Modelling the Impacts of Hydrogen–Methane Blend Fuels on a Stationary Power Generation Engine

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Modelling the Impacts of Hydrogen–Methane Blend Fuels on a Stationary Power Generation Engine

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Abstract: To reduce greenhouse gas emissions from natural gas use, utilities are investigating the potential of adding hydrogen to their distribution grids. This will reduce the carbon dioxide emissions from grid-connected engines used for stationary power generation, and it may also impact their power output and efficiency. Promisingly, hydrogen and natural gas mixtures have shown encouraging results regarding engine power output, pollutant emissions, and thermal efficiency in well-controlled on-road vehicle applications. This work investigates the effects of adding hydrogen to the natural gas fuel for a lean-burn spark-ignited four-stroke, 8.9 liter eight-cylinder naturally aspirated engine used in a commercial stationary power generation application via an engine model developed in the GT-SUITE™ modelling environment. The model was validated for fuel consumption, air flow, and exhaust temperature at two operating modes. The focus of the work was to assess the sensitivity of the engine’s power output, brake thermal efficiency, and pollutant emissions to blends of methane with 0–30% (by volume) hydrogen. Without adjusting for the change in fuel energy, the engine power output dropped by approximately 23% when methane was mixed with 30% by volume hydrogen. It was found that increasing the fueling rate to maintain a constant equivalence ratio prevented this drop in power and reduced carbon dioxide emissions by almost 4.5%. In addition, optimizing the spark timing could partially offset the increases in in-cylinder burned and unburned gas temperatures and in-cylinder pressures that resulted from the faster combustion rates when hydrogen was added to the natural gas. Understanding the effect of fuel change in existing systems can provide insight on utilizing hydrogen and natural gas mixtures as the primary fuel without the need for major changes in the engine.

Keywords: internal combustion engine; power generation; hydrogen–methane blends; engine modelling; low-carbon fuels; combustion modelling; emissions

1. Introduction

The environmental and human health impacts of pollutant emissions are driving increasing demand worldwide for near-term methods to reduce the use of fossil fuels. In particular, the medium- and long-term impacts of climate change have made reducing greenhouse gas (GHG) emissions from all sectors an urgent priority. Increasing electrification of personal transportation through electric vehicles and building heating through heat pumps offers the potential of substantial GHG emissions reductions depending on the carbon intensity of the electrical generation. However, these add significant load to electrical generation, transmission, and distribution infrastructure. Using the natural gas distribution network to transport low-GHG blends, including renewable natural gas (RNG) and renewably generated hydrogen (H₂), offers a near-term approach to reducing GHG emissions at the end use. Stationary power generation powered by internal combustion engines fed from the gas distribution network can be coupled with combined cooling, heat, and power (CCHP) systems, to help to meet peak electrical demand while supplying thermal energy for heating and cooling. While adding H₂ to the gas grid will reduce GHG emissions, the changes in fuel composition will have a direct impact on the performance...
of the engine. This work focuses on understanding the impacts of varying \( \text{H}_2 \) content in the gaseous fuel for a commercial building size stationary power generator and evaluates approaches to mitigate changes in the combustion process.

In commercial size stationary power applications (on the order of 100–300 kW output power), the engines are typically fueled with diesel, natural gas (NG), or propane. Of these fuels, NG is widely available through existing distribution networks and unlike diesel and propane does not require separate fuel deliveries. NG is composed primarily of methane (\( \text{CH}_4 \)), which gives it the lowest carbon dioxide (\( \text{CO}_2 \)) per unit of energy of any hydrocarbon fuel. It also generates relatively low particulate matter (PM) emissions, due to both the lack of carbon–carbon bonds in the fuel and the premixed combustion process employed in most NG engines [1,2]. Methane can also be generated from biological processes including organic municipal waste and farm waste, to form RNG. This results in “net” carbon emissions that can be very low or even negative based on feedstock over the full fuel production–consumption pathway characterized through mechanisms such as California’s low-carbon fuel standard.

While NG is widely used in stationary power generation applications, it does have some disadvantages. It has relatively low burning velocity, especially under lean conditions, that can cause high cyclic variation, longer combustion duration, potential for increased unburned fuel emissions, and reduced power output [3]. Unburned fuel emissions, which are predominantly methane, contribute to GHG emissions, with a global warming potential (GWP) of 28–34 on a 100-year horizon. Combined with the \( \text{CO}_2 \) emissions from the combustion of natural gas, there is a widely recognized need to further reduce GHG emissions from the NG grid in general and from stationary power generation applications in particular. One approach to reduce the greenhouse gas emissions from a gaseous fuel is to mix \( \text{H}_2 \) into the NG. \( \text{H}_2 \) has many desirable combustion characteristics: its combustion emits virtually no carbon monoxide (\( \text{CO} \)), \( \text{CO}_2 \), sulfur dioxide (\( \text{SO}_2 \)), hydrocarbons (\( \text{HC} \)), and PM compared with fossil fuels, and its main combustion product is water [3]. On the other hand, \( \text{H}_2 \) is hard to transport and store due to its low density [4].

Currently, the vast majority of \( \text{H}_2 \) is produced through steam-methane reforming (SMR) from fossil natural gas. With increasing demand for low-carbon energy, there is growing interest and demand for \( \text{H}_2 \) that is produced from cleaner sources, including SMR with carbon capture and storage or electrolysis of water using electrical energy generated from renewable or other low-GHG sources [5]. Currently, sources of such low-GHG \( \text{H}_2 \) are limited, infrastructure to efficiently deliver it to end uses is not yet developed, and end uses optimized for \( \text{H}_2 \) use are not widely available [6]. As a result, there are multiple barriers to realizing near-term benefits from \( \text{H}_2 \) as a fuel. One approach to distribute and use low-GHG \( \text{H}_2 \) is to mix it into natural gas to form hydrogen–natural gas blends (HCNGs). Not only does this offer a way to reduce the GHG intensity of the existing NG grid, the addition of \( \text{H}_2 \) has several positive impacts on NG end uses. Table 1 provides a comparison between methane, hydrogen, and HCNG blend properties based on the results of the developed model; of particular note is the increased flame speed with hydrogen addition, which is particularly noticeable at the higher temperature and pressure conditions seen in a typical internal combustion engine.

The most widely studied application of HCNGs is as a transportation fuel. On-road engines are generally optimized to operate over a wide range of speeds and loads and over transient cycles. They are also equipped with extensive sensors and actuators for both operational control and on-board diagnostics, along with an engine control unit that can provide real-time control over the engine performance. While stationary power generation units use the same basic combustion strategy—spark-ignited (SI) or compression ignition (CI) four-stroke cycle engines—the operating demands and installed capabilities are typically very different from on-road engines. Many current engines used in stationary power generation applications are focused on robustness, low net operating cost, and reliability. Transient demands are limited to relatively slow changes in load, and the power density of the engine is less of a barrier compared with vehicle engines. Many of these
engines retain carburetor-like fueling, in part because of the very low fuel pressures of some NG distribution networks. The engines are typically lean-burn and are not consistently equipped with sensors to monitor the air–fuel equivalence ratio. As such, while the fundamental effects on the combustion of H$_2$ addition to the NG will not change, the impacts of HCNGs on engine performance will differ significantly from those faced by more widely studied on-road engines.

### Table 1. Comparing properties of CH$_4$, H$_2$, and selected blends at environment temperature of 298 K and environment pressure of 1.01 bar.

<table>
<thead>
<tr>
<th>Properties</th>
<th>CH$_4$</th>
<th>HCNG10</th>
<th>HCNG20</th>
<th>HCNG30</th>
<th>H$_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric H$_2$%</td>
<td>0</td>
<td>10</td>
<td>20</td>
<td>30</td>
<td>100</td>
</tr>
<tr>
<td>Mass H$_2$%</td>
<td>0</td>
<td>1.4</td>
<td>3</td>
<td>5.1</td>
<td>100</td>
</tr>
<tr>
<td>Fuel energy * of H$_2$ fraction in the mixture</td>
<td>0</td>
<td>3.2</td>
<td>7.2</td>
<td>12.0</td>
<td>100</td>
</tr>
<tr>
<td>Gravimeter energy density * [MJ/kg]</td>
<td>50.0</td>
<td>51.0</td>
<td>52.1</td>
<td>53.6</td>
<td>120</td>
</tr>
<tr>
<td>Volumetric energy density * [MJ/m$^3$]</td>
<td>32.6</td>
<td>30.3</td>
<td>28.1</td>
<td>25.8</td>
<td>9.8</td>
</tr>
<tr>
<td>Stoichiometric air–fuel mass ratio</td>
<td>17.2</td>
<td>17.5</td>
<td>17.9</td>
<td>18.5</td>
<td>34.3</td>
</tr>
<tr>
<td>Energy * content for stoichiometric mixture [kJ/mol]</td>
<td>76.3</td>
<td>76.1</td>
<td>75.9</td>
<td>75.7</td>
<td>72</td>
</tr>
<tr>
<td>Laminar flame speed at ambient [cm/s]</td>
<td>33.7</td>
<td>35.8</td>
<td>38.5</td>
<td>42.1</td>
<td>120</td>
</tr>
<tr>
<td>Laminar flame speed at representative conditions (Tu<del>800 K, P</del>20 bar, EQR = 0.85)</td>
<td>38.3</td>
<td>40.6</td>
<td>48.75</td>
<td>59.4</td>
<td>-</td>
</tr>
</tbody>
</table>

* Fuel energy and energy density values are based on the lower heating value (LHV).

The addition of H$_2$ to NG has a direct impact on the fundamental combustion process. The flame velocity of H$_2$ is more than that of CH$_4$ and the other primary constituents of NG (C$_2$H$_6$ and C$_3$H$_8$); hence, adding H$_2$ to NG will increase the flame velocity in a premixed charge [7]. This is a result of both an increase in laminar flame speed [8] and increased wrinkling in the flame, leading to a larger flame surface area and a greater burning rate under both lean and stoichiometric conditions [9]. Another effect of adding hydrogen to natural gas is reduced quenching distance as the quenching distance of hydrogen is 1/3 of that of natural gas [7]. H$_2$ also has a higher auto-ignition temperature than NG; this leads to HCNGs having a stronger resistance to engine knock, which occurs when end gases auto-ignite ahead of the flame front in a premixed charge engine [10].

Using HCNGs as the fuel in SI engines impacts engine performance and engine-out emissions compared with NG. The increased flame velocity reduces the burn duration at a given premixed equivalence ratio, leading to the potential to improve efficiency but also increasing combustion noise and harshness through higher rates of pressure rise. Combined with lower unburned fuel emissions from shorter quench distances, these effects have led to most researchers reporting that adding H$_2$ to NG can increase thermal efficiency [11,12]. H$_2$’s stronger anti-knock propensity than NG offers potential to optimize the combustion phasing and to increase the compression ratios to achieve higher thermal efficiency. For a fixed spark timing, however, the increased burning speed of HCNG mixtures leads to more advanced combustion and higher end-gas temperatures and pressures. This increases the potential for damaging levels of engine knock [10], if the phasing of the combustion event is not carefully controlled.

For premixed NG engines, combustion instability and cycle-by-cycle variations limit the efficiency benefits that can be developed from lean premixed combustion. HCNGs’ higher flame speed leads to a wider flammability range, reducing cycle-by-cycle variations in lean burn engines [7], hence increasing combustion stability, potentially leading to an increase in output for the same fuel consumption [13].

Along with the combustion performance, adding H$_2$ to the NG fuel in a spark-ignition engine will also directly impact pollutant emissions. In addition to the lower CO$_2$ from the reduced carbon content of the fuel, oxides of nitrogen (NO$_x$) as well as combustion by-products (including unburned hydrocarbons, HC, and carbon monoxide, CO) are influenced. Researchers report that as the hydrogen fraction increases, a decrease in HC
and CO concentration in exhaust gases can be seen [13–17]; especially at high loads, HCNGs hardly emit any significant HC and CO [4]. Adding H₂ to CH₄ will reduce CH₄ emissions as it decreases the overall CH₄ in the fuel. In addition, the faster burn rate and shorter quench distances of H₂ should result in less unburned fuel emissions, hence lower CH₄ emissions for a given fuel–air equivalence ratio.

While the combustion improvement offered by HCNGs reduces combustion by-product emissions relative to NG, it also increases combustion temperature, leading to increased emissions of NOₓ [15,18,19], which are larger than would be expected as a result of the increase in adiabatic flame temperature alone [20]. By increasing the excess air ratio (leaner combustion, so lower temperature) to take advantage of HCNGs’ wider flammability range, the increases in NOₓ emissions can be minimized [21].

While there have been studies on the use of HCNGs in engines, especially in transportation applications, the use of hydrogen–methane blends in the power generation sector is less investigated. Like many jurisdictions around the world, the province of British Columbia in Canada has committed to reduce its GHG emissions by 80% compared with its levels in 2007 by 2050 [22], with H₂ having an important role to play [23] including through addition to the existing gas distribution network. While H₂ added to the NG network can reduce CO₂ emissions at the end use, its impact on the energy conversion devices needs to be well understood. NG fueled stationary and back-up generators are particularly sensitive to changes in fuel composition, so the impact of HCNGs needs to be carefully evaluated.

This work investigates the application of HCNGs in stationary engines used in CCHP systems. For these uses, it is important to understand the sensitivity of both power output and the thermal energy available (from both coolant and exhaust streams) to variations in H₂ content. The primary focus is on understanding the effects of changes in the fuel from pure methane to hydrogen–methane blends, up to 30% volumetric fraction of hydrogen (HCNG30) in a representative engine. The work includes the development of a new set of combustion calibration factors for HCNG mixtures for use in an industry-standard 1-D modelling environment (GT-SUITE™). The potential from spark timing and fuel/air ratio control to optimize the engine performance for increased thermal efficiency without impairing power output with HCNGs is then assessed. Finally, CO₂ emissions, engine performance, and energy budget plots are presented for use in CCHP or other system models.

2. Methodology

For this work, a 100 kW 9L commercial generator’s engine (Generac SG100, Generac Inc., Waukesha, WI, USA) was modelled and validated using available data. The configuration and calibration data for the engine were extracted from publicly available data sheets for this engine [24] (p. 1); key parameters are presented in Table 2.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value [Unit]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>V8</td>
</tr>
<tr>
<td>Displacement</td>
<td>8.9 [L]</td>
</tr>
<tr>
<td>Bore</td>
<td>114 [mm]</td>
</tr>
<tr>
<td>Stroke</td>
<td>108 [mm]</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>9.9:1</td>
</tr>
<tr>
<td>Brake mean effective pressure (BMEP)</td>
<td>775 [kPa]</td>
</tr>
<tr>
<td>Equivalence ratio</td>
<td>0.85</td>
</tr>
<tr>
<td>Intake air method</td>
<td>Naturally aspirated</td>
</tr>
<tr>
<td>Brake power</td>
<td>104 [kW]</td>
</tr>
<tr>
<td>Exhaust temperature</td>
<td>986 [K]</td>
</tr>
<tr>
<td>Firing order</td>
<td>1-8-7-2-6-5-4-3</td>
</tr>
</tbody>
</table>

1 Calculated from stated air and fuel flow rates at rated power condition [24] (p. 1).
The engine model was developed in the GT-SUITE™ modelling environment, release 2021.01. GT-SUITE™ is a widely used multi-physics simulation package that was originally developed to simulate internal combustion engines. For the purposes of the work reported here, the primary focus was on the engine and combustion system. Equations (1) and (2) were used to calculate the brake thermal efficiency (BTE) and equivalence ratio (EQR) in this work, in which m is mass, LHV stands for lower heating value, and brake power is the mechanical power generated by the engine and available at the crankshaft. The “stoich” suffix shows the values in stoichiometric conditions, while “actual” stands for the actual values used in the chemical equations.

\[
\text{BTE} = \frac{\text{Brake power}_{\text{Fuel}}}{\text{LHV}_{\text{Fuel}}} \quad (1)
\]

\[
\text{EQR} = \frac{\left( \frac{\text{Air}}{\text{Fuel}} \right)_{\text{stoich}}}{\left( \frac{\text{Air}}{\text{Fuel}} \right)_{\text{actual}}} \quad (2)
\]

2.1. Combustion Model

One of the main aims of the project was to provide an initial assessment of the effects of hydrogen addition on engine performance using a predictive combustion model. Of the various options available within GT-SUITE™, the SI-Turb model was identified as the most suitable for this work because it predicts the burn rate for homogeneous charge engines and incorporates fuel composition effects through the laminar flame speed. The SI-Turb model uses a two-zone combustion model in which the air–fuel mixture inside the combustion chamber consists of an unburned zone and a burned zone. The model simulates the flame as a turbulent mass entrainment process followed by a burn-up process behind the flame front. Flame propagation is a function of turbulent flame speed ($S_T$) and the laminar flame speed ($S_L$). Of these, the turbulent flame speed is a function of engine geometry and charge motion and is only weakly dependent on fuel composition. The laminar flame speed is strongly dependent on the composition of the fuel. The main formulas used in the SI-Turb model where fuel composition is incorporated are shown in Equations (3) to (5) [25]. Equation (3) demonstrates the mass entrainment rate of unburned gas, which is the rate at which the unburned mixture is transported across the interface of burned and unburned zones. It is dependent on the flame front area and the entrainment velocity, which is defined as the summation of the laminar and turbulent flame speeds [26].

\[
\frac{dm_e}{dt} = \rho_u A_e (S_T + S_L) \quad (3)
\]

In Equation (3), $m_e$ is the entrained mass, $A_e$ represents the surface area at flame front, $S_T$ is turbulent flame speed, $\rho_u$ is the unburned density, $S_L$ is laminar flame speed, and $t$ represents time. Once the rate at which unburned mixture is entrained into the flame has been established (Equation (3)), the next step is to identify the rate at which the entrained but unburned mass behind the flame front [26], divided by a time constant ($\tau$) representative of the combustion rate within the flame zone.

\[
\frac{dm_b}{dt} = \frac{(m_e - m_b)}{\tau} \quad (4)
\]

The entrained mass combustion (burn-up) process is a function of the rate at which a flame moves through the mixture (laminar flame speed) and the dissipation of energy due to turbulence; this is demonstrated in Equation (5) [26], in which $\tau$ is the time constant and $\lambda$ is the Taylor microscale length.

\[
\tau = \frac{\lambda}{S_L} \quad (5)
\]
Combining Equations (4) and (5) indicates that the rate of fuel consumption and associated energy release are increased for higher laminar flame speeds and reduced for longer Taylor length scales. The Taylor length scale relates to the viscosity of the charge and the structure of the local flow field, and as such, it will only be weakly impacted by changes in fuel composition. The SI-Turb model also includes other parameters—including spark kernel size—that are also only weak functions of fuel composition under the range of HCNG blends evaluated here. Similarly, the value of the turbulent flame speed ($S_T$ in Equation (3)) is dependent on the flow properties such as swirl and is not strongly dependent on the fuel composition.

2.2. Hydrogen–Methane Implementation

In the GT-SUITE\textsuperscript{TM} SI-Turb combustion model, the laminar flame speed is calculated using a modified version of the Keck and Metghalchi equation, shown in Equations (6) and (7)\cite{25}.

$$S_L = \left( B_m + B_\phi (\phi - \phi_m)^2 \right) \left( \frac{T_u}{T_{\text{ref}}} \right)^\alpha \left( \frac{P_u}{P_{\text{ref}}} \right)^\beta \cdot f(\text{Dilution})$$

$$f(\text{Dilution}) = 1 - 0.75 \times \text{DEM}(1 - (1 - 0.75 \times \text{DEM} \times \text{Dilution})^7)$$

In Equation (6), $S_L$ represents laminar flame speed, $B_m$ is the maximum laminar flame speed in a reference state ($T_{\text{ref}} = 298$ K, $P_{\text{ref}} = 101325$ Pa), $B_\phi$ is the laminar flame speed roll-off value, $\phi$ shows the in-cylinder equivalence ratio, $\phi_m$ represents the fuel/air equivalence ratio at maximum laminar flame speed, $T_u$ represents the temperature of the unburned gas, $\alpha$ shows the temperature exponent, $\beta$ is the pressure exponent, and $f(\text{Dilution})$ is the dilution effect. In Equation (7), DEM is the dilution effect multiplier.

To use Equation (8) in GT-SUITE\textsuperscript{TM}, knowing $B_m$, $B_\phi$, $\phi_m$, $\alpha$, and $\beta$ is necessary. While the base model includes these parameters for most fuels, they are not currently available for blends of $H_2$ and NG. In this work, the constants were calculated in CHEMKIN\textsuperscript{TM} using the Gri-Mech 3.0 mechanism. The validity of this approach is presented by Caputo et al.\cite{27} and Costa et al.\cite{28}. The constants used in this work were calculated for methane, HCNG10 (10% volumetric fraction of hydrogen), HCNG20, and HCNG30 by Costa et al.\cite{28}.

The temperature and pressure exponents were calculated based on formulations presented in different references. The temperature and pressure exponents defined in Equations (9) and (10) related to the “power law” equation at 1 atm and 298 K, which is the simplest and most widely used form of the empirical equation and is shown in Equation (8)\cite{29}. The same method, as described above, was used for calculating the pressure and temperature exponents of hydrogen shown in Equations (11) and (12) based on the work of Millo et al.\cite{30}. In the power law equation, $S_{L0}$ is the laminar flame velocity measured at $T_u = T_0$ and $P_u = P_0$ for a given equivalence ratio. Table 3 lays out the coefficients used in in Equations (9)–(12).

$$S_L(T_u, \phi, P_u) = S_{L0} \left( \frac{T_u}{T_0} \right)^\alpha \left( \frac{P_u}{P_0} \right)^\beta$$

$$\alpha(\phi)_{CH_4} = \left[ a_2 \phi^2 - a_1 \phi + a_0 \right]_{CH_4}$$

$$\beta(\phi)_{CH_4} = \left[ -b_2 \phi^2 + b_1 \phi - b_0 \right]_{CH_4}$$

$$\alpha(\phi)_{H_2} = \left[ a_2 \phi^2 + a_1 \phi + a_0 \right]_{H_2}$$

$$\beta(\phi)_{H_2} = \left[ b_2 \phi^2 + b_1 \phi + b_0 \right]_{H_2}$$
Table 3. Coefficients proposed for $\alpha$ and $\beta$ exponents for methane [29] and hydrogen [30].

<table>
<thead>
<tr>
<th>Fuel</th>
<th>$a_2$</th>
<th>$a_1$</th>
<th>$a_0$</th>
<th>$b_2$</th>
<th>$b_1$</th>
<th>$b_0$</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH$_4$</td>
<td>4.9199</td>
<td>10.287</td>
<td>6.9258</td>
<td>1.3712</td>
<td>2.6808</td>
<td>1.7492</td>
</tr>
<tr>
<td>H$_2$</td>
<td>0.105</td>
<td>-3.135</td>
<td>6.514</td>
<td>0.177</td>
<td>0.636</td>
<td>-1.449</td>
</tr>
</tbody>
</table>

To calculate the exponents for hydrogen–methane mixtures, Equations (13) and (14) presented by Farzam et al. [31] were used, where $x_i$ is the molar fraction of the fuels existing in the blend, and one set of sample results are presented in Table 4.

\[
\alpha_{\text{blend}} = \sum x_i \alpha_i \tag{13}
\]

\[
\beta_{\text{blend}} = \sum x_i \beta_i \tag{14}
\]

Table 4. Coefficients proposed for $\alpha$ and $\beta$ exponents for hydrogen–methane blends.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>$\alpha$</th>
<th>$\beta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH$_4$</td>
<td>1.56</td>
<td>-0.44</td>
</tr>
<tr>
<td>HCNG10</td>
<td>1.75</td>
<td>-0.46</td>
</tr>
<tr>
<td>HCNG15</td>
<td>1.85</td>
<td>-0.47</td>
</tr>
<tr>
<td>HCNG20</td>
<td>1.94</td>
<td>-0.48</td>
</tr>
<tr>
<td>HCNG25</td>
<td>2.04</td>
<td>-0.49</td>
</tr>
<tr>
<td>HCNG30</td>
<td>2.13</td>
<td>-0.5</td>
</tr>
</tbody>
</table>

2.3. Laminar Flame Speed Validation

To validate the blended fuel laminar flame speeds, the available literature data were compared with the values calculated in the methodology used in this work. As no literature data were found that covered laminar flame speeds under engine relevant temperatures and pressures, the comparisons were conducted under normal temperature and pressure (NTP) conditions. Representative correlations available in the literature were as presented in Equations (15) to (17) [32], in which $\phi$ is the EQR, $x$ represents the hydrogen blending ratio, and $S_L$ is calculated for NTP conditions.

\[
S_{L\text{CH}_4}(\phi) = -150.84\phi^3 + 287.6\phi^2 - 96.327\phi - 1.2924 \tag{15}
\]

\[
S_{L\text{H}_2}(\phi) = 51.902\phi^3 - 394.46\phi^2 + 835.14\phi - 267.07 \tag{16}
\]

\[
\frac{[S_{L\text{HCNG}}(\phi) - S_{L\text{CH}_4}(\phi)]}{[S_{L\text{H}_2}(\phi) - S_{L\text{CH}_4}(\phi)]} = 0.00737 \exp \left( \frac{x}{20.38} \right) + 0.00334 \tag{17}
\]

The results of Equation (17) were compared against the experimental results, and Ma et al. [29] reported that the results agreed well when working with low volumetric hydrogen fractions (RH) ($0 < \text{RH} < 30\%$) and high hydrogen fractions ($60\% < \text{RH} < 100\%$). Another correlation was tested by Ma et al. [32], presented in Equation (18), which proved to be accurate for lean and stoichiometric conditions regardless of the hydrogen content in the HCNGs. It is noteworthy that to be able to use a correlation in the GT model, it is necessary that they are presented in the form of Equation (6).

\[
S_{L\text{HCNG}}(\phi, x) = \frac{1}{x/S_{L\text{H}_2}(\phi) + (1-x)/S_{L\text{CH}_4}(\phi)} \tag{18}
\]

Equations (15) and (18) were compared against the laminar flame speed of methane and HCNGs under NTP conditions calculated using the approach in the current work; this comparison is presented in Table 5. The differences between the current approach and the Ma et al. correlations tended to increase as the hydrogen volumetric fraction increased.
but were still well within the acceptable range given the deviation in Ma et al. [32] from
the results published in the literature and shown in Table 5; these values are in general
agreement with other literature sources (e.g., [33]).

Table 5. Laminar flame speed validation for methane–air and HCNG–air mixtures with \( P = 1 \) atm,
\( T = 300 \) K, and \( \text{EQR} = 1 \) using results from GT-SUITE™ and available correlations.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>( S_L ) [m/s] This Work in GT-SUITE™</th>
<th>Correlation Presented by Ma et al. [32]</th>
<th>Relative Error [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{CH}_4 )</td>
<td>0.395</td>
<td>0.391</td>
<td>0.01</td>
</tr>
<tr>
<td>( \text{HCNG10} )</td>
<td>0.42</td>
<td>0.43</td>
<td>–2.3</td>
</tr>
<tr>
<td>( \text{HCNG15} )</td>
<td>0.43</td>
<td>0.45</td>
<td>–4.4</td>
</tr>
<tr>
<td>( \text{HCNG20} )</td>
<td>0.45</td>
<td>0.47</td>
<td>–4.25</td>
</tr>
<tr>
<td>( \text{HCNG25} )</td>
<td>0.47</td>
<td>0.5</td>
<td>–6</td>
</tr>
<tr>
<td>( \text{HCNG30} )</td>
<td>0.49</td>
<td>0.52</td>
<td>–5.8</td>
</tr>
</tbody>
</table>

2.4. Model Development

The work aims to evaluate the effects of \( \text{H}_2 \) blending in natural gas for a representative
generator; it was not necessary to precisely model a specific engine. To reliably validate
the model of the SG100 generator, publicly available specification sheets, owner’s manuals,
and engine drawings were evaluated; key values are presented in Table 1. The overall
configuration of the engine and key geometric parameters (e.g., intake manifold diameter of
7 cm) were based on SG100 drawings [34]. Based on the geometry collected from the engine
configuration data, a complete GT-SUITE™ model of the V-8 engine was developed. Where
information was missing, representative dimensions for similar engines were identified.
Key parameters used in the model included a connecting rod length of 220 mm; this was
assumed based on similar-sized engine dimensions. The geometry of the combustion
chamber was modelled as a flat piston with a horizontal cylinder head fire deck. The spark
plug was centrally mounted, and the engine was equipped with two valves per cylinder
(intake: 55 mm diameter, 11.6 mm maximum lift, exhaust: 55 mm diameter, 11.8 mm
maximum lift). The intake and exhaust port flow geometry was adapted from an existing
engine model for an equivalent displacement (intake runners part 1: 75 mm diameter,
50 mm long; intake runners part 2: 70 mm diameter, 30 mm long; exhaust runners: 70 mm
diameter, 85 mm long). The in-cylinder flow was modelled using the default parameter
values in GT-SUITE. To calculate the cylinder wall heat transfer, the Woschni model [35]
was used, which combined the radiation heat transfer in the convection heat transfer [25].

As a naturally aspirated engine, the air handling system development included an
intake air filter, fuel–air mixer, and throttle upstream of the intake manifold. Separate intake
manifolds for the two banks of the V-8 engine were simulated using volumes estimated
from publicly available data sheets. The data sheets and maintenance manuals identified
key dimensions including the throttle valve and air induction system diameter of 76.2 mm
(3”). A particular feature of the engine is that it has two separate exhaust systems, one
for each bank with the exhaust ducts (70 mm diameter). In the base engine, a downdraft
carburetor is used to add fuel to the intake air upstream of the throttle plate. To provide
more flexible control over the air and fueling rate, the model used a flow-control injector to
maintain a constant fueling flow rate throughout the cycle, ensuring homogeneous fuel
composition between the cylinders.

A simple control system was developed for the model. The model was run in speed
control mode, such that the simulation maintained a constant speed, but the torque varied
with fueling quantity and throttle position. In the base model, the air mass flow rate before
fuel induction was set as the primary control target, and the throttle valve was adjusted to
maintain this flow. When HCNGs were introduced, the stoichiometric air to fuel ratio was
calculated for each mixture, and using the constant air mass flow rate, the fuel mass flow
rate for each of the HCNG blends was calculated. The fueling rate was pre-defined for a
given air flow rate to maintain a target equivalence ratio (Equation (2)). It is noteworthy that this is not how most stationary power generation engines of this size are configured, but it allowed the fueling control to be more precise in the model.

2.5. Model Validation

In this work, the modelled engine was for a stationary application where the engine was designed to run at a single fixed speed. For validation, two load points at the fixed speed were available from specification sheet performance data. The available data were for two ratings: standby (short-duration maximum load) and prime (longer-duration operation at slightly reduced maximum power). For these two conditions, the fuel consumption, intake air flow rate, brake mean effective pressure (BMEP), and exhaust temperature were available and used to set the key parameters in the model. Specifically, the throttle valve was adjusted to achieve the target air flow, fueling was fixed at the equivalence ratio calculated from the specification sheet, and then spark timing was adjusted to achieve the target brake power. The exhaust temperature was not directly controlled and was used as an overall indicator of how well the model aligned with the experimental findings. Table 6 lists the results for the validation points. It is noteworthy that without detailed engine performance and in-cylinder pressure data for the specific engine, uncertainty remains on how closely the model represents the specific engine. However, the validation was sufficient for the purpose as it demonstrated equivalent performance for a representative generator to be used in a commercial power generation application. Furthermore, the model was only validated using natural gas data; the impacts of changing the fuel composition through the addition of H\textsubscript{2} were only predicted based on the modifications to the combustion model validated in Section 2.3.

Table 6. Modelling results comparison with SG100 specification sheet data for methane with equivalence ratio of 0.85 and 30° BTDC spark timing: brake power, BSFC, air mass flow rate, BMEP, and exhaust temperature with errors; values from SG100 specification sheet are rounded [24].

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Prime—Model</th>
<th>Prime—Spec Sheet</th>
<th>Prime—Error [%]</th>
<th>Standby—Model</th>
<th>Standby—Spec Sheet</th>
<th>Standby—Error [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brake power [kW]</td>
<td>105</td>
<td>104</td>
<td>0.96</td>
<td>117</td>
<td>115</td>
<td>1.74</td>
</tr>
<tr>
<td>BSFC [g/kWh]</td>
<td>215</td>
<td>218</td>
<td>−1.38</td>
<td>210</td>
<td>214</td>
<td>−1.87</td>
</tr>
<tr>
<td>Air mass flow rate [g/s]</td>
<td>125</td>
<td>125</td>
<td>0</td>
<td>132</td>
<td>138</td>
<td>−4.35</td>
</tr>
<tr>
<td>BMEP [kPa]</td>
<td>794</td>
<td>775</td>
<td>2.45</td>
<td>884</td>
<td>886</td>
<td>−0.22</td>
</tr>
<tr>
<td>Exhaust temperature [K]</td>
<td>1022</td>
<td>986</td>
<td>5.0</td>
<td>1031</td>
<td>1005</td>
<td>3.55</td>
</tr>
</tbody>
</table>

2.6. Model Utilization

A key aim of this work is to understand the effects of H\textsubscript{2} addition to evaluate approaches to offset the expected drop in performance from the reduction in fuel volumetric energy density (see Table 5). To understand these effects, a sequence of tests was conducted, as summarized in Table 7. To represent an actual carbureted engine, the model was run with no changes to fuel volumetric flow rate and spark timing for different fuel compositions: this is the baseline case in Table 7. Then, the fuel volume flow was increased so that the net equivalence ratio was maintained at its constant level. This could be achieved through the use of a wide-band oxygen sensor in the exhaust stream and using that to adjust the flow rate of fuel: this is the EQR control case. Finally, the effects of adjusting the phasing of the combustion to account for the changes in flame speed can be achieved through re-optimization of the spark timing. This is the fully optimized case, where the air and fuel flow are adjusted to maintain the target EQR and the spark timing is adjusted to achieve the maximum brake torque (MBT) phasing.
Table 7. Summary of studied cases.

<table>
<thead>
<tr>
<th>Case Name</th>
<th>Fuel Volumetric Flow Rate</th>
<th>Throttle Position</th>
<th>Spark Timing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline</td>
<td>Fixed</td>
<td>Set throttle to match air flow from NG</td>
<td>Fixed</td>
</tr>
<tr>
<td>EQR control</td>
<td>Adjust fueling to maintain 0.85 EQR</td>
<td>Set throttle to match air flow</td>
<td>Fixed</td>
</tr>
<tr>
<td>Optimized (EQR + MBT)</td>
<td>Adjust fueling to maintain 0.85 EQR</td>
<td>Set throttle to match air flow</td>
<td>Adjust timing to MBT</td>
</tr>
</tbody>
</table>

3. Results

The functional model was assessed with a focus on full load conditions, as these are most representative of the conditions to be encountered in the stationary power generation application. All the cases were studied for a fixed speed, 1800 RPM, and a targeted BMEP of 7.75 bars. Pure methane and blends of 10, 20, and 30% by volume H₂ in CH₄ were evaluated. As summarized in Table 7, three different sets of results were evaluated. Under the baseline conditions, the fuel volumetric flow rate was fixed. As the fuel volumetric energy density decreased with increasing H₂ content (Table 1), this resulted in a reduction in the chemical energy delivered to the engine. This can be quantified as the fuel power, which is the mass flow rate of fuel multiplied by its lower heating value, as shown in Figure 1. Fuel power (fuel energy rate) is defined as a measure of the total chemical energy delivered to the engine. As such, it is a function of the heating value of the fuel and the total mass of fuel.

![Figure 1](image-url). Fuel power as a function of fuel hydrogen content, at full load prime rated condition comparing the baseline, air flow control, and spark timing control cases.

To compensate for the reduction in fuel energy with fixed fuel volumetric flow rate, a simple control system could be used to maintain a constant EQR. This resulted in increasing the fuel volume for the HCNG blends and, as shown in Figure 1, nearly equivalent energy delivery to the engine. Under this condition, the throttle position was adjusted to maintain air flow, resulting in a slight increase in intake manifold pressure. As the testing was conducted under "prime" conditions, the engine did have some margin on intake pressure (up to the "standby" rating). Finally, the third stage investigated the MBT spark timing that allows spark timing adjustment based on the fuel in the model; however, this did not significantly impact the fuel energy delivered to the engine.

3.1. Air and Fuel Handling

Controlling the intake air, via throttle control, and fuel mass flow rate, via EQR control, can lay the foundations for solving complications that arise as hydrogen is added to the
fuel for the baseline engine. Figure 2 demonstrates the EQR and fuel mass flow rate as a function of fuel composition for the studied cases. As EQR was kept constant in the EQR control and spark timing control cases, the fuel mass flow rate was increased compared with that of the baseline case. Hence, the drop in power in the baseline case due to H\textsubscript{2} addition could be controlled.

![Figure 2](image1)

Figure 2. (a) EQR as a function of fuel hydrogen content, at full load prime rated condition comparing the baseline, air flow control, and spark timing control cases. (b) Fuel mass flow as a function of fuel hydrogen content, at full load prime rated condition comparing the baseline, air flow control, and spark timing control cases.

3.2. MBT Timing

To adjust the spark timing, a spark timing sweep, presented in Figure 3, was conducted to identify the MBT spark time of each studied fuel. As demonstrated in Figure 3b, as the hydrogen content increased, the MBT spark timing was delayed, moving closer to the top dead center for the piston motion. This was a result of the shorter burn duration and shorter ignition delay with hydrogen addition. This generally corresponded with the experimental findings in the work by Huang et al. [15].

![Figure 3](image2)

Figure 3. (a) Spark timing sweep as a function of fuel hydrogen content, at full load prime rated condition with air flow control. (b) MBT spark timing as a function of fuel hydrogen content, at full load prime rated condition with air flow control.
While the spark timing was adjusted to identify MBT (Figure 3), the fuel flow was constant. As a result, the torque output, brake power, and BTE increased slightly, as shown in Figure 4. Over the range of spark timings considered (5–10 °CA), at most, these differences were relatively small. However, the retarded spark timing had a substantial impact on in-cylinder temperature; without modification in timing, the peak HRR increased, and both burned and unburned gas temperatures increased as shown in Figure 5. As a result, the MBT timing should reduce both the knock propensity and the peak in-cylinder NO\textsubscript{X} concentration in the cylinder; these are strongly dependent on unburned end-gas temperature and combustion temperature, respectively. As the model was not validated for knock propensity and NO\textsubscript{X} emissions, the assessment was qualitative and speculative. However, understanding the mechanisms leading to knock and peak in-cylinder NO\textsubscript{X} concentration suggests that MBT will tend to mitigate both effects. In other words, as H\textsubscript{2} is added to the fuel, peak NO\textsubscript{X} concentration will increase without spark timing control. However, if the spark timing is controlled, the peak in-cylinder NO\textsubscript{X} concentration will be less than that of the case in which spark timing is not optimized for mixtures with more than 20% volumetric hydrogen. Further model development and validation should be conducted in the future to better understand these important factors.

![Figure 4](image-url)

**Figure 4.** Brake power as a function of fuel hydrogen content, at full load prime rated condition comparing the baseline, air flow control, and spark timing control cases.

![Figure 5](image-url)

**Figure 5.** Impacts of fixed spark timing at constant air flow and varying levels of fuel hydrogen content on in-cylinder temperatures. (a) Unburned zone temperature and (b) burned zone temperature, both as functions of crank angle degrees at full load prime rated power.
3.3. Combustion Process Evaluation

A key consideration for a stationary generator running at near full load is to avoid excessive cylinder pressures, significant increases in engine-out NO\textsubscript{x}, and increases in knocking propensity. While the model was not validated for either NO\textsubscript{x} or knock, the in-cylinder pressure and apparent heat release rate could be used to infer the relative effects of adding H\textsubscript{2} to the CH\textsubscript{4}. Figure 6 plots the HRR under all three studied cases—baseline, EQR control, and optimized. In the baseline case, the throttle angle, fuel volumetric flow rate (9.6 L/s), and spark timing (−30 CAD) were fixed.

![Figure 6](image_url)

**Figure 6.** Heat release rate (HRR) as a function of fuel hydrogen content and crank position at full load for (a) the baseline (fixed fuel volume) case, (b) the EQR control (increased fueling to maintain EQR) case, and (c) the fully optimized case with both EQR and spark timing control.

The heat release rate (HRR) plot for the modelled engine with EQR control, shown in Figure 6b, clearly indicates that as the H\textsubscript{2} content of the fuel was increased, the burn rate and the maximum HRR increased, while the ignition delay decreased. This could be explained with hydrogen’s faster laminar flame speed compared with that of methane. It is evident from Figure 6c that at MBT spark timing, the maximum HRR and burn rates were very similar for all the studied fuels. Based on the HRR graphs presented, initially, without compensating for the loss in fuel energy, the peak HRR was reduced as we added H\textsubscript{2} to the fuel, the ignition delay was increased, and the combustion rate decreased. This was due
to a combination of lower fuel energy inducted and a leaner overall mixture. As $H_2$ was added to $CH_4$, the maximum HRR reduced by more than 35% between $CH_4$ and HCNG30.

Along with the heat release rate, the in-cylinder pressure and temperature conditions were also of concern. The maximum cylinder pressure plot (Figure 7a) shows an ~35% increase as $H_2$ was added to the fuel in the EQR control case. By controlling the spark timing, the increase in peak cylinder pressure could be controlled and kept constant. Due to the drop in the total energy output for the baseline case, the peak cylinder pressure and temperature both declined. The peak in-cylinder temperature plot (Figure 7b) was expected to correlate with higher NO$_x$ formation and emissions. As $H_2$ was added to the fuel, the peak in-cylinder temperature could increase in the EQR control case. This increase in temperature could be controlled with spark timing control. It was expected that in the spark timing control case, the peak in-cylinder NO$_x$ among various fuels would be mostly higher than that of the EQR control case, yet it was almost constant. As a result of the studied cases, spark timing control was chosen as the optimum case that enabled keeping the brake power constant while controlling maximum NO$_x$ concentrations and maximum cylinder pressure.

![Figure 7](image.png)

(a) Maximum cylinder pressure as a function of fuel hydrogen content, at full load prime rated condition comparing the baseline, air flow control, and spark timing control cases. (b) Peak in-cylinder temperature as a function of fuel hydrogen content, at full load prime rated condition comparing the baseline, air flow control, and spark timing control cases.

3.4. Summary of Performance Outputs

In the baseline case, the EQR reduction as $H_2$ was added to the fuel (Figure 8) could be explained with a lower volumetric density of hydrogen. This caused a drop in brake power as the available fuel was reducing. The reduction in power is in agreement with other similar studies [36]. Hence, EQR needs to be controlled, which is an important consideration. EQR control can be achieved with a low-cost wide-band oxygen sensor.

The impact of increasing EQR was negligible on the intake manifold pressure, as demonstrated in Figure 9. Hence, to make an engine that is HCNG-compatible, the available fuel flow control needs to be introduced, and the intake air stream needs to be slightly oversized (for a naturally aspirated engine). In doing so, a small amount of throttling can be put on the engine when running on pure NG, and then as higher $H_2$ blends are introduced, the throttle can open more. If the manifold pressure in the EQR control case is kept constant and the same as the baseline case, a less than 1% reduction in torque for HCNG30 can be observed compared with the EQR control case results without manifold pressure control, which is in the acceptable uncertainty margins of the modelling work.
With fixing the EQR (0.85), the power drop caused by keeping the fuel’s volumetric flow rate constant can be avoided, and the brake power can be kept almost constant as hydrogen is added to the fuel. The same trend is visible in the BTE plot as shown in Figure 10a. EQR adjustment prevented the loss in brake power and slightly increased the BTE as H₂ was added to CH₄, as shown in Figures 8 and 10a, respectively. Adjusting the spark timing did not show a significant change the brake power and BTE. It (in addition to the EQR control) reduced the brake CO₂ emissions as presented in Figure 10b. (Brake-specific CO₂ emissions are the mass flow rates of pollutant per unit power output.) The H₂ content should reduce quench distances and lead to lower tailpipe CH₄ emissions (unburned fuel emissions were not investigated in this study), which is important from net GHG grounds due to its high GWP. Spark timing adjustment also helped keep maximum cylinder pressure approximately constant and avoided drastic changes in maximum NOₓ concentration as the H₂ content of the fuel increases, plotted in Figure 7.
3.5. Net energy Breakdown and Implementation in a CCHP System

Understanding the relative distribution of energy losses from the engine is important if the engine is to be combined with thermal energy recovery in a CCHP system. The energy losses based on the simulation work in this study are presented in Figure 11 under full load for the fully optimized case (MBT spark timing and fuel flow control to maintain EQR = 0.85). Pumping loss followed a descending trend as H\textsubscript{2} was added to the fuel, due to the increased throttle angle. The thermal energy available in the exhaust was decreased as the fuel became more H\textsubscript{2} rich. This could be explained with the faster burn rate of hydrogen, which leads to an increased in-cylinder temperature and hence higher in-cylinder heat transfer. The energy budget plot also reports the energy available to the CCHP system. The recoverable waste heat is a summation of the exhaust energy and the in-cylinder heat transfer. Based on Figure 11, the energy available to the CCHP system stayed at approximately 60% regardless of the fuel type.

Figure 11. Energy budget as a function of fuel hydrogen content at full load prime rated condition under EQR = 0.85 and MBT spark timing.

Based on the results of this study, brake power can be maintained as the H\textsubscript{2} content of the fuel increases, while the efficiency improves through the faster combustion rate. Due to the higher efficiency, the net waste energy—the combined energy lost through the exhaust...
and the engine coolant—is reduced. As shown in Figure 12, when H₂ was added to the fuel and the fuel flow and spark timing were controlled, the available thermal energy for recovery in a waste heat recovery system, such as CCHP, only decreased slightly. The predicted reduction in energy of less than 5% will not significantly impact the overall sizing or operation of a corresponding energy recovery system.

Figure 12. Thermal power available to a CCHP system as a function of fuel hydrogen content at full load prime rated condition under EQR = 0.85 and MBT spark timing.

To fully represent the performance of the stationary power generation unit, various modelling work was extended to include part-load operation. The corresponding fuel power (the lower heating value multiplied by the fuel mass) and the tailpipe CO₂ emissions are shown in Figure 13. As expected, with MBT timing and fixed EQR, the net fuel power required to achieve a target load was relatively insensitive to the H₂ content. As the carbon-to-energy ratio of the fuel was reduced, the CO₂ emissions decreased with higher H₂ content in the fuel but increased due to the reduced efficiency at part load. This work did not investigate the potential of the H₂ addition to allow a lower EQR before reaching the lean combustion limit. This offers the potential to reduce pumping losses, improving efficiency and further reducing CO₂ emissions in the part-load region.

Figure 13. (a) Fuel power as a function of fuel hydrogen content and various load at with fixed EQR (0.85) and MBT spark timing. (b) CO₂ emissions normalized by mechanical power output as a function of fuel hydrogen content and changing load at with fixed EQR (0.85) and MBT spark timing. The 100% load corresponds to the prime power rating of 104 kW.

The application of HCNG blends in a stationary generator is of great practical interest as many natural gas suppliers are anticipating adding relatively low concentrations of
H₂ to their gas distribution network. The results from this study have indicated that a properly configured engine, equipped with sensing and controls that would allow the EQR to be controlled and the spark timing to be adjusted, could maintain performance over a range of HCNG blends. Further work is needed to evaluate other important considerations, including NOₓ emissions and engine noise from the more rapid heat release. The potential to improve the efficiency through reduced throttling at part load and through higher compression ratios, enabled by HCNGs’ wider flammability range and higher knock resistance, respectively, should also be investigated. Finally, the material compatibility and durability of the fuel system components would need to be evaluated for HCNGs.

4. Summary and Conclusions

To assess the engine performance of a commercial power generation unit, a model in GT-SUITE™ environment was developed and presented in this work. The model’s performance under two operation modes (prime and standby) was validated against the product’s specification sheet. It was found that the errors for both validation points were less than 5%, which was considered acceptable for the evaluation to be conducted.

To implement the fuel change capability from methane to mixtures of hydrogen and methane, a methodology was developed using the works of previous researchers and the available literature. The developed methodology focused on calculating the laminar burning flame speed of methane and hydrogen–methane blends within the context of the two-zone combustion model used in the GT-SUITE™ simulation tool. The results for HCNGs were validated against the results of the available literature.

The engine evaluated in the work, which is representative of many commercial stationary engines, was fueled using a downdraft carburetor, which delivered a fixed volume of fuel. As the addition of H₂ reduced the volumetric energy density of the fuel, this resulted in a reduction in energy delivered to the engine and a corresponding loss in power: for 30% H₂ (by volume), the total fuel energy was reduced by 20%, and the power output from the engine decreased by 22% compared with NG.

To offset the negative effects of using HCNGs in the baseline engine, two methods can be used: (i) increasing the volumetric fuel flow to match EQR, which offsets the power loss without requiring significantly more air flow, and (ii) spark timing adjustment, which controls peak NOₓ concentrations and cylinder pressure. Controlling the equivalence ratio allowed a recovery of power but increases in cylinder pressures and temperatures due to more advanced combustion. Subsequently adjusting the spark timing enabled the optimization of the combustion phasing to maximize efficiency and avoid high peak cylinder pressures.

In summary, the following can be concluded from this study:

1. Compensation with EQR and MBT timing can enable the generator to retain equivalent performance (constant brake power and BTE as H₂ is added to CH₄) at lower CO₂ emission levels (more than 4.5% lower CO₂ emissions for HCNG30 fuel between the baseline and spark timing control cases);
2. Accounting for changes in the fuel volumetric density can offset the brake power reduction caused by H₂ addition by maintaining a constant global fuel–air ratio with a negligible impact on the intake manifold pressure;
3. Delaying the spark timing can be used to keep the cylinder pressure approximately constant and reduce the peak in-cylinder pressures and temperatures for HCNG blends;
4. To maintain good performance, minimize the peak in-cylinder NOₓ concentration increases, and avoid high cylinder pressures, knowledge of fuel composition (H₂ content) is needed to optimize the engine’s EQR and spark timing.

The results in this study were generated using a full engine system model adapted using generally accepted modelling approaches. Further work to validate the model over a wider range of conditions would be valuable, as would extending the work to provide accurate predictions of emissions—especially NOₓ—and engine knock propensity. As the deployment of H₂ in the NG grid increases, the potential for incorporating waste
heat recovery systems, such as a CCHP, should be assessed to quantify the overall energy efficiency and provide system-level insight. Finally, the impact of the implementation of the optimized engines as part of a CCHP system can then be evaluated to quantify their potential to reduce net GHG emissions as part of an integrated regional energy system that maximizes delivery of low-carbon energy through existing distribution networks.

**Author Contributions:** Conceptualization, G.P.-M.-C. and K.H.; methodology, K.H. and G.P.-M.-C.; software, K.H.; validation, K.H.; formal analysis, K.H.; resources, G.P.-M.-C.; writing—original draft preparation, K.H.; writing—review and editing, G.P.-M.-C.; visualization, K.H.; supervision, G.P.-M.-C.; project administration, G.P.-M.-C.; funding acquisition, G.P.-M.-C. All authors have read and agreed to the published version of the manuscript.

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**Abbreviations**

- $A_e$: Surface area at flame front
- AFR: Air to fuel ratio
- BTDC: Before top dead center
- BMEP: Brake mean effective pressure
- $B_{\phi}$: Laminar flame speed roll-off value
- CCHP: Combined cooling heat and power
- DEM: Dilution effect multiplier
- EQR: Equivalence ratio
- $f(Dilution)$: Dilution effect
- HCNG$x$: x percent volumetric hydrogen in the hydrogen–methane blend
- LHV: Lower heating value
- $m$: Mass
- $m_b$: Burned products mass
- $m_e$: Entrained mass
- MBT: Maximum brake torque
- NTP: Normal temperature and pressure
- $\Theta$CA: Degree crank angle
- $S_T$: Turbulent flame speed
- $S_L$: Laminar flame speed
- $S_{L0}$: Laminar flame speed at standard pressure and temperature
- $t$: Time
- $T_u$: Temperature of the unburned gas
- $x_i$: Molar fraction of the fuels existing in the blend
- $x$: Hydrogen blending ratio
- $\alpha$: Temperature exponent
- $\beta$: Pressure exponent
- $\tau$: Time constant
- $\lambda$: Taylor microscale length
- $\varphi$: In-cylinder equivalence ratio
- $\varphi_m$: Fuel/air equivalence ratio at maximum laminar flame speed
- $\rho$: Density
References


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