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Effect of Hydrogen on Natural Gas Fueled Direct Injection Engine, a Theoretical Study

[1,2]J. Zareei, [1]W.F. Wan Mahmood, [1]S. Abdullah, [1]Y. Ali

[1]*Center for Automotive Research, Faculty of Engineering & the Built Environment, University Kebangsaan Malaysia, 43600, Bangi, Selangor, Malaysia*
[2]*Department of Automotive Engineering, Faculty of Mechanical Engineering, Standard Research Institute, Tehran, Iran*

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ABSTRACT

Preparation of air-fuel mixture is considerably dependent on the fluid flow dynamics in order to achieve improved performance, efficiency, and engine combustion in the appearance of flow. In this study, effects on a spark ignition engine of mixtures of hydrogen and compressed natural gas have been numerically considered. This article presents the results of a direct injection engine with mixtures of hydrogen in methane of 0, 7, and 15% by volume. The result shows that the percentage of hydrogen in the CNG increases the burning velocity of CNG and come down the optimal ignition timing to obtain the maximum peak pressure of the engine running with blending of hydrogen to CNG. With adding hydrogen to natural gas, the peak heat release rates increase. For 15% hydrogen, the maximum values at crank angles for in cylinder temperature and heat release rate achieved at 8 degree of CA and the maximum temperature is about 150 K. Port injection gasoline was converted into direct injection by compressed natural gas fuel in this engine.

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INTRODUCTION

Due to the growing concerns about the harmful effects of conventional fossil fuel emissions have made natural gas a very attractive alternative fuel to internal combustion engine (ICE) and due to its advantages to be environmental friendly, clean burning, economical and efficient fuel. For light-duty engine applications in particular, direct injection (DI) of compressed natural gas (CNG) promises thermal efficiencies comparable to those accomplished by high compression ratio, unthrottled diesel engines, while maintaining the smoke-free operation of spark ignition (SI) engines and producing slightly lower NOx emissions (Zeng, K., *et al*., 2006). Natural gas is one of the most promising of alternative fuels, providing lower cost, cleaner emissions and is direct applicable to existing combustion systems. However, the use of natural gas as fuel in internal combustion engines can adversely affect engine performance (Park, J., 2011). Power of these engines is relatively less than gasoline- fueled engines because of the convergence of one or several factors.

Hydrogen gas is characterized by a rapid combustion speed, wide combustible limit and low minimum ignition energy. Such characteristics play a role to decrease engine cycle variation for the safety of combustion. However, it is frequently observed that the values of cycle variation for hydrogen-fueled engines with direct injection are higher than those of hydrogen-fueled engines with manifold injection or those of gasoline engines, due to a decrease in the mixing period by direct injection in the process of compressing hydrogen gas (Kim, J.M., 1995; Nakagawa, Y., 1982; Varde, K.S. and G.A. Frame, 1985).

The percentage of hydrogen in H_2CNG mixture increases the burning velocity of natural gas (Zareei, J., *et al*., 2012) and decreases the optimal ignition timing to obtain the maximum indicated mean pressure of the engine running with these mixtures. The indicated efficiency rises as the percentage of hydrogen in the NG increases (Tinaut, F., *et al*., 2011). Increasing the hydrogen fraction leads to variations in cylinder pressure and CO_2 emissions. Navarro *et al*. (2012) showed that the maximum cylinder pressure rises as the fraction of hydrogen in the blend increases.

Yilanci *et al*. (2009) and Weindorf *et al*. (2007) studied that hydrogen can be alternatively used in various governmental strategic plans as an energy carrier to achieve a sustainable energy system. Hydrogen is a good option due to various methods of hydrogen production, the long-term feasibility of a number of these methods (from nuclear power, from renewable energy: solar, wind, biomass, from fossil fuels etc.), high potential

Corresponding Author: J. Zareei, Center for Automotive Research, Faculty of Engineering & the Built Environment, University Kebangsaan Malaysia, 43600, Bangi, Selangor;
Tel: 60176332435; E-mail: ja_zareei@iust.ac.ir

efficiency at its use point, various energy production methods from hydrogen, and almost no dangerous emission (Shioji, M., 2004). Oxides formation and flame propagation were calculated for two engines with hydrogen fuel through an internal CFD code, with correlation of laminar burning velocity and a model of flame area evolution (Liu, D. and R. MacFarlane, 1983). The use of obtained laminar burning velocities through calculations of chemical kinetic was also investigated.

The results of a four-cylinder engine test with mixtures of hydrogen in methane of 0, 10, 20 and 30% by volume showed that HC, CO , CO2 emission values decrease and NO increase and brake thermal efficiency (BTE) values increase with increasing percentage of hydrogen (Akansu, S.O., 2007). The enhancement of combustion with hydrogen addition can be ascribed to the significant increase of H, O and OH in the flame as hydrogen is presented. The decrease of the mole fractions of CH2O and CH3CHO with hydrogen addition suggests a potential in the reduction of aldehydes emissions of methane combustion as hydrogen is added (Wang, J., 2009).

Hoekstra *et al*. (1994) observed that experimental results of natural gas fueled internal combustion engines claiming that hydrogen as additive in NG can strongly improve the performance of such engines, especially in terms of power, efficiency and emissions allowing the engine to work with leaner mixtures. Hydrogen-natural gas blends, commonly named HCNG, can be distributed using the natural gas infrastructures without significant modifications if hydrogen content is lower than 30% in volume. The results of research of Mariani *et al*. (2012) showed that HCNG blends improved engine brake efficiency, particularly at low loads and for the highest hydrogen content, with fuel consumptions on energy basis over NEDC 2.5%, 4.7% and 5.7% lower than CNG, for HCNG 10, 20 and 30 respectively.

Thermophysical Properties:

A variety of all thermo-physical properties of fluid(s), i.e. thermal conductivity, viscosity, density, specific heats and species diffusivities, as functions of concentration and temperature as well as other variables is provided by STAR-CD modeling framework. Alternative built-in dependencies are provided in some cases; but generally, there are facilities to insert user-specified property functions.

Fuel is combined with air after being introduced to the intake manifold. The process follows the subsequent chemical equation:

$$\varepsilon.\phi(v_1 CH_4(g) + v_2 H_2(g))_{(T_f P_f)} + (0.21O_2 + 0.79N_2 + \varpi H_2O)_{(T_i, P_i)}$$
$$\rightarrow (0.21O_2 + 0.79N_2 + \varpi H_2O)_{(T,P)} + \varepsilon.\phi.(v_1 CH_4(g) + v_2 H_2(g))_{(T,P)} \tag{1}$$

Chemical formulas for combustion of stoichiometric hydrogen–air and methane–air mixture are as follows:

$$CH_4 + 2(O_2 + 3.76N_2) = CO_2 + 2H_2O + 2\times 3.76N_2 \tag{2}$$

$$H_2 + 0.5(O_2 + 3.76N_2) = H_2O + 0.5\times 3.762N_2 \tag{3}$$

In these formulas, ε represents total number of fuel mole, ϕ is ratio of fuel/air equivalence, v_i corresponds to component i mole quantity per fuel mixture mole and ϖ stands for ratio of molar humidity. When pure hydrogen is used, $v_1 = 0$; but, if pure natural gas is used, $v_2 = 0$.

The first law of thermodynamics is applied to the process of mixing in order to calculate thermodynamic properties of the obtained air–fuel mixture. Potential energies and kinetic variations are ignored and it is assumed that no work is performed throughout the process. The equation would be:

$$\sum_{n=1}^{N} \varepsilon.\phi.v1 \left[\int_{T_0}^{T} C^0_{Pi}(T)dT - \int_{T_0}^{T_f} C^0_{Pi}(T)dT \right] + 0.21 \left[\int_{T_0}^{T} C^0_{PO2}(T)dT - \int_{T_0}^{Ti} C^0_{PO2}(T)dT \right]$$
$$+ 0.79 \left[\int_{T_0}^{T} C^0_{PN2}(T)dT - \int_{T_0}^{Ti} C^0_{PN2}(T)dT \right] + \varpi \left[\int_{T_0}^{T} C^0_{PH_2O}(T)dT - \int_{T_0}^{Ti} C^0_{PH_2O}(T)dT \right] - Q = 0 \tag{4}$$

where Q represents transfer of heat during the process, C^0_{Pi} is constant-pressure specific thermal capacity of an ideal gas (Poling *et al*. 2011), T_f stands for fuel temperature before injection, T_0 is reference temperature (298.15 K) and T_1 corresponds to temperature of the intake air.

The obtained air–fuel mixture is then introduced to the cylinder. The residual gas mixing process inside the cylinder and entering of air–fuel mixture into the cylinder occur during this phase.

Geometry Of Engine And Operating Condition:

According to Figure 1, the studied engine model was a typical CNGDI engine with a single cylinder and two exhaust and intake valves. As shown in Figure 2, it was provided with two pistons of different shapes. Two different piston crowns were deemed to examine the pattern and behaviour of turbulence, tumble and swirl intensity field inside the cylinder in the present study with the aim of attaining a suitable piston shape for the engine combustion process. The piston shapes indicate geometry model of real engine, which is usually operated to achieve higher ratio of compression, and the optimum process of combustion in a CNG-DI engine. Two different piston shapes were examined to verify the suitable piston crown for the analysis of heat transfer and turbulence characteristic and preparation of fuel mixing for the subsequent best combustion process. A bowl was positioned at the center of Piston A crown. The bowl volume of piston B was deeper; however, it was not positioned in the crown center.

Fig. 1: A schematic view of typical engine model with piston crown A.

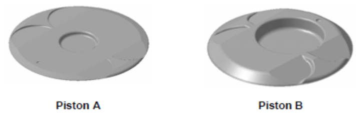

Fig. 2: Geometry of the combustion chambers.

MATERIALS AND METHODS

A single cylinder engine based on the Proton CAMPRO engine was modified into a CNG-fuelled engine with the direct injection system. The engine was operated in a wide open throttle condition with compression ratio of 10:1. Main specifications of the engine are given in Table 1. The experimental cases chosen for validation of the CFD model included both low and high engine speeds, i.e. 2000 and 6000 RPM, respectively, with certain variations in the intake temperature, injection timing, injection duration and spark ignition timing.

Table 1: Geometrical properties of CNG/DI engine.

Engine parameter	Unit	Value
Number of cylinders	4	-
Displacement volume	1596	cm^3
Bore	78	mm
Stroke	84	mm
Compression ratio	10:1	-
Intake valve opening	12	bTDC
Intake valve closing	48	aBDC
Exhaust valve opening	45	bBDC
Exhaust valve closing	10	aTDC
Maximum exhaust valve lift	7.5	mm

RESULTS AND DISCUSSIONS

Figure 3 presents the cylinder pressure versus of the crank angle. The peak cylinder pressures increased and appeared at advanced crank angles with addition of increasing amounts of hydrogen. Specifically, for 15% Hydrogen, the peak cylinder pressure was increased by about 11 bar compared to that for 0% Hydrogen in 2000 rpm, and appeared at an advanced crank angle, approximately 10 crank angle closer to top dead center and obtained at 10 degree crank angle after top dead center. As seen from Figure 3, when hydrogen addition to methane is increased, maximum peak pressure values are found to be close to the top dead center. The maximum peak pressure values are obtained at the engine speed of 6000 rpm.

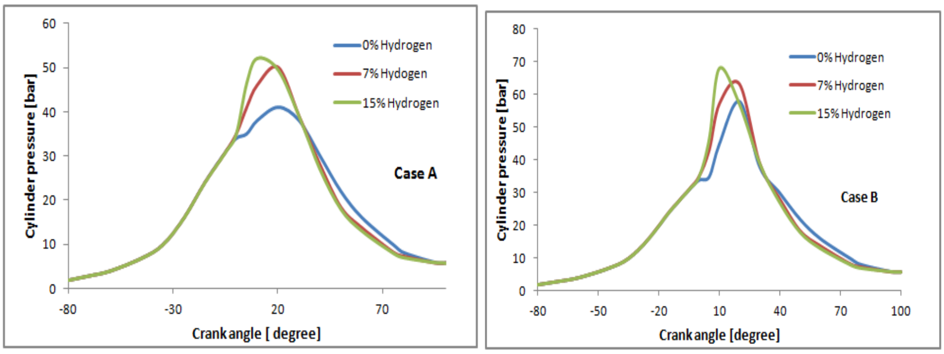

Fig. 3: Cylinder pressure as function of the crank angle at a fixed EAR of unit with different percentage of hydrogen and spark timing of 19CA BTDC at 2000 (case A) and 6000 Rpm (case B).

The in-cylinder temperature (refer Figure 4) increase due to the high adiabatic flame temperature. For 15%Hydrogen, the maximum values at crank angles for in cylinder temperature and heat release rate that were advanced by about 8 CA compared to the corresponding crank angles for 0% Hydrogen. The maximum temperature difference between the 0% H2 and 15% H2 cases was about 150 K. This is because the laminar flame speed of a methane and hydrogen blend increases with hydrogen addition for a fixed excess air ratio. Cylinder temperature is strongly related to engine thermal efficiency and it significantly depends on NOx emissions. This implies that increased heat release and in-cylinder temperature could lead to significantly increased NOx emissions with the addition of hydrogen.

The heat release rate is defined as the rate at which the chemical energy of the fuel is released by the combustion process. It is calculated from cylinder pressure versus crank angle as the energy release required creating the measure pressure. From the simulation results, the rate of heat release is directly extracted from the reactive species and their heat formation. By evaluating the heat release rate produced from the engine, the combustion duration can be predicted to maximize the work done when piston reaches the constant volume stage at the BDC position. Based on the theory of combustion, the combustion depends on the equivalence ratio, residual fraction, spark timing, laminar flame speed, and turbulence intensity of flow and combustion chamber shape (Fergusson, C.R. and A.T. Kirkpatrick, 2011).

Figure 5 shows the heat release rate versus of the crank angle for case A. With adding hydrogen to natural gas, the peak heat release rates increased and the crank angles at which they appeared were advanced.

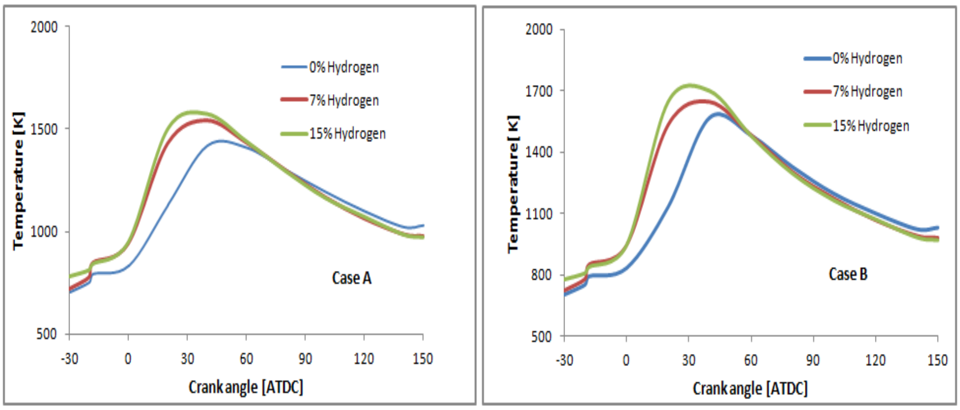

Fig. 4: Cylinder temperature versus the crank angle at a fixed EAR of unit and spark timing of 19CA BTDC at 2000 (case A) and 6000 Rpm (case B).

Conclusions:

In this study, the effects of the addition of hydrogen on a methane-fueled gas engine generator were numerically investigated in terms of engine performance. The results of this study can be summarized as follows:

a) The percentage of hydrogen in the CNG increases the burning velocity of CNG and come down the optimal ignition timing to obtain the maximum peak pressure of the engine running with blending of hydrogen to CNG. The indicated efficiency rises as the percentage of hydrogen in the NG increases. Therefore, as the Hydrogen fraction is increased, maximum peak pressure is found to be close to the top dead center.

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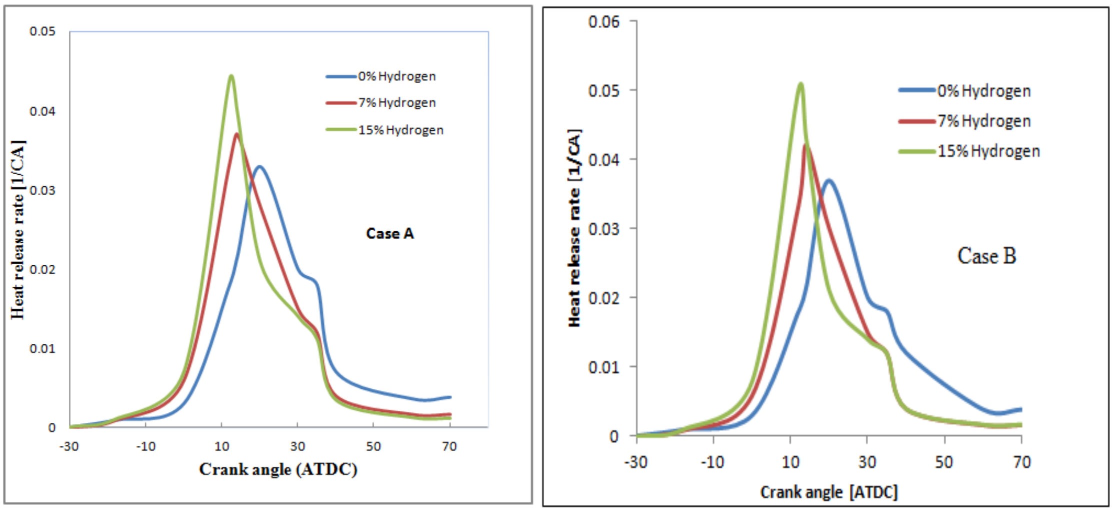

Fig. 5: Heat release rate versus the crank angle at a fixed EAR of unit and spark timing of 19CA BTDC at 2000 (case A) and 6000 Rpm (case B).

b) With adding hydrogen to natural gas, the peak heat release rates increased and the crank angles at which they appeared were advanced.
c) For 15% Hydrogen, the maximum values at crank angles for in cylinder temperature and heat release rate that were advanced by about 8 CA compared to 0% and 7% Hydrogen. The maximum temperature difference between the 0% H2 and 15% H2 cases was about 150 K.

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