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Improvement of performance and reduction of pollutant emission of a four stroke spark ignition engine fueled with hydrogen–gasoline fuel mixture

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Abstract

The effect of the amount of hydrogen/ethyl alcohol addition on the performance and pollutant emissions of a four stroke spark ignition engine has been studied. A detailed model to simulate a four stroke cycle of a spark ignition engine fueled with hydrogen–ethyl alcohol–gasoline has been used to study the effect of hydrogen and ethyl alcohol blending on the thermodynamic cycle of the engine. The results of the study show that all engine performance parameters have been improved when operating the gasoline S.I.E. with dual addition of hydrogen and ethyl alcohol.

It has been found that 4% of hydrogen and 30% of ethyl alcohol blending causes a 49% reduction in CO emission, a 39% reduction in NO_x emission, a 49% reduction in specific fuel consumption and increases in the thermal efficiency and output power by 5 and 4%, respectively.

When ethyl alcohol is increased over 30%, it causes unstable engine operation which can be related to the fact that the fuel is not vaporized, and this causes a reduction in both the brake power and efficiency.

Keywords: Combustion; Energy; Performance; Fuels; Thermodynamics; Hydrogen; Alcohol; Pollution; Fuel economy

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Nomenclature

A	cylinder heat transfer area
A_{fl}	flame front area
[Air]	molar concentration of air
B	cylinder bore
ff	turbulent flame factor
[F]	molar concentration of liquid fuel
[H]	molar concentration of hydrogen fuel
k	thermal conductivity
M_b	mass of burnt gases
N_{cr}	number of moles in crevice
X_b	mass fraction of burnt gases
T	gas temperature
$QC.R$	energy exchange across system boundary
$QH.T$	overall heat transfer rate to cylinder wall
BDC	bottom dead center
TDC	top dead center
$ATDC$	after top dead center
$BTDC$	before top dead center
$S.F.C$	specific fuel consumption
X_R	mole fraction of residual gas
Y_{H2}	amount of hydrogen addition
R	flame front radius
rpm	engine speed
S_L	laminar flame front speed
S_T	turbulent flame front speed
t	time
T_W	cylinder wall temperature
V_{cr}	crevice volume
V_{cyl}	cylinder volume
$e(T)$	specific internal energy at temperature T
R_{mol}	universal gas constant
U_P	mean piston speed
D_P	delay period
T_o	reference temperature
P_o	reference pressure

Greek symbols

θ	crank angle
μ	kinematic gas viscosity
ρ_u	density of unburnt gas mixture
ρ_b	density of burnt gas mixture
σ	Stefan–Boltzmann constant $= 5.67e^{-8}$
$\Delta\theta$	crank angle step
ϕ	equivalence ratio

1. Introduction

A large number of researches have been directed towards the development of alternative energy sources and alternative fuels. Hydrogen is considered an ideal alternative fuel. Many workers studied the effect of using hydrogen as a fuel (pure or mixed with gasoline) on engine performance and pollutant emissions [1–4].

A hydrogen–gasoline fueled engine generally develops lower maximum power and higher NO_x emissions compared to an equivalent gasoline engine, due to the restricted air flow and increase of the maximum temperature inside the cylinder, respectively. To decrease the amount of NO_x emissions, hydrogen is added at lean combustion operation. These conditions (lean mixture blended with hydrogen) give lower levels of NO_x emissions compared with that of a pure gasoline operation, but with more deterioration in engine power. In the present work, ethyl alcohol has been added with different volume ratios to the gasoline fuel to improve the output power and reduce the NO_x emissions of a hydrogen supplemented fuel engine. Consequently, a computer program is constructed and developed using a Quasi-Dimensional Model with a set of semi-empirical equations to simulate the combustion process in a reciprocating spark ignition engine which is fueled with a hydrogen–gasoline–alcohol fuel mixture. The advantage of the hydrogen supplemented fuel is that it requires a smaller quantity of hydrogen, which considerably reduces the problems connected with hydrogen storage in the automobile.

2. Fuel supply to the engine

The Ricardo E6/US single cylinder research engine has been used in this research. The technical details of the engine are given in Table 1. The liquid fuel (gasoline and ethyl alcohol) enters the air stream through the liquid fuel discharge tube in the carburettor body and is atomized and convected by the air stream past the throttle plate and into the intake manifold, Fig. 1. Liquid fuel evaporation starts within the carburettor and continues as fuel droplets move with the air stream. Hydrogen is mixed with the gasoline/ethyl alcohol air mixture before the throttle valve to enter the intake manifold. The engine power has been measured using an electrical dynamometer. The exhaust gas was analyzed for CO by a non-dispersive infrared analyzer, NDIR, and for NO_x by a chemiluminescent analyzer, CUSSONS. A high pressure

Table 1
The technical details of the engine

Type	Ricardo E6/US, spark ignition engine
Cycle	Four stroke
Number of cylinders	1
Cylinder bore	0.0762 m
Stroke	0.11 m
Connecting rod length	0.2413 m
Compression ratio	7.5
Speed	1500 rpm

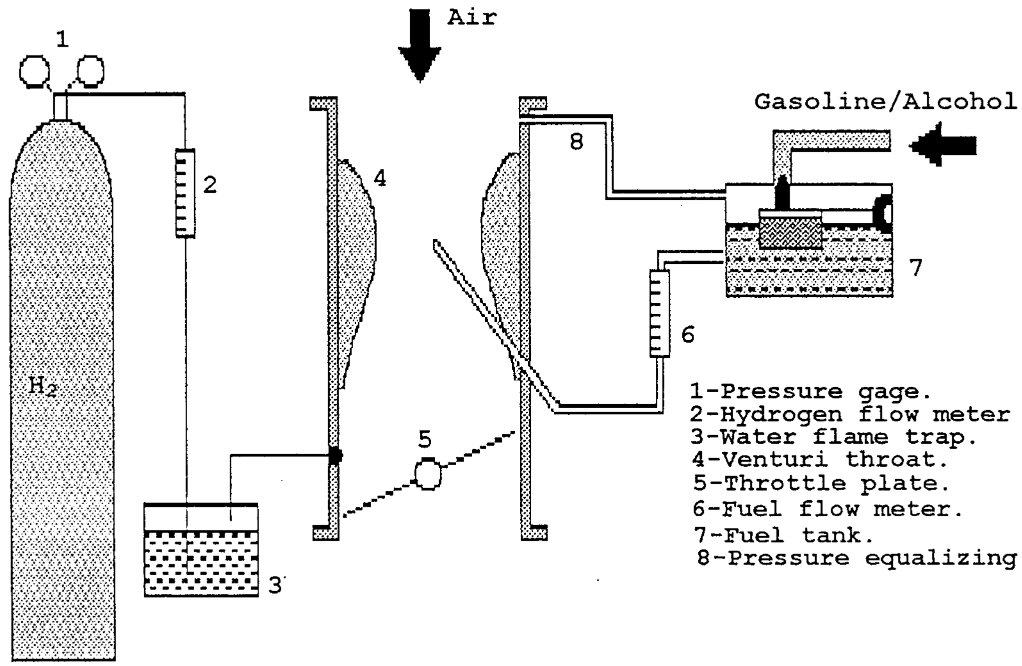

Fig. 1. Schematic of fuel supply to the engine.

transducer, type AVL-8QP, was used to record the cylinder head pressure. The transducer signal has been amplified by a CUSSONS-PIEZO channel amplifier and is then stored and presented on the display of a CRT Kikusui-COS5020-ST oscilloscope. A pick-up for an angle marker has been installed, and its signal is also presented on the oscilloscope display.

3. Theoretical consideration

In order to analyze the complete thermodynamic cycle with the objective of predicting the engine performance at different amounts of hydrogen and/or ethyl alcohol supplement, the following approach has been adopted.

The combustion chamber is generally divided into burnt and unburnt zones, separated by a flame front. The cylinder charge has been assumed to be composed of ideal gases, frozen in the unburnt zone and in chemical equilibrium (except for NO emission) in the burnt zone. The first law of thermodynamics, equation of state and conservation of mass and volume were applied to the burnt and unburnt zones. The pressure is assumed to be uniform throughout the cylinder charge. A system of first-order ordinary differential equations have been obtained for the pressure, mass, volume, temperature of the burnt and unburnt zones, heat transfer from the burnt and unburnt zones, and mass flow into and out of crevices. The crevices are the volumes between the piston, piston rings and cylinder wall, Fig. 2. Gases flow into and out of these volumes during the engine operating cycle as the cylinder pressure changes.

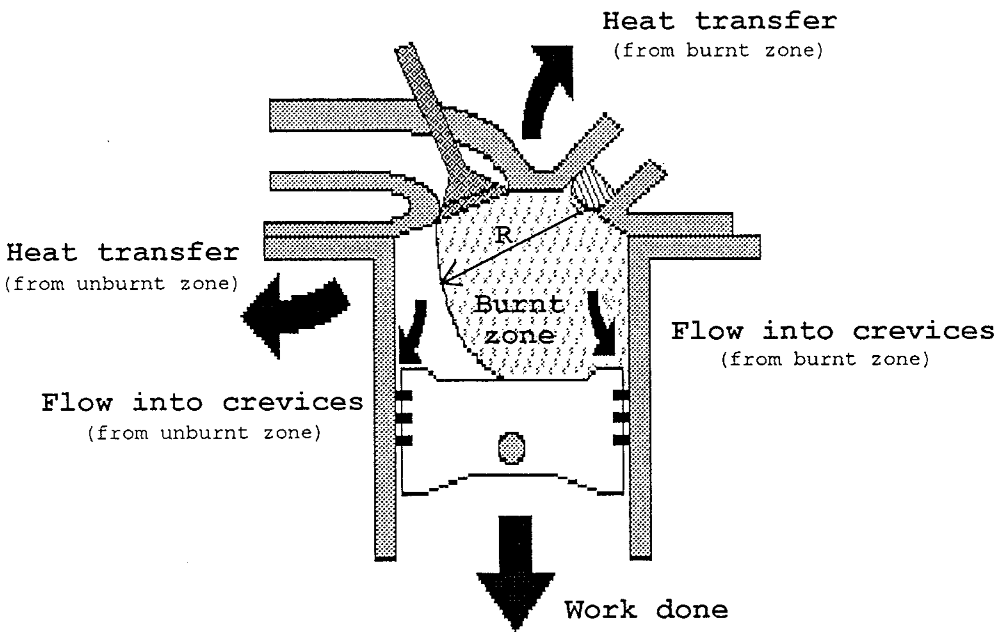

Fig. 2. Schematic of thermodynamic system.

The overall equivalence ratio for the dual fuel has been calculated using the following equation [2];

$$\phi = \frac{\left(\dfrac{[\mathrm{F}]}{[\mathrm{Air}] - \dfrac{[\mathrm{H}]}{([\mathrm{H}]/[\mathrm{Air}])_{\mathrm{st}}}}\right)}{\left(\dfrac{[\mathrm{F}]}{[\mathrm{Air}]}\right)_{\mathrm{st}}} \tag{1}$$

The mass burning rate was modeled by the following equation [5];

$$\frac{\mathrm{d}M_{\mathrm{b}}}{\mathrm{d}t} = A_{\mathrm{fl}} \cdot \rho_{\mathrm{u}} \cdot S_{\mathrm{T}} \tag{2}$$

The turbulent flame front speed (S_{T}) was calculated for the dual fuel using the following equation [6,7];

$$S_{\mathrm{T}} = S_{\mathrm{L}} ff \frac{(\rho_{\mathrm{u}}/\rho_{\mathrm{b}})}{[(\rho_{\mathrm{u}}/\rho_{\mathrm{b}}) - 1]X_{\mathrm{b}} + 1} \tag{3}$$

The laminar flame front speed for hydrogen and the hydrocarbon fuel was determined by using the following equation [6,7];

$$(S_{\mathrm{L}})_{\mathrm{C_nH_m+C_2H_6O+H_2}} = S_{\mathrm{Lo}}\left(\frac{T_{\mathrm{u}}}{T_{\mathrm{O}}}\right)^{\alpha_{\mathrm{o}}}\left(\frac{P}{P_{\mathrm{o}}}\right)^{\beta_{\mathrm{o}}}(1 - 2.06X_{\mathrm{R}}^{0.77}) + Y_{\mathrm{H_2}} M^{[\mathrm{m/s}]} \tag{4}$$

where M is a constant (0.83 m/s for hydrocarbon fuels) and Y_{H_2} is an indication of the relative amount of hydrogen addition, which was defined b y [2,7];

$$Y_{H_2} = \left(\frac{[H] + \frac{[H]}{([H]/[Air])_{st}}}{[G] + \left([Air] - \frac{[H]}{([H]/[Air])_{st}} \right)} \right) \tag{5}$$

$$\alpha_o = 2.18 - 0.8(\phi - 1) \tag{6}$$

$$\beta_o = -0.16 + 0.22(\phi - 1) \tag{7}$$

$$S_{L_o} = 0.2758 - 0.7834(\phi - 1.21)^2 \quad \text{for gasoline fuel [6]} \tag{8a}$$

$$S_{L_o} = 0.369 - 1.405(\phi - 1.11)^2 \quad \text{for ethyl alcohol fuel[6]} \tag{8b}$$

The flame front area (A_{fl}) calculations were based on the Annand model [8]. The flame front radius and delay period were calculated by the following equations;

$$R = \frac{ST \cdot \Delta\theta}{6\text{rpm}} \tag{9}$$

$$D_P = \left[\frac{6\text{rpm}}{S_T} \right] 3\sqrt{\left(\frac{0.0001 V_{cyl}}{\pi} \right)} \tag{10}$$

The instantaneous heat interaction between the cylinder content (burnt and unburnt zones) and its confining walls was calculated by using the empirical expression of Annand for a four stroke engine [2];

$$-\frac{dQ_{Ht}}{dt} = A\left[0.26\frac{k}{B}\left(\frac{U_P B}{\mu} \right)^{0.7} (T - T_w) + 0.69\sigma(T^4 - T_w^4) \right] \tag{11}$$

The instantaneous energy flows to the crevices was calculated by using the semi-empirical expression of Gatowski et al. [9] for a spark ignition engine;

$$\frac{\tilde{d} Q_{CR}}{d\theta} = (e(T) + R_{mol} \cdot T)\frac{dN_{cr}}{d\theta} \tag{12}$$

where $dN_{cr} > 0$ when flow is out of the cylinder into the crevice, $dN_{cr} < 0$ when flow is from the crevice to the cylinder.

$(e(T) + R_{mol} \cdot T)$ is evaluated at cylinder conditions when $dN_{cr} > 0$ and at crevice conditions when $dN_{cr} < 0$.

The cylinder pressure and the temperatures of the burnt and unburnt zones were predicted using energy, mass and volume balance equations and the equation of state. Twelve species were considered in the calculation of the combustion products concentrations. The following set of equations were used [10];

$$(1) \quad \tfrac{1}{2}H_2 \leftrightarrow H \tag{13}$$

$$(2) \quad \tfrac{1}{2}O_2 \leftrightarrow O \tag{14}$$

$$(3) \quad \tfrac{1}{2}N_2 \leftrightarrow N \tag{15}$$

$$(4) \quad 2H_2O \leftrightarrow 2H_2 + O_2 \tag{16}$$

$$(5) \quad H_2O \leftrightarrow OH + \tfrac{1}{2}H_2 \tag{17}$$

$$(6) \quad CO_2 + H_2 \leftrightarrow H_2O + CO \tag{18}$$

$$(7) \quad H_2O + \tfrac{1}{2}N_2 \leftrightarrow H_2 + NO \tag{19}$$

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

$$N + NO \leftrightarrow N_2 + O \quad K_1f = 3.1 \times 10^{10} \ \exp\frac{-160}{T} \quad [m^3/kmol\ S] \tag{20}$$

$$N + O_2 \leftrightarrow NO + O \quad K_2f = 6.4 \times 10^{6} \times T \times \exp\frac{-3125}{T} \quad [m^3/kmol\ S] \tag{21}$$

$$N + OH \leftrightarrow NO + H \quad K_3f = 4.2 \times 10^{10} \quad [m^3/kmol\ S] \tag{22}$$

where K_1f, K_2f and K_3f are the forward rate constants and were taken from Ref. [10].

4. Results

4.1. Measurements of the effect of hydrogen/alcohol blending on engine performance and emissions

Fig. 3 shows the measurements of the effect of hydrogen blending (0–20%), and/or ethyl alcohol blending (0–40%), on the performance and emissions of a Ricardo E6/US spark ignition engine operated with a stoichiometric mixture and optimum spark timing for best torque with 7.5 compression ratio and 1500 rpm. Each parameter studied is made dimensionless by relating it to its value when the engine is fueled with pure gasoline at the same conditions.

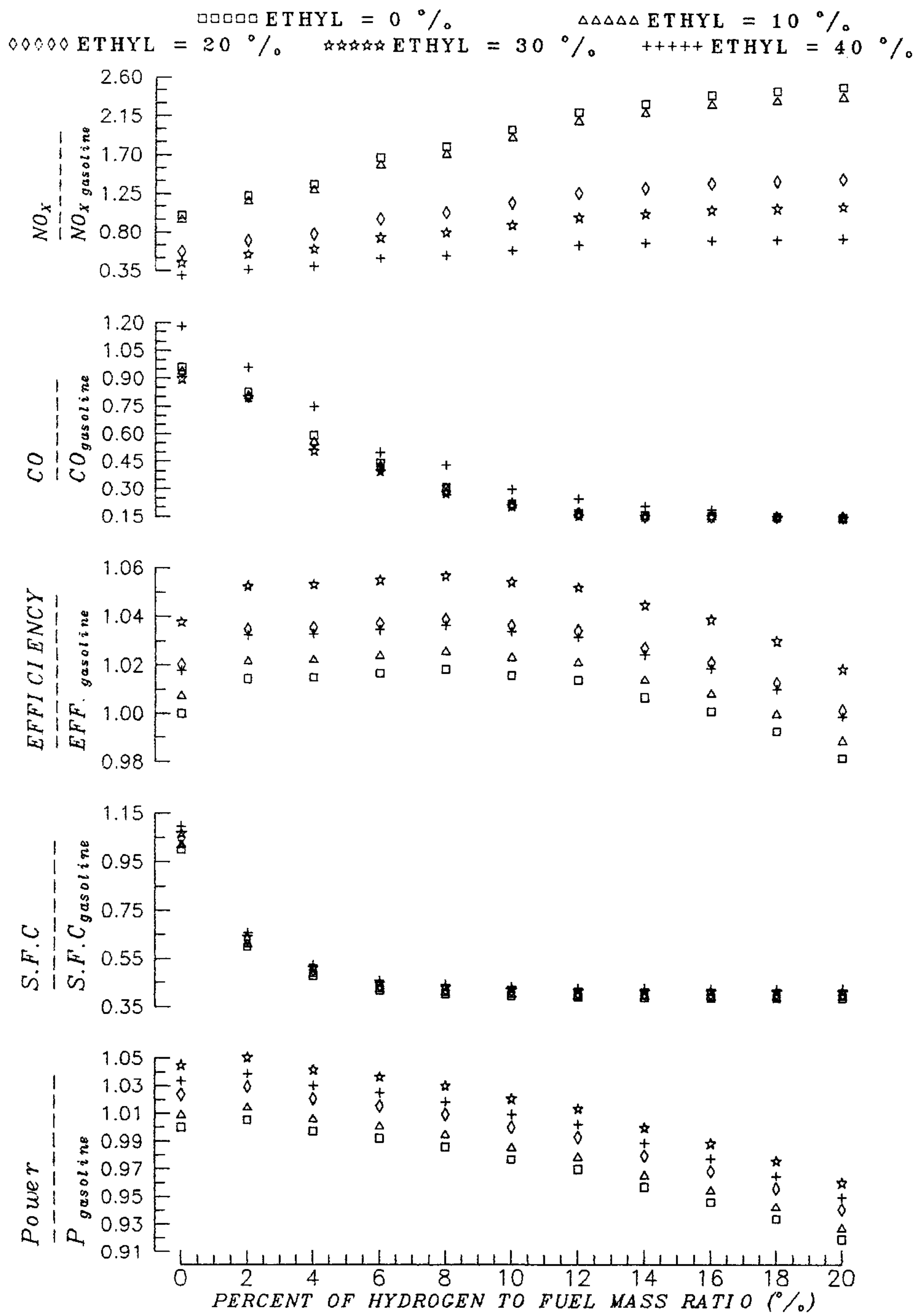

Fig. 3. Measurements of the effect of hydrogen/alcohol blending on the performance and emission of SI engine.

The results show that the engine power increases as the percentage of hydrogen blending is increased due to the high rate of mass burning of the hydrogen. When the percentage of blending is more than 2% the power decreases due to the reduction in mixture density and engine volumetric efficiency. The engine thermal efficiency is also improved as the percentage of hydrogen blending is increased, reaching a maximum at 8% blending. With further increase in hydrogen blending, the thermal efficiency decreases due to the drop in volumetric efficiency. The brake power and thermal efficiency increase with the increase of the alcohol percentage volume ratio in the gasoline up to 30% ethyl. At this percentage, the maximum brake power and maximum thermal efficiency have been obtained due to the increase in mixture density and engine volumetric efficiency. During the tests, it was noticed that when the ethyl alcohol percentage exceeds 30%, the carburettor is not capable of evaporating all the fuel supplied to the engine, and part of it enters the chamber without evaporation and combustion. This causes a reduction in the brake power and thermal efficiency as the ethyl alcohol percentage increases above the 30%.

The specific fuel consumption decreases as the percentage of hydrogen blending is increased until 6%. Then, the decrease in sfc seems to be marginal throughout the percentage range. The specific fuel consumption is slightly increased as the volume percentage of alcohol is increased in the mixture. This is due to the lower heating value of alcohol compared with gasoline.

The CO concentration decreases as the percentage of hydrogen addition increases. This is due to the reduction in carbon atoms concentration in the blended fuel and the high molecular diffusivity of hydrogen, which improves the mixing process and, hence, the combustion efficiency. When the alcohol has been added (0–30%), the CO emission remains practically unchanged, while at the 40% blending ratio, the CO emission is increased due to the incomplete combustion process inside the cylinder [11].

The NO_x concentration increases as the hydrogen mass ratio increases. This is due to the higher peak temperature and pressure and the reduction of the time required to dissociate NO to N_2 and O_2 [10]. The NO_x concentrations decrease as the percentage of alcohol addition increases. This is probably a result of the higher heat of vaporization of alcohol which reduces the peak temperature inside the cylinder.

As was observed from Fig. 3, the supplement of hydrogen 4% with 30% of alcohol to the gasoline produces a significant reduction in the SFC (49%), a significant reduction in both the CO and NO_x emissions (49%) and (39%), respectively, an increase in power (4%) and an increase in thermal efficiency (5%).

4.2. Predictions of the effect of hydrogen/alcohol blending on thermodynamic cycle parameters

Figs. 4a and 5a show some of the typical computed results of the effect of hydrogen and alcohol addition on the thermodynamic cycle of the gasoline engine for a fixed spark timing. The figures show that as the hydrogen fuel mass ratio increases, both peaks of the cylinder pressure and temperature increase, and the pressure diagram approaches closer to the ideal diagram, whereas the combustion period decreases. This is due to the increase in flame speed and, hence, increase in the rate of mass burning. Therefore, the time required for complete combustion is reduced, and this produces lower heat transfer to the cylinder walls, lower exhaust temperature, higher engine efficiency and lower tendency to knock. The figures also

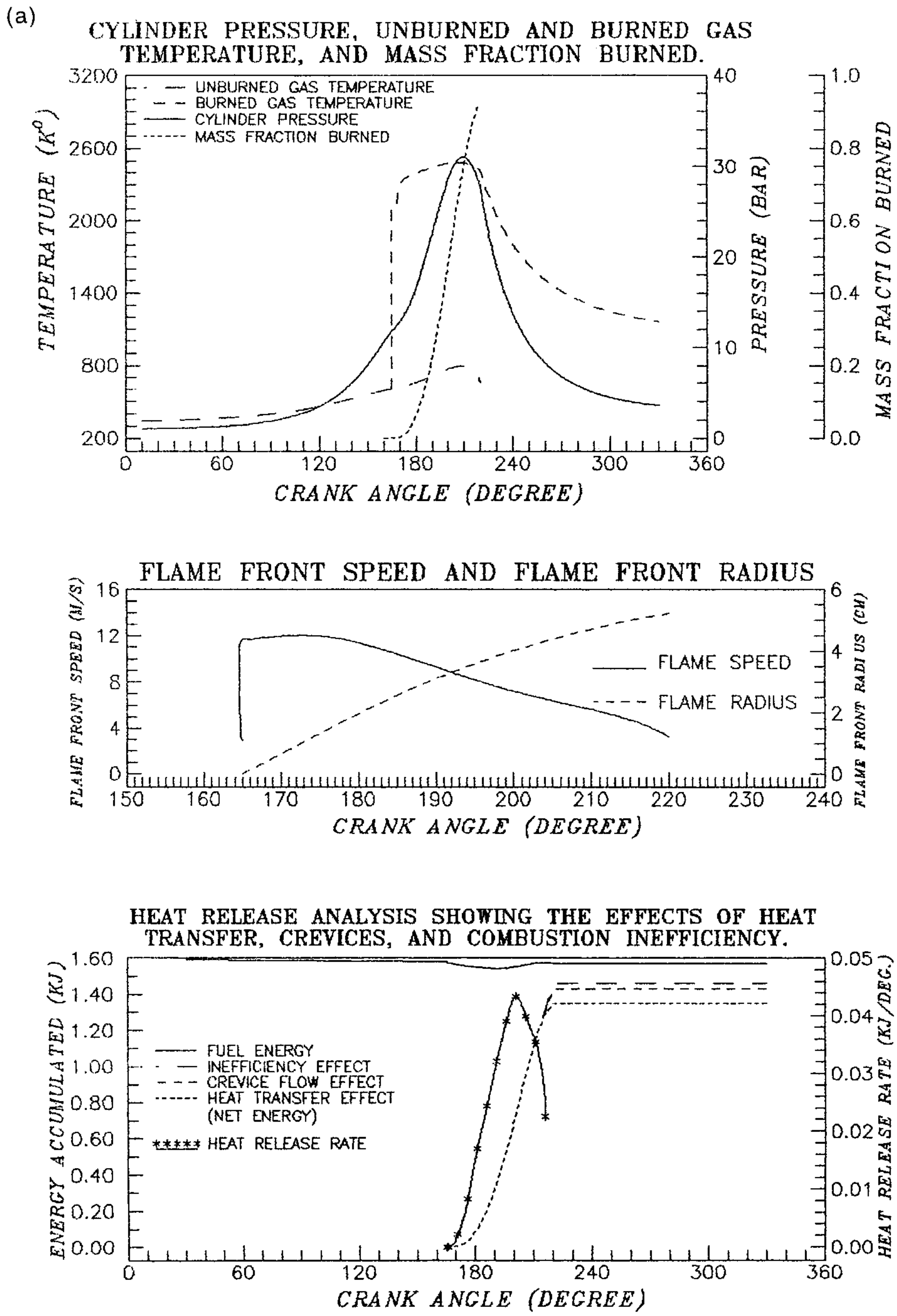

Fig. 4a. Variation of power cycle parameters with crank angle. Fuel mass ratio: 100% gasoline.

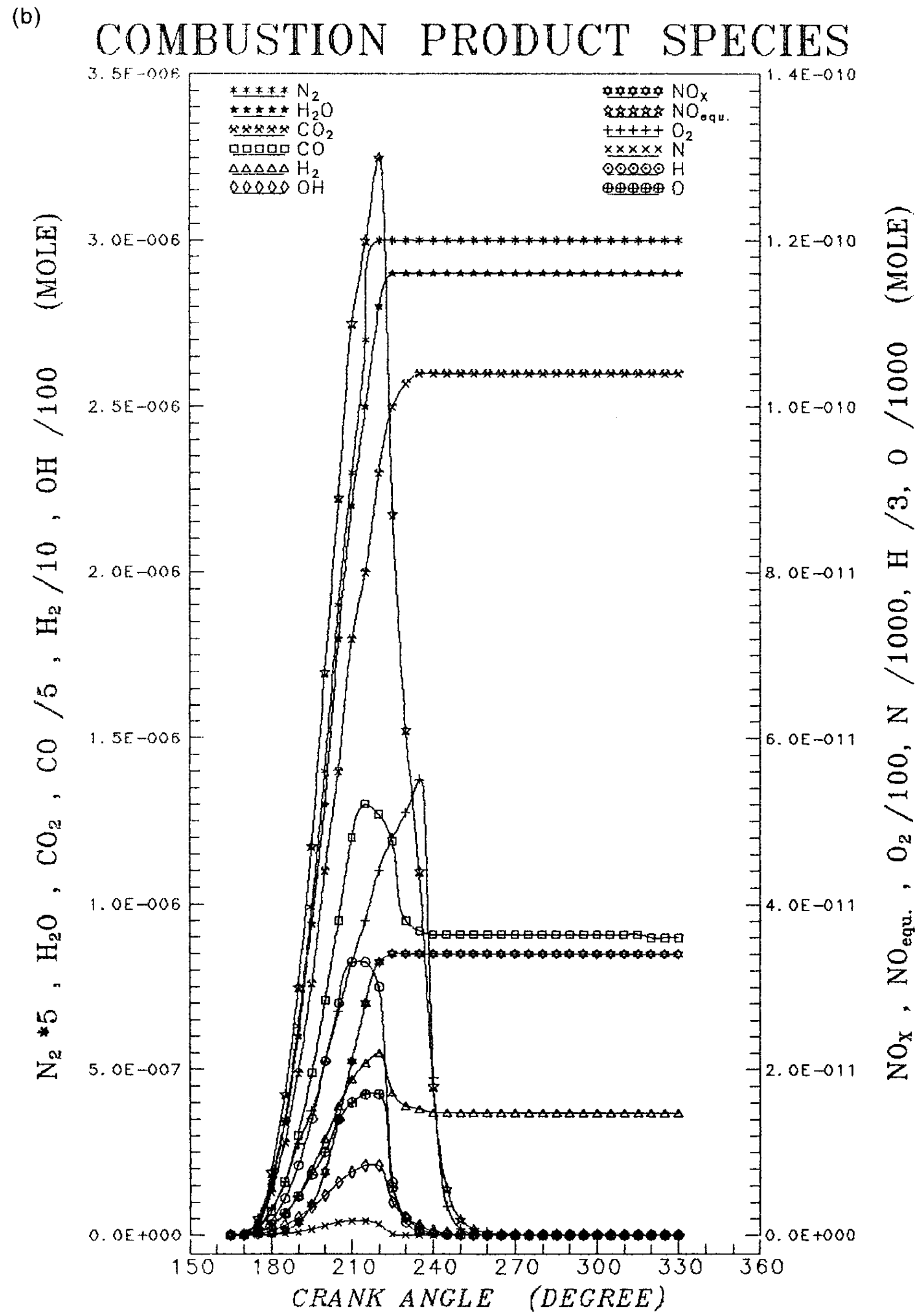

Fig. 4b. Variation of concentration of combustion products with crank angle. Fuel mass ratio: 100% gasoline.

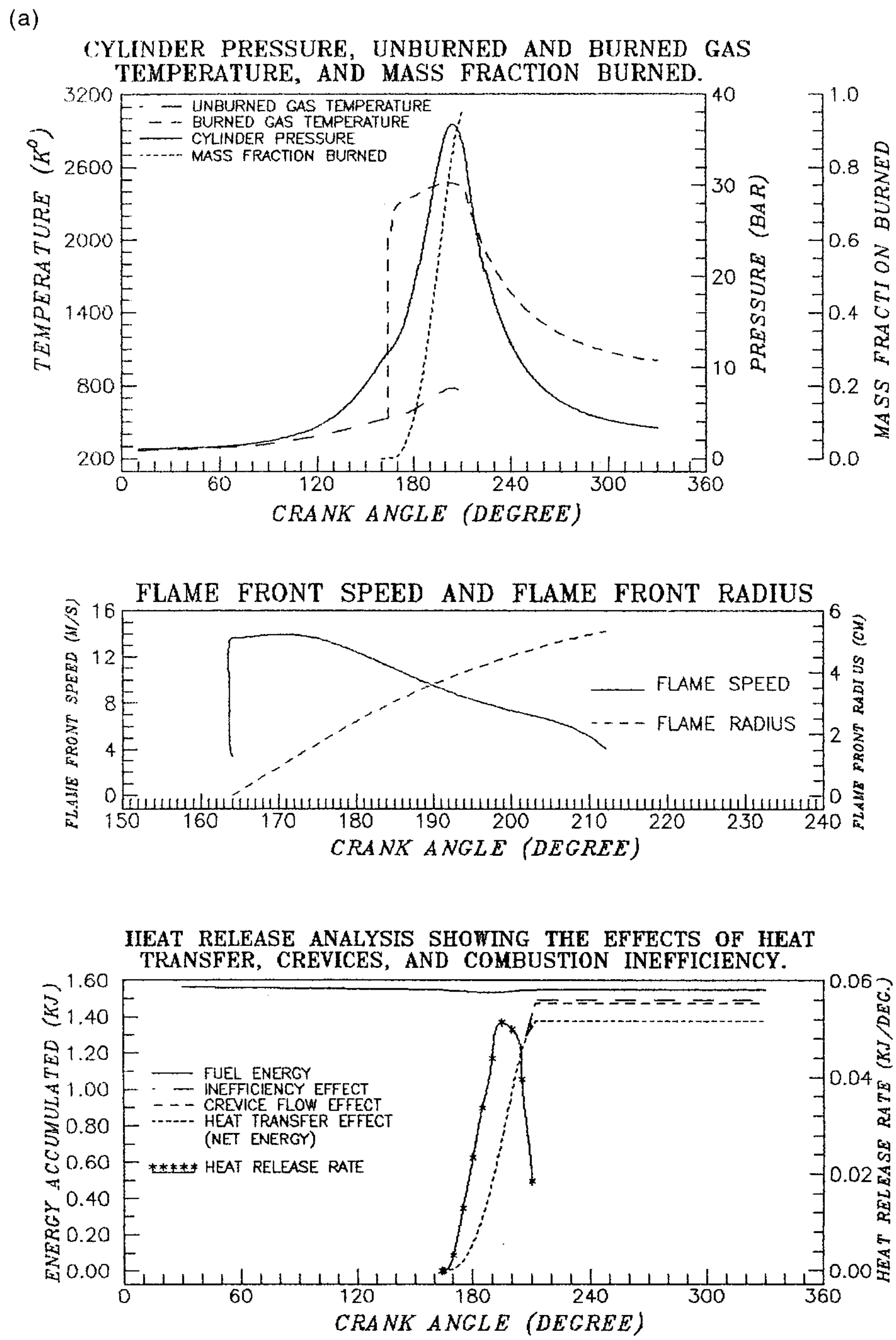

Fig. 5a. Variation of power cycle parameters with crank angle. Fuel mass ratio: 96% liquid fuel* + 4% hydrogen. *Liquid fuel consists of (30% ethyl and 70% gasoline by volume).

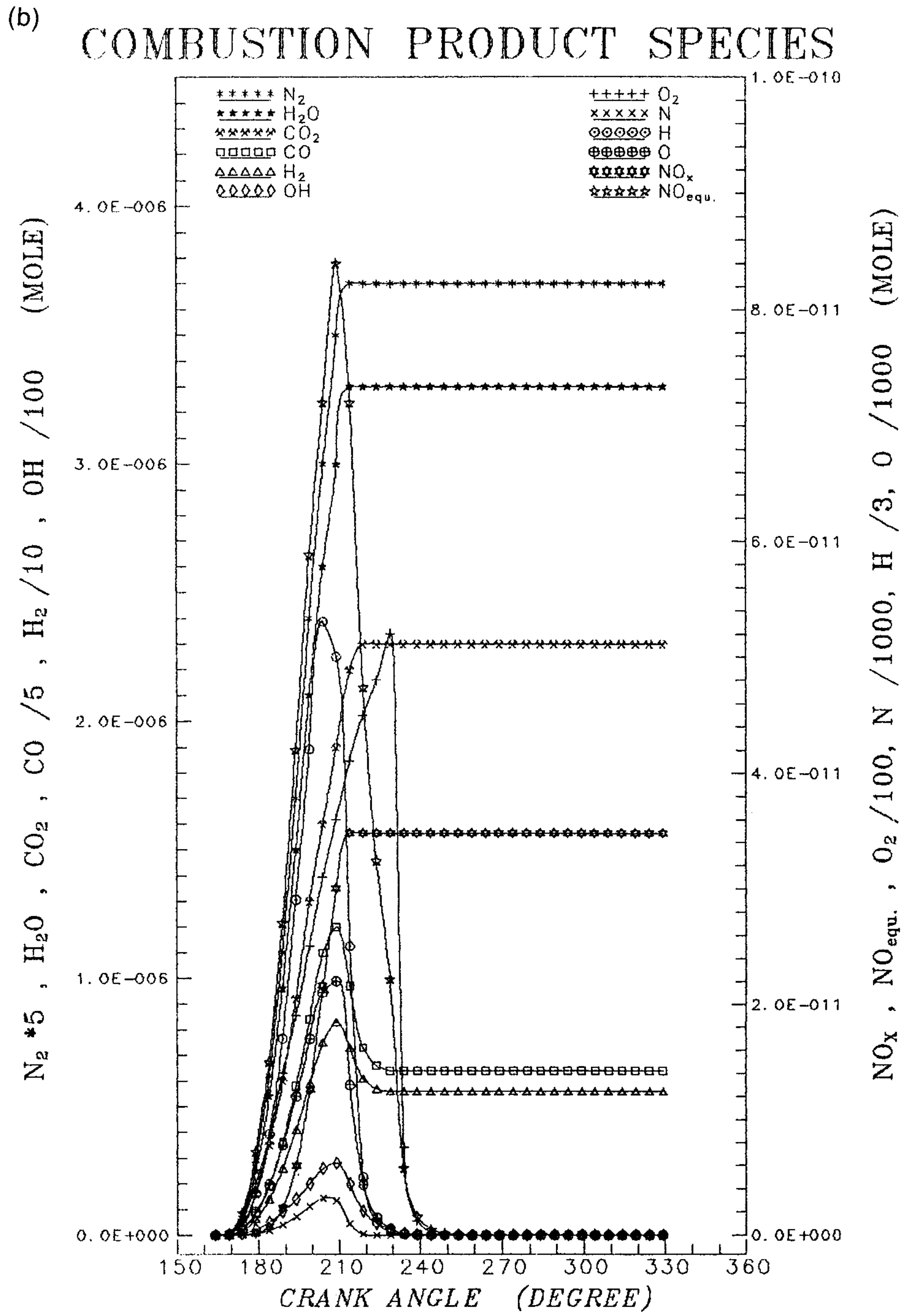

Fig. 5b. Variation of concentration of combustion products with crank angle. Fuel mass ratio: 96% liquid fuel* + 4% hydrogen. *Liquid fuel consists of (30% ethyl and 70% gasoline by volume).

show that the cylinder temperature is reduced as the alcohol fuel ratio increases, thus, increasing the thermal efficiency. This is due to the high heat of vaporization of ethyl alcohol. The local flame front speed and flame front radius increase with the increase of hydrogen and alcohol blending. This is due to the high burning speeds of the hydrogen–air mixture and the alcohol–air mixture which are higher than that of the gasoline–air mixture. The effects of heat transfer losses, crevice losses and combustion inefficiency are taken into account. The crevices volume is calculated to be 0.5% of the clearance volume. The hydrogen blending causes a reduction in both the total fuel energy and the energy flow into the crevices. This is due to the decrease in mixture density as a result of hydrogen blending, and hence, less energy per unit volume of mixture is obtained. Hydrogen and alcohol blending improve the combustion efficiency, therefore, the point of the maximum rate of heat release moves closer toward TDC. The maximum rate is also increased due to the faster flame front propagation. Figs. 4b and 5b show the effect of hydrogen and alcohol blending on the pollutant concentrations. It is clear that the CO_2 and CO concentrations decrease as the percentage of hydrogen and alcohol blending are increased. This is due to the reduction in carbon atoms concentration in the blended fuel and the high molecular diffusivity of hydrogen, which improves the mixing process and, hence, provides higher combustion efficiency [12]. The NO_x concentration increases as the hydrogen fuel mass ratio increases, but it decreases as the alcohol fuel ratio increases. This result demonstrates the high sensitivity of the NO_x emission level to the maximum temperature inside the cylinder, which is a typical behavior of chemical reactions [Eqs. 15–17].

5. Conclusion

The following conclusions are drawn from the present work.

1. Hydrogen can be used as a supplementary fuel in modern spark ignition engines without major changes, and it can help save a considerable part of the available oil and save our environment from toxic pollutants.
2. The blending of alcohol reduces the NO_x emission and peak temperature.
3. The thermal efficiency of the engine is increased until a hydrogen–fuel mass ratio of 8% and alcohol–fuel volume ratio of 30%.
4. The engine power is increased until a hydrogen–fuel mass ratio of 2% and alcohol–fuel ratio of 30%.
5. The blending of alcohol or hydrogen increases the heat release rate.
6. The blending of hydrogen reduces the specific fuel consumption, while alcohol blending increases the specific fuel consumption.
7. The exhaust temperature is reduced, and so is the crevices flow energy due to the blending of hydrogen or alcohol fuel.
8. The hydrogen added improves the combustion process, especially in the later combustion period, reduces the ignition delay, speeds up the flame front propagation, reduces the combustion duration and retards the spark timing.
9. The concentration of CO is reduced, and the concentration of NO_x is increased due to hydrogen blending.

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