

Fuel Economy Benefits and Aftertreatment Requirements of a Naturally Aspirated HCCI-SI Engine System

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ABSTRACT

This vehicle simulation study estimates the fuel economy benefits of an HCCI engine system and assesses the NO_x, HC and CO aftertreatment performance required for compliance with emissions regulations on U.S. and European regulatory driving cycles. The four driving cycles considered are the New European Driving Cycle, EPA City Driving Cycle, EPA Highway Driving Cycle, and US06 Driving Cycle. For each driving cycle, the following influences on vehicle fuel economy were examined: power-to-weight ratio, HCCI combustion mode operating range, driving cycle characteristics, requirements for transitions out of HCCI mode when engine speeds and loads are within the HCCI operating range, fuel consumption and emissions penalties for transitions into and out of HCCI mode, aftertreatment system performance and tailpipe emissions regulations. A key result of this work is that development priorities for attaining maximum fuel economy during urban driving cycles in the U.S. and Europe differ greatly due to the emissions regulations. Compliance with U.S. Tier 2 Bin 2 regulations requires lean NO_x aftertreatment system conversion efficiencies that approach those of a three-way catalyst.

INTRODUCTION

When a well-mixed mixture of fuel and air that is diluted with air and/or previously burned (residual) gases is compressed in an engine to a temperature that it autoignites, the engine operates in homogeneous charge compression ignition (HCCI) mode. To the automobile industry, HCCI combustion is an attractive alternative to spark ignition combustion because of the potential improvements in fuel consumption. However, numerous factors challenge the applicability of HCCI.

Heat release during an HCCI combustion event can be extremely rapid, lasting less than 12 crank angle degrees. A benefit of rapid heat release rates is the possibility of approaching ideal Otto-cycle (constant volume) combustion. However, controlling the state of

the in-cylinder mixture through the appropriate valve timing so that the HCCI combustion process occurs in a manner that does the optimal amount of work on the piston proves to be a complex controls issue. Additionally, the rapid rates of combustion generated pressure rise create noises and engine "knock" that can be alarming to the driver and detrimental to the structure of the engine over time.

To avoid overly rapid rates of pressure rise in the cylinder, the mixture used for HCCI combustion must be sufficiently diluted with air and/or residual gases. When too dilute, however, these mixtures will cause misfire if they are not sufficiently warmed via compression. For these reasons, HCCI combustion can currently comprise only part of the speed/load portion of an engine's operational range. This area for HCCI operation in a naturally aspirated engine is shown schematically in Figure 1. It is possible to operate the engine in HCCI mode at a relative air-fuel ratio of $\lambda=1$, but high dilution with residual gases is still required to limit the heat release rate and provide the thermal energy to ignite the mixture. The maximum load for HCCI in a naturally aspirated engine is lower than that of traditional SI operation. Raising the pressure in the intake manifold by boosting allows for higher loads to be attained in HCCI operation, but noise, engine knock and engine-out NO_x become a major concerns.

Figure 1 shows a 13% reduction in fuel consumption at 4.5 bar and moderate engine speeds and a 22% reduction in fuel consumption at idle loads and idle speeds. It should be noted that HCCI has the greatest impact on fuel consumption at the higher loads. A light truck consumes approximately 5 kg/hr of fuel at 4.5 bar and 3500 rpm and only 1 kg/hr at idle. Saving 13% of 5 kg/hr is more impactful on fuel consumption than saving 22% of 1 kg/hr.

A practical way of realizing 4-stroke HCCI operation is to trap large amounts of residual gases at the end of each cycle so that the autoignition temperature of the fuel is reached near the end of the compression stroke. The

trapping of residual gases therefore fulfills two prerequisites for HCCI combustion: dilution and high mixture temperature. The residual gas fraction typically comprises 40 – 60% of the in-cylinder mass. These levels of residual trapping can be accomplished by closing the exhaust valve early and opening the intake valve late (“negative valve overlap”, or NVO).

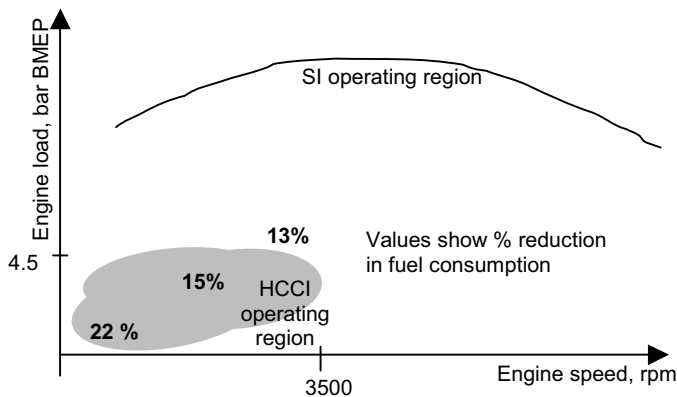


Figure 1: Schematic of the HCCI operating range as a portion of a naturally aspirated SI engine's operating range. Data obtained from Ford Motor Company.

During normal use in a passenger car, the engine is required to transition seamlessly between spark-ignited and HCCI operating modes. HCCI combustion is initiated when the autoignition temperatures are attained in the cylinder, before the piston reaches the top-center position of the compression stroke. To reach the thermal conditions required for HCCI combustion, the engine must be firing in a spark ignition mode before a switch into HCCI mode can occur. Operating the engine in HCCI mode while the engine is cold may not be possible due to this minimum temperature requirement and also because heat losses are too large both during the compression stroke and at top-center. Further, transitions into and out of HCCI combustion are required when the requested load falls outside of the HCCI operating region. In an NVO engine system, twin-independent cam phasing, intake and exhaust cam profile switching, flexible direct injection, and rapid throttle positioning enable the engine to operate at low loads with much reduced throttling and to execute transitions between combustion modes.

The autoignition timing of the HCCI combustion process is sensitive to the pressure and temperature of the in-cylinder mixture during the entire compression process. When the engine operates in SI mode, the start of combustion is determined by the spark timing. A successful switch into HCCI mode requires that the control parameters be estimated and set with precision. The change in fuel conversion efficiency, which is higher during HCCI combustion than during SI combustion, must also be estimated to prevent a discontinuity in torque during the transition. If the control parameters are not set correctly, the mode switch into, or out of, HCCI operation can fail or combustion can proceed with high cycle-to-cycle variation and/or knocking.

Another important challenge facing HCCI applicability in the context of passenger vehicles concerns the aftertreatment of lean exhaust emissions. More stringent emissions constraints place high demands on currently available technology for lean aftertreatment, and quantifying the extent of this challenge is a primary focus of this study.

CONTEXT AND SCOPE

A step toward bringing HCCI closer to applicability in the automotive industry is presenting an argument for net benefit in vehicle fuel economy under the tightening constraints of air-pollutant emissions regulations. Challenges facing an HCCI powertrain in a vehicle context include:

1. narrow operating range,
2. limited applicability in vehicles with moderate to low power-to-weight ratios,
3. robustness,
4. noise, vibration and harshness,
5. difficulty in controlling the HCCI combustion process,
6. appropriately controlling transitions between HCCI and spark-ignition (SI) modes of operation,
7. non-negligible fuel economy and NO_x penalties as a result of these transitions,
8. added hardware, added production and warranty costs,
9. increased engine-out hydrocarbons, increased tailpipe NO_x, and consequent increased difficulty in meeting near-zero emissions constraints.

These challenges must be addressed and shown to be outweighed by the benefit of reduced fuel consumption in order for the development of an HCCI-SI engine system to be worthwhile for vehicle application. The work reported here is a modeling exercise that examines how different emissions requirements and performance of the aftertreatment system affect the fuel economy benefits. It does not aim to explore how robustness, noise, vibration, and harshness could be removed from the list of factors that have potential in limiting HCCI implementation. Also, this work does not examine how the combustion process and transitions between combustion processes could be better controlled. Fuel and emissions penalties associated with transitions and stringent emissions requirements are, however, important factors that could directly prevent the HCCI process from realizing its full potential in reducing vehicle fuel consumption. These factors are explored in detail.

A Ford Ranger truck, with a 2.3 liter inline 4-cylinder engine, in the 3375 pound weight class, is used as the test vehicle. This work quantifies how changes in vehicle power-to-weight ratio and in HCCI implementation strategy could further improve vehicle fuel economy. Some of these changes are also shown to exacerbate the problem of meeting stringent emissions requirements.

Hydrocarbon emissions from HCCI combustion are higher than those from spark ignition. Hydrocarbons have been quantified but not treated as the major point of focus in the discussion of engine-out and tailpipe emissions. Meeting near-zero emissions standards for hydrocarbons is difficult with an SI engine coupled with a three-way catalyst for aftertreatment, and while HCCI implementation is not expected to make the challenge easier, it is also not expected to make the challenge notably more difficult. Complying with emissions regulations for carbon monoxide has not been difficult once the hydrocarbon and NOx targets have been reached with three-way catalyst systems, with and without HCCI implementation, provided rich mixtures are not used.

The impact of HCCI implementation on engine-out and tailpipe NOx, however, is central to the discussion of emissions and emissions compliance. While HCCI implementation markedly reduces engine-out NOx, the reduction is not so great as to eliminate the need for a lean NOx aftertreatment system. The engine exhaust during HCCI mode is lean of stoichiometric, creating an environment in the three-way catalyst that is not favorable for NOx conversion. One conclusion of this work is that additional lean NOx aftertreatment is required if the more demanding U.S. emissions regulations are to be met.

The emissions regulations considered in an analysis of the city driving cycle in the U.S. are those listed in Tier 2, Bins 5, 4, 3, and 2. For hydrocarbons and NOx, Tier 2, Bin 2 is equivalent to PZEV levels without the constraint on evaporative emissions. Euro 6 levels are considered in the analysis of the New European Driving Cycle. In the following table, the standard European units of g/km are converted to g/mi for the purpose of comparison.

Lowering engine exhaust emissions and achieving high catalyst conversion efficiencies are of crucial importance in meeting the more stringent of these regulations. Both engine-out and tailpipe emissions are estimated. Mathematical regressions based upon an extensive database of engine emissions are used to estimate engine exhaust gas temperature and engine-out emission indices for carbon monoxide, hydrocarbons, and nitrogen oxides. A simple catalyst model is then used to estimate the combinations of three-way catalyst light-off times and steady-state conversion efficiencies required to meet each of the various emissions constraints. The two parameters of this simple TWC model are the catalyst light-off time and the steady-state conversion efficiency. The catalyst light-off time, t_{50} , is defined as the time required to attain *half* of the steady-state conversion efficiency.

		mg/mi HC	mg/mi NOx	mg/mi CO	
U.S., Tier 2	Bin 5	std	90	70	4200
		target	67.5	52.5	3150
	Bin 4	std	70	40	2100
		target	52.5	30	1575
	Bin 3	std	55	30	2100
		target	41.25	22.5	1575
	Bin 2	std	10	20	2100
		target	7.5	15	1575
	PZEV (tp)	std	10	20	1000
		target	7.5	15	750

EU	Euro 5,6	std	108.8	96	1600
		target	81.6	72	1200

Table 1: Emissions regulations in the U.S. and in Europe. The development target for each regulation is 75% of the standard. The regulations for hydrocarbons are for non-methane organic gases (NMOG) in the U.S. and for non-methane hydrocarbons in Europe.

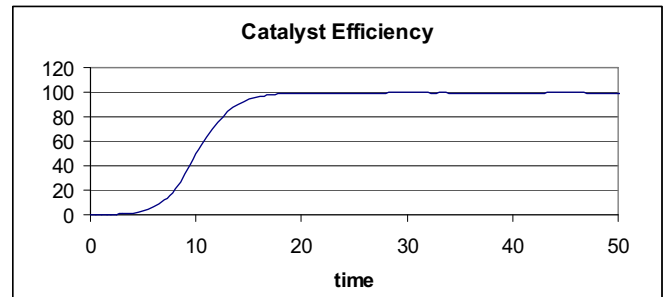


Figure 2: Illustration of the simple catalyst model providing conversion efficiency as a function of light-off time and steady-state conversion efficiency. In this case, t_{50} is approximately 10 seconds and η_{ss} is close to 100%.

MODELING APPROACH

The vehicle with its standard engine is modeled with Ford-internal programs. Several driving cycles are chosen and half second-by-half second data are obtained for each using these engine and vehicle simulation programs. The data include fuel flow rates, engine speeds and engine torques. These data are used to estimate fuel consumption over the driving cycle, engine-out emissions using pre-existing regressions, and tailpipe emissions using the simple catalyst model. Aftertreatment requirements are then discussed in terms of the combinations of catalyst light-off times and steady-state efficiencies needed to meet various emissions standards. The process is repeated with increasingly more complex engines including an HCCI capable engine system. Engines modeled include:

1. 2.3L port fuel-injected,
2. 2.3L direct-injected,
3. 2.3L direct-injected with variable valve timing, and
4. 2.3L HCCI-SI engine system.

For the sake of exploring options, the first three engines were modeled with and without a fixed EGR schedule (provided by Ford). The HCCI-SI engine system was modeled only with EGR. The vehicle was first modeled at a test weight of 3375 pounds and then again with at 2375 pounds to explore how power-to-weight ratio effects fuel economy and HCCI applicability. Data collected from the models of the first three engines, with and without an EGR schedule, in each of the heavier and lighter vehicles can be found in the referenced doctoral thesis [1]. This paper focuses on the fuel consumption and NOx emission over a driving cycle associated with each of a number of different HCCI implementation strategies. The baseline comes from the analysis of the heavier vehicle with Engine #3, a 2.3L direct-injected engine with variable valve timing and EGR. The four driving cycles considered are:

1. U.S. EPA City Driving Cycle
2. New European Criving Cycle (NEDC)
3. U.S. Highway Driving Cycle
4. US06 Driving Cycle

Engine #3, DI + TI-VCT		heavy = 3375 lb ETW							
		city		highway		NEDC		US06	
		no EGR	EGR	no EGR	EGR	no EGR	EGR	no EGR	EGR
FE	mpg	25.6	25.7	33.8	34.1	24.8	24.8	20.6	20.8
	g/mi	110.3	109.8	83.5	82.9	114.1	113.9	136.8	135.8
HC	g/mi	1.7	1.8	1.0	1.1	1.7	1.8	1.5	1.6
	% fuel	1.5	1.6	1.2	1.4	1.5	1.6	1.1	1.2
NOx	g/mi	3.9	1.9	3.1	1.6	4.3	2.4	6.3	5.6
	% fuel	3.5	1.7	3.8	2.0	3.8	2.1	4.6	4.1
CO	g/mi	9.2	9.2	7.0	6.9	9.6	9.6	11.4	11.3
	% fuel	8.4	8.4	8.4	8.4	8.4	8.4	8.4	8.4

Table 2: Summary of key results for Engine #3 in the heavier (standard) vehicle over the four driving cycles.

Engine #3, DI + TI-VCT		light = 2375 lb ETW							
		city		highway		NEDC		US06	
		<i>no EGR</i>	<i>EGR</i>	<i>no EGR</i>	<i>EGR</i>	<i>no EGR</i>	<i>EGR</i>	<i>no EGR</i>	<i>EGR</i>
FE	<i>mpg</i>	28.1	28.3	34.4	34.8	27.5	27.7	23.5	23.6
	<i>g/mi</i>	100.6	99.8	82.1	81.1	102.8	102.0	120.4	119.7
HC	<i>g/mi</i>	1.6	1.7	1.0	1.1	1.6	1.7	1.4	1.5
	<i>% fuel</i>	1.6	1.7	1.2	1.4	1.5	1.7	1.1	1.2
NOx	<i>g/mi</i>	3.3	1.6	3.7	1.5	3.6	1.9	6.2	4.1
	<i>% fuel</i>	3.2	1.6	4.5	1.8	3.5	1.9	5.2	3.5
CO	<i>g/mi</i>	8.4	8.4	6.9	6.8	8.6	8.6	10.1	10.0
	<i>% fuel</i>	8.4	8.4	8.4	8.4	8.4	8.4	8.4	8.4

Table 3: Summary of key results for Engine #3 in the lighter vehicle over the four driving cycles.

Fuel economy and emissions estimates were calculated for each engine/vehicle/driving cycle combination. These estimates for the heavier and lighter vehicles with the 2.3L DI engine with variable-valve timing are summarized in Tables 2 and 3.

SI ENGINE EMISSIONS COMPLIANCE

The simple catalyst model is applied to the half second-by-half second engine emission data obtained from Ford internal programs, and tailpipe emissions are thereby estimated. Tables 4 and 5 show cumulative tailpipe emissions of the heavy (standard) vehicle executing the EPA City and New European Driving Cycles with Engine #3 with EGR.

3375 lb ETW

EPA CITY

Light-off time, t₅₀

		5	10	15	20
99.7	ave η	99.5	99.3	99.1	98.9
	HC, mg/mi	11.9	18.4	24.6	30.9
	NOx, mg/mi	6.1	6.4	6.8	8.8
	CO, mg/mi	49.2	60.0	71.6	86.8
99.8	ave η	99.6	99.4	99.2	99.0
	HC, mg/mi	10.1	16.6	22.8	29.1
	NOx, mg/mi	4.2	4.4	4.9	6.9
	CO, mg/mi	40.0	50.9	62.4	77.7
99.9	ave η	99.7	99.5	99.3	99.1
	HC, mg/mi	8.3	14.8	21.1	27.3
	NOx, mg/mi	2.3	2.5	3.0	5.0
	CO, mg/mi	30.9	41.7	53.3	68.6
100	ave η	99.8	99.6	99.4	99.2
	HC, mg/mi	6.5	13.0	19.3	25.6
	NOx, mg/mi	0.4	0.6	1.1	3.0
	CO, mg/mi	21.7	32.6	44.2	59.5

steady-state catalyst efficiency, %

Table 4: Cycle-averaged TWC conversion efficiencies and weighted cumulative tailpipe emissions for the heavier vehicle executing the city driving cycle with Engine #3 with EGR. The cycle-averaged efficiency is *not* weighted according to the three phases of this driving cycle.

Significantly longer light-off times are allowed by the Euro 6 standard for the NEDC driving cycle than by the more stringent North American standards for the city driving cycle. Table 1 shows that the Euro 6 standard requires hydrocarbon levels below 109 mg/mi, whereas Tier 2 Bin 5 requires hydrocarbon levels below 90 mg/mi. The much more stringent Tier 2 Bin 2 standard requires hydrocarbon levels below 10 mg/mi. Similarly, the Euro 6 standard requires NOx levels below 96 mg/mi whereas the Tier 2 Bins 5 and 2 standards require NOx levels below 70 and 20, respectively.

Light-off time, t_{50}

		17	18	19	20	
steady-state catalyst efficiency, %	99.7	ave	98.2	98.1	98.0	98.0
		HC, mg/mi	54.5	56.5	58.4	60.2
		NOx, mg/mi	14.7	15.2	15.7	16.2
		CO, mg/mi	152.1	156.7	161.1	165.3
	99.8	ave	98.3	98.2	98.1	98.1
		HC, mg/mi	52.8	54.7	56.6	58.4
		NOx, mg/mi	12.2	12.8	13.3	13.8
		CO, mg/mi	142.6	147.2	151.6	155.8
	99.9	ave	98.4	98.3	98.2	98.2
		HC, mg/mi	51.0	52.9	54.8	56.6
		NOx, mg/mi	9.8	10.4	10.9	11.4
		CO, mg/mi	133.2	137.8	142.2	146.4
	100	ave	98.5	98.4	98.3	98.3
HC, mg/mi		49.2	51.1	53.0	54.8	
NOx, mg/mi		7.4	8.0	8.5	9.0	
CO, mg/mi		123.8	128.4	132.8	137.0	

Table 5: Cycle-averaged TWC conversion efficiencies and cumulative tailpipe emissions for the heavy (standard) vehicle executing the New European Driving Cycle with Engine #3 with EGR.

HCCI ENGINE

Estimates for engine-out emissions, fuel consumption and penalties associated with transitions into and out of HCCI mode are based on data collected using HCCI capable Ford engines. Single cylinder and multi-cylinder, naturally aspirated, four-stroke engines were operated in HCCI mode with commercial gasoline in spark ignition and NVO HCCI combustion modes. The valve-timing was adjusted in situ allowing for optimization of HCCI operation at different engine speed and engine load conditions. HCCI engine operation requires that the fuel-air mixture be very lean or highly diluted with residual gases from the combustion event of a previous cylinder cycle. The dilution provides the thermal energy required for autoignition and slows chemical kinetics during combustion and prevents heat from being released too rapidly and causing excessive engine knock.

Several tests were performed with different combinations of speed and load conditions while varying the valve timing and the inlet manifold air pressure. Starting with conventional SI combustion, the negative valve overlap was varied until HCCI combustion was optimized. The maximum engine loads for which data have been collected is 4.5 bar BMEP at low speeds, and the maximum engine speed was 3500 rpm. Figure 3 shows the Ford engine [8], and Table 6 lists the engine specifications.

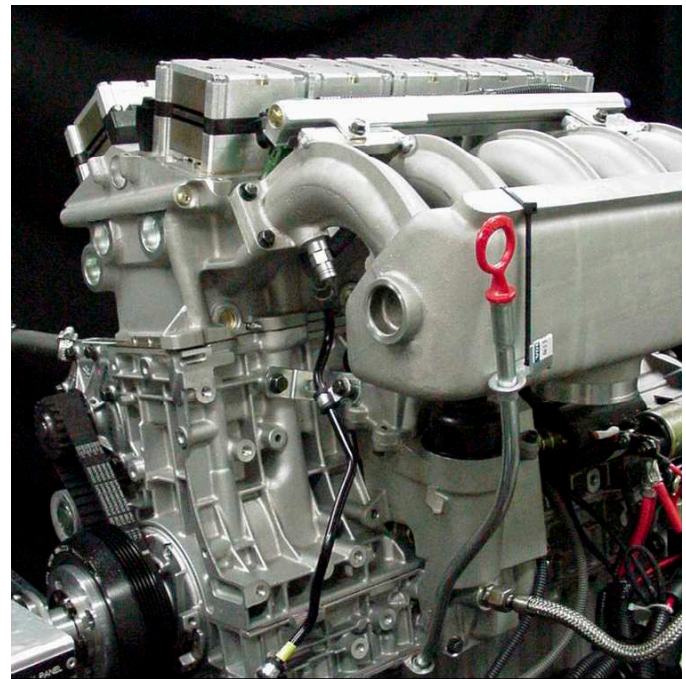


Figure 3: The Ford EVC engine used for HCCI testing.

	Single Cylinder	Multi-cylinder
Bore	83 mm	
Stroke	90 mm	
Compression ratio	13.2	11
Displaced volume	487 cc	2435 cc
Fuel (CEC legislative fuel)	RF-08-A-85, 97 RON	
Inlet temperature	25 deg C	

Table 6: Ford engine specifications.

HCCI PERFORMANCE APPROXIMATIONS - The HCCI engine operating characteristics are in the form of regressions that come from data collected at Ford. These regressions provided the following information:

1. Air-fuel ratio as a function of brake engine torque (converted to brake mean effective pressure in bar)
2. Percent fuel consumption reduction as a function of brake engine torque (converted to BMEP in bar)
3. COEI as a function of engine speed and brake engine torque (converted to BMEP in bar)
4. HCEI as a function of engine speed and brake engine torque (converted to BMEP in bar)
5. NOxEI as a function of BMEP
6. Exhaust gas temperature at the exit of the exhaust manifold as a function of engine speed and brake engine torque (converted to BMEP in bar)

Details about spark timing, camshaft profiles, exhaust valve lift, and fueling strategies are included in the regressions, which are "end-result" expressions for the amount of fuel going into the engine, the air-fuel ratio (as $\lambda = \text{AFR}/14.6$), the emissions from the engine, and the temperature of the exhaust gas stream. Details of the engine and how successful operation was achieved can be found in the cited literature [2, 8, 3, 2, 2].