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Performance prediction of spark-ignition engine running on gasoline-hydrogen and methane-hydrogen blends

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HIGHLIGHTS

- One-dimensional model is developed with gasoline–hydrogen and methane–hydrogen blends.
- Experimental measurements were implemented to verify the developed model.
- Parametric analysis was performed in terms of mass fraction, engine speed and equivalence ratio.
- Methane–hydrogen blend seemed more attractive than gasoline–hydrogen blends.
- The benefits of hydrogen addition are considerably stronger than the limitations.

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ABSTRACT

Hydrogen is a strong candidate as an alternative fuel and energy carrier which could address problems of environmental pollution, emissions, and geo-political tensions. The aim of this paper is to compare the performance of hydrogen fuel with other fuels and to investigate the power and performance penalty when adding different fractions of hydrogen fuel to the other fuels. A one-dimensional model is developed for an engine with hydrogen and gasoline–hydrogen and methane–hydrogen blends. These models have been calibrated and validated against experimental works and the findings of previous studies. The validation of the pressure trace and the torque showed the predictive capability of the model. Furthermore, the penalty and benefits from hydrogen enrichment were clarified. It was shown that adding small controllable mass factions of hydrogen (<10%) to gasoline enhances the burning velocity and combustion process in the low speed range. However, a small reduction in the output power (<6%) was documented. Adding hydrogen to methane showed greater advantages due to the extremely low burning velocity of methane. The benefits of hydrogen addition are considerably stronger than the limitations. Methane–hydrogen blend seemed more attractive than gasoline–hydrogen blends. It can be seen that the developed simulation codes are powerful tools for the H2ICE community.

1. Introduction

Increased energy use is the universal driver for raising the quality of life in all societies, from developing to developed countries. However, the present reliance on energy from fossil fuels produces unwanted side effects. These effects include environmental pollution which threatens human health, carbon dioxide emissions which accelerate global warming, and geo-political tensions arising from the non-uniform distribution of fossil resources throughout the world. Interest in energy systems based on hydrogen is growing rapidly [1–6]. The challenge is to find highly efficient ways to produce, deliver, and use energy that enhance quality of life but do not threaten the environment and climate or strain geo-political relations. The energy-carrier hydrogen is an alternative to fossil fuels and has the potential to achieve these goals [7–11]. Apparently, there is a need to focus on the positive features as well as the limitations and drawbacks that need to be solved for hydrogen to become an accepted fuel for internal combustion engine applications. As an attempt to overcome the obstacles, several operation strategies have been proposed. Using hydrogen as a dual fuel and running the engine with hydrogen

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blends are the most commonly applied strategies. Dual fuel operation describes any combination of hydrogen and liquid fuels in which several mixture preparation devices are used [4,12]. These devices use separate storage and fuel systems for the different fuels. This operation strategy can reduce the on-board storage problem [13] and can be used in the transition period before hydrogen becomes the sole fuel [14–16]. On the other hand, blended fuel operation refers to combinations of hydrogen with one or several other fuels. Typically, the fuel is already stored and delivered to the engine in blended form using a single carburetion or fuel-injection system [12]. In spark-ignition (SI) engines, methane–hydrogen and gasoline–hydrogen are the most investigated blends.

Methane has a unique tetrahedral molecular structure with larger C–H bond energies. This demonstrates some unique combustion characteristics such as high ignition temperature and low burning velocity [17], leading to the poor lean-burn ability and long combustion duration. Consequently, cycle-by-cycle variations are increased due to the presence of partial burn cycles in natural gas fuelled engines. An effective approach with regard to solving this problem is to mix the natural gas with hydrogen to improve the ignition performance and increase the burning velocity. Previous studies showed that hydrogen-enriched natural gas can promote flame propagation and combustion stability [18–20], leading to a fast burning cycle [21,22] and low cycle-by-cycle variations [23–25]. Similarly, the unique combustion properties of hydrogen can improve performance and reduce emissions from gasoline fuelled engines. Firstly, the flame limit range with hydrogen in the air is 4.1–75% by volume, which is much wider than that of gasoline (1.5–7.6% on a volume basis). Thus, gasoline–hydrogen blends are able to work under much leaner conditions. Moreover, since the ignition energy of hydrogen is ten times lower than that of gasoline, gasoline–hydrogen mixture can be more easily ignited compared to pure gasoline. Therefore, the hydrogen-enriched gasoline engine can achieve a smooth start and good operating stability under lean conditions. It has been commonly agreed that proper lean combustion is an effective way to improve engine thermal efficiency and emissions performance [26]. At the same time, the lower combustion temperature under lean conditions also contributes to a decrease in cooling and exhaust losses. Furthermore, NO_x emissions can also be eased by the reduced in-cylinder temperature [27]. The diffusion coefficient of hydrogen ($0.61\ cm^2/s$) is about four times that of gasoline ($0.16\ cm^2/s$), which improves the mixing process of fuel and air and also helps in improving the homogeneity of the combustible mixture [28]. The adiabatic flame speed of hydrogen (237 cm/s) is five times as large as that of gasoline (42 cm/s), which may contribute to improving engine operating stability [29–31]. However, since the gravimetric density and energy density of hydrogen on a volume basis are much lower than those of gasoline, a reduction in the power densities is expected. In this paper a one-dimensional model is developed, calibrated, and validated for a single-cylinder engine with a port injection feeding system. With this model it is possible to run the engine with two blended fuels: gasoline–hydrogen blends and methane–hydrogen blends.

2. Modeling of dual fuel engine

Using hydrogen as a dual fuel and running the engine with hydrogen blends are the main strategies used to overcome barriers to the use of hydrogen fuel. The dual fuel strategy can reduce the on-board storage problem [13] and can be used in the transition period before hydrogen becomes the sole fuel [14]. Adding hydrogen as a blended fuel is another attractive solution. Methane–hydrogen and gasoline–hydrogen are the most investigated blends. Previous studies have shown that methane–hydrogen can promote flame propagation and combustion stability, leading to a fast burning cycle and low cycle-by-cycle variations [14,18]. Similarly, the unique combustion properties that hydrogen fuel possesses can improve the performance and reduce the emissions from gasoline fuelled engines (GICE) [26,27,32–34]. A one-dimensional model is developed for a single-cylinder engine with a port injection feeding system. With this model, it is possible to run the engine with three pure fuels, namely hydrogen, gasoline, and methane, and also gasoline–hydrogen and methane–hydrogen blends. In these blends, hydrogen is added in controllable fractions on a mass basis of up to 20%.

A model for a single-cylinder four-valve engine with a total swept volume of 149.8 cc and a compression ratio of 10.4:1 was developed as shown in Fig. 1. This engine is a motorbike engine manufactured by Yamaha and made in Malaysia with a port gasoline fuel injection system (model Yamaha FZ150i). The details of the engine parameters and specifications are listed in Table 1. This engine was modeled with dual fuel operation so that it can operate with hydrogen, gasoline, methane, and blends of gasoline–hydrogen as well as methane–hydrogen. Hydrogen was added as a supplementary fuel with controllable mass fractions of up to 20%. The boundary conditions of pressure, temperature, and composition in the engine inlet were defined as 1 atm and 300 K, with air being the intake gas. The effects of the ram air velocity were taken into account, so that the total pressure at the inlet boundary, p_{total}, is calculated as in Eq. (1):

$$p_{total} = p_{atm} + \frac{1}{2} C_{pc} \rho v^2 \quad (1)$$

where C_{pc} is the ram air pressure coefficient, which was set as a calibration parameter. The velocity, v, of the inlet air is specified by the motorbike velocity and the wind velocity. Wind speed increases the ram air speed at the engine inlet if the motorbike drives in the opposite direction to the wind and vice versa.

The air is then taken to the air cleaner, which was modeled as a pipe 0.04 m in diameter and 0.2 m in length with some restriction in the way of the intake air. The velocity of the intake air was monitored continuously throughout every simulation. Then, air is passed to the throttle body by a joint pipe 0.04 m in diameter and 0.144 m in length. The intake manifold, which is located downstream of the throttle orifice, transfers the intake air to the intake port, which was modeled as a pipe 0.038 m in diameter and 0.06 m in length. As mentioned, the intake port is the passage inside the head between the manifold and the cylinder.

In addition, two sequential pulse injectors have been used to inject the fuels in this port. In the intake system, a flow-split was used to divide the intake charge between the two intake valves. The diameter of these valves is 0.0195 m. Another flow split was used to collect the expelled exhaust gases in the exhaust port, which was modeled as a pipe 0.033 m in diameter and 0.06 m in length. The diameter of the exhaust valves is 0.017 m. The exhaust system was modeled as a series of straight and rounded pipes. The muffler at the end of the exhaust system was modeled as a pipe 0.1 m in diameter and 0.4 m in length. For the liquid gasoline fuel, the port injection of the fuel was modeled taking into account the puddling of the fuel on the port walls. The injected fuel mass is separated into vapor fuel, liquid fuel droplets entrained in the air flow, and liquid fuel, which is deposited directly onto the port wall, forming a fuel film, which is illustrated in Fig. 2. The build-up and evaporation of the liquid film are modeled with the transport of the film due to shear forces. In addition, the evaporation of the entrained liquid droplets in the air was also modeled. Several conditions, constants, and parameters have been defined for the injection model. The conditions and parameters used in the injection model are summarized in Table 2.