G.Ü. Fen Bilimleri Dergisi,
16(4):811-824, 2003
ISSN 1303-9709
G.U. Journal of Science
16(4):811-824, 2003

A METHODOLOGY FOR A HCCI-GASOLINE ENGINE DESIGN USING THE PARAMETERS OF DOUBLE INJECTION AND INTAKE AIR TEMPERATURE

Mustafa ÇANAKÇI*, Cenk SAYIN, Necati ÖZSEZEN
Kocaeli University, Technical Education Faculty, Department of Mechanical Education, 41100, Izmit/Turkey, canakci@kou.edu.tr, cenk@kou.edu.tr, nozsezen@kou.edu.tr

ABSTRACT

Homogeneous charge compression ignition (HCCI) is receiving attention as a new low emission engine concept. Little is known about the optimal operating conditions for this engine operation mode. Combustion at homogeneous, low equivalence ratio conditions results in modest temperature combustion products, containing very low concentrations of NO_x (nitrogen oxides) and PM (particulate matter) as well as providing high thermal efficiency. However, this combustion mode can produce higher HC (hydrocarbon) and CO (carbon monoxide) emissions than those of conventional engines. An electronically controlled Caterpillar single-cylinder oil test engine (SCOTE), originally designed for heavy-duty diesel applications, was converted to a HCCI direct-injection gasoline engine. The engine features an electronically controlled low pressure common rail injector with a 60°-spray angle that is capable of multiple injections. The use of double injection was explored for emission control, and the engine was optimized using fully-automated experiments and a micro-genetic algorithm (GA) optimization code. The variables changed during the optimization had included the intake air temperature, start of injection timing, and split injection parameters (percent mass of the fuel in each injection, dwell between the pulses). The engine performance and emissions were determined at 700 rev/min with a constant fuel flow rate at 10 MPa fuel injection pressure. The results showed that significant emissions reductions are possible with the use of optimal injection strategies.

Key Words: HCCI engine, engine performance, exhaust emissions

KADEMELİ ENJEKSİYON VE EMME HAVA SICAKLIĞI DEĞİŞKENLERİNİN KULLANIMI İLE BENZİNLİ BİR HCCI MOTORUNUN TASARIMI İÇİN BİR YÖNTEM

ÖZET

Homojen doldurmalı sıkıştırmayla ateşleme (HCCI), düşük emisyonlu motor tasarımı olarak dikkat çekmektedir. Bu motor tipinin optimum çalışma şartları hakkında çok az bilgi bulunmaktadır. Düşük yakıt eşdeğerlik katsayısı koşullarındaki homojen karışımın yanması, çok yüksek olmayan sıcaklıklarda olmakta ve geleneksel motorlara göre düşük NOx (nitrojen oksit) ve PM (partikül) konsantrasyonları içeren yanma ürünleri ortaya çıkmaktadır. Bununla birlikte; yüksek ısıl verim elde edilen bu yanma tipinde daha fazla HC (hidrokarbon) ve CO (karbon monoksit) emisyonları oluşabilmektedir. Bu çalışmada, elektronik kontrollü, tek silindirli, direk püskürtmeli bir ağır yük dizel deney motoru, HCCI motoruna dönüştürülmüştür. Bu motor üzerine elektronik kontrollü, düşük basınçlı, 60° püskürtme açılı ve kademeli püskürtme özelliğine sahip, "common rail" tipi bir enjektör monte edildikten sonra emisyon kontrolü için kademeli püskürtme kullanımı uygulanmış ve mikro-genetik algoritma optimizasyon kodu ile bilgisayar kontrollü deneyler yapılarak motorun çalışması optimize edilmiştir. Optimizasyon kapsamındaki değişkenler; emme havası giriş sıcaklığı, püskürtme başlangıç zamanı, ve her bir püskürtmedeki yakıt yüzdesinde değişim ve iki püskürtme aralığında değişim gibi farklı püskürtme parametreleridir. Motor performansı ve emisyonlar, 10 MPa yakıt basıncında ve 700 d/dak motor hızında ölçülmüştür. Sonuçlar optimum püskürtme stratejisinin kullanımı ile önemli emisyon azalmalarının olabileceğini göstermektedir.

Anahtar Kelimeler: HCCI motor, motor performansı, egzoz emisyonları

1. INTRODUCTION

The Environmental Protection Agency (EPA) and California's Air Resources Board (CARB) have proposed regulations for reducing the exhaust emissions from stationary and mobile IC engines. The exhaust emissions standards in 2007 for heavy-duty engines will be 0.25 g/kW-hr NO_x+HC and 0.01 g/kW-hr PM (1). It will not be easy to reach these low emissions levels for heavy-duty engines. Therefore, engine exhaust emissions reduction studies are almost at the heart of all research today on IC engines.

In spark ignition (SI) engines, a spark plug ignites the air fuel mixture in the cylinder, creating high local temperatures and resulting in high NO_x emissions. In compression ignition (CI) engines, after taking air into the cylinder and compressing it, the start of combustion is controlled by the injection of fuel into the hot and high-pressure air. This system creates a combustion pattern that produces a high-temperature combustion zone and a fuel-rich zone, which yield NO_x and particulate emissions, respectively (2).

In direct injection spark ignition (DISI) gasoline engines, the fuel is injected directly into the cylinder and stratified near the spark plug. This allows for a flame to propagate, and fuel can be consumed with excess oxygen around the flammable fuel charge. The high temperature flame produces NO_x emissions similar to th ose of a homogeneously charged engine. Due to the fuel stratification, locally fuel rich regions can produce particulate matter (PM) emissions (3). Therefore, the current limitations on NO_x and particulate emissions of DISI engines have stimulated further investigation on alternative combustion systems like Homogeneous Charge Compression Ignition (HCCI), which produces little NO_x emissions and minimal particulate emissions by operating overall lean (4). However, only a few studies on DI gasoline engines have been performed using the HCCI combustion mode. The study of Marriott and Reitz (5) showed that significant reductions in NO_x were possible with the use of double injection in HCCI combustion. In that study, the first injection took place early in the cycle (e.g., during the intake stroke) to allow time for mixing and the formation of a lean homogeneous mixture. This mixture was such as to be too lean for self-ignition during the compression stroke. The second injection occurred during the compression stroke and provided a locally rich mixture to serve as the ignition source. To further explore this concept, in this study, a HCCI direct-injection gasoline engine was studied to investigate and optimize the effects of double injections and intake air temperature on the engine performance and emissions using fully-automated experimental micro-genetic algorithms (GA).

2. REVIEW OF HCCI

The HCCI mode is an alternative combustion concept for the internal combustion engine. Like in SI engines, the fuel is homogeneously premixed with air if early injection is used but with a high proportion of air, and when the piston reaches top dead center (TDC), this lean mixture autoignites like in a diesel engine. In SI and diesel engines the start of combustion timing can be controlled with spark timing and fuel injection timing, respectively, but the HCCI engine does not have a direct method to control the start-of-combustion timing. The following parameters affect the combustion phase of the HCCI engine: fuel characteristics, intake air temperature, air-fuel ratio, fuel injection timing, multiple pulse fuel injection, engine speed, and boost and back pressure. In addition, the engine performance is influenced by the injector spray geometry, internal exhaust gas recycling (IEGR), external exhaust gas recycling (EEGR), variable valve timing, swirl ratio and supercharging, compression ratio, and piston-cylinder geometry.

One of the prerequisites in HCCI combustion is to prepare a suitably homogeneous mixture. The auto-ignition characteristics of the fuel are one of the dominant parameters effecting the HCCI combustion phase. The vaporization temperature of the fuel also plays a significant role to form a well-mixed charge in the cylinder. HCCI researchers have investigated different fuels such as isooctane, ethanol (6-10), natural gas (6, 7, 8, 11), hydrogen (12), gasoline (3, 7, 13-17), diesel fuel (3, 7, 14, 18), n-heptane (7, 10, 18), methanol (10) and propane (19). These studies have shown that fuels with higher volatility like hydrogen and ethanol mix better with air to form homogeneous mixtures. Christensen et al. (6, 7) compared gasoline, diesel, isooctane and n-heptane under engine conditions with a compression ratio of 11:1 and an inlet air temperature below 90°C. They concluded that, with diesel fuel, the combustion became very poor due to poor mixture and vaporization.

After selecting a particular fuel, many researchers have investigated the effect of different engine parameters to control combustion in HCCI engines. One of the main parameters to control HCCI combustion is the intake air temperature, which affects the time-temperature history of the mixture. The intake air temperature is commonly varied with an electrical heating element (3, 6, 8, 14, 15). However, in practical applications, exhaust waste heat would be used to control the intake air temperature using a heat exchanger. The results show that higher intake temperatures advance the start of combustion but cause a reduction in the volumetric efficiency.

The effect of compression ratio on the ignition timing has also been discussed (7, 9, 12). Christensen et al. (7) varied the compression ratio from 10:1 to 28:1 with a constant air/fuel equivalence ratio of 3.0 using a port injection system. They concluded that gasoline required a compression ratio of 22.5:1 for satisfactory operation without the use of inlet air preheating. The results showed that no smoke was generated and that NO_x emissions were very low with increased compression ratio. Since the combustion efficiency decreased with increased compression ratio, the indicated efficiency did not improve with increased compression ratio.

The fuel/air equivalence ratio of the mixture has a significant effect on the timing of auto-ignition since it alters the maximum charge temperature in the cylinder. HCCI combustion is practical only at low fuel/air equivalence ratios since NO_x emissions increase with high heat release rates and high local temperatures. When the fuel/air equivalence ratio is increased at high loads, large EGR rates have been used to decrease the reaction kinetics and to reduce engine knocking (6, 8, 11, 14, 16). The mixture reactivity in the cylinder is altered using EGR since it contains species with high specific heats (CO_2 and H_2O) that lower the compression temperature. Another method to control the combustion is by using variable exhaust valve timing. Law et al. (13) observed the effect of internal EGR (IEGR) ratio between 36% and 59% on controlled auto-ignition (CAI) combustion using a Lotus active valve train (AVT). In that study, a gasoline engine was used with a compression ratio of 10.5:1. They concluded that the heat release rate curve shifted toward TDC with increasing IEGR and that there was no need to pre-heat either the intake air or the intake air-fuel mixture if IEGR is used.

The effect of supercharging on HCCI combustion has been examined and it has been found to increase HCCI engine power output (9, 15).

Fuel injection timing has also a significant effect on homogeneous charge formation and on controlling the combustion mechanism of HCCI. Both Gold et al. (17) and Marriott (3) conducted a study on the effect of fuel injection timing using a direct injection (DI) gasoline HCCI engine. Gold et al. tested the engine at 1500 rpm with a constant 22:1 air/fuel ratio by varying the start of injection (SOI) between 0 and 90° ATDC with increments of 30°. Their results showed that as the time between injection and ignition increases, the variability in combustion stability improves. Marriott (3, 20) studied the effect of SOI on HCCI combustion for different engine speeds and equivalence ratios, and concluded that SOI would be an effective and useful needed parameter to control ignition timing for transient engine applications. In this study, the effect of double injection was also studied and it was found that multiple-injections were able to reduce NO_x emission levels by up to 80% compared to that of single injections with increased engine speed and constant intake air temperature.

HCCI operation with in-cylinder injection is only reported in the literature for diesel engines. However, fuel wall impingement is a severe problem when injecting heavy fuels into low-density air charge, some researches have expended significant effort in minimizing wall impingement by developing low-penetration fuel injectors and have obtained significant improvements. However, wall impingement is likely to remain a problem for a cylinder injection of diesel fuel into low density environments. In-cylinder fuel injection is likely to be come to preferred method for fueling HCCI engines in future as better mixture preparation techniques are developed (21).

3. EXPERIMENTAL SETUP

A fully instrumented Single Cylinder Oil Test Engine (SCOTE), which is a single-cylinder version of the Caterpillar 3400 series heavy-duty diesel engine, was used after converting it to a HCCI direct-injection gasoline engine. The engine is connected to a 68 kW Westinghouse direct current dynamometer. The SCOTE is capable of producing 62 kW at 1800 rpm when operated as a diesel engine. The basic specifications of the engine are provided in Table 1 and a schematic of the engine test unit is shown in Figure 1.

A data acquisition system was used to generate all digital information regarding the engine experiments by averaging 20 engine cycles in 0.5° CA increments. For the purpose of running the automated genetic algorithm experiments in the laboratory, specialized data acquisition and analysis software (AutoOpt GA) was developed by Thiel (24) using National Instruments LabWindows/ CVI programming software. To control the engine and the engine systems for the automated experiments, both RS-232 serial and analog communication was established between a laboratory PC, which ran the acquisition, analysis, and control code, and each of the process controllers and measurement instruments, including the dynamometer controller; the injector controller; the intake and exhaust tank pressure controller; the gaseous emission analyzer; the AVL DPL particulate analyzer; the temperature scanner; the EGR pump drive; and various pressure transducers. A detailed description about AutoOpt GA can be found in the thesis of Thiel (24).

During the experiments, a temperature controlled intake surge tank was used, as shown in Figure 1. The intake air temperature is regulated by a closed-loop control system operating the two 220 V three-phase 4.5 kW intake air heaters placed in series within the intake air flow stream. All engine temperatures were monitored using type K dual element thermocouples. Engine fluid pressures were measured using a variety of gages and transducers.

A prototype low-pressure Gasoline Direct Injection (GDI) common rail system supplied by Delphi Energy and Engine Management Systems was used in the present study. The injector creates a hollow cone sheet spray for good atomization. The injector was designed to operate with a rail pressure of 10 MPa and for injections into a gas with pressures less than 2 MPa. This restriction allows for an end of injection (EOI) timing as late as -25° ATDC for the SCOTE at normally aspirated operating conditions. The basic specifications are shown in Table 2. Detailed information about the fuel injection system is given in Ref. 3. The fuel used in the present study was Amoco Indolene. Fuel analysis results provided by the manufacturer are given in Table 3.

Table 1. Engine specification of the Caterpillar SCOTE (Single Cylinder Oil Test Engine)

Bore × Stroke	137.2 mm × 165.1 mm
Compression Ratio	16.1:1
Displacement	2..44 liters
Connecting Rod Length	261.62
Squish Height	1.57 mm
Combustion Chamber	In-piston Mexican Hat with sharp edged crater
Piston	Articulated
Valve Train (4-Valve)	EVC= -355° ATDC, IVC= -143° ATDC, EVO= 130° ATDC, IVO= 335° ATDC

Table 2. Fuel system specifications

Injector Type	DISI Electronically Controlled Low-Pressure Common Rail
Injection Pressure	10 Mpa
Nozzle Type	Pressure Swirl Atomizer
Spray Geometry	Hollow Cone
Spray Orientation	Coaxial with engine cylinder
Spray Angle (included)	60°

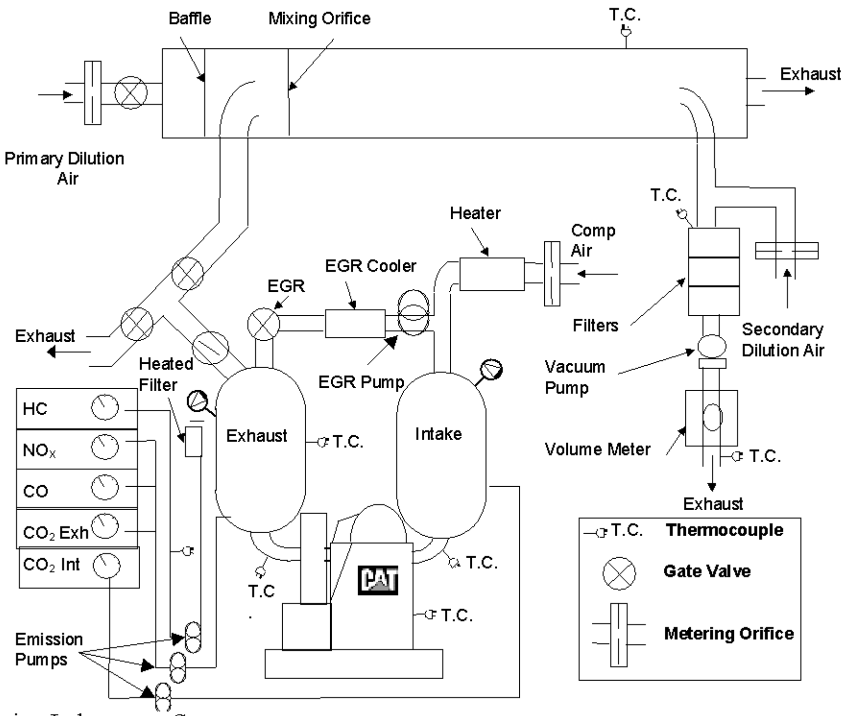

Figure 1. Engine Laboratory Setup

Emissions data recorded during the engine tests include UHC, CO, CO_2, NO_x and PM. Calibration of each analyzer was done before the tests. The NO_x, CO, and CO_2 emissions were measured on a dry basis after a particulate filter and ice bath. The UHC emission was measured after a heated filter, heated transport line, and heated diaphragm pump, all of which were maintained at 190°C to prevent UHC condensation. The instruments used in the measurements of the engine exhaust emissions are summarized in Table 4.

Table 3. Fuel Analysis Results

Fuel	Amoco Indolene
Carbon (by mass)	86.14 %
Hydrogen (by mass)	13.57 %
C/H Ratio	1.877
Lower Heating Value (LHV)	43.00 MJ/kg
Sulfur Content	3 ppm
Research Octane Number (RON)	96.8
Motor Octane Number (MON)	88.7
API Gravity @ 60°F	59.0
Reid Vapor Pressure, kPa	62.05
Distillation, (K)	
Initial Boling Point	302
10 % Evaporation	327
50 % Evaporation	377
90 % Evaporation	432
End point	468

Table 4. Exhaust Emission Measurement Equipment.

NO/NO_x	Chemiluminescent detector/California Analytical Inc., Model 400-CLD
CO	Infrared gas analyser / California Analytical Inc., Model 3300A
CO_2	Infrared gas analyser / Horiba, Model VIA-510
UHC	Flame ionization detector (FID) /Siemens, Model FIDAMAT 5E-IM
Soot	Dynamic Particulate analyser /AVL, Model DPL 482

4. GENETIC ALGORITM AND OPTIMIZATION METHODOLOGY

In this section, the baseline design, the parameters of interest, the objective function, and its evaluation will be described. The genetic algorithm code used in the experimental optimization study was originally developed by Senecal and Reitz (22). The code specifically uses the micro-genetic algorithm (GA) technique based on the GA code of Carroll (25). The optimization methods used in this experimental work are discussed in detail in Refs. 22 and 24.

4.1. Baseline Design and Parameters of Interest

The baseline engine specifications and operating conditions are presented in Table 5. The baseline engine cases for 700 rpm were taken from the double injection study on the HCCI gasoline engine of Marriott (3). The baseline case of 700 rpm includes an intake air temperature of 99°C, 71.1% of fuel mass injection in the first injection with an equivalence ratio of 0.26. The start of injection timings (SOI) for the first and second injections was -289 and -106° CA ATDC, respectively.

Figure 2 describes the nomenclature used in multiple injections. The injection shown in the Figure 2 is a 50 (110) 50 split injection (50% of the fuel injected in the first pulse followed by a 110 degree CA dwell and 50% of the fuel injected in the second pulse).

As stated earlier, for the present study the optimization parameters were the intake air temperature, start-of-injection timing, and split injection parameters (percent mass of fuel in each injection, and the dwell between the pulses). The considered engine optimization factors and ranges are presented in Table 6. The optimization target values are given in Table 7. These values were chosen as 80% of the EPA's 2004 emission regulations for heavy-duty engines.

Table 5. Operating Conditions for the Baseline Engine Case

Engine Speed (rpm)	700
Intake Pressure (kPa)	101.325
Exhaust Pressure (kPa)	101.325
Injection Pressure (Mpa)	10
Equivalance Ratio	0.26
Intake Temperature (°C)	99
1st SOI (°CA ATDC)	-289
2nd SOI (°CA ATDC)	-106
Fuel Mass in First Pulse (%)	71.1

Table 6. Engine Optimization Factors and Ranges for 700 rpm

Inlet Air Temperature (°C)	80→120
Start of injection	-320→-115
Fuel mass in first pulse (%)	10→90
Dwell between pulses (%)	5→160

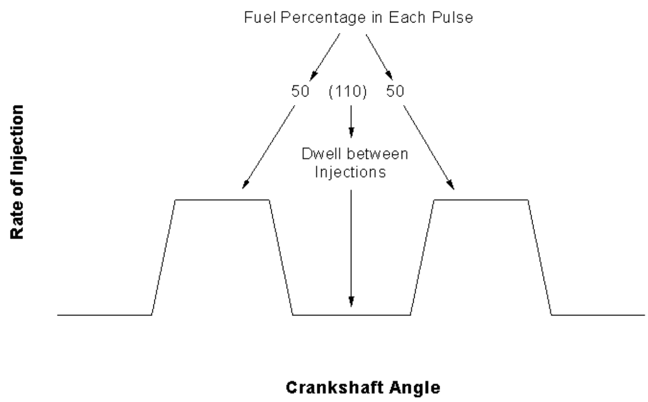

Figure 2. Description of Multiple Injection Nomenclature

4.2. Objective Function and Its Evaluation

The purpose of the present study was to reduce the emissions of the HCCI gasoline engine by varying engine operating parameters. Therefore, the objective (merit) function should contain exhaust emission measurements, including NO_x, HC, CO and PM, as well as fuel consumption. Similar objective functions were used for diesel optimization in Refs. 22, 23, and 24.

$$f(X) = \frac{10^3}{R_1^2 + R_2^2 + R_3^2 + R_4} \quad [1]$$

where

The objective function used in this study is: and the parameter vector X is the array of factors (SOI, air temp, dwell, etc.). For the current study, the target values were 2.68 g/kW-hr (NO_x + HC), 16.63 g/kW-hr CO and 0.107 g/kW-hr PM, as given in Table 7. The BSFC target value was 200 g/kW-hr, which is a baseline fuel consumption of the SCOTE.

$$R_1 = \frac{NO_x + HC}{(NO_x + HC)_t} \quad [2]$$

$$R_2 = \frac{CO}{CO_t} \quad [3]$$

$$R_3 = \frac{PM}{PM_t} \quad [4]$$

$$R_4 = \frac{BSFC}{BSFC_t} \quad [5]$$

After inserting the measured values in Equation 1, the optimum search provides a higher merit value for members of each generation with lower exhaust emissions and fuel consumption. It is more convenient to work with integer merit values. Therefore, the factor of 103 was included in the numerator of the function.

An alternative optimization was also used where given by

$$f(x) = \frac{10^3}{R_1^2 + R_4} \quad [6]$$

The results obtained using these objective functions deemphasize the role of particulates in the engine optimization. The physical constraints on the engine included a maximum intake pressure of 276 kPa, a maximum exhaust temperature of 1023 K, and peak combustion pressure of 15.2 MPa. Cases that violated these constraints were rejected.

5. RESULTS and DISCUSSION

To begin the optimization using fully-automated experiments and the micro-genetic algorithm (GA) optimization code (AutoOpt GA), ranges were determined for each factor. The values used in the present study are given in Table 6. Starting with a random number seed, the algorithm was run for 16 generations (i.e., 16x4=64 test conditions) over the course of 5 days. As a comparison, Figure 3 and Table 8 present, respectively, the in-cylinder combustion characteristics and operating conditions of the baseline and optimum cases.

The optimum conditions were an intake air temperature of 110°C, and 31.3% of fuel mass injected in the first injection at the same equivalence ratio of the baseline case. The SOI for the first and the second injections were -143.5 and -116° CA ATDC, respectively.

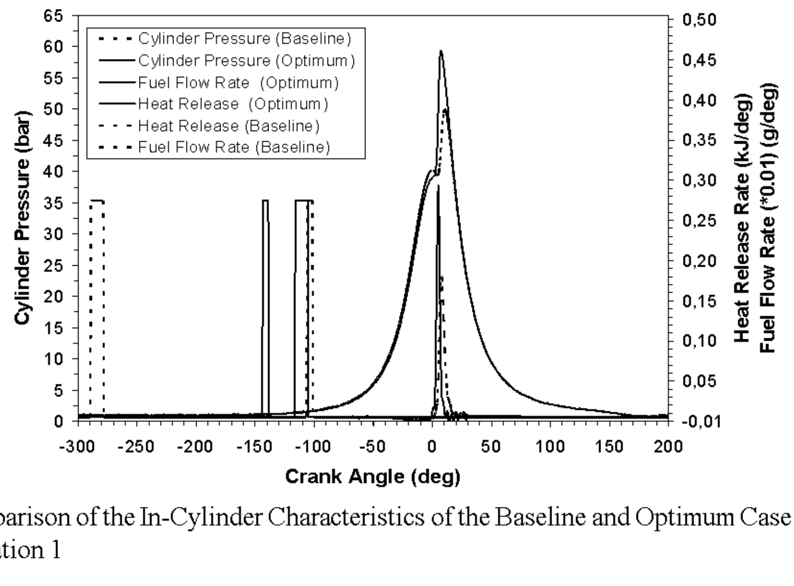

Figure 3. Comparison of the In-Cylinder Characteristics of the Baseline and Optimum Case with Merit from Equation 1

As seen in Figure 3, the peak cylinder pressure and heat release rates for the optimum case are higher than those of the baseline case as a result of the different operation conditions. The air-fuel mixture is less homogeneous than in the baseline case since the first SOI is 145.5° later than the first SOI of the baseline case and there is a shorter dwell between injections. In second injection, 68.7% of the total fuel injected was used. However, the higher intake air temperature helps the vaporization process.

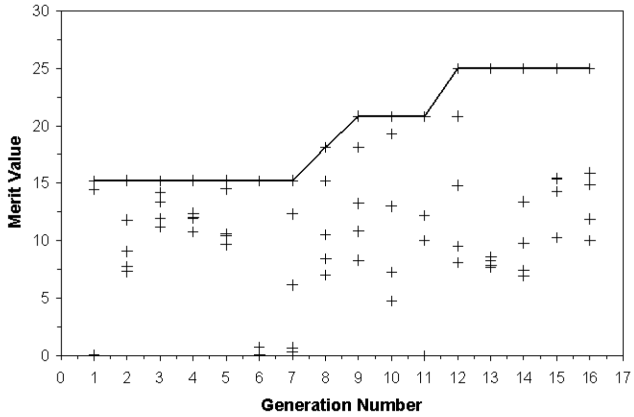

Figure 4. Merit Value vs. Generation Number for Merit Determined from Equation 1

Table 7. Optimization Target Values

$(NO_X + HC)_t$	2.68 g/kW-hr
CO_t	16.63 g/kW-hr
PM_t	0.107 g/kW-hr
$BSFC_t$	200 g/kW-hr

Table 8. Operating Conditions and Emission Results for the Baseline and Optimum Engine Cases with Merit Determined from Equation 1.

Operating Conditions	**Baseline**	**Optimum**
Intake Temperature (°C)	99	110
1st SOI (°CA ATDC)	-289	-143.5
2nd SOI (°CA ATDC)	-106	-116
Fuel Mass in First Pulse (%)		
Emissions (g/kW-hr)		
NO_x	0.015	0.73
HC	10.851	7.09
NO_x +HC	10.866	7.82
PM	-	0.580
CO	21.24	17.31
BSFC	261.65	254.24

Figure 4 shows the merit value of each citizen as a function of generation number where the maximum merit value for the generation is shown by the solid line. Throughout the optimization, the maximum merit value increased only two times, suggesting that convergence has not yet been achieved. As shown in Table 9, the unburned HC and particulate is much higher than the target values.

Table 9. Comparison of Target and Optimum Values

Emissions (g/kW-hr)	**Target**	**Optimum from**	
		Eq. 1	**Eq.6**
NO_x	2.14	0.73	0.73
HC	0.54	7.09	3.24
NO_x +HC	2.68	7.82	3.97
PM	0.107	0.580	0.237
CO	16.63	17.31	17.85
BSFC	200	254.23	286.31

Table 10. Operating Conditions and Emission Results for the Baseline and Optimum Engine Cases with Merit Determined form Equation 6.

Operating Conditions	Baseline	Optimum
Intake Temperature (°C)	99	120
1st SOI (°CA ATDC)	-289	-142.5
2nd SOI (°CA ATDC)	-106	-121
Fuel Mass in First Pulse (%)		
Emissions (g/kW-hr)		
NO_x	0.015	0.73
HC	10.851	3.24
NO_x +HC	10.866	3.97
PM	-	0.237
CO	21.24	17.85
BSFC	261.65	286.31

Figure 5 and 6 present the BSFC vs. NO_x and PM vs. NO_x data points, respectively, for all the runs of the present study. The data show that the choice of merit function is very important, since in this case PM is reduced at the expense of an increase in NO_x. Note that very high levels of unburned HC (relative to the target value) are found in this engine. The unburned HC and CO amounts in the exhaust directly affect BSFC. This suggests that the objective (merit) function should contain only NO_x, and BSFC for improved optimization. Figure 6 shows that very high PM's were found in the optimization. However, the AVL dynamic particulate analyzer results showed that around 95% of the measured value was soluble organic fraction (SOF) in the PM. Therefore, the exhaust gases contain very low insoluble organic fraction (ISOF). This is consistent with the high measured HC concentration, as seen in Table 9.

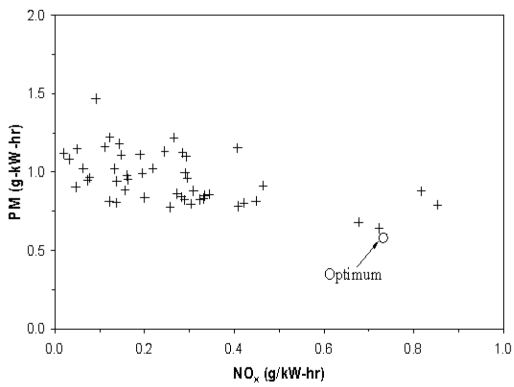

Figure 5. BSFC vs. NO_x Data with Merit from Equation 1.

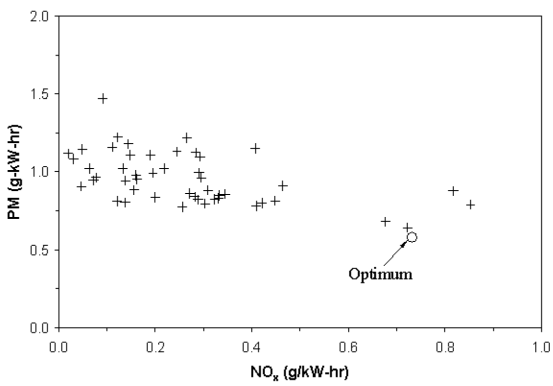

Figure 6. PM vs. NO_x Data with Merit from Equation 1.

The relationship between combustion efficiency and NO_x+HC emission is shown in Figure 7. Because of the choice of merit function, the optimum point has higher NO_x than the baseline (see Figure 5), even though it has a higher combustion efficiency. This is because other parameters also appear in the merit function of Equation 1 (i.e., species other than NO_x), and their influence is included in the optimization. Table 9 presents a comparison of the target and optimum values. As can be seen, the target values were not reached. Accordingly, a revised optimization was started.

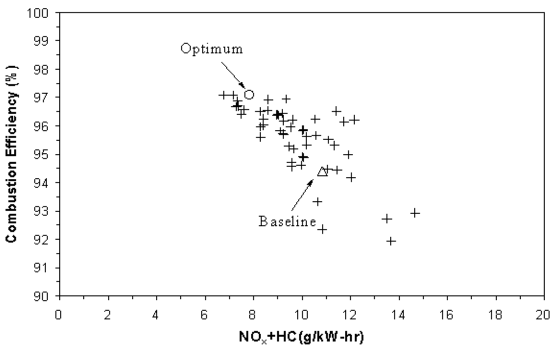

Figure 7. Combustion Effiency vs. NO_x+HC Data with Merit from Equation 1.

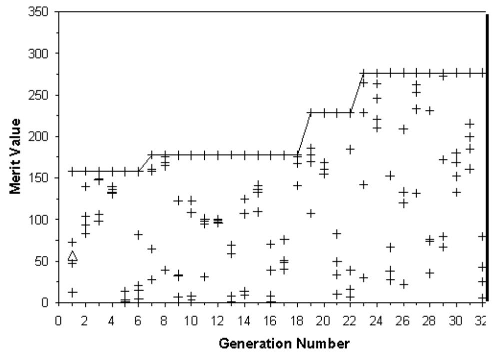

Figure 8. Merit Value vs. Generation Number from Merit Determined from Equation 6.

As mentioned earlier, the merit function needs to be selected carefully since the optimum reached depends on the choice of parameters selected. Therefore, as an alternative, the merit function was changed to deemphasize the contribution of unburned HC and soluble particulates, as shown in equation 6. Using the same baseline case, the algorithm was run for 33 generations (i.e., 33x4=132 test conditions) over the course of 10 days. As a comparison, Figure 8 presents the progress of the merit value. The relationship between combustion efficiency and NO_x+HC is shown in Figure 9. As can be seen by comparing Figure 9 with Figure 7, lower NO_x+HC emission was obtained in the optimum case using Equation 6.

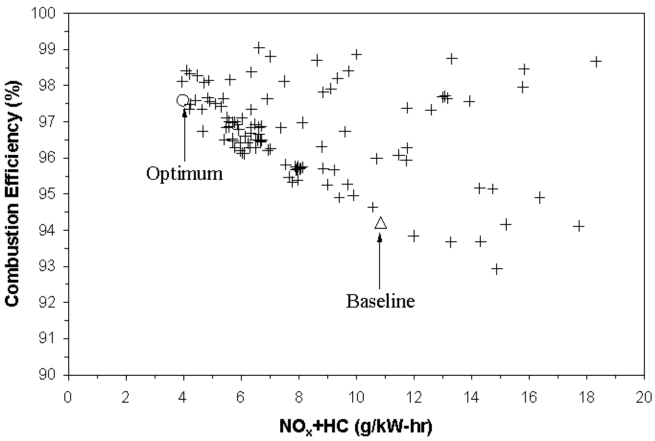

Figure 9. Combustion Effiency vs. NO_x+HC Data with Merit from Equation .6

Table 10 compares the baseline and optimum cases. For the optimum case, the conditions were an intake air temperature of 120°C, and 58% of fuel mass injected in the first injection. The SOI for the first and second injections were -142.5 and -121° CA ATDC, respectively.

When comparing the optimum case to the baseline case, the air-fuel mixture is less homogeneous since the first SOI is 146.5° later than the first SOI of the baseline case and there is a shorter dwell (10° CA) between injections. 42% of the total fuel injected was used in the second injection.

As seen in Table 9 and 10, the use of the merit function of Equation 6 leads to reduced PM and NO_x+HC emissions over those obtained from Equation 1 at the expense of increased fuel consumption.

6. CONCLUSIONS

An electronically controlled Caterpillar single-cylinder oil test engine (SCOTE) was configured as a HCCI direct-injection gasoline engine. The use of double injection was explored for emission control, and the engine was optimized using fully automated experiments and a micro-genetic algorithm (GA) optimization code. The variables changed during the optimization included the intake air temperature, start of injection timing, and split injection parameters (percent mass of the fuel in each injection, dwell between the pulses). The engine performance and emissions were determined at 700 rev/min with a constant fuel flow rate at 10 MPa fuel injection pressure. The results show that significant emissions reductions are possible with the use of optimal injection strategies. The fully automated experiments and optimization code provides a useful tool for engine designers investigating the effects of a large number of input parameters on emissions and performance.

This technique efficiently determined a set of input parameters resulting in significantly lower emissions and BSFC compared to baseline case. However, the results also show that the appropriate selection of the merit function is crucial in the search for optimum engine parameters for improved emissions and BSFC in HCCI combustion.

ACKNOWLEDGMENTS

This research funded by Caterpillar Inc. and the DOE/Sandia National Laboratory was done in Engine Research Center in UW-Wisconsin. The authors would like to thank Dr. Rolf D. Reitz for his valuable advises.

REFERENCES

1. EPA Document, "Heavy-Duty engine and vehicle standards and highway diesel fuel sulfur control requirements", ***EPA Document*** #: EPA420-F-00-057 (2000).
2. Kelly-Zion, P. L., and Dec, J., "A Computational study of the effects of fuel-type on ignition time in HCCI engines", Proceedings of the 28th International Symposium on Combustion, ***Combustion Institute***, 28: 1187-1194 (2000).
3. Marriott, C. D., "An experimental investigation of direct injection for homogeneous and fuel-stratified charge compression ignited combustion timing control", ***M.Sc. Thesis, University of Wisconsin***, Madison, WI (2001).
4. Kong, S-C., Marriott, C. D., Reitz, R. D., and Christensen, M., 2001, "Modeling and experiments of HCCI engine combustion using detailed chemical kinetics with multidimensional CFD", ***SAE Paper***, 2001-01-1026 (2001).
5. Marriott, C. D., and Reitz, R. D., "Experimental investigation of direct injection-gasoline for premixed compression ignited combustion phasing control", ***SAE Paper***, 02-01-0418 (2002).
6. Christensen, M., Johansson, B., and Einewall P., "Homogeneos charge compression ignition (HCCI) using isooctane, ethanol, and natural gas - A comparison with spark ignition operation", ***SAE Paper***, 2874 (1997).
7. Christensen, M., Hultqvist, A., and Johansson, B., "Demonstrating the multi fuel capability of a homogeneous charge compression ignition engine with variable compression ratio", ***SAE Paper***, 1999-01-3679 (1999).
8. Christensen, M., and Johansson, B., "Influence of Mixture Quality on Homogeneous Charge Compression Ignition", ***SAE Paper***, 982454 (1998).
9. Christensen, M., and Johansson, B., AmnJus, P., and Mauss, F., "Supercharged homogeneous charge compression ignition", ***SAE Paper***, 980787 (1989).
10. Richter, M., Franke, A., Alden, M., Hultqvist, A., and Johansson, B., "Optical diagnostics applied to a naturally aspirated homogeneous charge compression ignited engine", ***SAE Paper***, 1999-01-3649 (1999).
11. Morimoto, S. S., Kawabata, Y., Sakurai, T., and Amano, T., "Operating characteristics of a natural gas-fired homogeneous charge compression ignition engine (Performance improvement using EGR)", ***SAE Paper,*** 2001-01-1034 (2001).
12. Goldsborough, S., and Van Blarigan, P., "A numerical study of a free piston IC engine operating on

homogeneous charge compression ignition combustion", ***SAE Paper***, 1999-01-0619 (1999).

13. Law, D., Kemp, D., Allen, J., Kirkpatrick, G., and Copland, T., "Controlled combustion in an IC-Engine with a fully variable valve train", ***SAE Paper***, 2000-01-0251 (2000).
14. Thring, R. H., "Homogeneous-charge compression- ignition (HCCI) engines", SAE Paper, 892068 (1989).
15. Aoyama, T., Hattori, Y., and Mizuta, J., "An experimental study on premixed-charge compression ignition gasoline engine", ***SAE Paper***, 960081 (1996).
16. Oakley, A., Zhao, H., and Ladommatos, N., "Experimental studies on controlled auto-ignition (CAI) combustion of gasoline in a 4-stroke engine", ***SAE Paper***, 2001-01-1030 (2001).
17. Gold, M., Stokes, J., Morgan, R., Heikel, M., De-Sercey, G., and Begg, S., "Air-fuel mixing in a homogeneous charge DI gasoline engine", ***SAE Paper***, 2001-01-0968 (2001).
18. Gray, A., and Ryan, T., "Homogeneous charge compression ignition (HCCI) of diesel fuel", ***SAE Paper***, 971676(1997).
19. Aceves, S. M., Flowers, D. L., Martinez-Frias, J., Smith, J. R., Dibble, R., Au, M., and Girard, J., 2001, "HCCI combustion: analysis and experiments", ***SAE Paper,*** 2001- 01-2077 (2001).
20. Marriott, C. D., Kong, S-C., and Reitz, R. D., 2002, "Investigation of hydrocarbon emissions from directInjection-gasoline premixed charge compression ignited engine", ***SAE Paper***, 2002-01-0419 (2002).
21. Stanglamaier, R., Roberts, C., "Homogenous charge compresion ignition (HCCI): Benefits, compromises, and future engine applications", ***SAE Paper***, 1999-01-3682 (1999).
22. Senecal P. K., and Reitz, R. D., "Simultaneous reductions of engine emissions and fuel consumption using genetic algorithms and multi-dimensional spray and combustion modeling", ***SAE Paper***, 2000-01-1890 (2000).
23. Montgomery, D. T., and Reitz, R. D., "Effects of multiple injections and flexible control of boost and EGR on emissions and fuel consumption of a heavy-duty diesel engine", ***SAE Paper***, 2001-01-0195 (2001).
24. Thiel, P. T., "Application of automated experiments to the optimization of a Heavy-duty direct injected diesel engine for the simultaneous reduction of NO_x and particulate emissions", ***M.Sc. Thesis, University of Wisconsin***, Madison, WI (2001).
25. Carroll, D. L., "Genetic algorithms and optimizing chemical oxygen-iodine lasers", ***Developments in Theoretical and Applied Mechanics***, 18: 411 (1996).

Accepted:24.03.2003 *Received:11.07.2003*