

Article

A Comprehensive Analysis of Hydrogen–Gasoline Blends in SI Engine Performance and Emissions

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Abstract: This study investigates the influence of adding hydrogen as an additive to gasoline in a four-stroke engine, utilizing comprehensive thermodynamic comparative analysis conducted with self-developed engine model. This research aims to assess the performance, emissions, and efficiency of the engine when using gasoline–hydrogen blends, and to provide insights into the potential benefits of this approach. First, the engine performance and emissions under different hydrogen blending levels were examined. A range of different air/fuel ratios (rich to lean) and varying percentages of hydrogen were considered. This systematic variation allowed for a detailed evaluation of the influence of hydrogen content on combustion efficiency, power output, and emissions characteristics. The analysis results included key parameters such as indicated specific fuel consumption and mean effective pressure. Additionally, the study focused on the range prediction of nitrogen oxide (NO_x) emissions, which are a critical environmental concern associated with internal combustion engines. The analysis of pressure and temperature profiles throughout the engine cycle shed light on the combustion characteristics and efficiency improvements associated with hydrogen addition. In terms of emissions, the study projected that all emissions were reduced except NO_x, which is highly dependent on hydrogen percentage, and might be reduced in some cases, but with the higher temperatures and pressures associated with hydrogen addition, in most cases, there is actually a NO_x increase, especially at higher engine loads.

Keywords: gasoline–hydrogen blends; emission reduction; nitrogen oxide (NO_x) emissions; combustion engine modeling



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1. Introduction

To successfully deal with the trend of combating climate change as laid out in the recommendations of the Paris (COP21 [1]) and Glasgow (COP26 [2]) Climate Agreements, propulsion technologies have to be capable of achieving major CO₂ reductions in increasingly shorter time frames [3]. In order to achieve this ambitious goal, the use of electric powertrains powered by batteries is not only essential, but also serves as the primary focus for research conducted by academic and industrial organizations. Nevertheless, this technology is incapable of fulfilling all of the objectives for individual transportation, ecological accountability, and feasibility. As a result, internal combustion engines (ICE) powered by non-fossil hydrocarbons and hydrogen (H₂) will still play a significant role in the future [4,5]. A comprehensive review of the positive and negative aspects of various alternative fuels, taking into account the current state of scientific and technological development as well as the needs of modern society, show that hydrogen is one of the most promising fuels—a gas with both good energy and environmental performance, as well as sufficient thermodynamic and technical compatibility with existing combustion engines. Particularly attractive industries for its use are automotive, trucking, and bus fleets. It also

has prospects in power generation, maritime shipping, and agriculture [6,7]. The concept of using hydrogen as an engine fuel is not new. There are several papers [8–13] that address various issues of utilizing hydrogen in internal combustion engines (ICEs). One of the primary challenges is the occurrence of abnormal combustion phenomena like pre-ignition, knock, and backfiring. These are caused by hydrogen's unique properties such as its low minimum ignition energy, wide flammability limits, and high flame speed. Pre-ignition occurs when the hydrogen–air mixture ignites prematurely before the spark, leading to inefficient and erratic engine operation. Knock combustion, caused by the spontaneous ignition of the end-gas region, can severely damage the engine due to high-pressure waves. Backfiring is the phenomenon of the mixture igniting in the intake manifold and propagating back toward the air intake, posing a risk of damage. Despite hydrogen's high energy density by mass, its low volumetric density under ambient conditions results in a lower energy content of the hydrogen–air mixture in the cylinder, leading to reduced power output compared with that of gasoline engines. Hydrogen combustion under stoichiometric conditions leads to high combustion temperatures, which can cause increased NO_x emissions, a significant pollutant from hydrogen ICEs. Optimizing fuel induction strategies, appropriate injection systems, valve timing strategies, and air/fuel ratio control is crucial for hydrogen ICEs to address these issues. In terms of engine performance, employing hydrogen as a gaseous fuel has several advantages over the use of liquid fuels like diesel and gasoline, such as the easier cold start, lower emissions of pollutants, and less contamination from lubricating oil [14,15]. Another advantage is that hydrogen may be produced from both renewable and nonrenewable sources, such as coal, biomass, water, and natural gas, using technologies such as gasification, electrolysis, and steam reforming. In general, hydrogen possesses various qualities that are very desirable for improving engine in-cylinder combustion and consequently engine efficiency. Examples of those properties are its high flame propagation speed, heating value, diffusivity, and short quenching distance [16]. Furthermore, when used in ICEs, hydrogen outperforms all other fossil fuels in terms of emissions, combustion stability, and leanness limit. Hydrogen, being a non-toxic and carbon-free gas, also does not contribute to emissions of unburned hydrocarbon (HC) and carbon oxides, resulting in significant pollution reductions [17]. H₂-fueled ICEs (H₂ICEs) is the sole option for keeping ICEs in future production because they do not produce any tank-to-wheel CO₂ emissions at the tailpipe. Additionally, unlike fuel cell powertrains, H₂ICEs can be fueled with non-purified hydrogen, resulting in a considerably lower cost of hydrogen fuel. H₂ICEs can also make use of current advanced engine and combustion control technologies, such as direct injection [18], the Miller cycle [19], lean/diluted combustion [20], pre-chamber ignition [21], etc. However, the maximum engine load with hydrogen operation is restricted by pre-ignition and autoignition. If not controlled, these can develop into more serious knocks, which can cause irreversible damage to engine components [22]. Typically, the engine peak power output is reduced by 35–50% because of knock, which restrict the overall equivalence ratio to values from 0.6 to 0.8 [23]. In addition, engine heat losses, aided by the short quenching distance of hydrogen, rise disproportionately during high-equivalence-ratio operation, reaching up to 45% of the total fuel heating value [24], significantly restricting engine efficiency and durability.

In contrast to conventionally fueled engines, hydrogen–gasoline or hydrogen–diesel blend engines can achieve reduced harmful emissions and improved fuel efficiency [25,26] without requiring extensive engine modifications. In general, converting an ordinary gasoline engine into a hydrogen–gasoline one requires installing hydrogen injection and storage systems; hydrogen can be stored in hydrogen tanks or supplied by the onboard hydrogen generator, which is typically delivered to the hydrogen injectors via a stainless steel hydrogen supply system. Furthermore, to successfully carry out hydrogen injection, the engine's electronic control unit software must be appropriately updated. Thus, hydrogen blending is a possible option to improve engine performance in the near future. Because of its distinct qualities, hydrogen is more suited for usage in spark ignition (SI) engines as opposed to compression ignition (CI) engines [27]. According to [28], adding hydrogen

to the gasoline–ethanol blend boosted the combustion process and combustion efficiency, therefore eliminating the detrimental effect of ethanol on thermal efficiency. Increased hydrogen percentages resulted in much lower CO, HC, and CO₂ emissions. However, it was also shown that increasing the hydrogen fraction in the blend increases NO_x levels. In [29], it was shown that the use of hydrogen in SI engines along with a passive prechamber ignition system gives significant benefits in the main chamber combustion process by improving the thermo-chemical characteristics of the mixture, boosting flame speed, and improving flame structure.

The modeling of these phenomena has become increasingly harder. It is necessary to integrate fluid dynamics, heat transport, thermodynamics, and chemical kinetics at different degrees of complexity in order to forecast engine operating parameters, combustion, and emissions. Zero-dimensional models are the most basic of them as they do not need all the details of the combustion process. This makes this particular type of combustion modeling simpler to carry out, but it may also lead to decreased precision and reliability [30]. The Wiebe function is one method that is frequently used in engine combustion models to calculate the mass fraction burnt [31]. The range of Wiebe function's S-shaped curve is from zero, which indicates the beginning of combustion, to one, which indicates the conclusion of combustion. Depending on the intensity of the charge motion, engine operating state, and designating shape factor that influences the shape of the mass fraction burnt, the function's form varies with a representative efficiency factor. A double-Wiebe function combustion model was developed in [32] to study the combustion characteristics of a dual-fuel engine. The authors claimed that there is a good correlation between the predicted function results and experimental findings. It is demonstrated in [33] that a triple-Wiebe function can accurately reflect every stage of dual-fuel combustion for both diesel-only operation and dual-fuel operation. After completing an in-depth literature analysis of the gasoline–hydrogen blend, it was evident that an investigation of the parameters most influential on engine characteristics is necessary.

This study delves into the effects of hydrogen addition to gasoline in a four-stroke engine through a detailed thermodynamic analysis using a model based on the two-zone Wiebe function. It evaluates the engine's performance, emissions, and efficiency with various hydrogen blends, providing insights into the thermodynamic behavior of these blends and their potential advantages. This research encompasses a broad range of air/fuel ratios, compression ratios, and hydrogen concentration levels, offering a comprehensive assessment of hydrogen's impact on combustion efficiency, power output, and emission profiles in the complete engine operating range, with a particular focus on predicting NO_x emissions, a significant environmental concern regarding internal combustion engines.

2. Methodology

The study in this paper was conducted using an in-house MATLAB R2017a model of a four-stroke internal combustion engine. The model was specifically designed to enable dual-fuel operation, using hydrogen and gasoline as fuels, and to provide an in-depth thermodynamic analysis of the engine's dynamics, with a particular focus on the complexities of the combustion process, emission characteristics, and improvements in efficiency resulting from the incorporation of hydrogen into the gasoline–air mixture. A schematic representation of model structure is shown in Figure 1.

The presented model integrates many interconnected components that replicate the actual processes occurring within the engine. The air intake system incorporates a boundary condition that represents the surrounding air, together with an air filter model that accurately calculates the pressure drop, closely resembling real-world circumstances. The throttle and exhaust restriction component control the flow of both air and exhaust, simulating various pressure drops in the intake and exhaust manifolds and tailpipe.

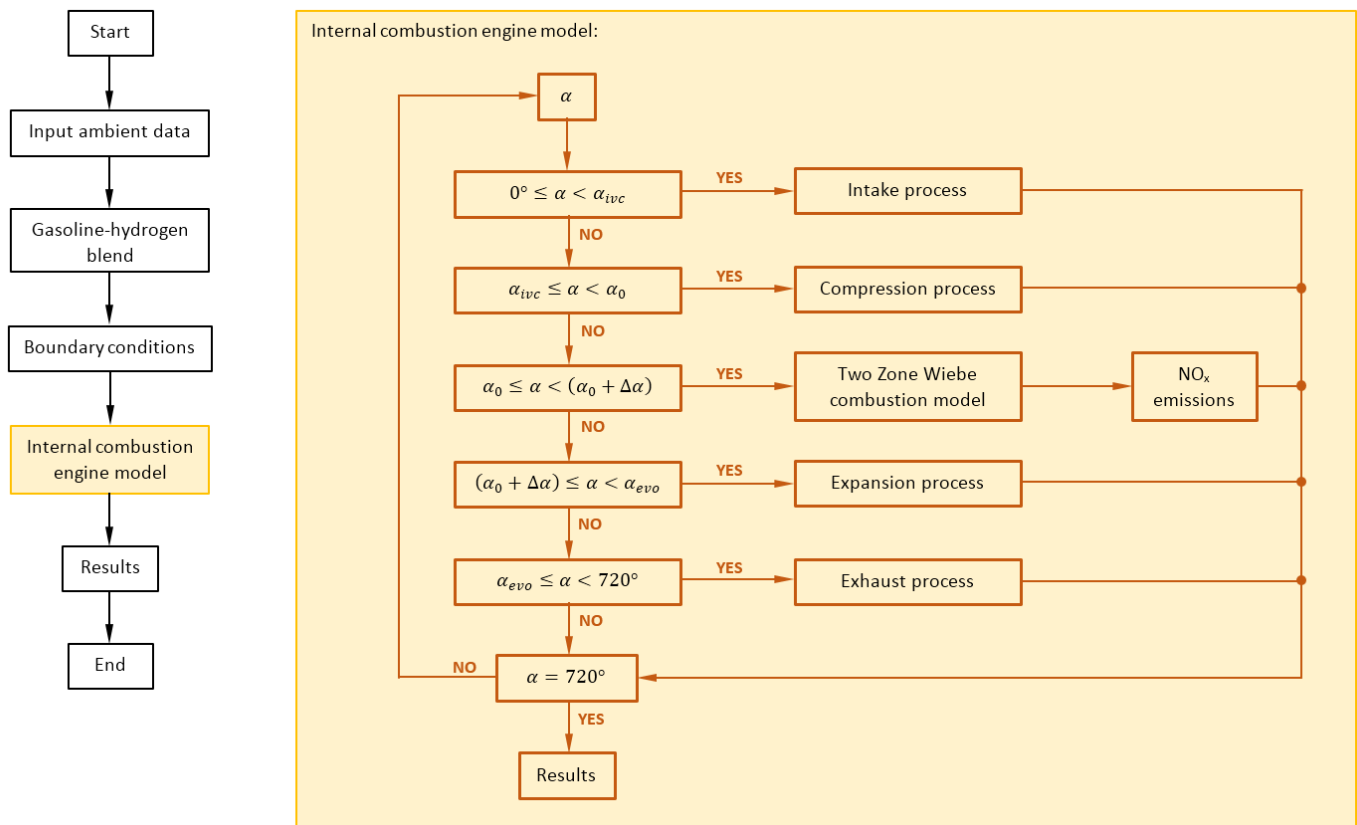


Figure 1. Schematic representation of model structure.

2.1. Fuel System Model

The fuel system is designed with two separate storage units for hydrogen and gasoline, enabling precise control over blend ratios. The central components responsible for controlling and monitoring the model's operations are the common rail model and control unit. These oversee the fuel injection and ignition timing processes.

The common rail model is an essential part of modern direct injection systems. The high-pressure fuel reservoir ensures a constant supply of fuel at the right pressure for injection into the engine's cylinders. A calculation is performed to determine the pressure within the rail for the control system. The model contains a detailed computation of the mass flow rate entering and exiting the rail in order to precisely assess the pressure dynamics. The formula for calculating the mass flow rate of rail pressure takes into account variables such as fuel density, the size of the intake and outlet orifices, and the pressure differential across these orifices, and can be fully seen in Equation (1):

$$\frac{dp_{\text{rail}}}{dt} = \frac{E}{V_{\text{rail}}} \cdot \frac{1}{\rho_{\text{fuel}}} \cdot \left(\frac{dm_{\text{pump}}}{dt} + \frac{dm_{\text{inj}}}{dt} \right) \quad (1)$$

In Equation (1) p_{rail} is the pressure inside the rail, E is bulk modulus for the working fluid, V_{rail} is the rail volume, ρ_{fuel} is the density of the fluid inside the rail, and $\frac{dm_{\text{pump}}}{dt}$ and $\frac{dm_{\text{inj}}}{dt}$ are mass flows through the pump and injector.

Injection mass flow is determined by considering the area of the nozzle hole and the velocity of injection. Velocity is determined by applying the Bernoulli equation. The derived equation contains the fuel density and the discharge coefficient. The discharge coefficient estimates the amount of frictional losses occurring across the entire injection

system. The flow through an orifice, assuming the fluid is incompressible, can be described by the following equation:

$$\frac{dm_{inj}}{dt} = A_{NH} \cdot c_D \cdot \sqrt{2 \cdot \rho_{fuel} \cdot (p_{rail} - p_{cyl})} \quad (2)$$

where A_{NH} is area of the nozzle holes, c_D is the discharge coefficient, and p_{cyl} is the pressure inside the cylinder.

2.2. Control Unit Model

The control unit was designed to compute the fueling multipliers for gasoline and hydrogen in a dual-fueled internal combustion engine. It considers operating characteristics such the engine's rotational speed, the volumetric air flow rate, the mass ratio of hydrogen to gasoline (a_1), and the intended air/fuel equivalence ratio (λ). The computed multipliers, one for gasoline (M_h) and one for hydrogen (M_g), establish the necessary quantities of each fuel to produce combustion under various conditions.

$$AF_{st} = (1 - a_1) \cdot 14.7 + a_1 \cdot 34 \quad (3)$$

In cases when the engine's rotational speed is less than 700 RPM, the gasoline multiplier is calculated using a specific equation that assumes a constant stoichiometric ratio for gasoline, without taking into account the hydrogen content or the air/fuel equivalence ratio. This was carried out to ensure that idle control remained as basic as possible. On the other hand, when the RPM is 700 or higher, the function uses a more dynamic approach to calculate the multipliers for both gasoline and hydrogen, which can be seen in Equation (4):

$$M_g = \frac{(1 - a_1) \cdot \dot{m}_{air} \cdot 60 \cdot Z}{rpm \cdot AF_{st} \cdot \lambda \cdot N} \quad (4)$$

$$M_h = M_g \frac{a_1}{(1 - a_1)}$$

These calculations modify the fueling rates by taking into account the current engine speed, the blended stoichiometric ratio, and the target air/fuel equivalence ratio. This approach guarantees that the fuel blend achieves optimal combustion by dynamically adjusting to the engine's operational conditions. The gasoline multiplier is calculated initially, taking into account the quantity of gasoline in the blend. Consequently, the hydrogen multiplier is derived by calculating the gasoline multiplier and then modifying it based on the ratio of the hydrogen to gasoline mass. This approach allows for fine control over the fueling of both gasoline and hydrogen, supporting efficient combustion throughout a range of operational situations.

2.3. Combustion Model

At the model's core is the engine block where combustion is initiated, linked to the air intake and exhaust systems. The Wiebe model is a method used to simulate the combustion process within an internal combustion engine, offering a simplified approach that hinges on specifying the rate of heat release based on the engine's crank angle rather than time. This method efficiently translates cylinder pressure data into a combustion heat release rate by plotting the former against crank angle degrees, which can be readily obtained from engine tests and through a literature search. The model employs what is known as the Wiebe function, a mathematical representation that captures the characteristics of an engine's heat release. Parameters within this function, including the mass fraction burned and crank angle, along with shape parameters, dictate the form of the heat release curve.

The formula for determining the rate of heat release calculated with the Wiebe function can be seen in Equation (5):

$$\frac{dQ}{d\alpha} = Q \cdot \frac{c}{\Delta\alpha_c} \cdot (m+1) \cdot \left(\frac{\alpha - \alpha_o}{\Delta\alpha_c} \right)^m \cdot e^{-c \left(\frac{\alpha - \alpha_o}{\Delta\alpha_c} \right)^{m+1}} \quad (5)$$

where Q is the fuel heat input, c is the Wiebe parameter for combustion completeness, $\Delta\alpha_c$ is the combustion duration, m is the shape parameter, α is the crank angle, and α_o is the start of combustion. The integral of this function over the crank angle gives the cumulative mass of fuel burned since combustion started, providing a continuous picture of the combustion process. To add complexity and enhance the realism of the simulation, the Wiebe two-zone model is used in this article, which divides the combustion chamber into two zones—burned and unburned. Mathematical expressions in this model ensure the conservation of mass and volume between the two zones, reflecting the physics of the combustion process more closely. In a single-zone analysis of combustion, the energy equation considers only the overall state of the cylinder, which averages out the temperature and the composition of the species and applies the perfect gas law to those averaged properties of the gas mixture. In contrast, the two-zone model divides the cylinder's contents into burned and unburned zones, applying the perfect gas law separately to each zone. These zones display marked differences in temperature and species composition. The variable nature of gas properties such as the gas constant (R) and specific heats (c_v , c_p), which depend on temperature and species composition, means that the amount of fuel needed to achieve a particular pressure target will differ between single-zone and two-zone approaches during combustion. In the case of uneven combustion, such as that found in direct injection engines, the two-zone model requires an additional assumption. The local air/fuel ratio changes over the course of the combustion, starting out rich and becoming leaner, which affects the expansion rate of the burned zone and its temperature trajectory. In this article, the air/fuel ratio is modeled to increase linearly in proportion to the ratio of fuel burned.

2.4. NO_x Model

One more thing that needs more detailing is the modeling of NO_x reactions. The NO_x formation model implemented in this model is based on the well-known Zeldovich mechanism, which is a kinetic scheme for describing how NO_x is formed during high-temperature combustion.

In Table 1, six reactions (R1 to R6) are shown for each reaction along with their rate constants (k_0) and temperature exponents (a), as well as the reference temperatures (T_A) at which these constants are valid. Each reaction rate (r) is expressed as a product of the concentration of reactants, the rate constant (k), and a temperature function. In the context of a two-zone model, which separates the cylinder contents into burned and unburned zones, the NO_x model focuses on the burned zone, as indicated by the variable T , representing the local combustion temperature in the burned zone. The concentration of N_2O , an intermediate in NO_x formation, is calculated using a specific formula that considers the concentration of the partial pressure of oxygen and the local combustion temperature, as shown in Equation (6):

$$c_{\text{N}_2\text{O}} = 1.1802 \cdot 10^{-6} \cdot T^{0.6125} \cdot e^{\left(\frac{9471.6}{T}\right)} \cdot c_{\text{N}_2} \cdot \sqrt{p_{\text{O}_2}} \quad (6)$$

The final rate of NO production/destruction (r_{NO}) is then given by Equations (7) and (8), which combine the effects of the individual reaction rates with tuning parameters ($C_{\text{PostProcMult}}$ and $C_{\text{KineticMult}}$) that allow for the calibration of the model based on empirical data or the desired simulation accuracy.

$$r_{\text{NO}} = C_{\text{PostProcMult}} \cdot C_{\text{KineticMult}} \cdot 2.0 \cdot (1 - \alpha^2) \frac{r_1}{1 + \alpha \cdot AK_2} \frac{r_4}{1 + AK_4} \quad (7)$$

$$\alpha = \frac{c_{NO,act}}{c_{NO,equ}} \cdot \frac{1}{C_{PostProcMult}} AK_2 = \frac{r_1}{r_2 + r_3} AK_4 = \frac{r_4}{r_5 + r_6} \quad (8)$$

where α is the fraction of NO.

Table 1. Six reactions considered in the Zeldovich mechanism.

	Reaction Equation	Rate $k_i = k_{0,i} \cdot T^a \cdot e^{\left(\frac{-T_A}{T}\right)}$	k_0 (cm ³ , mol, s)	a (-)	T_A (K)
R1	$N_2 + O = NO + N$	$r_1 = k_1 \cdot c_{N_2} \cdot c_O$	4.93×10^{13}	0.0472	38,048.01
R2	$O_2 + N = NO + O$	$r_2 = k_2 \cdot c_{O_2} \cdot c_N$	1.48×10^8	1.5	2859.01
R3	$N + OH = NO + H$	$r_3 = k_3 \cdot c_{OH} \cdot c_N$	4.22×10^{13}	0	0
R4	$N_2O + O = NO + NO$	$r_4 = k_4 \cdot c_{N_2O} \cdot c_O$	4.58×10^{13}	0	12,130.6
R5	$O_2 + N_2 = N_2O + O$	$r_5 = k_5 \cdot c_{O_2} \cdot c_{N_2}$	2.25×10^{10}	0.825	50,569.7
R6	$OH + N_2 = N_2O + H$	$r_6 = k_2 \cdot c_{OH} \cdot c_{N_2}$	9.14×10^7	1.148	36,190.66

3. Simulation Results

Internal combustion engine performance was simulated using an in-house MATLAB simulation code. The simulations were performed for four different gasoline–hydrogen blend levels, ranging from 0 to 20% hydrogen, different excess air coefficients, ranging from 1 to 2.5, and different engine compression ratio values, ranging from 8 to 14.

For the purpose of this study, a four-cylinder engine with a bore and stroke of 75 mm and 85 mm, respectively, was considered.

Figure 2 shows the effective power simulation results as a function of the compression ratio and excess air coefficient for different gasoline–hydrogen blend levels at a 3000 rpm point of operation. The effective power resulted in the highest values for a 0% hydrogen blend level for all considered excess air coefficients and compression ratios. With the addition of hydrogen into the fuel blend mixture, the effective power dropped gradually by 3.24% on average for an excess air coefficient value of 1 and by 0.41% on average for an excess air coefficient value of 2.5, with each level of hydrogen increase. The overall average drop in effective power for a 20% hydrogen blend level, compared with that for a 0% hydrogen blend level, were 9.45% and 1.22% for excess air coefficients 1 and 2.5, respectively. A similar trend in a drop of power can also be observed for other crankshaft rotation speeds, as can be seen in Figure 3.

Figure 4 shows the specific fuel consumption simulation results as a function of the compression ratio and excess air coefficient for different gasoline–hydrogen blend levels at a 3000 rpm point of operation. The specific fuel consumption resulted in the highest values for the 0% hydrogen blend level for all considered excess air coefficients and compression ratios. With the addition of hydrogen into the fuel blend mixture, the specific fuel consumption dropped gradually by 8.75% on average for all excess air coefficients considered. The average drop in specific fuel consumption values for a 20% hydrogen level, compared with that for a 0% hydrogen level, was 24.03%. The average specific fuel consumption difference between an air coefficient of 1 and 2.5 was 5% for all hydrogen blend levels considered. Figure 5 shows the specific fuel consumption simulation results for other crankshaft rotation speeds. As the hydrogen concentration level increases within the gasoline–hydrogen blend, so does the overall air/fuel ratio, so accordingly, the air and fuel blend mixture becomes leaner for the same excess air coefficient values. Therefore, the overall gasoline–hydrogen blend mass on the intake is smaller with each increase in hydrogen level, resulting in lower specific fuel consumption values as well as lower effective power. However, the reduction in specific fuel consumption values was not affected by a drop in corresponding effective power values, as the difference in input energy for different gasoline–hydrogen blends due to the lack of an overall blend mass was compensated for by the better lower heating value of the gasoline–hydrogen blend. Furthermore, the leaner fuel blend mixture also resulted in lower CO₂ emissions with

each level of hydrogen. With the addition of hydrogen into the gasoline mixture, the CO₂ emission level dropped gradually by 11.72% on average, while the overall average drop in the CO₂ emission level for a 20% hydrogen level, compared with that for a 0% hydrogen level, was 31.21%.

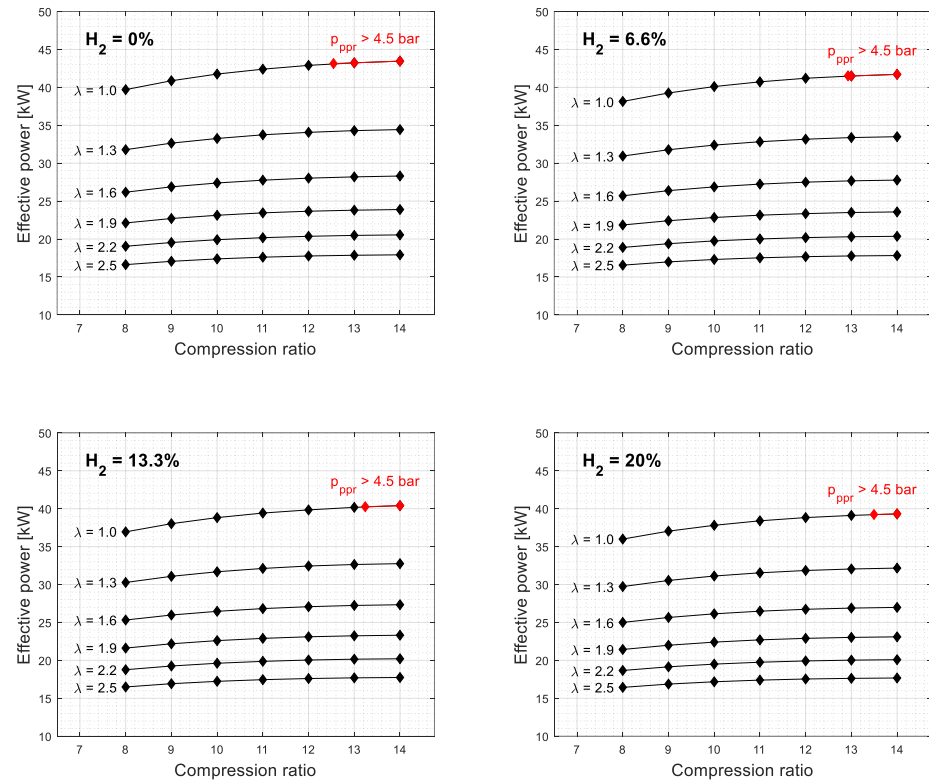


Figure 2. Effective power simulation results for 3000 rpm operation point.

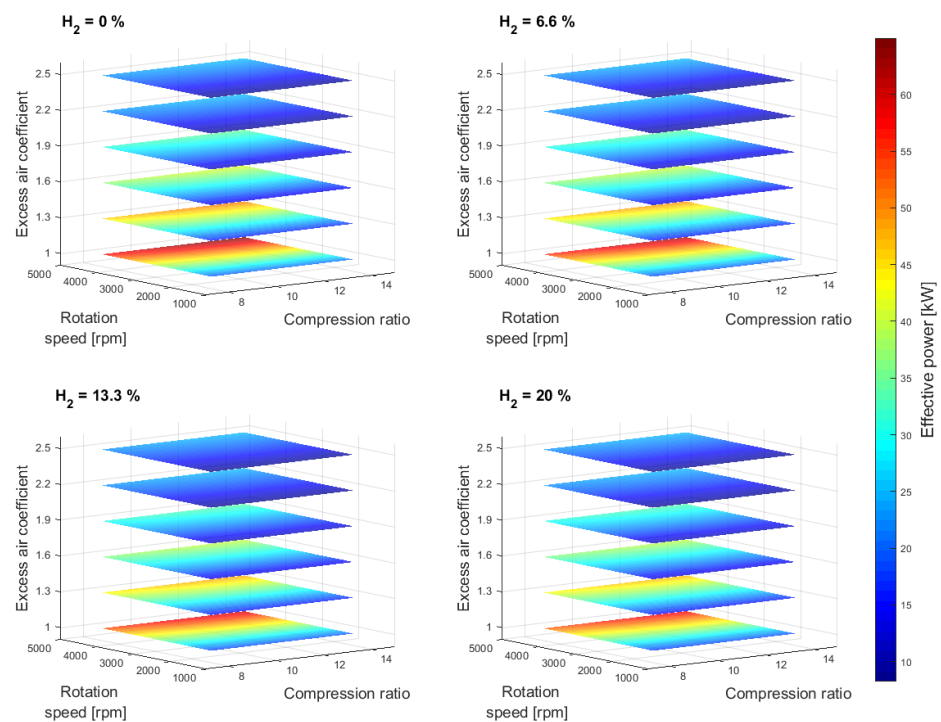


Figure 3. Effective power simulation results for various crankshaft rotation speeds.

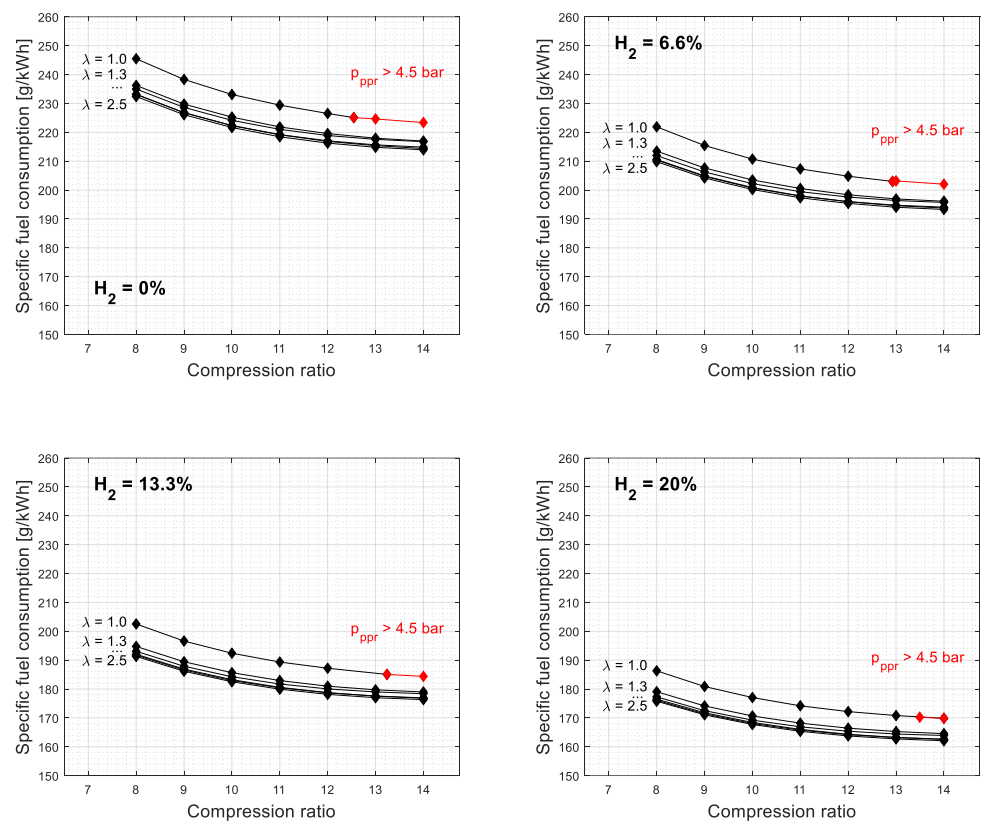


Figure 4. Specific fuel consumption simulation results for 3000 rpm operation point.

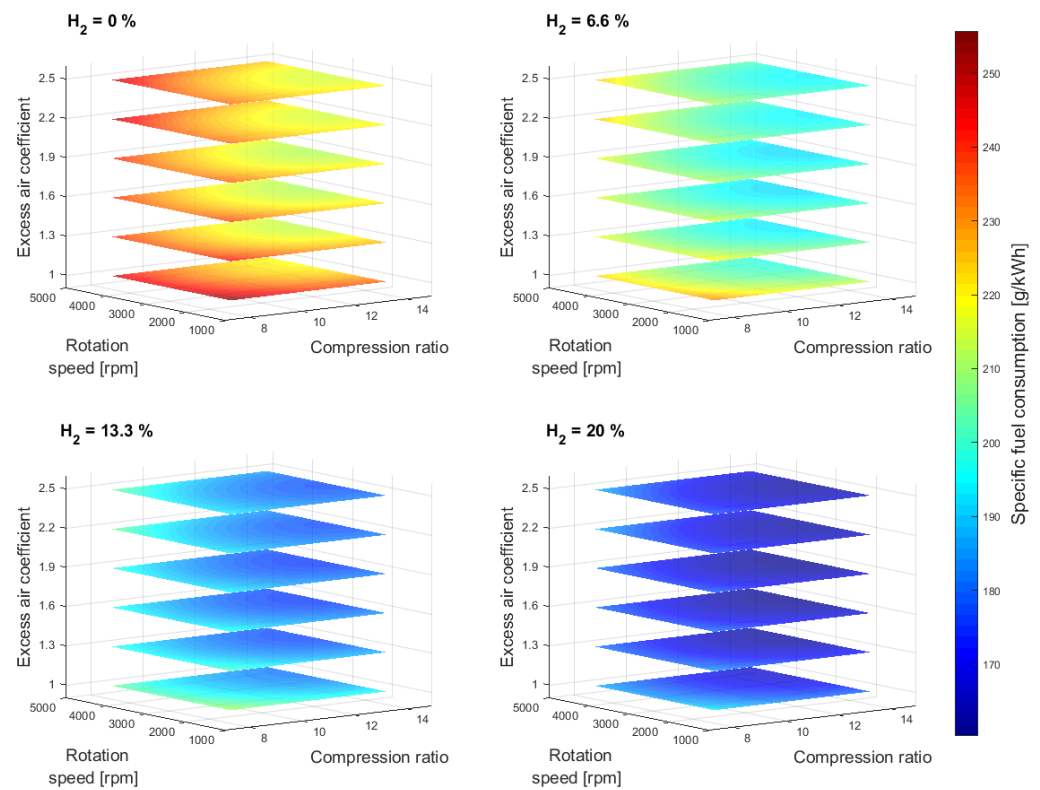


Figure 5. Specific fuel consumption simulation results for various crankshaft rotation speeds.

At higher compression ratio values, the effective power resulted in higher values and the specific fuel consumption resulted in lower values. However, the results also show an unfavorable engine operation area at higher compression ratio values for an excess air coefficient of 1, as shown in Figures 2 and 4. Furthermore, with the addition of hydrogen into the gasoline mixture, the unfavorable area of engine operation as a function of the compression ratio was reduced. A similar trend can also be observed for other rotation speed values in Figure 6. It should also be noted that the elevated peak pressure rise to above 4.5 bar was not recorded for excess air coefficient values ranging from 1.3 to 2.5. Since the higher excess air coefficient values were achieved via a reduction in the overall gasoline–hydrogen blend mass, the lower overall energy input through the fuel blend mixture resulted in lower effective power values and lower levels of average pressure at the end of the combustion process, as can be seen in Figure 7. However, if the higher excess air coefficient values for the constant fuel blend mass was achieved via turbocharging, the higher intake pressure and higher overall energy input would also affect the average temperature and pressure values at the end of the combustion process, which would create unfavorable conditions for engine operation.

Nitrogen oxides (NO_x) represent a major challenge in terms of emissions when using hydrogen fuel in internal combustion engines. NO_x formation is promoted by the high combustion temperatures that occur during hydrogen combustion compared with those of hydrocarbon fuels like gasoline. In spark-ignition hydrogen engines, NO_x emissions tend to be higher than those produced by gasoline engines for the same reason. NO_x levels peak around stoichiometric air/fuel ratios where the combustion temperatures are the highest. Operating an engine with ultra-lean mixtures reduces combustion temperatures and NO_x formation accordingly. This behavior can be seen in the following figures. Figure 8 shows the specific NO_x emission simulation results as a function of the compression ratio and excess air coefficient for different gasoline–hydrogen blend levels at a 3000 rpm point of operation. The specific NO_x emissions resulted in higher values with the addition of hydrogen into the fuel blend mixture. The overall average rise in the specific NO_x emissions for the 20% hydrogen concentration level, compared with that for the 0% hydrogen concentration level, corresponding to excess air coefficient 1, was 440%, even though the overall average rise in temperature at the end of the combustion process was only 1.67%, for a hydrogen concentration level between 0% and 20%, for same considered conditions, as can be seen in Figure 9. However, Wiebe's two-zone model separately determines the temperatures of unburned and burned zones. The burned zone, as expected, resulted in considerably higher temperatures compared with the unburned zone due to the thermal energy released during the simulation of the combustion process. Even though the overall fuel blend mass was reduced with the addition of hydrogen, the lack of fuel was compensated for by the better overall fuel blend calorific value. Therefore, adding hydrogen increased the fuel blend calorific value, resulting in higher temperatures within the burned zone and creating better conditions for NO_x generation. Furthermore, adding hydrogen increased the potential for NO_x generation at an excess air coefficient of 1.3. With each level of hydrogen increase, specific NO_x emissions corresponding to an excess air coefficient of 1.3 exceeded specific NO_x emissions corresponding to an excess air coefficient of 1 at a lower compression value. This would suggest that the beneficial conditions for NO_x generation are easily created by adding hydrogen to the gasoline and with a small amount of excess oxygen. Figure 10 shows the specific NO_x emission simulation results for other crankshaft rotation speeds.

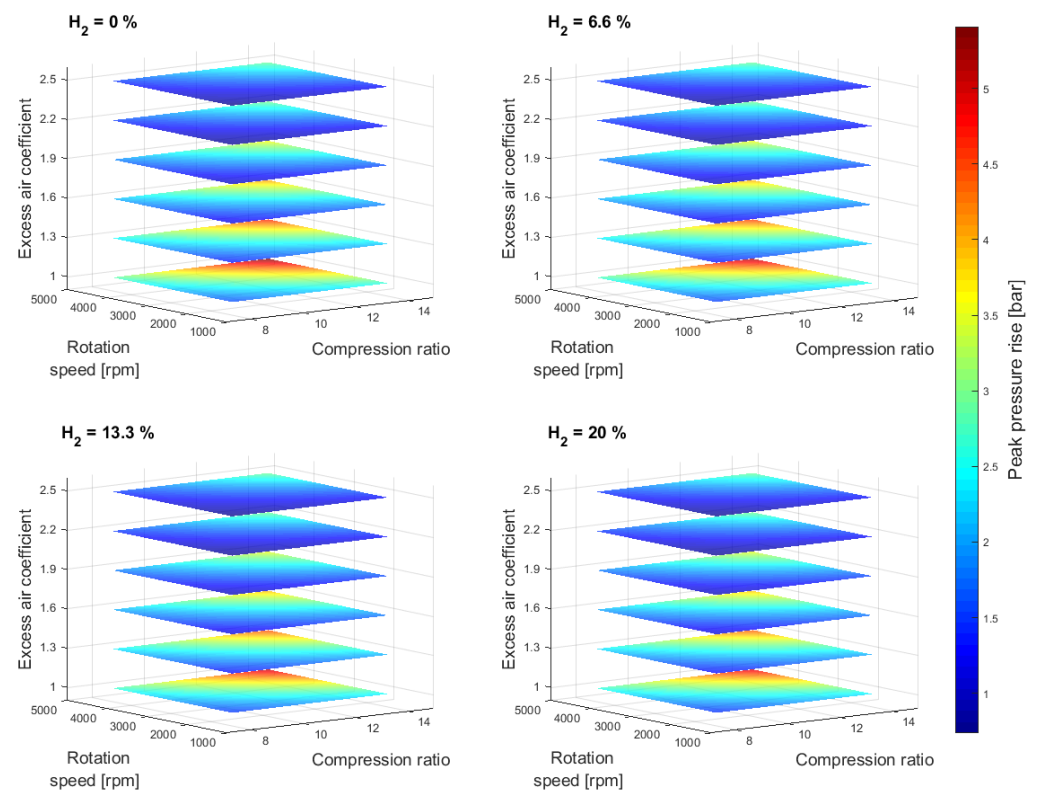


Figure 6. Peak pressure rise simulation results for various crankshaft rotation speeds.

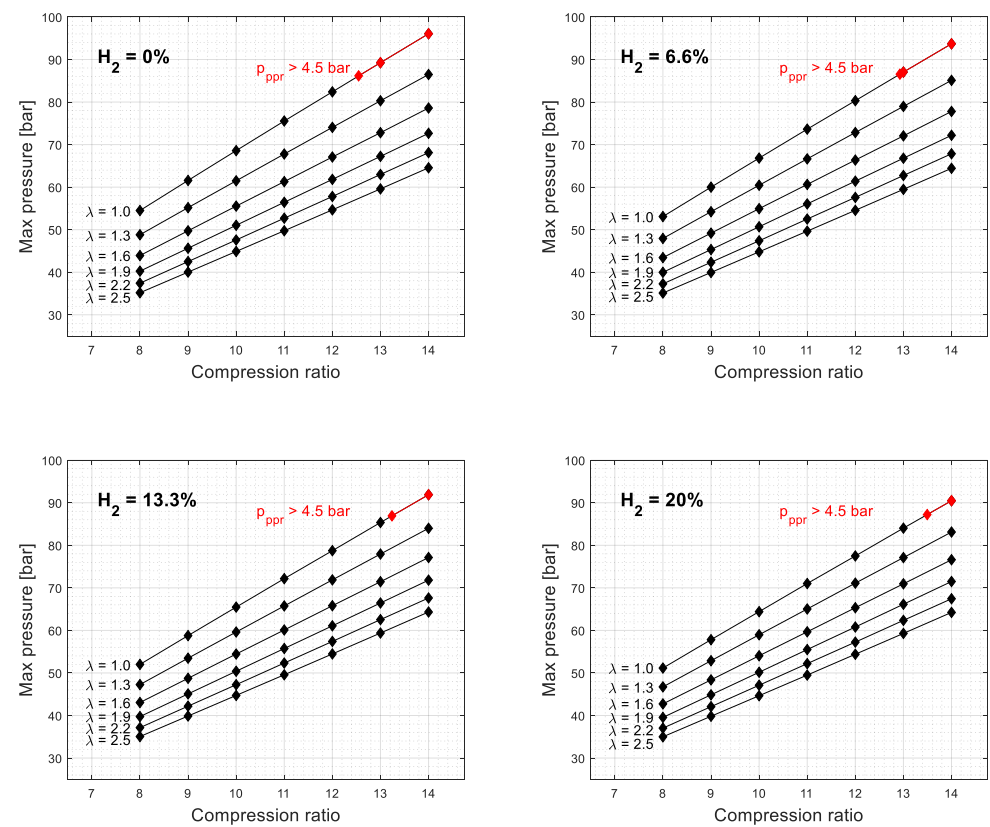


Figure 7. Pressure at the end of combustion process; simulation results for 3000 rpm operation point.

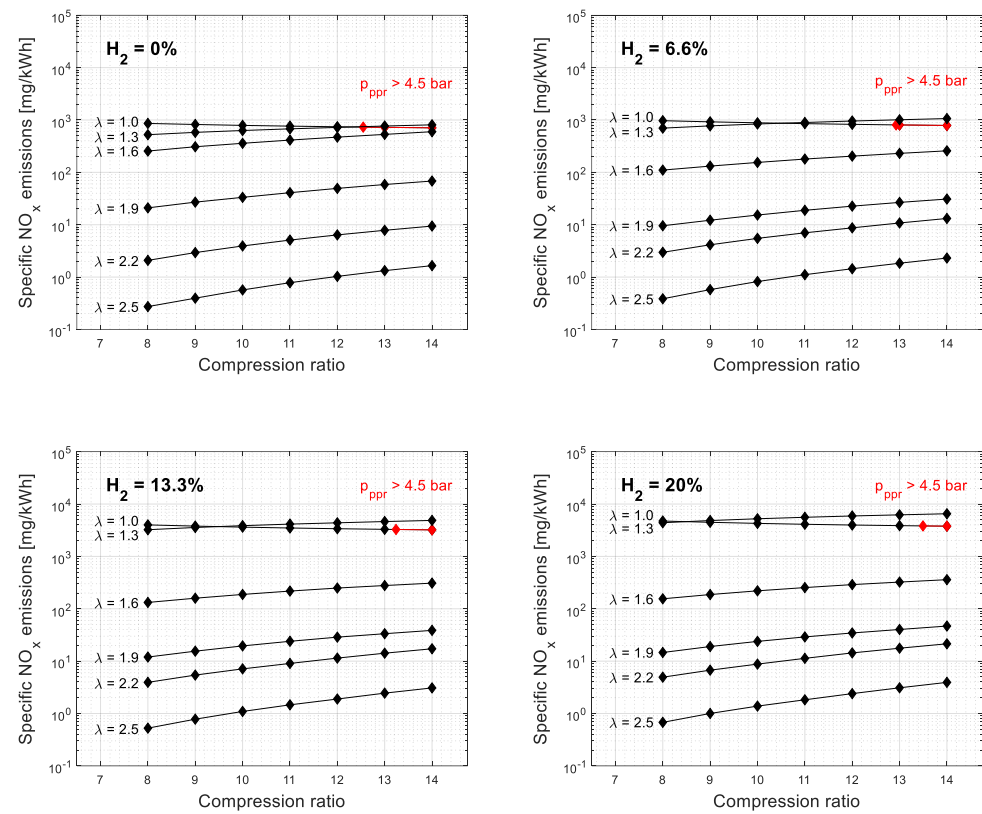


Figure 8. NO_x simulation results for 3000 rpm operation point.

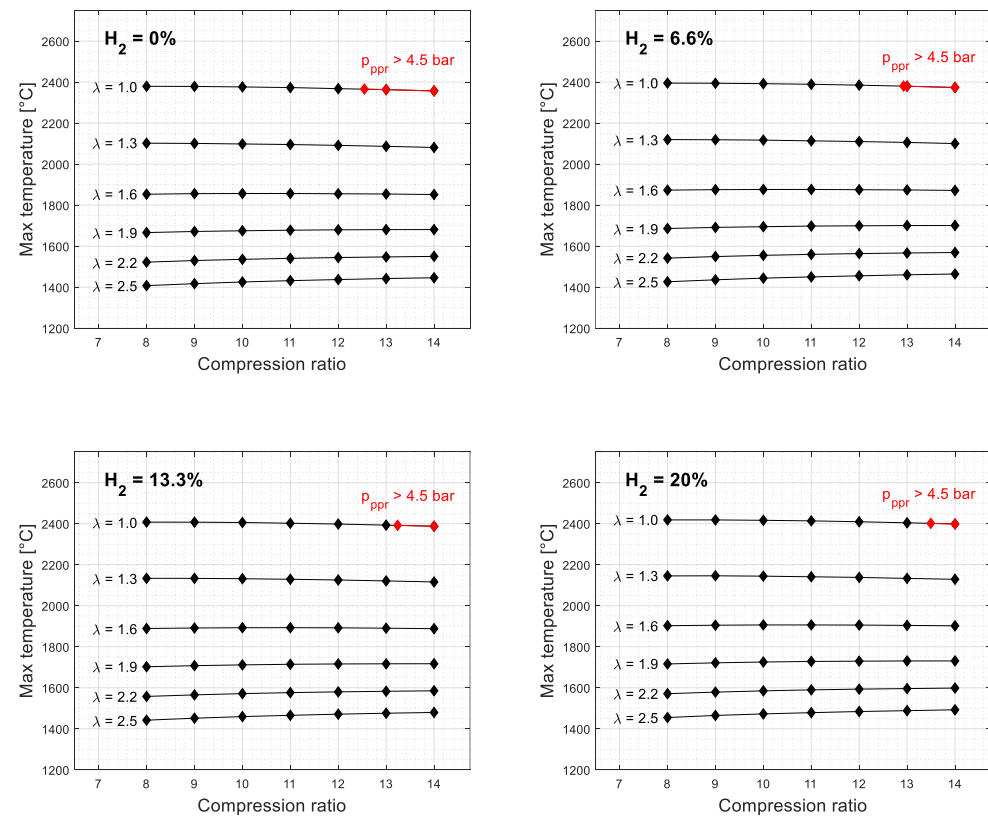


Figure 9. Average temperature at the end of the combustion process; simulation results for a 3000 rpm operation point.

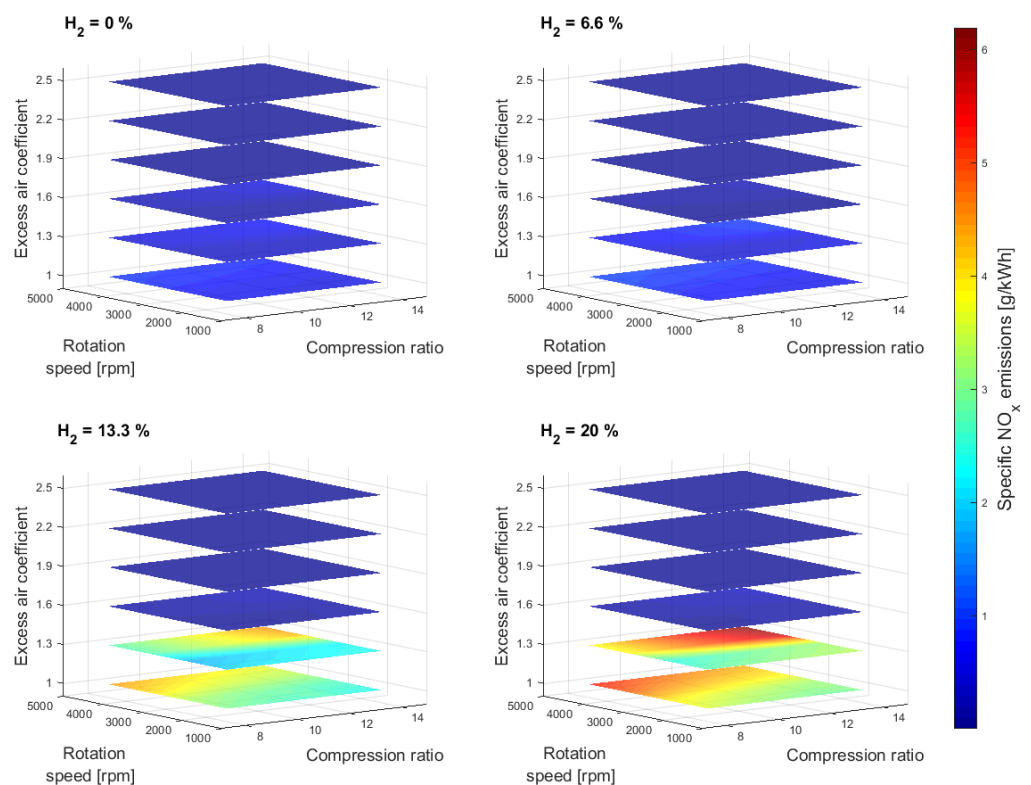


Figure 10. NO_x simulation results for various crankshaft rotation speeds.

4. Conclusions

This research explored the feasibility of utilizing blends of hydrogen and gasoline within internal combustion engines, evaluating different hydrogen ratios up to 20%. The scope of the study extended across various engine operational points to assess how hydrogen enrichment affects engine performance and pollutant emissions. The findings indicated that adding hydrogen to a fuel blend generally leads to a decrease in the engine's effective power, with a significant reduction of 9.45% observed for blends containing 20% hydrogen. This reduction is attributed to the resulting leaner air–fuel mixture when hydrogen is added.

On the other hand, there was a notable decrease in specific fuel consumption, which fell by an average of 24.03% for the 20% hydrogen mixture. This improvement primarily stems from the greater heating value of the hydrogen–gasoline mixtures. Likewise, a considerable decline in carbon dioxide emissions was recorded, with an average cutback of 31.21% at the maximum level of hydrogen blending. Investigations into compression ratio effects demonstrated that higher ratios could enhance power and diminish fuel consumption. Nonetheless, overly high compression might induce unfavorable combustion conditions, which can be partially mitigated by the introduction of hydrogen.

However, this research also uncovered a significant issue: a dramatic increase in nitrogen oxide (NO_x) emissions, which escalated by an average of 440% with the 20% hydrogen mixture under certain conditions. This rise in NO_x emissions correlates with the elevated temperatures in the combustion zone, another effect of hydrogen's higher heating value. The introduction of hydrogen into gasoline for use in internal combustion engines affects not only efficiency and emissions but also alters combustion dynamics, leading to leaner combustion. Although lean combustion offers several advantages, such as improved fuel efficiency and lowered CO₂ emissions, it necessitates the implementation of sophisticated exhaust aftertreatment technologies to counteract the increased NO_x emissions, including strategies like exhaust gas recirculation, selective catalytic reduction (SCR) [34], lean NO_x traps [34], and NO_x adsorbers. Furthermore, trends in NO_x emissions

with varying injection timings are complex. The literature says that, at low loads with early direct injection creating a homogeneous mixture, NO_x is low but rises rapidly beyond air/fuel ratios of ~0.5 up to 0.8 before falling slightly under stoichiometric conditions. In contrast, late direct injection near the spark timing with stratified mixtures causes elevated NO_x levels at low loads but significantly lower NO_x levels at high loads, compared with those under early injection. Various strategies to manage NO_x, such as lean burn combustion, variable air/fuel ratio operation, EGR dilution, high boost pressures, as well as exhaust aftertreatment systems for capturing NO_x emissions, are all valid. Optimizing the injection timing, pressure, number of injections, and overall combustion strategy is key to minimizing NO_x formation from hydrogen engines across the operating range and could be part of future work.

In summary, integrating hydrogen with gasoline emerges as a viable strategy to boost engine efficiency and lower CO₂ emissions. However, this method introduces the challenge of controlling increased NO_x emissions, requiring a delicate optimization of engine operational parameters to leverage the advantages while overcoming the negatives. This investigation provides a foundation for subsequent research and development aimed at refining hydrogen utilization in internal combustion engines, underscoring the intricate balance between efficiency improvements and emission management. Future research will cover the control of parameters influencing the combustion process of electronically controlled dual-fuel engines.

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