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Predicting the performance and exhaust NO_X emissions of a spark-ignition engine generator fueled with methane based biogases containing various amounts of CO_2

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The effect of methane–CO_2 blends with varying CO_2 concentrations (corresponding to various biogas compositions) on combustion and emission characteristics were investigated numerically for a gas engine generator. CO_2 variation in a methane-based biogas fuel produced effects similar to the CO_2 effects of exhaust gas recirculation (EGR), decreasing the cylinder temperature, pressure, heat release rate, and flame speed. By design of experiment (DOE) analysis, the maximum brake torque (MBT) timing was calculated for various values of the spark timing and excess air ratio (EAR). As the EAR increased from 1.2 to 1.8 and the CO_2 content decreased, the MBT timing was far advanced from top dead center (TDC). At 50% CO_2 content, incomplete combustion phenomena were detected in the leaner region at EAR values above 1.6. In spite of the performance degradation of the engine itself, the addition of CO_2 up to 50% in volumetric content improved the electrical generating efficiency by an average of 6.4%, with lower NO_X formation.

1. Introduction

Sustainable and renewable energy sources are now being employed in mechanical systems. In particular, bio-fuels are being used as alternative fuels for mechanical systems, either alone or blended with conventional fossil fuels or additives (Sudheesh and Mallikarjuna, 2010; Gumus and Kasifoglu, 2010; Mbarawa, 2010; Kang et al., 2012; Crookes, 2006). Methane-based fuels such as natural gas, shale gas, biomethane and biogas known as landfill gas (consisting mainly of methane (CH_4) and carbon dioxide (CO_2)) are among the most promising alternative fuels, offering substantial benefits that include lower fuel costs, cleaner exhaust emissions, relatively higher thermal efficiency with lean-burn capability, and direct applicability to the most powerful of existing combustion systems. Due to the high octane number of methane portion of biomethane or biogas, it can be applied to not only SI engines as it is, but also CI engines as blends with diesel fuel. By applying methane-based fuel such as biomethane or biogas on conventional SI and CI engines, reduced emission characteristics in terms of NO_X and soot could be achieved. On the other hand, engine performance and thermal behavior including torque, power, pressure, and thermal efficiency have degraded and more optimization of whole system have to be performed by field engineers. A point of view with electrical generator connected to conventional engine, however, performance degradation of engine part could be allowed with acceptable electricity production from engine power (Crookes, 2006; Porpatham et al., 2008; Bedoya et al., 2012; Chandra et al., 2011; Park et al., 2010a,b).

The raw biogases from real-world production site have been upgraded to enrich the methane portion and to trace CO_2 and organic compounds in biogas by purification for different biogas utilization applications. However, there are trade-offs between costs for purification system developments and energy efficiencies after purification. In that point of view, increasing efforts to apply raw biogas directly in engine-based power systems have led to research related to biomethane or biogas composition verification, since gaseous fuel composition varies according to national, regional, and onsite environmental conditions.

A few studies have been carried out by varying the concentration of methane or CO_2 (Porpatham et al., 2008; Chandra et al., 2011). However, these studies have not been directly applicable to

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real-world sites due to a lack of data for wide engine system operating ranges, which presents difficulties and limitations in finding optimized conditions solely by experimental methods. And time and cost are also important issues for engine research. Therefore, demands for numerical analysis have been required to save time and cost. The typical simulation studies related to engine with alternative fuels have followed the conventional method. Those methods are usually focused on the three-dimensional combustion analysis with KIVA, fluidic analysis with CFD tools, and kinetic studies to provide detailed chemistry. For engine cyclic analysis, cycle simulation, which can provide prediction of whole engine operation processes with conventional fossil fuel, have been used by commercial engine manufacturers (Park et al., 2010a,b; Gogoi and Baruah, 2010). However, there have been relatively few applications of cycle simulation to engine analysis using alternative fuels, such as methane-based fuels, and improvements in the optimization techniques are still needed. One type of engine optimization employs design of experiment (DOE) methods for a gas engine generator fueled with a methane–hydrogen blend (Park et al., 2012). In that research, the predicted peak cylinder pressures of the gas engine were in good agreement with experimental results.

In the present study, the performance and NO_X emission characteristics of a gas engine powered by methane-based fuels with varying CO_2 concentrations are examined via cycle simulation. Dominant parameters were the CO_2 content in methane-based biogas, spark timing, and EAR and were independent variables at a constant engine speed of 1820 rpm under stoichiometric and fuel lean conditions. The fundamental effects of CO_2 content on combustion are investigated. To determine the maximum performance of a gas engine generator, the MBT timing is determined for various EAR and CO_2 content, using the full factorial DOE method. Moreover, the generating efficiency is calculated at a constant electrical power of 15 kW.

2. Experiments

Experiments were performed to have accurate validation results of the numerical model and confirm the modeling accuracy in terms of maximum cylinder pressure. The experimental setup and procedures were described in detail in our previous research on gas engine generators fueled with natural gas and biogas (Park et al., 2012). The experiments were performed at an EAR of 1.2, an engine speed of 1820 rpm, and a boost pressure of 1.2 bars. Table 1 shows the engine specifications and experimental conditions for model validation. The experiments were conducted using natural gas at spark timings of 18°–13° crank angle (CA) before top dead center (BTDC), and biogas at a spark timing of 16° CA BTDC, which are the operating ranges suggested by the engine manufacturer. The seven experimental data of maximum cylinder pressure were compared to numerical results obtained by the DOE method.

Table 1
Engine specifications and operating conditions.

Item	Specification
Engine type	Vertical water-cooled spark ignition engine
Bore	88 mm
Stroke	94 mm
Displacement	2300 cc
Compression ratio	13:1
Intake system	Turbocharger
Maximum power	30 kW
Engine operating speed	1820 rpm
Boost pressure	1.2 bar
Electrical power	15 kW
Test fuel	Methane: 100% & Model biogas: 60% CH_4 + 40% CO_2 (by vol.%)

3. Modeling and validation overview

In this section, modeling and validation overview based on our previous work is introduced briefly. The numerical analysis was carried out with GT-POWER, which is designed for engine cycle simulations based on thermodynamics. More detailed explanation used in this study can be found in our previous study (Park et al., 2010a, 2012, 2014). Fuel supplies (methane and biogas) were placed at the intake pipe and mixed with ambient air. The turbocharger performance data was input from the turbocharger map. The 1.2 bar of boost pressure was maintained by the proportional–integral–derivative control method.

The combustion process was analyzed with a two-zone model, which was used in our previous numerical analysis of a gas engine generator fueled with methane–hydrogen blends (Park et al., 2012). The energy, mass, and momentum conservation equations were solved separately for each time step in each zone (Park et al., 2012, 2014; Heywood, 1988):

$$\frac{d(m_u e_u)}{dt} = -p\frac{dV_u}{dt} - Q_u - \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right) + \frac{dm_{f,i}}{dt}h_{f,i} \quad \text{for the unburned zone} \tag{1}$$

$$\frac{d(m_b e_b)}{dt} = -p\frac{dV_b}{dt} - Q_b + \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right) \quad \text{for the burned zone} \tag{2}$$

The following equations below describe mass entrainment rate into the flame front and the burn rate:

$$\frac{dM_e}{dt} = \rho_u A_e (S_T + S_L) \tag{3}$$

$$\frac{dM_b}{dt} = \frac{(M_e - M_b)}{\tau} \tag{4}$$

The entrainment rate for unburned mass, M_e, depends on the flame front area, A_e and an entrainment velocity, S (formed from the sum of the turbulent (which is mainly applied to a swirling motion) and laminar flame speeds, S_T and S_L). The burn rate behind the flame front is assumed to be proportional to the unburned mass behind the flame front, resulting in the following rate equation for the burned mass, M_b. Since the burn is postulated to take place at the laminar flame speed and over a length scale typical of the microscale of turbulence, the time constant is expressed as following:

$$\tau = \frac{\lambda}{S_L} \tag{5}$$

The laminar flame speed S_L for conventional fuel is usually calculated by the equation

$$S_L = \left(B_m + B_\phi(\phi - \phi_m)^2 * \left(\frac{T_u}{T_{ref}}\right)^\alpha \left(\frac{p}{p_{ref}}\right)^\beta \times \left(1 - 2.06(\text{Dilution})^{\text{DEM}*0.77}\right)\right) \tag{6}$$