INFLUENCE OF HYDROGEN AS A FUEL ADDITIVE ON COMBUSTION AND EMISSIONS CHARACTERISTICS OF A FREE PISTON ENGINE

by

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It has been shown that using fuel additives play an important role in enhancing the combustion characteristics in terms of efficiency and emissions. In addition, free piston engines have shown capable in reducing energy losses and presenting more efficient and reliable engines. In this context, the objective of the present work is to investigate the effect of using hydrogen as a fuel additive in natural gas homogeneous charge compression ignition free piston engine. To this aim, two models have been iteratively coupled: the combustion model that is used to calculate the heat release of the combustion and the scavenging model that is employed to determine the in-cylinder mixture state after scavenging in terms of its homogeneity and species mass fractions and to obtain the finial pressure and temperature of the in-cylinder mixture. In the former model, the 0-D approach through Cantera toolkit has been considered due to the fact that homogeneous charge compression ignition combustion is very rapid and the fuel-air mixture is well-homogenous, whereas in the latter model, 3-D-CFD approach through ANSYS FLUENT software is considered to ensure precise calculations of the species exchange at the end of each engine cycle. The effect of hydrogen as a fuel additive has been quantified in terms of the combustion characteristics (e. g., ignition delay, heat release rate, engine overall efficiency and emissions, etc.). It has been shown that hydrogen addition reduces ignition delay time, decreases the in-cylinder peak pressure, while allowing the engine to operate with higher mechanical efficiency as it has high heat release rate, increases the NO_x emission levels of the engine, but decreases the CO levels

Key words: *hydrogen, free piston engine, mixture homogeneity, emission, homogeneous charge compression ignition combustion, 3-D scavenging*

Introduction

In the last few decades, a great development in energy saving and conversion fields including the improvements and modifications of internal combustion (IC) engines, has been shown. Looking for high efficiency and low emissions in IC engines emphasize many researchers to look for more reliable and efficient engine configurations than those of conven-

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tional engines. In this regard, free piston engines are among the modifications that aim to reduce energy losses and present more efficient and reliable engines. The absence of the complicated parts, such as the crankshaft, allows free piston engines to work with less frictional losses compared to conventional engines [1]. Free piston engines can operate on variable compression ratio (CR) due to the fact that the piston motion is not restricted along the piston stroke ends. This increases the applicable fuels which can be used in the engine and allows employing different ignition modes [2]. Free piston engines and conventional engines are extremely different in terms of piston motion profile [3]. The expansion stroke in the free-piston engine is much faster than this of conventional engine where piston motion is restricted by the crankshaft motion. Also, the piston motion in conventional engines during the one cycle cannot be changed due to the high system inertia. However, in free-piston engines, the piston motion profile is proportional to the instantaneous forces resultant over the piston. This makes the combustion process, presented by the pressure force, is the dominant factor which influences the speed of expansion stroke. A further consequence of this is that the piston motion profile may differ between different operating conditions [2]. One advantage of the fast expansion stroke in the free-piston engine is the reduction in heat losses during the stoke and, therefore, the formation of temperature-dependent emissions is reduced.

The mentioned features of free piston engines are extremely desired when homogeneous charge compression ignition (HCCI) combustion is employed. In HCCI engines, the combustion is completely controlled by the chemical kinetics of the fuel-air species. In fact, the ignition occurs when the fuel-air mixture reaches the auto-ignition temperature, which achieved by the effect of compressive heating [4]. In addition, the nature of combustion is lean and with relatively low heat release rates due to the presence of excess air [4]. The interest in HCCI combustion is based on the fact that this type of combustion has low emissions when compared to other combustion types. Also, the nature of HCCI combustion extends the fuel operation range of the engine where fuels like gasoline and diesel can be used [4, 5]. Therefore, by presenting HCCI combustion in free piston engines, it is expected to observe higher efficiency and low emissions, especially when appropriate control of the process is applied.

To move toward cleaner and efficient energy producing methods, fuel additives have been considered as one of the effective ways used for enhancing the combustion characteristics in terms of both efficiency and emissions. Natural gas (NG) is one of the suitable fuels for IC engines operation due to its affordable prices and reasonable heating values, but it has many limitations when compared to other fossil fuels. For instance, low combustion efficiency and narrow flame propagation are examples of NG combustion limitations [6].

One effective method that is used to enhance the performance of NG engines is using hydrogen as a fuel additive. Hydrogen combustion is clean and produces no emissions, except NO_x. In addition, hydrogen combustion includes fast flame velocity and low ignition energy, which extend the operational limits of the engine and increase its efficiency [7]. Hydrogen also allows the engine to run at relatively lean conditions and therefore, achieving high engine efficiency and low emissions [8].

The present work discusses the effect of using hydrogen as a fuel additive particularly in HCCI free piston engine fueled originally with NG. Different combustion characteristics, such as ignition delay and heat release rate, have been investigated. Also, engine overall efficiency and emissions outcomes have been considered in the study.

Review of literature

By reviewing the literature, it was noted that few studies focused on investigating the hydrogen addition on HCCI engines performance and none of them considered its effect on free piston engines. However, some studies focused on studying the addition of hydrogen in conventional crankshaft engines as this done by Yab *et al.* [9] In their study, Yab *et al*. [9] found that hydrogen addition promotes engine stability and lowers the required intake temperature of combustion. The experimental results also showed that hydrogen addition reduce NO_x levels, especially at low engine loads.

Later on, Askari *et al.* [10] examined the influence of hydrogen addition on the performance of NG HCCI engine. Their study showed that hydrogen addition by different percentages can advance the combustion process and maximize the peak combustion temperature. In terms of emissions, Askari *et al.* [10] found that CO, CO_2, and unburned HC decrease significantly, as hydrogen percentage increases.

After that, Guo and Neill [11] investigated the addition of hydrogen to n-heptane HCCI engine. The findings of their research came with the conclusion that hydrogen tetrads combustion phasing and reduce its duration. Also, hydrogen decreases unburned HC levels, but at the same time a noticeable rise in NO_x emissions had been observed.

Another study by Hu *et al.* [12] analyzed the effect of hydrogen addition to dimethyl ether (DME) HCCI engine. The results in Hu *et al.* [12] work indicated that hydrogen addition can delay the ignition timing of the fuel-air mixture, especially at high mixture temperatures (above 1000 K).

Finally, Hammound *et al.* [13] concluded that hydrogen addition with a small fraction to CH_4 fueled HCCI engine has a significant effect on the combustion phasing and can be a controlling parameter that maintain low emissions and peak cylinder pressure.

It was mentioned that piston motion in free piston engines is totally controlled by the combustion pressure force which by itself depends on the chemical reactions of the fuel species with air. Therefore, it is expected to observe variations in the performance of the free piston engine, especially in terms of its dynamics, when hydrogen is used as a fuel additive. For this reason, this study focuses on investigating the effect of hydrogen addition on the performance of the free piston engine, particularly. In fact, this investigation could help in understanding the relation between engine dynamics and thermodynamics which plays a significant role in achieving an optimal engine operation.

Methodology and model

In this study, a single cylinder two-stroke engine has been modeled using MATLAB, Cantera toolkit and ANSYS FLUENT. The model includes linear spring that works as energy storing device to return the piston during the compression stroke. The model also includes, electrical generator that converts the linear motion of the piston to electrical power by moving coil against permanent magnets. The loading force applied through the electrical generator on the engine is designed to be variable through the engine cycles. Figure 1 describes different kind

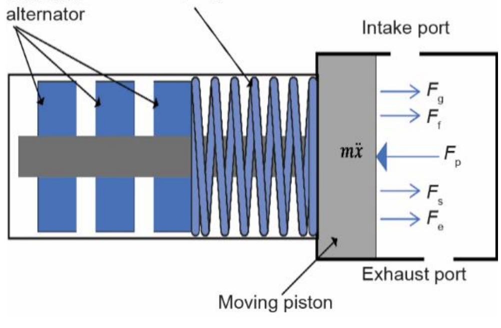

Figure 1. Free piston engine model

of forces that have been considered in the engine model including, spring force, F_s, gravity force, F_g, friction force, F_f, generator force, F_e, and pressure force, F_p. The pressure force is the result of fuel-air combustion process which causes rapid increasing in the in-cylinder pressure and therefore causes the piston to move down along the power stroke. The pressure force depends on the fuel-air ratio and the nature of combustion process, *i. e.*, it is a function of the fuel-air chemical kinetics, which describes the nature of HCCI combustion. On the other hand, the generator force, F_e, is working against the pressure force by absorbing the kinetics energy of the moving piston. This force depends on piston position, x, strength of magnetic field and coil turns number and diameter. The generator force can be changed to sustain the steady-state operation. In this study, it has been considered as shown in eq. (1).

Combustion model

In this model, Cantera toolkit [14] has been employed to calculate the heat release rate of the combustion process. The GRI-Mech 3.0 [15] has been used to consider the fuel-air species reactions. The GRI-Mech 3.0 incorporates 53 species and 325 reactions and can demonstrate adequately hydrogen and NG reactions in terms of HCCI combustion. The mechanism calculates NO_x and CO emission by considering their species and reactions which allowed the current study to investigate briefly the effect of hydrogen addition on the engine emissions. The 1-D approach has been considered to simulate the species reactions, due to the nature of HCCI ignition which assumes that all in-cylinder gas mixture is in well-homogenous state and the ignition happens everywhere, in a very rapid manner.

Table 1. Engine parameters

Parameter	Symbol	Value
Bore	B	28 mm
Maximum stroke	S_{max}	51 mm
Exhaust port height	X_{Ex}	16.2 mm
Intake port height	X_{In}	12.2 mm
Piston mass	m	0.385 kg
Spring stiffness	k	245 N/m
Spring free location	x_s	0.02 m
Engine speed	ω	125 Hz (7500 rpm)

The engine model parameters are reported in tab. 1 and the mathematical representations of all forces acting on the piston are presented in eqs. (1)-(5):

$$F_e = -C_g \dot{x} \operatorname{sign}(\dot{x}) \tag{1}$$

$$F_p = A_p P \tag{2}$$

$$F_f = -\beta \dot{x} \operatorname{sign}(\dot{x}) \tag{3}$$

$$F_g = mg \tag{4}$$

$$F_s = k(x - x_s) \tag{5}$$

where P is the instantaneous in-cylinder pressure, A_p – the piston area, g – the gravity acceleration, β – the combined friction coefficient which considers both static and the viscous friction forces as mentioned in [16], C_g – the generator load coefficient, k and x_s – the spring stiffness and the spring free length, respectively.

The resultant of the mentioned forces that causes piston acceleration is shown in eq. (6), where m is piston mass and $\ddot{x}$ is the piston acceleration:

$$m\ddot{x} = \sum F \tag{6}$$

To solve for piston acceleration, the First law of thermodynamics and the ideal gas law are solved for each piston position using finite difference as a numerical method, as shown, respectively:

$$m_g \frac{du}{dt} = \frac{dQ_{comb}}{dt} - P\frac{dV}{dt} - \frac{dQ_{loss}}{dt} \tag{7}$$

$$V\frac{dP}{dt} + P\frac{dV}{dt} = m_g R \frac{dT}{dt} \tag{8}$$

where m_g is the in-cylinder gas mixture mass, u – the specific internal energy, V – the in-cylinder volume, Q_{loss} and Q_{comb} – the heat loss by engine cylinder and the heat released by combustion, respectively. The heat transfer from the engine is found using a modified version of Woschini's correlation [17]:

$$h = 3.26 \cdot B^{-0.2} P^{0.8} T^{-0.55} V_P{}^{0.8} \tag{9}$$

$$\dot{Q}_{loss} = hA_{wall}(T - T_{wall}) \tag{10}$$

where h is the heat transfer coefficient, T and T_{wall} – the in-cylinder instantaneous and wall temperatures, respectively, V – the instantaneous volume, A_{wall} – the instantaneous surface area of the cylinder, the average piston speed is V_p, which is empirically correlated to the gas mixture velocity inside the combustion chamber.

Scavenging model

By the end of the expansion stroke, the scavenging model starts so the gas exchange process between the fresh charge and the in-cylinder residual gases is modeled. The 3-D engine model as this in fig. 2 has been used to simulate the scavenging process using CFD analysis. The aim of this model is to determine the in-cylinder mixture state after scavenging in terms of its homogeneity and species mass fractions. Using ANSYS FLUENT software, the percentage of the remaining residual gas can be found, this is necessary in the combustion model where residual gases can play significant role in defining different combustion phases such as maximum peak pressure and temperature. In addition, in the scavenging model; the finial pressure and temperature of the in-cylinder mixture can be obtained. As HCCI combustion is

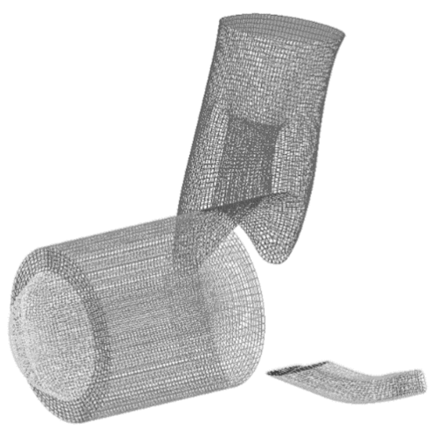

Figure 2. The 3-D CFD mesh of the scavenging model

influenced by the self-ignition conditions of the fuel-air mixture, the initial pressure and temperature can affect the ignition significantly and therefore overall piston dynamics. The complex gas exchange process that happens during scavenging requires employing the 3-D approach in order to simulate the process precisely as many factors should be included such as species thermal and mass diffusion and the effect of flow swirling. Such factors cannot be modeled using the 0-D approach as they depend not only on time but also on the geometry of the engine and the location of its inlet and exhaust ports.

The engine mesh which has been used in the scavenging simulation is shown in fig. 2. A total number of 129141 is the number of cells that have been considered where 106597 of them are of HEXA_8 type and 22544 are QUAD_4 cells. Sum mesh configuration minimize and limit cell distortion and numerical instabilities during the process calculations. The cells that consist the piston part in the mesh configuration move along the stroke length through the layering technique implemented in ANSYS FLUENT menu. A user defined function has been employed to describe the piston motion profile and different engine parameters like the fuel-air mass fraction and the in-cylinder intake pressure, and temperature have been also entered. To model the gas turbulence and the species transport during the scavenging simulation, k-ε and species transport models have been used. To be more convenient, equivalent crank angle (ECA) and equivalent speed (ES) notations is preferred in describing engine motion. Equations (11) and (12) [18] show the conversion from time and frequency scales to crank angle (CA) and rotation speed scales, where t_0 and t are the initial and instantons times, respectively, and f is the engine frequency:

$$ECA = 360(t - t_0)f \tag{11}$$

$$ES = 60f \tag{12}$$

Simulation methodology

The solution process includes coupling both the combustion model and the scavenging model through an iterative process, where both models feed each other back and forth until a certain tolerance is obtained. In the combustion model, 1-D approach has been assumed due to the fact that the HCCI combustion is very rapid and the fuel-air mixture is well-homogenous, which is verified in the scavenging model. However, the scavenging model has been simulated using 3-D-CFD approach to ensure precise calculations of the species exchange at the end of each engine cycle. The simulation procedure starts by feeding the combustion model with fuel and air mass fractions, which always maintain constant equivalence ratio (ϕ = 0.35). Other initial conditions like intake pressure and temperature, initial piston position, and velocity are also included. The combustion model solves for the fuel-air combustion products and for the pressure and temperature profiles. Once the exhaust port is opened, the scavenging model starts and it continues until the same port get closed. By the end of the scavenging process, the resulted in-cylinder species mass fractions, pressure, and temperature are fed again into the combustion model. The same process will continue, again, until the difference between the new and the old outputs (*i. e.*, mass fractions, pressure, and temperature) of each cycle are within a tolerance of 10^{-8}. The final pressure profile is, then, used to find the pressure force and other forces along the engine cycle. New piston position, velocity and acceleration are obtained and inserted back to the combustion model to start a new engine cycle or to restart the cycle if the difference between the new piston position and the old one is greater than a certain tolerance (10^{-8}).

Results and discussions

The investigations in this study have been performed by studying the hydrogen addition in different percentages to the NG-air mixture and modeling the engine cycle for the resulted mixture. Different combustion phases, like combustion peak pressure and temperature, have been compared at different hydrogen-NG ratios. Engine power, efficiency and emissions have been discussed as well, briefly.

In order to validate the current model, the study results have been compared to other results published by Sandia National Labs SNL [19]. The results agreed when two cases that includes pure NG and pure hydrogen are assumed in this study model. The comparison between the current numerical results and SNL experimental results are reported in tab. 2. The variations in the experimental results had been discussed by SNL people within the discussion section and they refer to operating temperature and CR as the main reasons of such variations. In addition, a similar free piston engine model to current one has been validated and published in [20] for DME fuel with including exhaust gas re-circulation (EGR) effect where the numerical results showed a very good agreement with the experimental data.

Table 2. A comparison between the current model and SNL model

	Current model		SNL model	
	CR	$\eta_{thermal}$ [%]	CR	$\eta_{thermal}$ [%]
NG (ϕ = 0.365, T_{in} =340 K)	37.19	48.7	30-54	50-55
Hydrogen (ϕ = 0.319, T_{in} =340 K)	21.8	41.4	17-50	40-55

A comparison between the free-piston engine motion profile and the conventional one is presented in fig. 3. Engine speed and intake air pressure and temperature, have been set identical in both engine models to ensure fare comparison. It is obvious that the free piston motion differs during both of compression and expansion strokes. In the case of the conventional engine, the profile looks axisymmetric around the TDC meaning that the engine piston spends the same time in its two strokes. However, by locating the TDC of the free piston engine profile, the expansion stroke seems shorter than the compression stroke. The short residence time around the TDC in the case of the free-piston engine cause the expansion stroke to be faster than in the case of the conventional engine where the piston is restricted by the crank rotation speed.

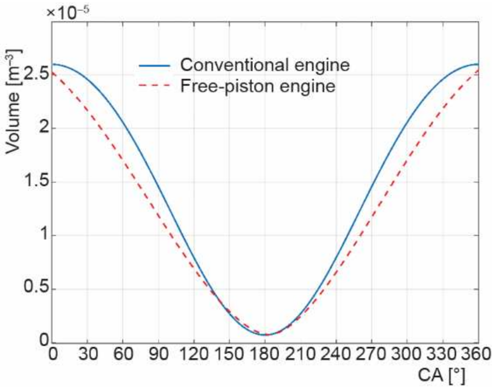

Figure 3. In-cylinder volume profile (free piston engine *vs.* conventional engine)

The variations between the piston motion profile between the free piston engine and the conventional engine will cause variations in the CR and therefore variations in the pressure and temperature profile. Also, as HCCI ignition is controlled by primary by piston dynamics and the species chemical kinetics; the performance of the two-engine type is expected to vary significantly from each other when hydrogen is added to the fuel-air mixture. The following discussion will consider the effect of hydrogen addition to the free piston performance. The results can be compared to those of the conventional engines with hydrogen addition mentioned in the literature review.

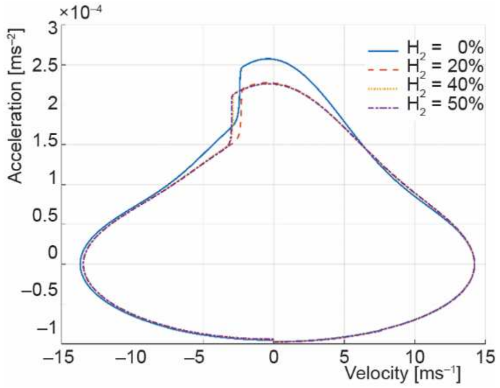

Figure 4. Piston acceleration *vs.* velocity at different hydrogen ratios

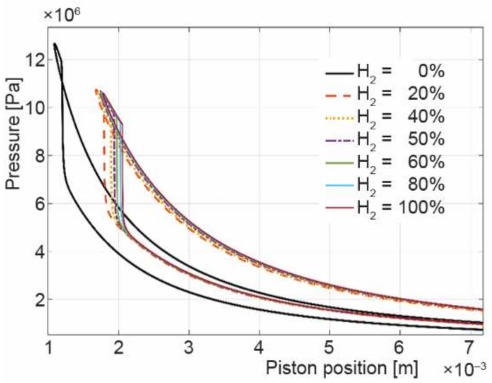

Figure 5. Pressure-position diagram at different hydrogen-NG ratios

It was mentioned that free piston engines differ mainly from conventional engine in their piston dynamics nature, fig. 4, the acceleration *vs.* the velocity of the piston are presented at different hydrogen addition ratios. The addition of hydrogen causes piston declaration near the TDC the piston motion as the figure indicates where adding more hydrogen will end with no change in the piston acceleration. The rate of change in the piston acceleration explains any changes could happen in the engine power output as the last is related directly to the combustion pressure force. The more the reduction in the piston acceleration, the more of the energy available to the engine generators as the piston will store less kinetic energy.

Figure 5 shows the pressure-position diagram of the free piston engine at different percentages of hydrogen addition. When comparing the case of pure NG (H_2 = 0.0) with other cases, that include hydrogen addition, one can recognize that the difference is significant in both of in-cylinder peak pressure and piston location near the TDC. In the case of pure NG, the piston at the TDC is closer to the cylinder head and the in-cylinder peak pressure is much higher. However, when hydrogen is added by 20% for example. A large drop in the peak pressure is observed and the piston is shifted away from the cylinder head. By considering greater hydrogen to NG ratio, the change in the peak pressure and piston location becomes much smaller and for relatively high hydrogen to NG ratios (>60%); the change becomes even negligible. As hydrogen requires less heat to reach the ignition point compared to NG, variations in the peak pressure and piston location are noticed between the different hydrogen addition cases. Small amounts of hydrogen can change the total fuel-air combustion characteristics, especially in terms of fuel-air ignition conditions, thus the peak pressure and piston location will vary. However, the amount of adverse work (*i. e.*, the inversion on the top of pressure-position diagram) is increased as hydrogen is presented in the engine fuel-air mixture. This indicates that the piston cannot convert the heat energy into mechanical energy in the same rate of converting the chemical energy of the fuel to heat energy. As the piston reaches the exhaust port, the gas exchange process will take place by replacing the in-cylinder residual gases by a new fresh charge.

The pressure variations during this process are presented in fig. 6 in terms of pressure against the equivalent CA. By comparing two cases, one of pure NG and another with 50% hydrogen addition, the difference in the pressure is not significant during the process and it is not negligible by the end of the scavenging. The fact that the species transport during scavenging is influenced by the pressure and temperature of the involving gases and not by their reactions, explains why adding hydrogen has no effect on the gas exchange process.

In fig. 7, the difference in both of initial pressure and initial temperature is presented against hydrogen-NG ratio. The values of initial pressure and temperature have been calculated once the exhaust port is closed by the end of the scavenging process. The addition of hydrogen seems has no effect on the pressure and temperature values of the gas mixture, which means that these two parameters are mostly independent on the hydrogen addition (*i. e.*, chemical content). On the other hand, other scavenging conditions, such as the induction pressure and engine speed, are dominant factors in defining the initial pressure and temperature of the engine cycle.

The effect of hydrogen addition on the heat release rate is shown in fig. 8. Two main combustion aspects can be compared for each case of hydrogen addition. The first is the ignition delay and the second is the heat release rate. It is obvious from fig. 8 that adding hydrogen will reduce the ignition delay noticeably, as hydrogen percentage is more than 20%. Ignition delay, which is defined also by resistance toward knocking, becomes shorter, because hydrogen requires less energy to reach the self-ignition point. At the same time, hydrogen oxidation rate is much faster than NG gas, which includes long chain HC reactions. On the other hand, hydrogen has much higher heat release rate than NG gas, which explains why adding more hydrogen results with higher rates of heat. In terms of combustion duration, NG gas has longer burning period than hydrogen, due to the fact that the first has more resistance toward ignition and more complex molecular ponds than hydrogen.

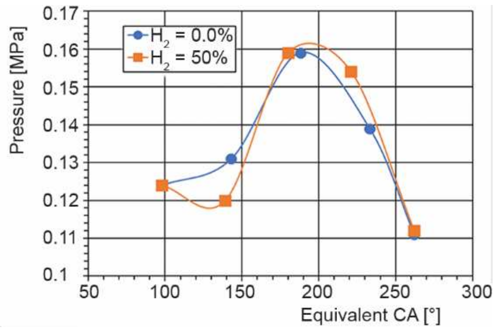

Figure 6. Pressure-position diagram at different hydrogen-NG ratio

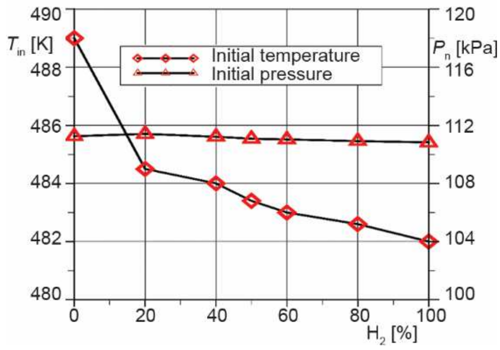

Figure 7. Initial pressure and temperature by the end of the scavenging process

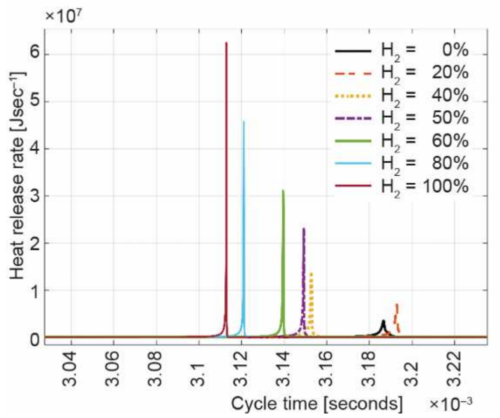

Figure 8. Heat releasing rate at different hydrogen-NG ratios

Due to the change in ignition delay and combustion duration, the engine overall efficiency, presented by the ratio between the power of the electrical generator to the total heat release amount, is expected to vary with hydrogen addition. In fig. 9, a significant increase in the overall efficiency of the engine is observed as hydrogen is added by 20% to NG. For hydrogen-NG ratio above 20%, the engine effective efficiency is within 40% with a rise of 4% from

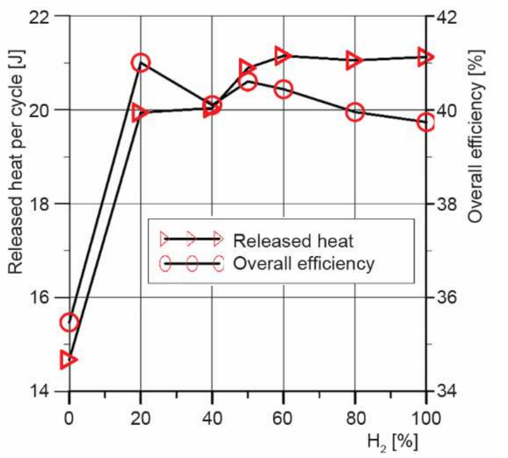

Figure 9. Overall efficiency and accumulated heat at different hydrogen-NG ratio

the pure NG case. The increase in the overall efficiency is explained by the increase in the heat of combustion as hydrogen percentage is increased which is indicated in fig. 10. The presence of high amount of combustion energy allows the piston to deliver more energy to the generator that will increase the system overall efficiency. For relatively high hydrogen-NG percentages (> 40%), heat release does not vary significantly, but the total efficiency decreases slightly due to the increase in the adverse work as indicted in fig. 5.

As design parameters, maximum in-cylinder pressure and temperature should be considered in any engine modeling process. High cylinder pressure can cause engine failure by increasing the wear at the piston rings. At the same time, high cylinder temperatures are proportional to engine emission especially NO_x emissions, which increase as the peak cylinder temperature increases. In fig. 10, the peak in-cylinder pressure and temperature against different hydrogen-NG ratios are demonstrated. The peak cylinder pressure drops by 2 MPa as hydrogen is added in 20%, but very slight change in its value is observed for ratios above 20%. The change in compression explains such drop in the peak cylinder pressure.

Figure 11 indicates that there is a significant reduction in the CR as long as hydrogen is added to the pure NG. The fuel-air mixture has low resistance to reach the ignition state as hydrogen species are implemented. However, a gradual rise in the cylinder peak temperature is obtained as hydrogen-NG ratio is increased, as can be concluded from fig. 11. As the heating value of hydrogen is high when compared with pure NG, the cylinder peak temperature is expected to increase in the presence of higher hydrogen-NG ratios.

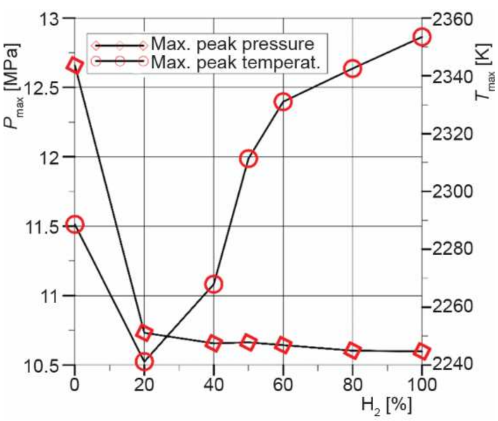

Figure 10. In-cylinder peak pressure and temperature at different hydrogen-NG ratios

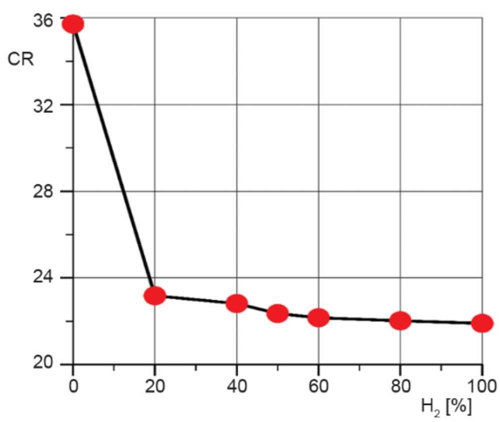

Figure 11. Engine CR at different hydrogen-NG ratio

In fig. 12, residual gases are presented in terms of CO_2 and H_2O species for different hydrogen-NG ratios. Increasing the ratio of hydrogen-NG will decrease CO_2 levels by reducing the amount of carbon mass available for oxidation. On the other hand, H_2O mass fraction increases due to the excess of hydrogen content available for the combustion process. Residual gases have a significant effect on the combustion characteristics of the engine as they have higher heat capacity than the fuel-air species, which can lower the heat release rate and, therefore, decrease the peak temperature in the combustion chamber. Exhaust gas re-circulation (EGR) principle applies basically the idea of re-circulation some of the residual gases in order to decrease engine emissions, such as NO_x.

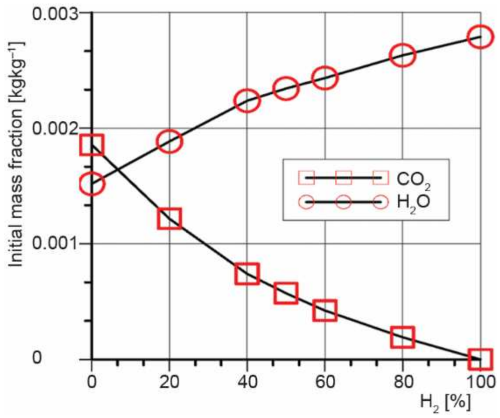

Figure 12. Initial residual gas mass fraction (H_2O, CO_2) at different hydrogen-NG ratios

In the current study, the effect of hydrogen addition on free piston engine emissions has been also investigated. The study focuses on two main emissions, CO and NO_x. The CO is a toxic gas and has many healthy and hazardous issues and it is caused due to the uncomplete oxidation of the fuel-air mixture [21]. The NO_x emissions, on the other hand, is one of the most dangerous emissions that has many healthy and environmentally effects [21]. Hydrogen addition changes the nature of the fuel-air combustion and, therefore, it is expected to have effects on the engine emissions. As hydrogen-NG ratio increases, the engine produces less CO emissions as indicated in fig. 13. Obviously, the reduction in carbon content due to the increase in hydrogen percentage is the reason behind the drop in CO emissions. On contrast, NO_x emissions start to increase as hydrogen amount increases in the fuel-air mixture. Since NO_x is usually coupled to in-cylinder peak temperature, the high heat release rate attached to the high hydrogen content causes the rise in NO_x levels of the engine.

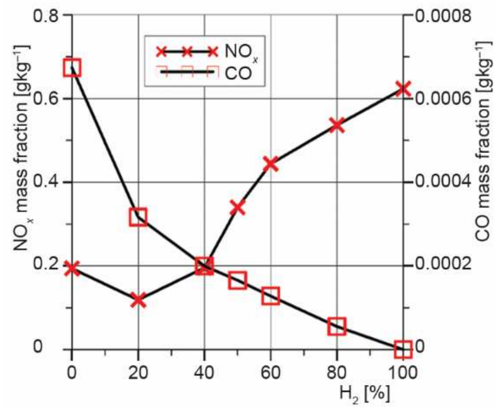

Figure 13. The CO and NO_x emission mass fractions at different hydrogen-NG ratios

The 1-D approach used in simulating the combustion process of fuel-air mixture assumes that the fuel-air mixture is homogenous and it has no significant variations along the combustion chamber. As HCCI combustion requires fuel and air species to be in homogenous and well mixing state, the variations in the fuel distribution through the combustion chamber have been investigated and described as shown in fig. 14. Three different hydrogen-NG ratios have been chosen and compared in terms of the hydrogen mass distribution in the combustion chamber. Figure 14 indicates a slight difference between the hydrogen concentration near the inlet port and this near the exhaust port. Adding hydrogen in higher percentages almost does not change the hydrogen distribution along the region between the engine ports. Therefore,

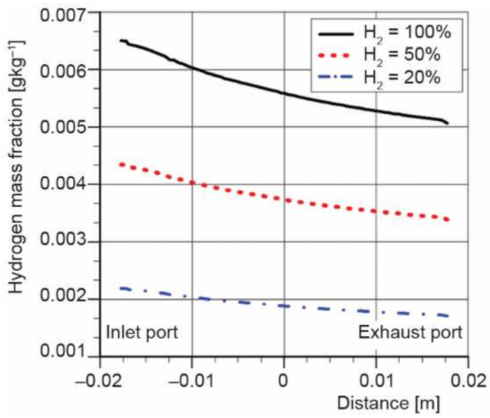

Figure 14. Hydrogen mass distribution within the combustion chamber at different hydrogen-NG ratio

using the 1-D approach is fairly enough for describing the fuel-air reactions especially that chemical kinetics is of high accuracy when compared to other empirical approaches.

Conclusions

The current study presents interesting investigations regarding the use of hydrogen as a fuel additive in free piston engines that employ HCCI combustion mode. Numerical approach, including 0-D and 3-D-CFD analysis, has been used to obtain the study results. The simulation process has been described by using two main models, the combustion model which is responsible about calculating the heat release rate of combustion. In this model, chemical kinetics approach has been used through Cantera toolkit to estimate heat release rate and emissions of the free piston engine under HCCI conditions. The other model is the scavenging model, in which the gas exchange process between the combustion products and the fresh charge is simulated using 3-D-CFD tools available in ANSYS FLUENT software. The results show that hydrogen addition has the ability to influence free piston engine dynamics by affecting the combustion characteristics of the engine. Hydrogen reduces ignition delay time and combustion duration of the air fuel-mixture and, therefore, allows the engine to work with lower CR . Hydrogen also decreases the in-cylinder peak pressure, but at the same time allows the engine to operate with higher mechanical efficiency as it increases the heat release rate. However, hydrogen addition causes the in-cylinder peak temperature to increase, which increases NO_x emission levels of the engine. On the other hand, hydrogen addition decreases CO levels due to the reduction in the carbon content in the fuel-air mixture. Finally, hydrogen has no significant effect on the initial pressure, temperature and gas mixture species as they all depend on different scavenging parameters mainly. The distribution of the fuel-air species through the combustion chamber is almost homogenous with slight variations between the inlet and exhaust ports and no effect on the distribution is observed as hydrogen-NG ratio is changed.

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