

Combustion Modeling of Landfill Gas Fueled Spark Ignition Engine Performance

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Abstract

Landfill gas (LFG) is a waste-product resource composed primarily of methane and carbon dioxide that can be collected and used to produce power either by extracting the methane or using the landfill gas directly in an internal combustion engine in what amounts to a net-negative greenhouse gas emission process. The carbon dioxide component of LFG dilutes the fuel and absorbs some of the heat of combustion, but also suppresses knock. A model is developed, using KIVA-4 code, to simulate engine performance at various operating conditions and evaluate the benefits of methane purification and direct use of LFG as a fuel.

It was found that landfill gas used directly at higher compression ratios than can be used for pure methane fuel produces higher fuel efficiency than can be achieved using pure methane. Furthermore the study has shown that the fuel efficiency of the engine can be achieved within 5% of that of pure methane fuel for landfill gas, diluted up to 1:1 with carbon dioxide by simply adjusting the spark timing of the engine. A model for combustion duration is also developed which can estimate the combustion duration in an internal-combustion engine for varying fuel compositions.

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Introduction

There is 75 billion Nm³ of methane gas produced in landfills annually worldwide, and less than 10% of this fuel is captured and utilized [1]. Methane is a potent greenhouse gas with 72 times the global warming potential of carbon dioxide when averaged over 20 years and 24 times that of CO₂ when averaged over 100 years [1, 8, 9]. In large landfills in the United States, the Environmental Protection Agency (EPA) requires landfill gas to be collected and flared to prevent the release of methane into the atmosphere, but the methane fuel can profitably be collected and utilized in a landfill-gas-to-energy (LFGTE) operation to produce power and or heat. In large landfills, gas turbines are used to produce energy because of their high power density, but in addition to these, the US EPA has identified hundreds of candidate sites for LFGTE; most with smaller amounts of waste in place. For these operations, an economical internal combustion engine generator presents an attractive option. LFGTE projects are among the greenest technologies. Combusting the methane in

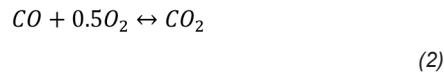
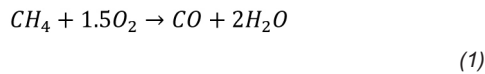
landfill gas is a net-negative greenhouse gas emission operation, and in this case energy production can be considered a "free" byproduct of greenhouse gas reduction. From an economic perspective, LFGTE projects are very attractive. In 2005, it was estimated that the break-even price for an electricity-generating LFGTE project was between \$0.03-0.04/kWh which is far below the current cost of electricity [6].

Landfill gas (LFG) is produced in an anaerobic environment and consists primarily of methane and carbon dioxide with small amounts of nitrogen and other non-methane organic compounds. An internal combustion engine can produce power using either purified methane fuel or raw LFG. Economic analyses have determined that LFGTE projects can be very profitable, so technologies which help to utilize this largely untapped resource will produce both economic and environmental benefits particularly reducing green house gas effects.

Modeling Theory

KIVA-4 code from Los Alamos National Lab was utilized for simulating a spark-ignition engine. KIVA is a CFD program designed specifically to simulate internal combustion engine performance and has been successfully employed to simulate methane combustion in a spark-ignited engine by Tsao and Han and others [7]. For the sake of simplicity, only the compression and power stroke were modeled - from timing at intake valve closure to exhaust valve opening. The fuel/air mixture was assumed to be homogenous and included residual exhaust gases from the previous cycle.

A two-step chemical kinetic mechanism proposed by Westbrook and Dryer [2] was used for the combustion of methane. The Westbrook-Dryer mechanism includes one irreversible and one reversible steps as shown below.

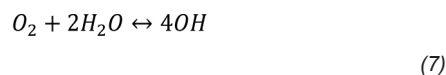


The reaction rates used in the Arrhenius form are shown in Table 1 with units of cm, s, cal, mol.

Table 1. Reaction Constants (units are cm, s, cal, mol)

Reaction	A	β	Ea	Reaction Order
1f	1.59×10^{13}	0	47800	$[\text{CH}_4]^{0.7}[\text{O}_2]^{0.8}$
2f	3.98×10^{14}	0	40700	$[\text{CO}][\text{O}_2]^{0.25}[\text{H}_2\text{O}]^{0.5}$
2b	5.0×10^8	0	40700	$[\text{CO}_2]$

In addition, following five equilibrium reactions were used:



KIVA also employs a mixing controlled chemistry model based on the eddy-dissipation model developed by Magnussen and Hjertager [3]. This model produces a flame speed dependence on the square root of k (turbulent kinetic energy) [5]. In the Magnussen and Hjertager model for premixed turbulent flames, the rate of fuel combustion for the fuel oxidation reaction is limited by the smallest value of any of three limitations which describe dissipation of eddies which may limit the oxidation of fuel. These limitations are based on dissipation of fuel and oxygen eddies, the flame speed in regions of high fuel concentration and low oxygen concentration, and the combustion rate based on dissipation of hot eddies of products of combustion.

Using KIVA, the flame front can be seen to propagate as a temperature gradient from the spark location at the top-center of the cylinder toward the outside. Figure 1 and 2 show temperature gradient of a slice from centerline to the cylinder wall at various stages of combustion.

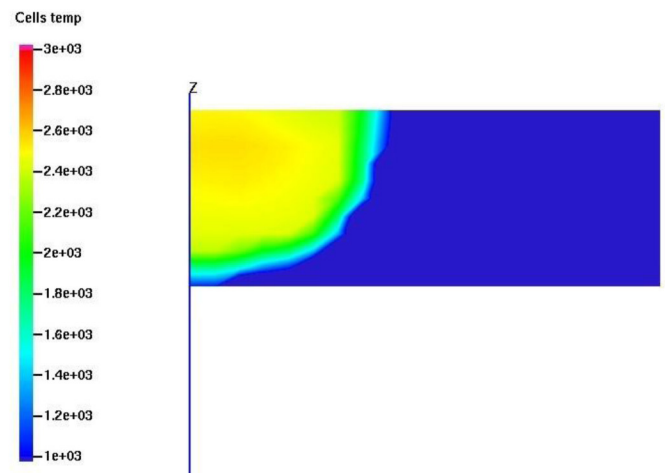


Figure 1. Temperature Gradient at 5% of Fuel Consumed (K)

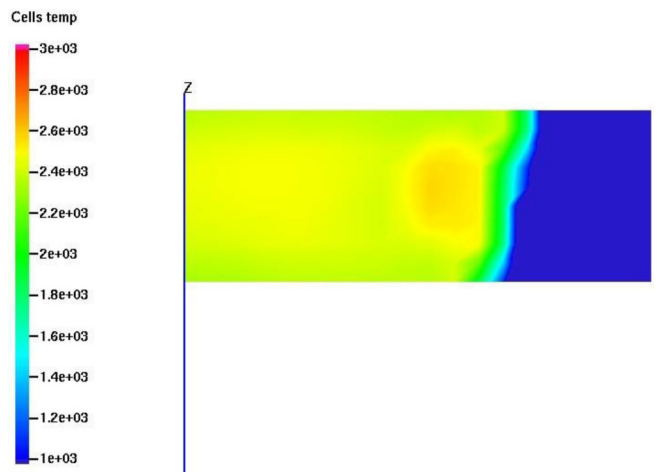


Figure 2. Temperature Gradient at 25% of Fuel Consumed (K).

Model Verification

Engine performance was modeled using the characteristics of a cooperative fuel research (CFR) spark-ignition engine. The cylinder dimensions, stroke length, valve timing, spark timing, spark plug location, connecting rod length, engine speed, initial temperature and pressure, fuel composition, and equivalence ratio were all taken from the CFR engine with known performance characteristics. The static engine characteristics are shown in [Table 2](#) below.

Table 2. Engine Characteristics

Variable	Value	Units
Bore	8.26	cm
Stroke	11.43	cm
Connecting Rod Length	25.4	cm
Valve Close	-146	deg ATDC
Valve Open	140	deg ATDC

Engine performance, as shown in [Table 3](#) and [4](#), was matched to experimental results produced by the CFR engine by Narayana [8]. This was accomplished by iteratively modifying the input values for spark energy addition to match the experimental value for ignition delay and the input value of turbulent kinetic energy in Magnussen and Hjertager model [3] which impact flame speed in order to match the experimentally determined value for combustion duration [9]. The resulting model produced combustion duration, power, and peak pressure trends that compared generally favorably with experimental values except that the simulation does not suffer from ring or valve leakage as a real engine does and so simulated values for peak pressure and power are somewhat higher than experimental values. Ignition delay did not produce good agreement or follow trends for diluted fuel; however, since a range of spark timings were examined, the model is still valid except in that actual spark timing required to produce similar results may vary from the model. Moreover, the ignition delay generally influences minimum on the performance parameter of the engine. Additionally, the simulations used for modeling were run at 1,200 rpm instead of 600 rpm used for verification, because an engine speed of 1,200 rpm represents a common operating speed for an electrical generator. The ignition delay and combustion duration were determined as 5% and 95% of the mixture mass burned for all the calculations.

Results

[Figure 3](#) demonstrates the effect of increasing compression ratio on power output. For the same equivalence ratio (ER), power increases with compression ratio as expected. In addition, optimum spark timing retards with increasing

compression ratio because raising the compression ratio also raises the gas temperature which leads to higher flame speeds and shorter combustion duration.

When carbon dioxide is introduced into the fuel, the results are consistent with predictions. [Figure 4](#) shows pressure changes in the compression and power strokes in a motored cycle, a stoichiometric mixture of air and methane fuel, and a stoichiometric mixture of air and a fuel consisting of 50% methane and 50% carbon dioxide by volume basis. In fact, all compositions of the fuel mixtures were presented in molar basis if not mention otherwise explicitly. And in [Figure 5](#) it can be seen that increasing the amount of CO₂ in the fuel has a degrading effect on power output. In addition, the lower flame speed advances the spark timing associated with optimum power generation as the percentage of carbon dioxide in the fuel increases.

Table 3. Comparison With Experimental Values

Parameters	Simulated	Experimental	Difference
Ignition Delay [deg]	3	3	0%
Combustion Duration [deg]	25	27	8%
Peak Pressure [MPa]	6.7	5.1	24%
Indicated Power [kW]	2.86	1.9	34%

Methane, CR=8.5, ER=0.9, 600 rpm, 98 kPa inlet pressure, spark timing 30 deg BTDC, initial/inlet temp 303K

Table 4. Comparison With Experimental Values

Parameters	Simulated	Experimental	Difference
Ignition Delay [deg]	5	15	200%
Combustion Duration [deg]	30	33	10%
Peak Pressure [kPa]	4.89	3.9	20%
Indicated Power [kW]	2.71	2.0	26%

50% Methane, 50% CO₂, CR=8.5, ER=1.0, 600 rpm, 98 kPa inlet pressure, spark timing 19 deg BTDC, initial/inlet temp 303K