ derived from the universal bound $|\theta|_{\text{end}}^{\text{end}}$ and the vertical dash line corresponds to $\theta_{chid} = 2|\theta|_{\text{end}}^{\text{max}}$ derived from Lemma 1.

Engine speed of 1500 RPM			
MAF (kg/hr)	62.5	79.7	99.6
θ_{chid} (%)	9.5	8.2	9.0
l_p (%)	47.5	43.8	46.3
α_{op}	0	0	0

Engine speed of 2000 RPM				
MAF (kg/hr)	84.1	103.6	152.8	176.3
θ_{chid} (%)	1.3	4.3	6.4	12.3
l_p (%)	6.3	20.5	29.2	78.3
α_{op}	0	0	0	1

Engine speed of 2500 RPM					
MAF (kg/hr)	82.3	106.2	127.1	150.2	167.8
θ_{chid} (%)	11.8	10.7	9.1	8.2	6.8
l_p (%)	90.8	71.8	97.0	51.7	33.2
α_{op}	1	1	1	0	0

Table 23: Residual threshold and LHV-based estimation reliability tuning matrix with $|\theta|_{\text{end}}^{\text{end}} = 0.20$ and $l_{p, chid} = 0.70$

Table 23 summarizes the computed residual threshold θ_{chid} , the conditional probability l_p and the reliability flag α_{op} at different operating conditions corresponding to averaged data points collected from the experiment, with the tuning parameters $|\theta|_{\text{end}}^{\text{end}} = 0.20$ and $l_{p, chid} = 0.70$. The discriminator α_{op} in (E21) could be designed using Table 23. In the particular engine, it is possible to bound the ethanol estimation error by 20% and be 70% confident that the standard SAFR estimation will improve using the developed integrated LHV estimation. The resetting event allows us simultaneously to check the expected injector shift and issue an injector maintenance alert if the drift exceeds some specified limits.

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C. Results

C.1. Simulations

Simulation of the proposed algorithm was performed with the turbo-charged SIDI VVT engine model in [16]. Since the fuel injector drifts and the error accumulation in the estimates are the phenomena of very large time scale, an exaggerated engine operating scenario was simulated with fast ethanol changes and injector drifts within a compressed time window.

Step changes in the throttle input and the engine speed input were simulated as in Figure 88(a) to create a dynamic environment emulating real engine operation. Figure 88(b) shows the simulated load change induced by the throttle and speed inputs. A 5% bias in the mass air flow (MAF) sensor was actually introduced and the compensation algorithm in [11] was utilized for accurate cylinder air flow estimation so that we can concentrate on the ethanol estimation under challenging scenario of injector drifts.

Figure 88: Simulated throttle and speed inputs and induced load

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Figure 89: Simulated ethanol content and fuel injector drift estimations

Figure 90: Simulated ethanol estimation error and fault detection residual

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Figure 89 shows the simulated estimation results. The actual ethanol content was simulated to change from 0 to 0.5 and the actual injector drift was simulated to change from 0 to 0.2 while large amount of drift is introduced during the ethanol estimation period when the drift estimation is off ($\alpha_d = 1$) in order to cause good amount of error accumulation in short time. After the ethanol estimate is reset as the LHV-based estimation is determined to be reliable, the injector drift estimation converges close to the actual value as the ethanol estimate is more accurate than before by resetting. For the gradual resetting of the ethanol content estimate, a quantic polynomial profile in time was utilized with 15 seconds or resetting period.

Figure 90 shows the simulated absolute ethanol content estimation error and the simulated fault detection residual. Resetting occurs as the fault detection residual s_e in (E25) becomes greater than the pre-determined threshold when the estimation difference is sufficiently large with $\alpha_{LHV} = 1$.

Figure 91: Simulated system flags

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Figure 91 shows the system flags. First the refueling flag $\alpha_R = 1$ initiates the SAFR ethanol estimation algorithm (E11)-(E17) which stops at the 160th second. Due to the large injector shift (introduced unrealistically fast here for demonstration purposes) the difference between the estimated ethanol and the LHV-prediction $|\hat{\epsilon} - \hat{\epsilon}_{LHV}|$ grows beyond the ethanol fault detection threshold $\epsilon_{crit} = 20\%$ as early as the 142nd second, but the resetting flag β_{FH} turns on at the 175th second when, in addition to the large difference between the two ethanol estimations $\hat{\epsilon}$ and $\hat{\epsilon}_{LHV}$, the engine operates under conditions with high confidence levels for the accuracy of the LHV estimation $\hat{\epsilon}_{LHV}$. Indeed at the 170th second the engine speed is raised to 2500 rpm and the concurrent load satisfies the reliability examiner flag α_{op} in (E21) as calculated by Table 23. After 5 seconds any possible transient effects can be neglected and the reliability history is long enough to trigger the ethanol detection fault flag β_{FH} .

The injector fault is reported ($\alpha_{FI} = 1$) at around 188 sec as the estimated injector drift $\hat{f}_{ef}$ from (E11)-(E18) using the reset value of the ethanol content estimate grows out of the allowable range, [-0.15, 0.15] in this simulation. Note that the actual injector fault has occurred much earlier as the actual drift is out of the allowable range as early as 155 second shown in Figure 89. Were it not for resetting based on the proposed algorithm, even this late report and possible maintenance alert would not be possible as large ethanol estimation errors would continue to accumulate.

Figure 92: Simulated detection feature and RAFR

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Figure 92 shows the simulated detection feature and λ . Although the actual feature $\tilde{f}$ in (E19) from [13] shows large cycle-to-cycle variability, the estimated feature based on the regression model is relatively clean and the variability is filtered out in the LHV-based ethanol content estimate in Figure 89 as the dynamic or filtered inversion is used as in (E19). Overall, the RAFR, λ , is well-regulated at unity, i.e., the stoichiometric condition even during the switching periods of the ethanol estimators at the 175th second, followed by the switching to the fuel injector estimation and simultaneous throttle transients. We do not attempt to paint a rosy picture in this simulation by highlighting the fact that there is a remaining 7% ethanol error and an associated error in the fuel injector drift estimation at the end of the simulated scenario. As it has been highlighted before, the error arises from the mismatch between the model and the observed LHV based ethanol estimation indicated in (E10) and shown in Figure 85. Note again that the remaining 7% error in the proposed ethanol estimation scheme should be compared to the 28% error observed without the proposed integrated method.

C.2. LHV-based Detection Feature Behaviors

In order to parameterize the regression model for the LHV-based detection feature, the flex-fuel optimized engine was tested at the engine dynamometer over various fuel blends E0... E85 under regularly visited operation conditions with engine speeds of 1500RPM-2500RPM and relative load 40%-80%. Please note that a physical ethanol sensor was installed at the high pressure pump inlet to measure the ethanol content. During the experiments, the LHV-based detection features have demonstrated a monotonic correlation with the ethanol content over most of the operation conditions.

Directly derived from the regression model of the LHV-based detection feature, the ethanol estimation error during the dynamometer testing, illustrated in Figure 93, indicates an average error of around 5% in volumetric ethanol concentration up to E80.

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(a) 1500RPM

(b) 1750RPM

(c) 2000RPM

(d) 2500RPM

Figure 93: Ethanol Estimation Error using LHV-based detection feature via regression model

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C.3. Fuel System Adaptation and Diagnosis Experiments

The proposed integrated algorithm was also experimentally validated by the engine dynamometer test with the algorithm implemented in the engine control unit (ECU). For the experiment, the latest turbo-charged SIDI VVT flex-fuel engine was used, which is the most updated prototype and different from the engine associated with the previous sections. The detection feature model (E7) was re-parameterized and the residual threshold and other parameters were retuned as well, accordingly.

Figure 94: Experimental results of the ethanol content estimation and RAFR control

The actual fuel of E55 was used and the ethanol content estimate states were reset at the beginning of the test to show the performance of convergence of each estimation and to emulate the refueling event as shown in the first plot of Figure 94. The 10% fuel injector error was fictitiously introduced at the very beginning of the test to emulate the fuel injector drifts, which would happen with very long time period in real operation. The test was performed at the fixed speed and load, 1500 RPM and 57 kg/hr mass air flow, and no mass air flow sensor error was considered.

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Figure 95: System flags during the experiment of the integrated algorithm

Figure 94 shows the ethanol content estimation, the measured LHV-based detection feature and the measured λ , and Figure 95 shows the system flags during the test. During the period of the ethanol content estimation ($\alpha_e = 1$), the ethanol estimate deviates from the actual value, 55%, due to the injector error, and it remains fixed before the ethanol estimate reset at around 120th second while the ethanol content estimation is off ($\alpha_e = 0$). The reliability flag α_{LHV} is switched on at around 120th second as $\bar{\alpha}_e$ is switched off at the moment and α_{op} is always on. The flag $\bar{\alpha}_e$ was tuned to switch off with $t_f = 100$ sec as the LHV-based estimation was much slower than the SAFR-based estimation to filter out the cycle-to-cycle variability in the measured detection feature in the second plot of Figure 94. The trigger β_{fit} is triggered at around 120th sec as α_{LHV} is switched on and the difference between the ethanol content estimate and the LHV-based ethanol content estimate is quite large at the moment. The resetting of the ethanol estimate is performed during 120-130 seconds starting with the trigger of β_{fit} with the quantic polynomial profile in time. The measured λ in the third plot of Figure 94 is well-regulated at the stoichiometric condition as the feedforward control is smooth with the smooth transition of the ethanol content estimate.

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D. Conclusion

In this report, a robust methodology of estimating fuel ethanol content in a flex-fuel SIDI engine under fuel injector drifts is proposed. The LHV-based ethanol detection feature, indicating the charge cooling effects of the fuel injected during the compression stroke, is calculated using cylinder pressure measurements under single and split injection modes. The proposed approach enables the use of LHV-based detection feature in an appropriate confined operation regime where strict monotonicity with the ethanol content is observed. A large difference between the ethanol estimation and the LHV-based ethanol estimation is monitored and is used to correct estimation errors, and at the same time, trigger an injector fault detection flag. Criteria for setting the residual threshold and assessing the reliability of the LHV-based estimation to trigger the resetting event are developed as well. Verification of the algorithm and the fault detection performance are demonstrated not only by simulation with a validated engine model under simplified operating scenario with fuel injector drifts but also by experiment with a dynamometer test with the algorithm implemented in the engine control unit (ECU).

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Appendix I: Nomenclature

Signal	Description
N	measured engine speed
I_k	measured knock intensity
$(p_b)^*, p_b$	desired and measured boost pressure
p_{cyl}	measured cylinder pressure
$(p_f)^*, p_f$	desired and measured fuel rail pressure
p_{im}	measured intake manifold pressure
W_a	measured intake mass air flow rate
$(W_{cyl})^*, W_{cyl}$	desired and estimated cylinder charge
$(T)^*$	desired engine torque
$(\theta_{th})^*, \theta_{th}$	desired and commanded throttle positions
$(\phi_{VVT})^*, \phi_{VVT}$	desired and commanded variable valve positions (intake valve open, IVO, and exhaust valve closing, EVC)
$(\phi_s)^*, \phi_s$	desired and commanded spark timing
$(u_{wg})^*, u_{wg}$	desired and commanded wastegate duty cycle
$(\lambda)^*, \lambda$	desired and measured exhaust gas oxygen concentration
$E_{toh}\%$	estimated ethanol concentration
COV_{IMEP}	computed coefficient of variance (COV) of indicated mean effective pressure (IMEP)
$(CA50)^*, CA50$	desired and computed crank angle at 50% mass fraction burn
dp_{max}	computed maximum pressure rise rate

Table 24: Nomenclature

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Appendix II: Element and Function List in Robert Bosch standard Simulation Libraries (RBSL)

Domains	Elements and Functions
EMachines	Direct current machines, electronically commuted machines, asynchronous machines, auxiliary blocks for electric machines
Hydraulic	Containers, pipes, pistons, pumps, resistors, valves
Mechanics	Gears, moving masses, coupling elements, springs and dampers
Pneumatics	Chambers, engine, heat exchangers, orifices, turbo machinery
PowerElectronics	Logic blocks, power stages, auxiliary blocks
Powertrain	Clutches, coupling elements, engine gear interface, masses and shafts, vehicle, wheel
Control	Collection of compensator blocks
Signal	Filters and signal generators
CAE Toolchain	Provide a structured collection of commands that allow to carry out all necessary steps during the parameter exchange between different simulation tools, optimization of mechatronic systems and computer-aided test
Optimization	Provide functions and GUI tools to perform computer-based parameter
Toolbox	optimization of existing simulation models
ParIS	Provide a GUI for parameters variation and evaluation of models from various simulation tools

Table 25: List of modeling domains and functions in Robert Bosch Standard simulation Libraries

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Appendix III: Abbreviations

AFR	Air fuel ratio
ATDC	After top dead center
BMEP	Brake mean effective pressure
BSFC	Brake specific fuel consumption
BTDC	Before top dead center
CA50	Crank angle of 50% mass fraction burnt
CAD	Computer aided design
CAE	Computer aided engineering
CVT	Continuously variable transmission
DEM	Discrete Event Model
DOHC	Double overhead camshaft
ECU	Engine Control Unit
EGO	Exhaust Gas Oxygen sensor
EGR	Exhaust Gas Recirculation
EVC	Exhaust valve closing angle
FDR	Final drive ratio
FFV	Flex-Fuel Vehicle
FTP75	Federal Test Procedure
HDEV	High pressure fuel injector
HDP	High pressure fuel pump
HFM	Hot-film mass air meter
HSP	Homogeneous split injection mode
IMEP	Indicated mean effective pressure
ISFC	Indicated specific fuel consumption
IVO	Intake valve opening
LHV	Latent heat of vaporization
LIVC	Late intake valve closing
LNf	GM Ecotec 2.0L DI Turbo engine
MAF	Mass air flow
MBT	Maximum break torque
MED	Bosch Motronic ECU with Electronic Throttle and Direct Injection
MVM	Mean Value Model
NSFC	Net specific fuel consumption
PMEP	Pumping mean effective pressure
RAFR	Relative air-to-fuel ratio

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RBSL	Robert Bosch standard Simulation Libraries
SAE	Society of Automotive Engineers
SAFR	Stoichiometric air-to-fuel ratio
SULEV	Super Ultra Low Emission Vehicle
TDC	Top dead center
THC	Total hydrocarbons
ULEV	Ultra Low Emission Vehicle
VVT	Variable valve timing
WOT	Wide open throttle

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