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Technical note

Effect of compression ratio, equivalence ratio and engine speed on the performance and emission characteristics of a spark ignition engine using hydrogen as a fuel

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Abstract

The present energy situation has stimulated active research interest in non-petroleum and non-polluting fuels, particularly for transportation, power generation, and agricultural sectors. Researchers have found that hydrogen presents the best and an unprecedented solution to the energy crises and pollution problems, due to its superior combustion qualities and availability. This paper discusses analytically and provides data on the effect of compression ratio, equivalence ratio and engine speed on the engine performance, emissions and pre-ignition limits of a spark ignition engine operating on hydrogen fuel.

These data are important in order to understand the interaction between engine performance and emission parameters, which will help engine designers when designing for hydrogen.

Keywords: Fuel economy; Pollution; Hydrogen; ICE; Power generation; Clean energy; Alternative fuel

1. Introduction

The present energy situation has stimulated active research interest in non-petroleum, renewable, and non-polluting fuels. The world reserves of primary energy and of raw materials are obviously limited. The enormous growth of the world

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Nomenclature

A	cylinder heat transfer area
A_{fl}	flame front area
B	cylinder bore
EH	activation energy = 83740 J mole/k
K	thermal conductivity
M_b	mass of burned gases
N_{cr}	number of mole in crevice
P	cylinder pressure
R_{mol}	universal gas constant
Rpm	engine speed
S	stroke
S_T	turbulent flame front speed
T	gas temperature
T_b	gas temperature of burned zone
T_u	gas temperature of unburned zone
T_w	cylinder temperature
U_p	mean piston speed
X_f	mole fraction of fresh mixture
θ	crank angle
μ	kinematics gas viscosity
ρ	density of gas mixture
ϕ	equivalence ratio
δ	Stefan–Boltzman constant = $5.67e^{-8}$

population during the last decades, the strongly increased technical development and standard of living in the industrial nations have led to an intricate situation in the field of energy supply. Much of the present world's energy demand may still be supplied by exhaustible fossil fuels (natural gas, oil, and coal), which are also the material basis for the chemical industry. It is well known that combustion of fossil fuel causes air pollution in cities and acid rains which damages forests, and leads to the build-up of carbon dioxide, changing the heat balance of the earth.

Everything, which can be done to save our environment, is important. Fossil fuel can be substituted in part, by an alternative energy source, such as hydrogen. This fuel is an attractive substance for the role of the energy vector in many practical applications. While conventional energy sources such as natural gas, oil and coal are non-renewable, hydrogen can be coupled to renewable energy sources.

The study described in this paper provides data on the effect of compression ratio, equivalence ratio and engine speed on the engine performance and emissions of a spark ignition engine operating on hydrogen fuel. An analytical model was developed, tested and verified against the experimental data of the engine. The model is used to study and provide data on the effect of compression ratio, equivalence ratio and engine speed on engine power, optimum spark timing, thermal efficiency, specific fuel consumption, exhaust gas temperature and NO_x emission of a carbureted engine operating on hydrogen fuel. These data help in the understanding of the interaction between engine performance and emission parameters, which will help engine designers while designing for hydrogen.

2. Availability and suitability of hydrogen as an SI engine fuel

The use of hydrogen as an automotive fuel appears to promise a significant improvement in the performance of a spark ignition engine. Besides being the cleanest burning chemical fuel, hydrogen can be produced from water (using non-fossil energy [1]) and conversely, on combustion forms water again by closed cycle [2,3]. The self-ignition temperature of the hydrogen/air mixture is greater than that of the other fuels and, therefore hydrogen produces an antiknock quality of fuel. The high ignition temperature and low flame luminosity of hydrogen makes it a safer fuel than others, it is also non-toxic. Hydrogen is characterized by having the highest energy–mass coefficient of the chemical fuel and in terms of mass energy consumption it exceeds the conventional gasoline fuel by about three times, alcohol five to six times [4]. Therefore, the results clearly establish that hydrogen fuel can increase the effective efficiency of the engine and reduce the specific fuel consumption. A small amount of hydrogen mixed with air produces a combustible mixture, which can be burned in a conventional spark ignition engine at an equivalence ratio below the lean flammability limit of gasoline/air mixture. The resulting ultra lean combustion produces low flame temperature and leads directly to lower heat transfer to the walls, higher engine efficiency and lower exhaust of NO_x [5–7]. The burning velocity of hydrogen/air mixture is about six times higher than that of the gasoline/air mixture. As the burning velocity rises, the actual indicator diagram approaches closer to the ideal diagram and a higher thermodynamic efficiency is achieved [8,9]. The high molecular diffusivity of hydrogen into the air improves the mixture uniformity and hence the combustion efficiency and cycle-to-cycle variation [10]. The use of gaseous fuel (rather than a liquid fuel) for short periods during cold start and warm-up, avoids problems of cold fuel evaporation, uneven distribution of the fuel to the different cylinders due to the presence of a liquid film on the walls of the intake manifold and to unwanted large variations in supplied air–fuel ratio during transient conditions such as acceleration and deceleration [10,11]. The combustion of hydrogen in oxygen produces only water with no CO, CO_2, HC or fine particles, but in air, it produces some oxides of nitrogen. These features make hydrogen potentially an excellent fuel to meet the ever increasingly stringent environmental controls of exhaust emissions from combustion devices, including the reduction of green house gas emissions [12–14].

3. Brief description of the model and validation

The simulation program used in the present work is based on the theory developed by researchers [5–18] and is an extension of the work of Sadiq Al-Baghdadi [6,7]. This has been modified to cover a wide range of engines. A computer quasi-one dimensional model simulating the compression, combustion and expansion processes of the spark ignition engine cycles with all species of exhaust emissions has been developed for hydrogen fuel. The combustion chamber was divided into burned and unburned zones separated by a flame front. The first law of thermodynamics, equation of state and conservation of mass and volume were applied to the burned and unburned zones. The pressure was assumed to be uniform throughout the cylinder charge. A system of first order ordinary differential equations were obtained for the pressure, mass, volume, temperature of the burned and unburned zones, heat transfer from burned and unburned zone, and mass flow into and out of crevices.

The mass burning rate was modeled by the following equation [15];

$$\frac{\mathrm{d}M_\mathrm{b}}{\mathrm{d}t} = A_\mathrm{fl} \cdot \rho \cdot S_\mathrm{T} \tag{1}$$

The turbulent flame front speed (S_T) was modeled by the following semi-empirical formula suggested by Fagelson[16];

$$S_\mathrm{T} = 5000\left(0.1\ \mathrm{rpm} \cdot B \cdot S \cdot P/T_\mathrm{b}^{1.67}\right)^{0.4} \cdot \left(T_\mathrm{b}^{0.41} \cdot T_\mathrm{u}^{1.25}\right) \cdot \left(\frac{R_\mathrm{mol}}{EH}\right) \times \left(\frac{X_\mathrm{f} \cdot \left(1 - \phi \cdot \left(1 - \frac{R_\mathrm{mol} \cdot T_\mathrm{b}^2}{EH \cdot (T_\mathrm{b} - T_\mathrm{u})}\right)\right)}{\phi}\right)^{0.5} \times \exp\left(\frac{-EH}{2 \cdot R_\mathrm{mol} \cdot T_\mathrm{b}}\right) \tag{2}$$

The instantaneous heat interaction between the cylinder content (burned and unburned zones) and its walls were calculated by using the semi-empirical expression for a four stroke engine [9,10];

$$-\frac{\mathrm{d}Q_\mathrm{ht}}{\mathrm{d}t} = A\left[0.26\frac{k}{B}\left(\frac{U_\mathrm{p} \cdot B}{\mu}\right)^{0.7}(T - T_\mathrm{w}) + 0.69\sigma\left(T^4 - T_\mathrm{w}^4\right)\right] \tag{3}$$

The crevices are the volume between the piston, piston rings and cylinder wall. Gases flow into and out of these volumes during the engine operating cycle as the cylinder pressure changes. The instantaneous energy flows to the crevices was calculated by using the semi-empirical expression of Gatowski et al. [18] for a spark ignition engine;

$$\frac{\mathrm{d}Q_\mathrm{cr}}{\mathrm{d}\theta} = (e + R_\mathrm{mol} \cdot T) \cdot \frac{\mathrm{d}N_\mathrm{cr}}{\mathrm{d}\theta} \tag{4}$$

where $\mathrm{d}N_\mathrm{cr} > 0$ when flow is out of the cylinder into the crevice; $\mathrm{d}N_\mathrm{cr} < 0$ when flow is from the crevice to the cylinder; and $(e + R_\mathrm{mol} \cdot T)$ is evaluated at cylinder conditions when $\mathrm{d}N_\mathrm{cr} > 0$, and at crevice conditions when $\mathrm{d}N_\mathrm{cr} < 0$.

The cylinder pressure and the temperatures of burnt and unburned zones were predicted using energy, mass and volume balance equations and the equation of state.

Ten species were considered in the calculation of combustion product concentrations. The following equations were used;

$$\frac{1}{2}H_2 \leftrightarrow H \tag{5}$$

$$\frac{1}{2}O_2 \leftrightarrow O \tag{6}$$

$$\frac{1}{2}N_2 \leftrightarrow N \tag{7}$$

$$2H_2O \leftrightarrow 2H_2 + O_2 \tag{8}$$

$$H_2O \leftrightarrow OH + \frac{1}{2}H_2 \tag{9}$$

$$H_2O + \frac{1}{2}N_2 \leftrightarrow H_2 + NO \tag{10}$$

The calculations were based on the equilibrium assumption except for NO_x formation where the extended Zeldovich mechanism was used.

$$N + NO \;\leftrightarrow\; N_2 + O \qquad K1_f = 3.1 \times 10^{10} \exp\left(\frac{-160}{T}\right) \tag{11}$$

$$N + O_2 \;\leftrightarrow\; NO + O \qquad K2_f = 6.4 \times 10^{6} \cdot T \cdot \exp\left(\frac{-3125}{T}\right) \tag{12}$$

$$N + OH \leftrightarrow\; NO + H \qquad K3_f = 4.2 \times 10^{10} \tag{13}$$

where $K1_f$, $K2_f$ and $K3_f$ are the forward rate constants and were taken from Ref. [10].

The Ricardo E6/US single cylinder four stroke research engine with cylinder bore of 76.2 mm, connecting rod length of 241.3 mm and stroke of 110 mm was used in this research for the validation of the program. Hydrogen mixed with air before throttle valve to enter the intake manifold. The engine power was measured using an electrical dynamometer. The exhaust gas was analyzed for NO_x by chemiluminescent analyzer, CUSSONS equipment. A high-pressure transducer, type AVL-8QP was used to record the cylinder head pressure. The transducer signal was amplified by a CUSSONS-PIEZO channel amplifier, and is stored and presented on the display of a CRT kikusui-COS5020-ST oscilloscope. A pick-up for an angle marker was installed and its signal presented on the oscilloscope display. The tests were performed with variable compression ratios i.e. (6, 7, 8, 9, 10, 11, and 12), at equivalence ratios from rich to lean limit i.e. (1.2, 1.1, 1, 0.9, 0.8, 0.7, and 0.6), and at variable engine speed i.e. (15, 20, 25, 30, 35, 40, and 45 rps). The results of the mathematical model were then verified against the experimental data of the engine, as shown in Fig. 1, which shows that the results predicted by the mathematical model are very close (within 3%) to the experimental results. This verifies that the model developed can be used to a great degree of accuracy.

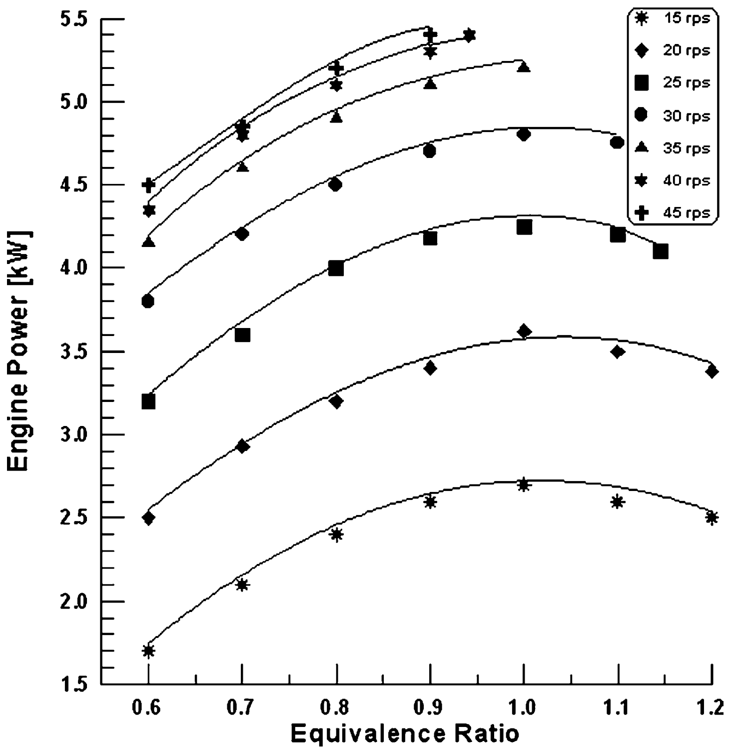

Fig. 1. Measurements (dots) and predictions (solid lines) of the effect of equivalence ratio and engine speed on the engine power.

Hydrogen requires a very low ignition energy, which leads to uncontrolled pre-ignition problems, and this causes a reduction in both brake power and efficiency. Fig. 2 shows the variations of operational limits for pre-ignition with compression ratio changes at optimum spark timing for best torque and 25 rps engine speed. The figure shows that the operational limits for pre-ignition decrease as the compression ratio increases. This is because of the increase in the temperature inside the cylinder. Fig. 3 shows the variations of operational limits for pre-ignition with engine speed changes at optimum spark timing for best torque and HUCR. The figure also shows that the operational limits for pre-ignition decrease as the engine speed increases. The increase of engine speed causes an increase in the turbulence inside the cylinder, which causes an increase in flame speed and hence, the increase in the rate of mass burning. Therefore, the time required for complete combustion is reduced and this produces higher peak pressure and temperature. Hence, higher tendency to unstable combustion [19]. This result demonstrates the high sensitivity of the turbulent flame front speed to the engine speed, pressure and temperature inside the cylinder (Eq. (2)) [6–16]. As was observed from Figs. 2 and 3, the mathematical model is valid within the closed area of these figures.

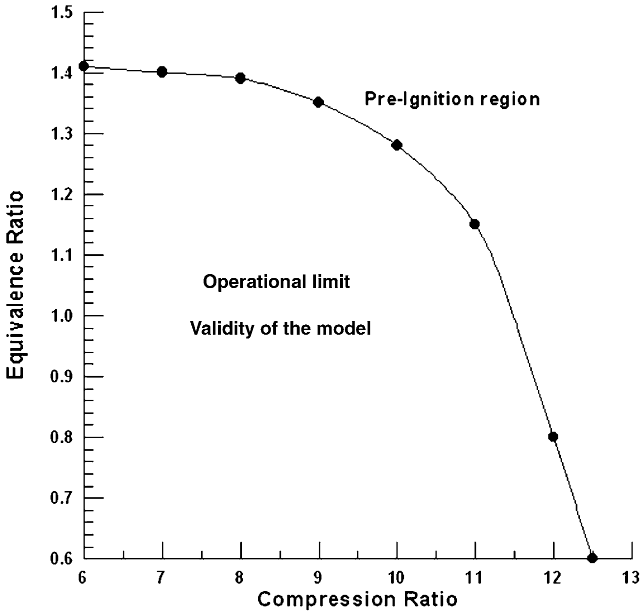

Fig. 2. The variations of operational limits for pre-ignition with compression ratio changes.

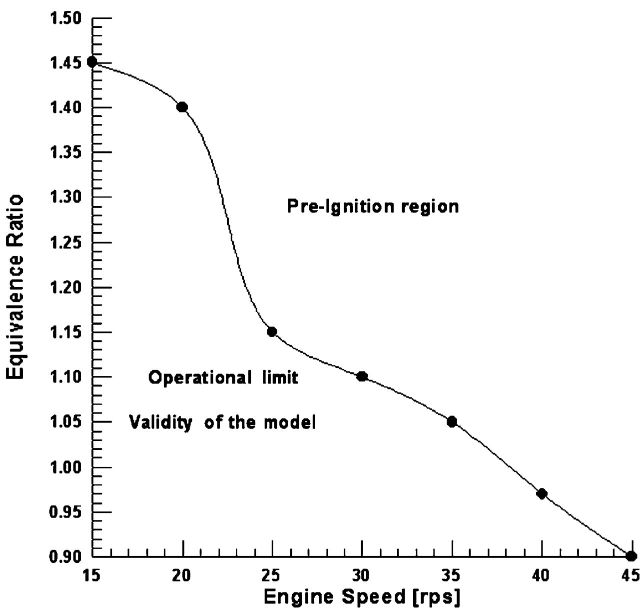

Fig. 3. The variations of operational limits for pre-ignition with engine speed changes.

4. Engine's performance parameters

4.1. Engine power

Fig. 4 shows the effect of compression ratio and equivalence ratio on the engine power, at optimum spark timing for best torque and 25 rps engine speed. The figure shows that the high useful compression ratio (HUCR), which gives the highest power, occurred at the compression ratio (CR) of 11:1. With further increase in compression ratio, the engine power decreases due to unstable combustion. However, the maximum power output of a hydrogen engine is limited by the loss of combustion control, usually described as "pre-ignition". Referring to Fig. 4, it can be observed that operating at lean or rich mixtures tends to decrease the engine power for all compression ratios. Air-to-fuel ratios rich of stoichiometric decrease engine power due to decreasing combustion efficiency. Air-to-fuel ratios lean of stoichiometric decrease engine power due to a reduction in the volumetric lower heating value of the intake mixture, despite increasing combustion efficiency. Fig. 5 shows the effect of engine speed on the engine power, at optimum spark timing for best torque and HUCR. It is clearly shown that as the engine speed increases the engine power of the engine increases.

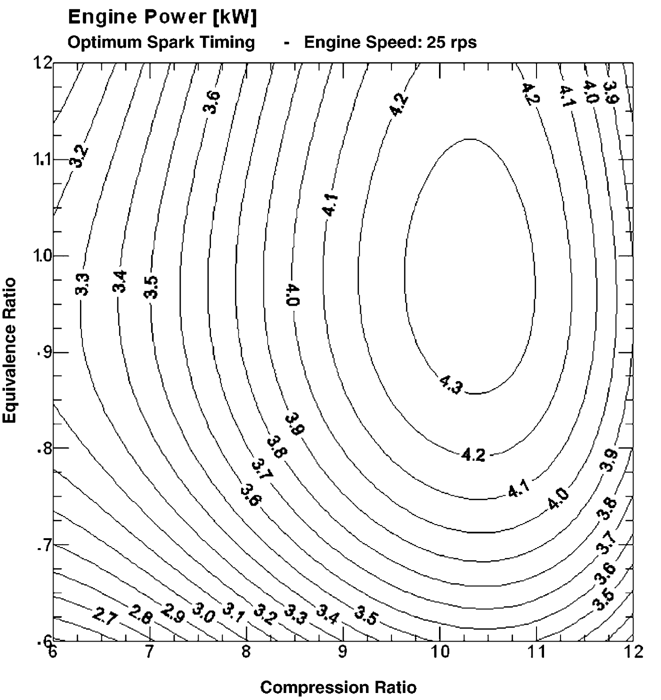

Fig. 4. Effect of compression ratio and equivalence ratio on the engine power.

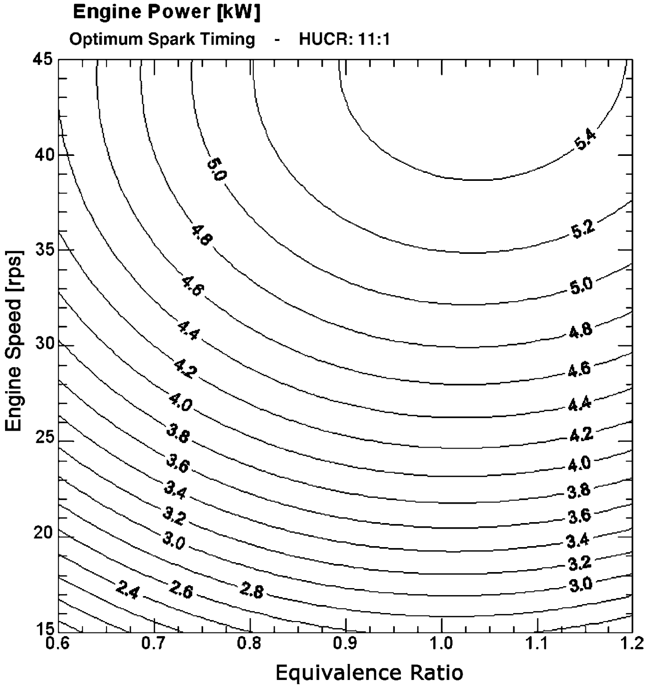

Fig. 5. Effect of engine speed and equivalence ratio on the engine power.

4.2. Optimum spark timing

Fig. 6 shows the effect of compression ratio and equivalence ratio on the optimum spark timing, at 25 rps engine speed. The optimum spark timing decreases as the compression ratio is increased. This is because of the increase in the end-of-compression temperature and pressure and decrease in the fraction residual gases. This creates a favorable condition for the reduction of ignition lag and increase in the flame speed. Referring to Fig. 6, it can be observed that operating at lean or rich mixtures tends to increase the optimum spark timing for all compression ratios. This is because of the lesser thermal energy liberated from the leaner mixtures, which increases the ignition delay and slows the flame propagation. The flame temperature is low at lean and rich mixtures. Further, the incomplete combustion due to oxygen deficiency at rich mixtures also has an adverse effect over the flame speed. Fig. 7 shows the effect of engine speed on the optimum spark timing, at HUCR. It is clearly shown that as the engine speed increases the optimum spark timing increases due to a shorter time in which the flame can burn. This is a clear effect of turbulence. As the engine speed increases, the turbulence inside the cylinder increases, leading to increase the flame front speed [9,10].

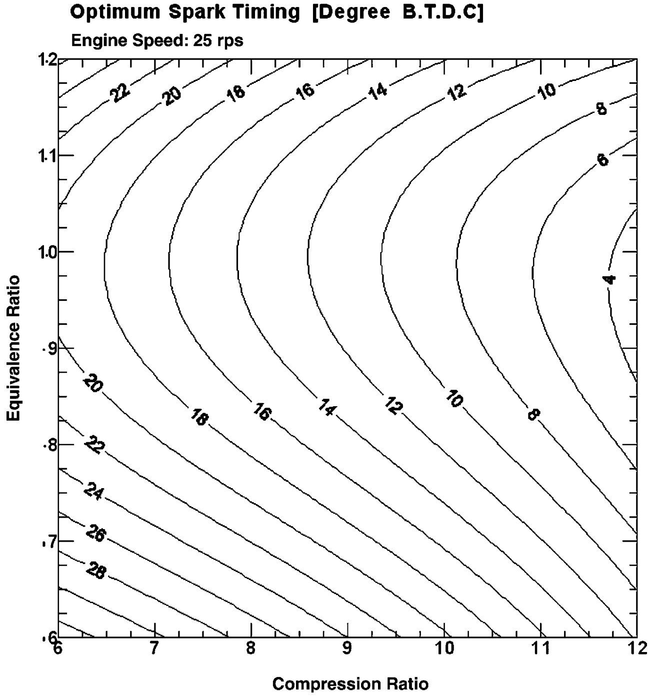

Fig. 6. Effect of compression ratio and equivalence ratio on the optimum spark timing.

4.3. Indicated thermal efficiency

Fig. 8 shows the effect of compression ratio and equivalence ratio on the indicated thermal efficiency, at optimum spark timing for best torque and 25 rps engine speed. Indicated thermal efficiency is partially dependent on combustion efficiency. Combustion efficiency is high when using lean equivalence ratios, as shown by the small amount of products of incomplete combustion. Combustion efficiency decreases for rich equivalence ratios, as there is insufficient oxygen to complete combustion [15]. Indicated thermal efficiency, as seen in Fig. 8, shows an increase in the proportion of energy captured from the fuel as the equivalence ratio decreases for all compression ratios. The figure also shows that the indicated thermal efficiency is improved as the compression ratio is increased, reaching maximum at compression ratios between 10:1 and 11:1. This improvement in the thermal efficiency is attributed to the improvement of the combustion process and hence, improvement in combustion efficiency. With further increase in compression ratio, the thermal efficiency decreases due to unstable combustion.

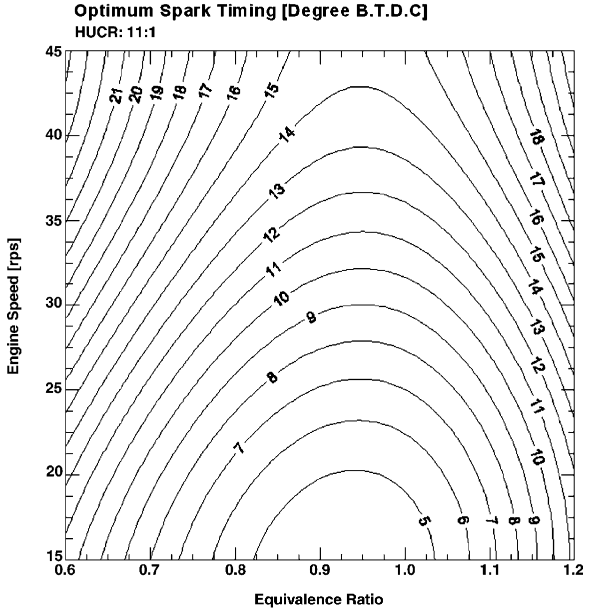

Fig. 7. Effect of engine speed and equivalence ratio on the optimum spark timing.

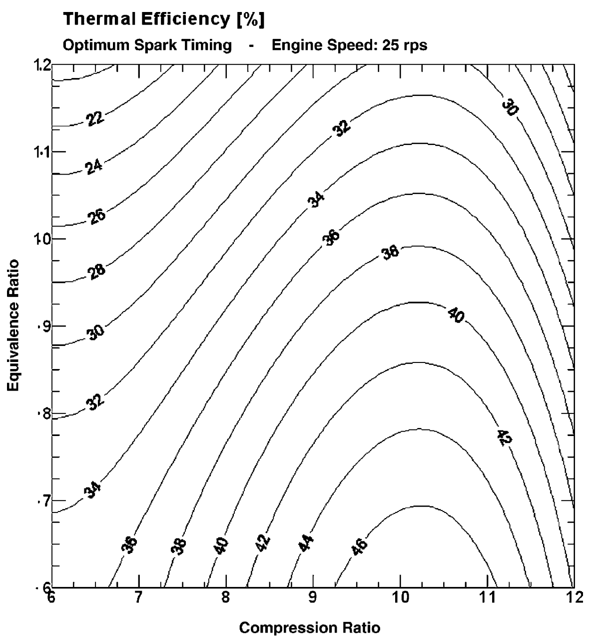

Fig. 8. Effect of compression ratio and equivalence ratio on the indicated thermal efficiency.

4.4. Specific fuel consumption

Fig. 9 shows the effect of compression ratio and equivalence ratio on the specific fuel consumption, at optimum spark timing for best torque and 25 rps engine speed. The specific fuel consumption is decreased as the compression ratio is increased until CR nearly equal to 11:1 where the thermal efficiency is highest and the engine power produced is proportionally high. When the compression ratio is more than 11:1, the specific fuel consumption increases due to unstable combustion. The figure also shows that the specific fuel consumption increases as the equivalence ratio increases from 0.6 to 1.2 for all compression ratios due to reduction in the combustion efficiency and hence, reduction in the proportion of energy captured from the fuel.

4.5. Exhaust gas temperature

Fig. 10 shows the effect of compression ratio and equivalence ratio on the exhaust gas temperature, at optimum spark timing for best torque and 25 rps engine speed. The exhaust gas temperature decreases as the compression ratio

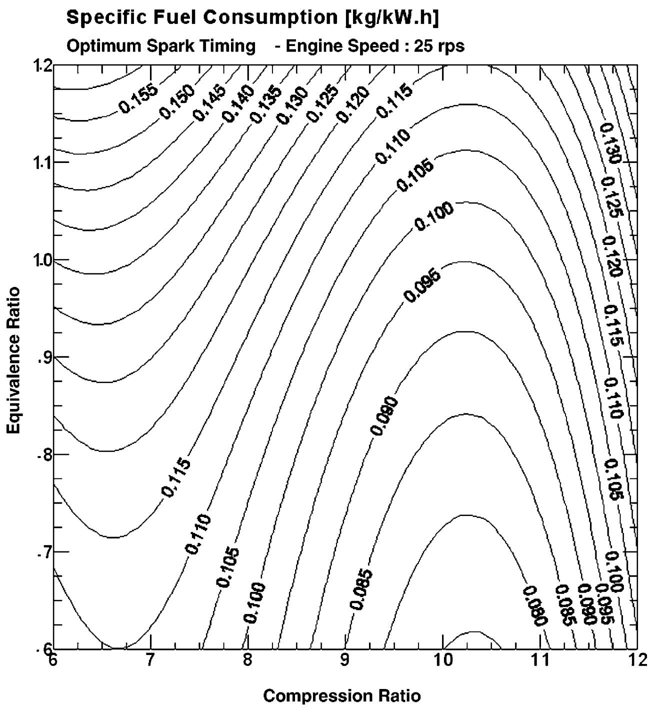

Fig. 9. Effect of compression ratio and equivalence ratio on the specific fuel consumption.

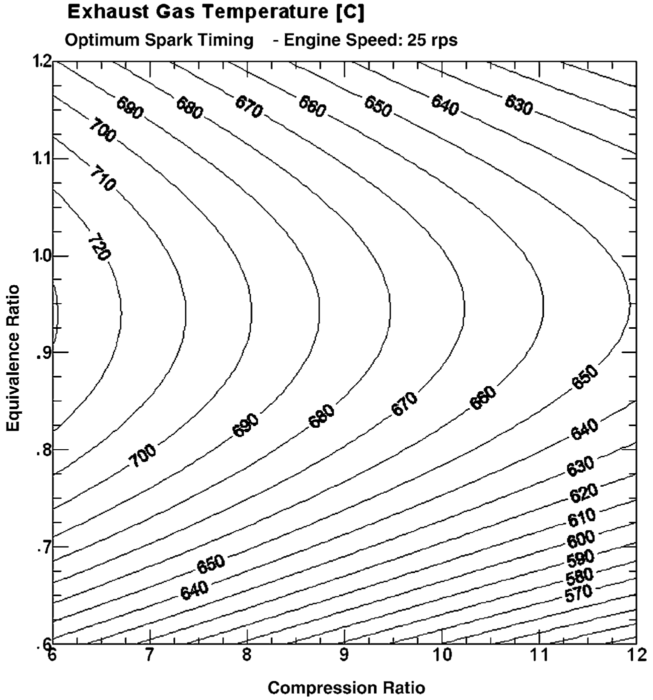

Fig. 10. Effect of compression ratio and equivalence ratio on the exhaust gas temperature.

increases for all equivalence ratios. The increase of the compression ratio causes an increase in burning velocity and hence, the increase in the rate of mass burning. Therefore, the time required for complete combustion is reduced and this produces lower exhaust gas temperature [9,10]. Also from this figure it can be seen that the exhaust gas temperature is maximum at equivalence ratios around 0.9–1.0, for all compression ratios. This is due to the improvement in combustion efficiency around this range, and this produces higher combustion temperature [12].

5. Engine emissions

Fig. 11 shows the effect of compression ratio and equivalence ratio on the NO_x emission, at optimum spark timing for best torque and 25 rps engine speed. The NO_x emission increases as the compression ratio increases, for all equivalence ratios less than 0.8 due to the high combustion temperature with abundance of oxygen. But, the NO_x emission decreases as the compression ratio increases, for all equivalence ratios more than 0.8 (although combustion temperatures are still high), due to a decreasing amount of oxygen.

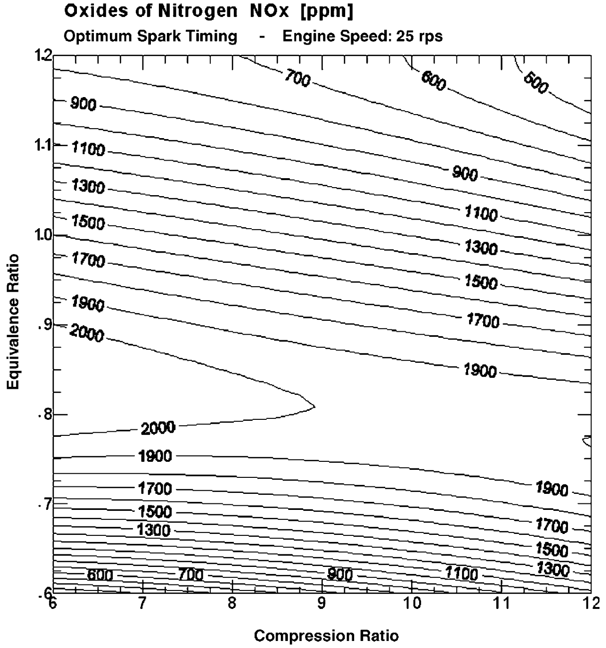

Fig. 11. Effect of compression ratio and equivalence ratio on the NO_x emission.

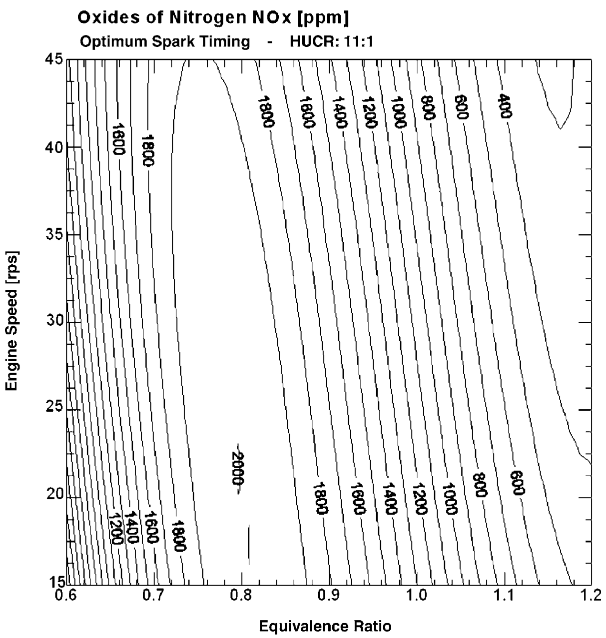

Fig. 12. Effect of engine speed and equivalence ratio on the NO_x emission.

Referring to Fig. 11, it can be observed that the NO_x emission reaches a maximum value at equivalence ratio equal to 0.8. This is probably a result of the higher combustion temperature prevailing inside the cylinder with abundance of oxygen. From peak production, the concentration decreases for richer equivalence ratios (although combustion temperatures are still high), due to a decreasing amount of oxygen. For leaner equivalence ratios, the decrease in NO_x is primarily a reflection of decreasing combustion temperature. It is notable that the chemical processes controlling the net NO emissions evolved to different stages for different mixture stoichiometries. For equivalence ratios less than 0.8 the NO is controlled by quenching of formation reactions during the expansion stroke, while for richer mixtures the NO emissions are primarily determined by the quenching of decomposition reactions during the expansion process [16].

Fig. 12 shows the effect of engine speed on the NO_x emission, at optimum spark timing for best torque and HUCR. The NO_x emission increases as the engine speed increases, for all equivalence ratios less than 0.8 due to the increases in the maximum temperature in the cycle with abundance of oxygen in addition to reduction of the time required for dissociating NO to N_2 and O_2 [9]. But, the NO_x emission decreases as the engine speed increases for all equivalence ratios more than 0.8 due to a decreasing amount of oxygen.

6. Conclusions

1. Engine operating parameters have to be carefully chosen by the designer, taking into account their effect on the engine performance and emission.
2. Any attempt to control emissions by operating the engine at leaner mixtures has to take into account the effect on other variables like power.
3. Compression ratio and equivalence ratio have a significant effect on both performance and emission characteristics of the engine and have to be carefully designed to achieve the best engine performance characteristics.
4. Higher engine rotational speeds can be used in lean mixtures to increase the power output of an engine operating on hydrogen while maintaining high efficiency and pre-ignition free operation.
5. The variation in spark timing with hydrogen is very effective in controlling the combustion process.
6. Higher compression ratios can be applied satisfactorily to increase the power output and efficiency, mainly because of the relatively fast burning characteristics of the hydrogen–air mixtures.

References

[1] Veziroglu TN, Barbir Frano. Solar-hydrogen energy system: the choice of the future. Environ. Conserv. 1991;18(4):304–12.
[2] Veziroglu TN, Gurkan I, Padki MM. Remediation of greenhouse problem through replacement of fossil fuels by hydrogen. Int. J. Hydrogen Energy 1989;14(4):257–66.
[3] Veziroglu TN, Barbir FH. Hydrogen, the wonder fuel. Int. J. Hydrogen Energy 1992;17(6):391–404.

[4] Veziroglu TN, Petkov T, Sheffield JW. An outlook of hydrogen as an automotive fuel. Int. J. Hydrogen Energy 1989;14(7):449–74.
[5] North DC. Investigation of hydrogen as an internal combustion fuel. Int. J. Hydrogen Energy 1992;17(7):509–12.
[6] Al-Baghdadi Maher ASadiq, Al-Janabi Haroun AShahad. A prediction study of the effect of hydrogen blending on the performance and pollutants emission of a four stroke spark ignition engine. Int. J Hydrogen Energy 1999;24(4):363–75.
[7] Al-Baghdadi Maher AS, Al-Janabi Haroun AS. A prediction study of a spark ignition supercharged hydrogen engine. Energy Conversion Manage. 2003;44(20):3143–50.
[8] Desoky AA, El-Emam SH. A study on the combustion of alternative fuel in spark ignition engines. Int. J. Hydrogen Energy 1985;10(8):497–504.
[9] Sher E, Hacohen Y. Measurements and predictions of the fuel consumption and emission of a spark ignition engine fueled with hydrogen-enriched gasoline. Proc. Instan. Mech. Engrs 1989;203:155–62.
[10] Sher E, Hacohen Y. On the modeling of a si 4-stroke cycle engine fueled with hydrogen-enriched gasoline. Int. J. Hydrogen Energy 1987;12(11):773–81.
[11] Al-Baghdadi Maher AS. A study on the hydrogen–ethyl alcohol dual fuel spark ignition engine. Energy Conversion Manage. 2002;43(2):199–204.
[12] Ghazi AK. Hydrogen as a spark ignition engine fuel. Int. J. Hydrogen Energy 2003;28(5):569–77.
[13] Jehad AA, Yamin HN, Gupta BBB, Srivastava ON. Effect of combustion duration on the performance and emission characteristics of a spark ignition engine using hydrogen as a fuel. Int. J Hydrogen Energy 2000;25(6):581–9.
[14] Das LM, Rohit Gupta, Gupta PK. Performance evaluation of a hydrogen-fuelled spark ignition engine using electronically controlled solenoid-actuated injection system. Int. J. Hydrogen Energy 2000;25(6):569–79.
[15] Heywood JB. Internal Combustion Engine Fundamentals. McGraw-Hill; 1989.
[16] Fagelson JJ, Mclean WJ, De Boer PCT. Performance and NO_x emissions of spark ignited combustion engines using alternative fuels-quasi one-dimensional modeling. J. Combustion Sci. Tech. 1978;18:47–57.
[17] Al-Baghdadi MAS, Al-Janabi HAS. Improvement of performance and reduction of pollutant emission of a four stroke spark ignition engine fueled with hydrogen–gasoline fuel mixture. Energy Conversion Manage. 2000;41(1):77–91.
[18] J.A. Gatowski, E.N. Balles, F.E. Nelson, J.A. Ekchian, J.B. Heywood, Heat release analysis of engine pressure data, SAE, 1985; paper no. 841359. pp. 5.961–5.977.
[19] A.M. Douaud, P. Eyzat, Four-octane-number method for predicting the anti-knock behavior of fuels and engines. SAE, 1979; paper no. 780080, pp. 294–308.