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A prediction study of the effect of hydrogen blending on the performance and pollutants emission of a four stroke spark ignition engine

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Abstract

Considering energy crises and pollution problems today, investigations have been concentrated on decreasing fuel consumption by using alternative fuels and on lowering the concentration of toxic components in combustion products. In the present work a quasi-dimensional model was developed to study the effect of hydrogen blending on fuel consumption and pollutant concentrations. The results of the study show that the maximum improvement in engine thermal efficiency occurs at 8% hydrogen blending. The results also show that 10% hydrogen blending reduces CO concentration by 73.8% but the NO concentration increases by 100%. However the problem of increasing NO concentration was solved by operating the engine with lean mixture. Hydrogen blending also reduces the specific fuel consumption until about 6% blending, then the effect becomes marginal.

Nomenclature

A	cylinder heat transfer area
A_{fl}	flame front area
[Air]	molar concentration of the air
B	cylinder bore
ff	turbulent flame factor
[G]	molar concentration of the hydrocarbon fuel
[H]	molar concentration of the hydrogen fuel
k	thermal conductivity
M_b	mass of burnt gases
N_{cr}	number of mole in crevice
R	the flame front radius
rpm	engine speed
SL	laminar flame front speed
ST	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
X_b	mass fraction of burnt gases
T	gas temperature
Q_{CR}	energy exchange across the system boundary
Q_{HT}	overall heat transfer rate to the cylinder wall
BDC	bottom dead center
TDC	top dead center
ATDC	after top dead center
BTDC	before top dead center
SFC	specific fuel consumption
XR	mole fraction of residual gas
Y_{H2}	amount of hydrogen addition
UP	mean piston speed
DP	delay period
T_0	reference temperature
P_0	reference pressure
θ	crank angle
μ	kinematics gas viscosity
ρ_u	density of unburnt gas mixture
ρ_b	density of burnt gas mixture
σ	Stefan–Boltzman constant = $5.67e^{-8}$
$\Delta\theta$	crank angle step
ϕ	equivalence ratio

1. Introduction

A large amount of research has been directed towards the development of alternative energy sources and alter-

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native fuels. Hydrogen is considered as an ideal alternative fuel. Many workers studied the effect of using hydrogen as a fuel (pure or mixed with gasoline) on engine performance and pollutants emissions.

The aim of the present work is to study the effect of hydrogen blending on the performance and emission of a spark ignition engine. Consequently, a computer programme is constructed and developed using a quasi-dimensional model with a set of semiempirical equations to simulate the combustion process in a reciprocating spark ignition engine, which is fueled with hydrogen–gasoline 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.

Desoky and El-Emam [1] studied the combustion properties of hydrogen–air mixture and found that it has higher self-ignition temperature than a gasoline–air mixture and this enhances its knock resistance. They also found that it has the highest energy mass coefficient. Hacohen and Sher [2] found that using gaseous fuel (hydrogen) avoids uneven distribution of fuel to the different cylinders and the problems of cold fuel evaporation. The only toxic products of hydrogen combustion are nitric oxides [1, 3]. Hacohen and Sher [3] performed a modeling study for a four stroke cycle S.I. engine fueled with hydrogen enriched gasoline and found that a conventional S.I. engine fueled with a mixture of hydrogen and gasoline can operate with ultra-lean mixture, which gives best fuel economy and less pollutant emissions. Parks [4] examined experimentally the emission characteristics of a hydrogen rich fuel of a single cylinder S.I. engine. The results showed that increasing the hydrogen energy fraction, H.E.F. from 0.13 to 0.48 at an equivalence ratio of 0.8, reduced the HC emission by 15–60%, respectively and increased NOx emissions. Chunhuang and Xiaolong [5] investigated experimentally the performance of a spark ignition engine fueled with hydrogen and carbon monoxide produced from the decomposition of methanol in an on-board reactor. Their results showed that the thermal efficiency is improved by 30% compared with that of the same engine when fueled with gasoline. They also found that the maximum specific power and torque remained constant and the exhaust emissions were greatly reduced.

2. Present model

The present work was directed towards the development of a mathematical model to study the effect of the percentage of hydrogen blending on the performance and pollutant emissions of a four stroke spark ignition engine. The combustion chamber was generally divided into burnt and unburnt zones separated by a flame front. The cylinder charge was 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 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 burnt and unburnt zones, heat transfer from burnt and unburnt zone, and mass flow into and out of crevices. The crevices are the volumes between the piston, piston rings and cylinder wall (Fig. 1). Gases flow into and out of these volumes during the engine operating cycle as the cylinder pressure changes.

The overall equivalence ratio for the dual fuel was calculated using the following equation [3];

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

The mass burning rate was modeled by the following equation [6];

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

The turbulent flame front speed (ST) was calculated for the dual fuel using the following equation [7, 8];

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

The laminar flame front speed for hydrogen and hydrocarbon fuel was accounted for by using the following equation suggested by Yu et al. [8];

$$(SL)_{\mathrm{CnHm+H_2}} = SL_0\left(\frac{T_{\mathrm{u}}}{T_0}\right)^{\alpha_0}\left(\frac{P}{P_0}\right)^{\beta_0}(1 - 2.06X_{\mathrm{R}}^{0.77}) + Y_{\mathrm{H_2}}M[\mathrm{m/s}] \tag{4}$$

Where M is a constant (≈ 0.83 m/s for hydrocarbon fuels) and $Y_{\mathrm{H_2}}$ is an indication of the relative amount of hydrogen addition, which was defined by [3, 8];

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

$$\alpha_0 = 2.18 - 0.8(\phi - 1) \tag{6}$$

$$\beta_0 = -0.16 + 0.22(\phi - 1) \tag{7}$$

$$SL_0 = 0.2758 - 0.7834(\phi - 1.11)^2 \tag{8}$$

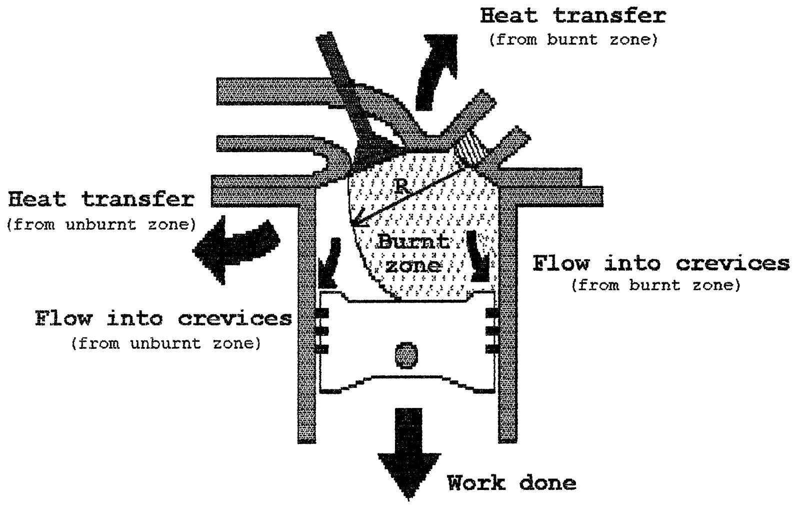

Fig. 1. Schematic of thermodynamic system.

The flame front area (A_{fl}) calculations were based on Annand model [9]. The flame front radius and delay period were calculated by the following equations;

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

$$DP = \left[\frac{6\,\text{rpm}}{ST}\right] \cdot 3\sqrt{\left(\frac{0.001.V_{cyl}}{\pi}\right)} \tag{10}$$

The instantaneous heat interaction between the cylinder content (burnt and unburnt zones), and its confined walls was calculated by using the empirical expression of Annand for a four-stroke engine [3];

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

The instantaneous energy flows to the crevices was calculated by using the semi-empirical expression of Gatowski et al. [10] for a spark ignition engine;

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

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

The cylinder pressure and the temperatures of 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 combustion products concentrations. The following equations were used [11];

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

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

$$\tfrac{1}{2}N_2 \leftrightarrow N \tag{15}$$

$$2H_2O \leftrightarrow 2H_2 + O_2 \tag{16}$$

$$H_2O \leftrightarrow OH + \tfrac{1}{2}H_2 \tag{17}$$

$$CO_2 + H_2 \leftrightarrow H_2O + CO \tag{18}$$

$$H_2O + \tfrac{1}{2}N_2 \leftrightarrow H_2 + NO \tag{19}$$

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

$N + NO \leftrightarrow N_2 + O$

$$K_{1f} = 3.1\times10^{10}\exp\left(\frac{-160}{T}\right)[\text{m}^3/\text{kmol}\cdot\text{S}] \tag{20}$$

$N + O_2 \leftrightarrow NO + O$

$$K_{2f} = 6.4\times10^{6}T\exp\left(\frac{-3125}{T}\right)[\text{m}^3/\text{kmol}\cdot\text{S}] \tag{21}$$

$N + OH \leftrightarrow NO + H$

$$K_{3f} = 4.2\times10^{10}[\text{m}^3/\text{kmol}\cdot\text{S}] \tag{22}$$

where K_{1f}, K_{2f} and K_{3f} are the forward rate constant and were taken from reference [11].

3. Results and discussion

As mentioned earlier the aim of this research is to study the effect of hydrogen blending on the performance and pollutants emission of a four stroke S.I. engine. Therefore the study is divided into two parts. The first part is to study the effect of hydrogen blending on the thermodynamic cycle parameters at a fixed spark timing for all test. The second part is to study the effect of hydrogen blending on the performance and pollutants emission of the spark ignition engine operated at the optimum spark timing for best torque. The engine specification parameters are given in Table 1.

3.1. Effect of hydrogen blending on the thermodynamic cycle parameters

The study showed that a spark advance of 20° C.A. BTDC and 35° C.A. BTDC gives maximum torque for stoichiometric mixture ($\phi = 1.0$) and lean mixture ($\phi = 0.8$) respectively when the engine fueled with gasoline–air mixture at 1500 rpm and 7.5 compression ratio. Therefore the study was concentrated on these two conditions.

3.1.1. Cylinder pressure and temperature

Cylinder pressure and temperature are given as functions of crank angle for each state of hydrogen blending (Figs 2–4). It is clearly shown that as the hydrogen fuel mass ratio increases, peak cylinder pressure and peak temperature increase, and the pressure diagram approaches closer to the ideal diagram, whereas combustion period decreases. The increase of hydrogen blending from 0% to 10% by mass, increases the peak pressure from 31.043 bar to 45.482 bar and peak temperature from 2482 K to 2634 K, for stoichiometric equivalence ratio, and increases the peak pressure from 28.425 bar to 45.244 bar and peak temperature from 2306 K to 2489 K for 0.8 equivalence ratio. This is due to the 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 lower heat transfer to cylinder walls, lower exhaust temperature, higher engine efficiency, and lower tendency to knock. Also it is noticed that the rates of both pressure and temperature rise for the lean operation conditions, (i.e. $\phi = 0.8$) are greater than these of stoichiometric operation condition. This is due to the increase of the flame front speed, which enhances the turbulence inside the cylinder, hence improves the combustion efficiency.

Table 1

Type	Recardo E6, spark ignition four stoke
Cycle	Four-stroke
Number of cylinder	1
Cylinder bore	0.0762 m
Stroke	0.11 m
Connecting rod length	0.2413 m
Compression ratio	7.5
Speed	1500 rpm

3.1.2. Flame front speed and flame front radius

Figures 2–4 show also the variation of flame front speed and flame front radius with crank angle for each state of hydrogen blending. It is shown that both local flame front speed and radius increase with increasing the hydrogen blending. This is due to high burning speed of hydrogen–air mixture which is higher than that of the gasoline–air mixture. Higher cylinder pressure and unburnt zone temperature also enhance flame front speed. It is also shown that the local flame front speed decreases as the equivalence ratio decreases, which is due to the lower combustion temperature.

3.1.3. Rate of heat release and accumulated heat release

The rate of heat release and the accumulated heat release are plotted against the crank angle as shown in Figs 2–4 for each state of hydrogen blending. The effects of heat transfer losses, crevices losses and combustion inefficiency are accounted for. The crevices volume was calculated to be 0.5% of the clearance volume. The results show that the percentage of hydrogen blending is increased from 0–10%, the total fuel energy and the energy flow into the crevices decrease by 3.758% and 56.626% respectively for the stoichiometric mixture and by 2.349% and 57.432%, respectively for the lean mixture. This is due to the decrease in mixture density as a result of hydrogen blending and hence less energy per unit volume of mixture.

Hydrogen blending also improves combustion efficiency and therefore the point of maximum rate of heat release moves closer toward TDC. Also the maximum rate is increased due to faster flame front propagation. The accumulated heat release increase initially as a result of hydrogen blending due to high rate of mass burning but with further increase in the percentage of hydrogen the accumulated heat release decreases due to the reduction in total fuel energy.

3.1.4. Concentration of pollutant species

Figures 5–7 show the effect of hydrogen blending on pollutants concentration. It is clear that CO_2 and CO concentrations decrease as the percentage of hydrogen blending is increased. This is due to the reduction in carbon atoms concentration in the blended fuel and the high molecular diffusivity of hydrogen which improves mixing process and hence combustion efficiency.

The figures also show that the reduction in CO_2 con-

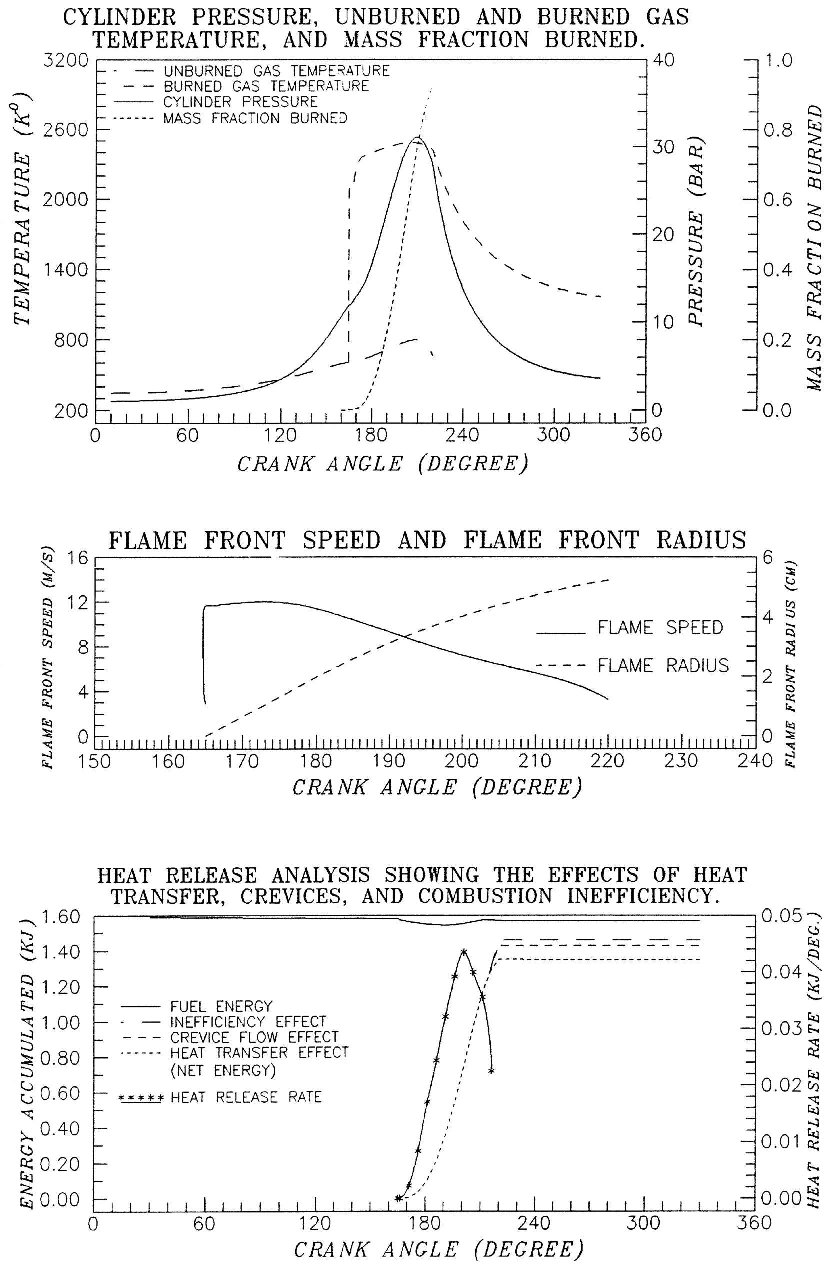

Fig. 2. Variation of power cycle parameters with crank angle. Fuel mass ratio: 100% gasoline; equivalence ratio: 1.0; compression ratio: 7.5; engine speed: 1500 rpm; spark advance: 20 BTDC; turbulence flame factor 3.7.

centration is greater than that in CO concentration. This is attributed to higher cylinder temperature caused by hydrogen blending which enhances the combination of CO_2 and H_2 to form CO and H_2O.

The NOx concentration increases by 182% for the stoichiometric mixture and by 383% for the lean mixture, ($\phi = 0.8$), as the percentage of hydrogen blending is increased from 0–10% for a fixed spark timing. This result demonstrates the high sensitivity of the NOx emission level to the maximum temperature inside the cylinder which is a typical behavior of chemical reactions (equations 20–22).

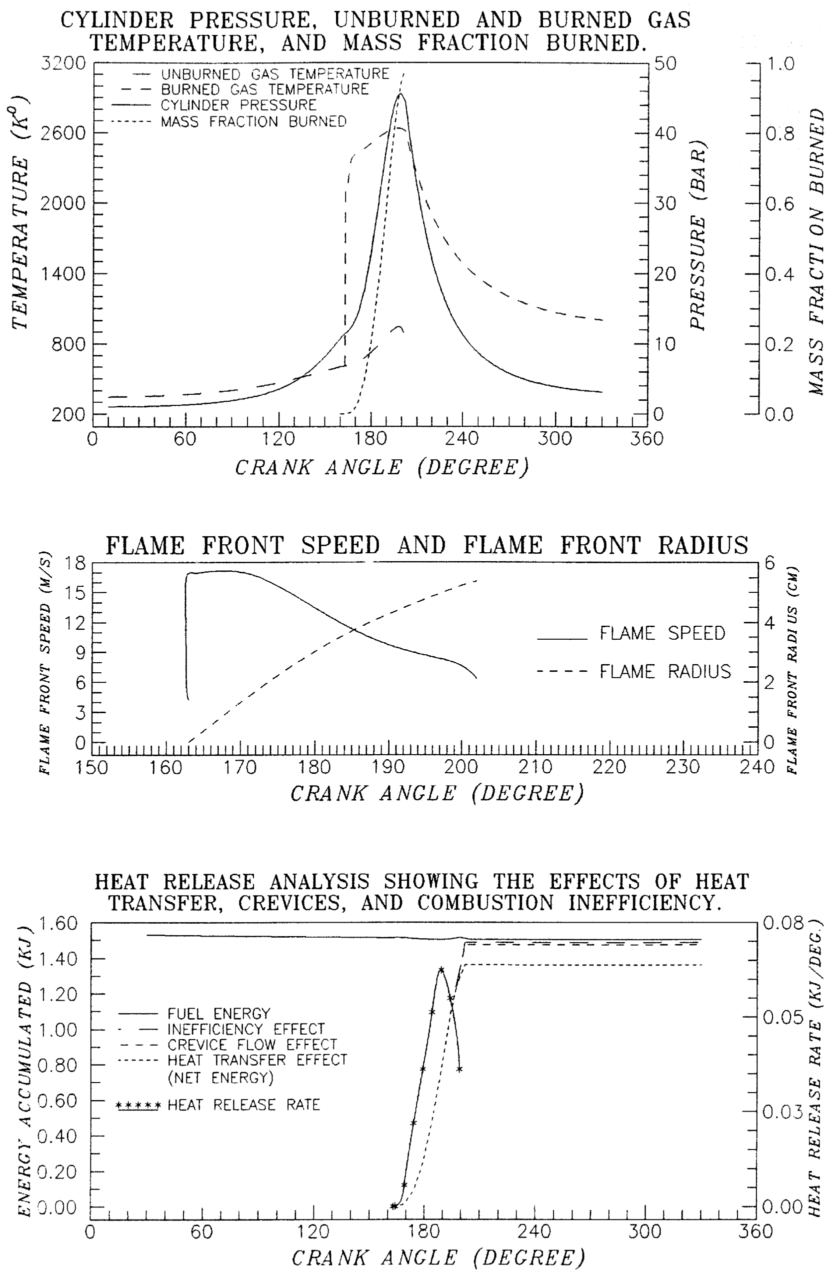

Fig. 3. Variation of power cycle parameters with crank angle. Fuel mass ratio: 90% gasoline + 10% hydrogen; equivalence ratio: 1.0; compression ratio: 7.5; engine speed: 1500 rpm; spark advance: 20 BTDC; turbulence flame factor 3.7.

3.2. Effect of hydrogen blending on engine performance and emission when operated with stoichiometric mixture

Figure 8 shows the effect of percentage of hydrogen blending on the performance and emission of a spark ignition engine operated with 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 a pure gasoline at the same conditions.

The results show that the engine power increases as the

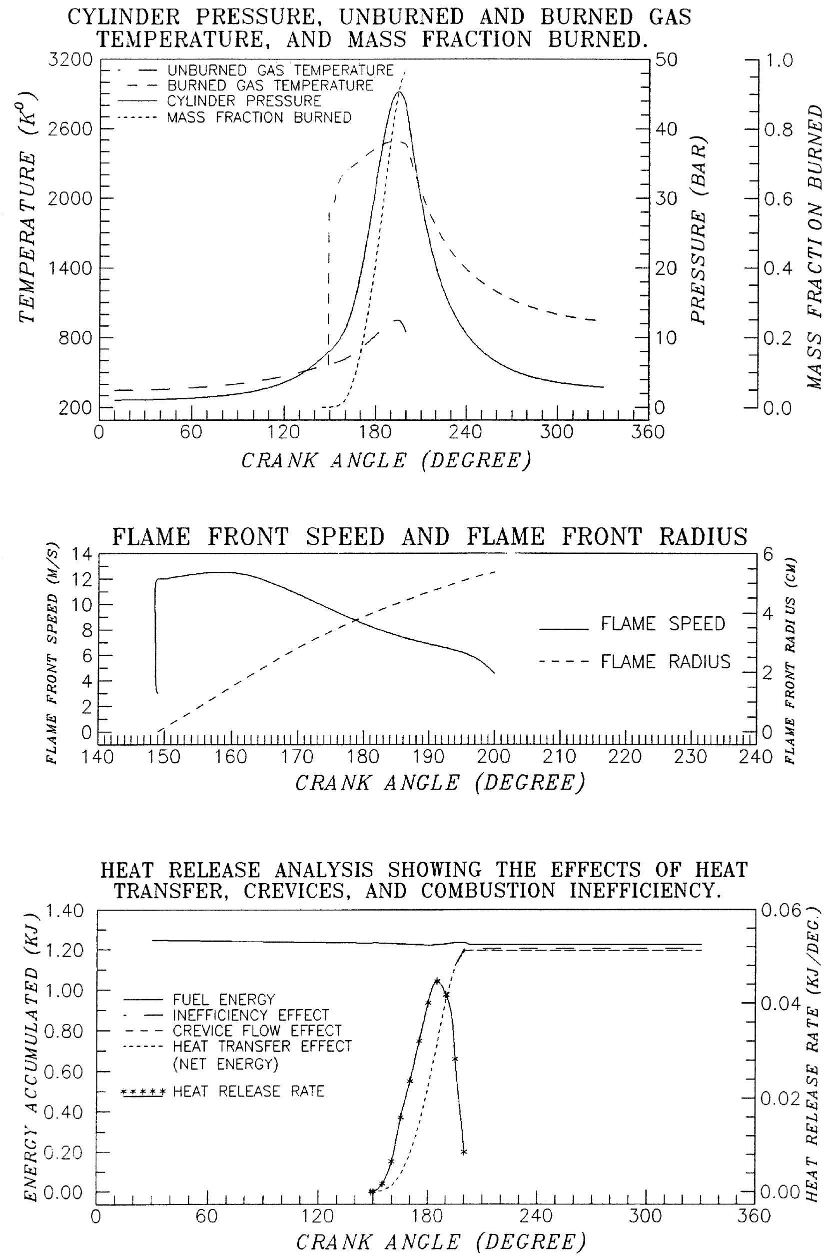

Fig. 4. Variation of power cycle parameters with crank angle. Fuel mass ratio: 90% gasoline + 10% hydrogen; equivalence ratio: 0.8; compression ratio: 7.5; engine speed: 1500 rpm; spark advance: 35 BTDC; turbulence flame factor: 3.7.

percentage of hydrogen blending is increased due to the high rate of massing burning of 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 specific fuel conception decreases as the percentage of blending is increased until 6% then the decrease in sfc seems to be marginal throughout the percentage range.

The engine thermal efficiency is also improved as the percentage of hydrogen blending is increased reaching maximum at 8% blending. With further increase in hydrogen blending the thermal efficiency decreases due to the drop in volumetric efficiency.

The volumetric efficiency decreases as the hydrogen mass ratio increases. This is because the density of hydrogen is less than that of gasoline therefore, the addition of

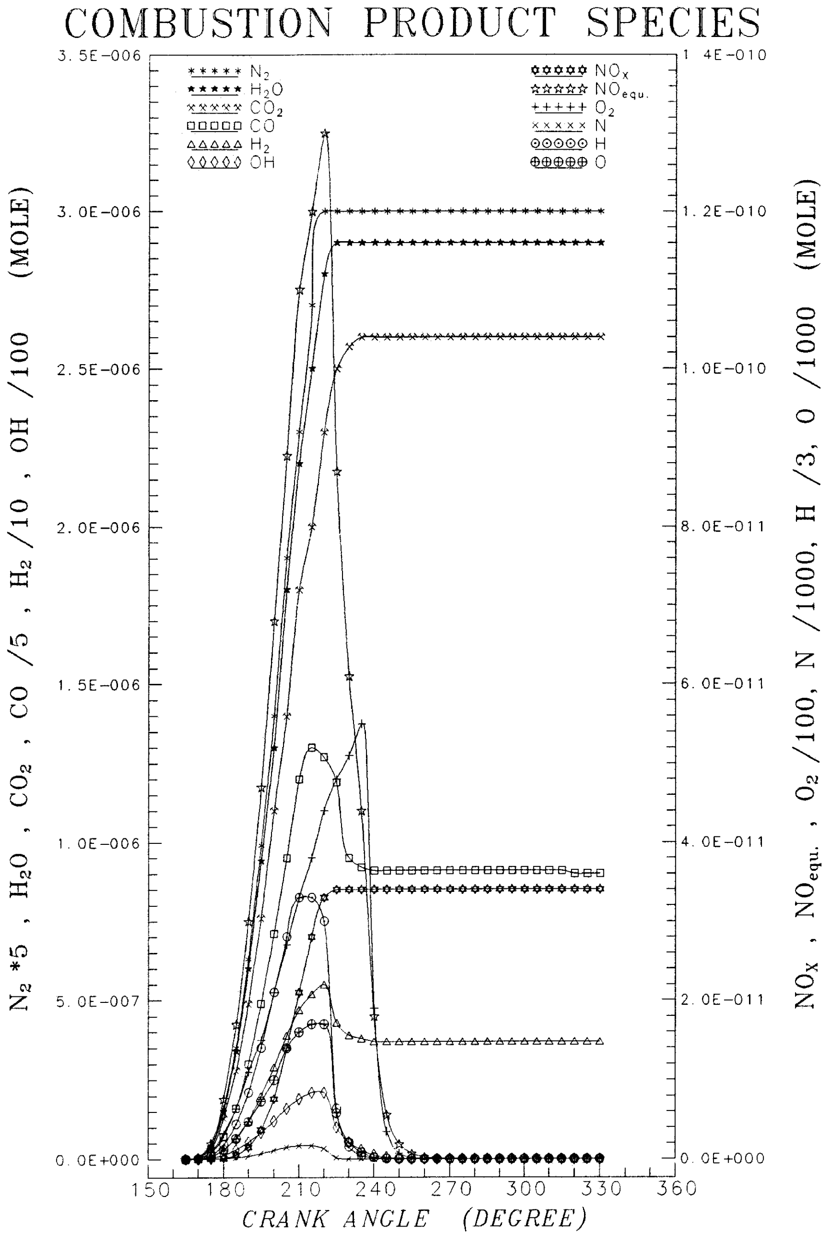

Fig. 5. Variation of concentration of combustion products with crank angle. Fuel mass ratio: 100% gasoline; equivalence ratio: 1.0; compression ratio: 7.5; engine speed: 1500 rpm; spark advance: 20 BTDC; turbulence flame factor 3.7.

hydrogen causes a reduction in the mixture density, and hence a reduction in volumetric efficiency. Also the theoretical air/fuel ratio by volume of gasoline fuel is about 59.5, while for hydrogen it is 2.38, hence as the hydrogen addition increases, the volume of incoming air decreases and causes a reduction in the volumetric efficiency.

The combustion duration decreases as the percentage of hydrogen blending increases. This is due to the increase in the flame front speed, which makes the time required to complete the combustion shorter. The results show that as the percentage of hydrogen blending increased by 10%, the flame front speed increased by 46%. And the combustion duration is reduced by 24%. The reduction in the combustion duration is very important, since it leads directly to lower heat transfer to cylinder walls, higher engine efficiency, lower specific fuel consumption, lower tendency to knock, and lower HC emission.

As was observed from Fig. 8, the supplemental of

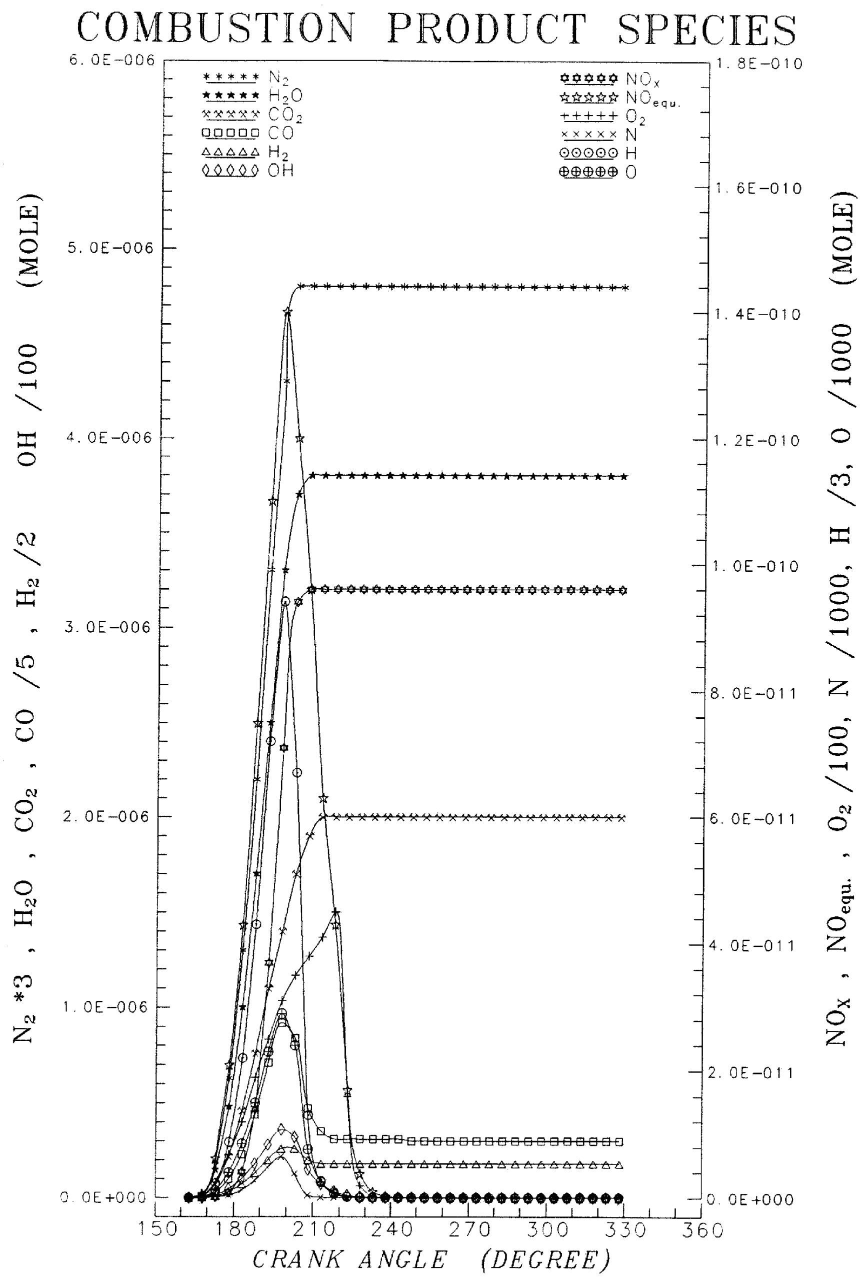

Fig. 6. Variation of concentration of combustion products with crank angle. Fuel mass ratio: 90% gasoline + 10% hydrogen; equivalence ratio: 1.0; compression ratio: 7.5; engine speed: 1500 rpm; spark advance: 20 BTDC; turbulence flame factor 3.7.

hydrogen with stoichiometric operation condition produces a significant reduction in the SFC (38%), a significant reduction in the CO emission (15%), an increase in power (1%), and an increase in thermal efficiency (1.5%) with hydrogen to fuel mass ratio of 2%. On the other hand, the only disadvantage of this operation is the increase of NOx emission by the order of 25%. Therefore, to avoid the increase of NOx emission, hydrogen is added at lean combustion operation. These conditions (lean mixture blended with hydrogen) gives lower level of NOx emission compared with that of a pure gasoline operation. This is due to less combustion energy and hence

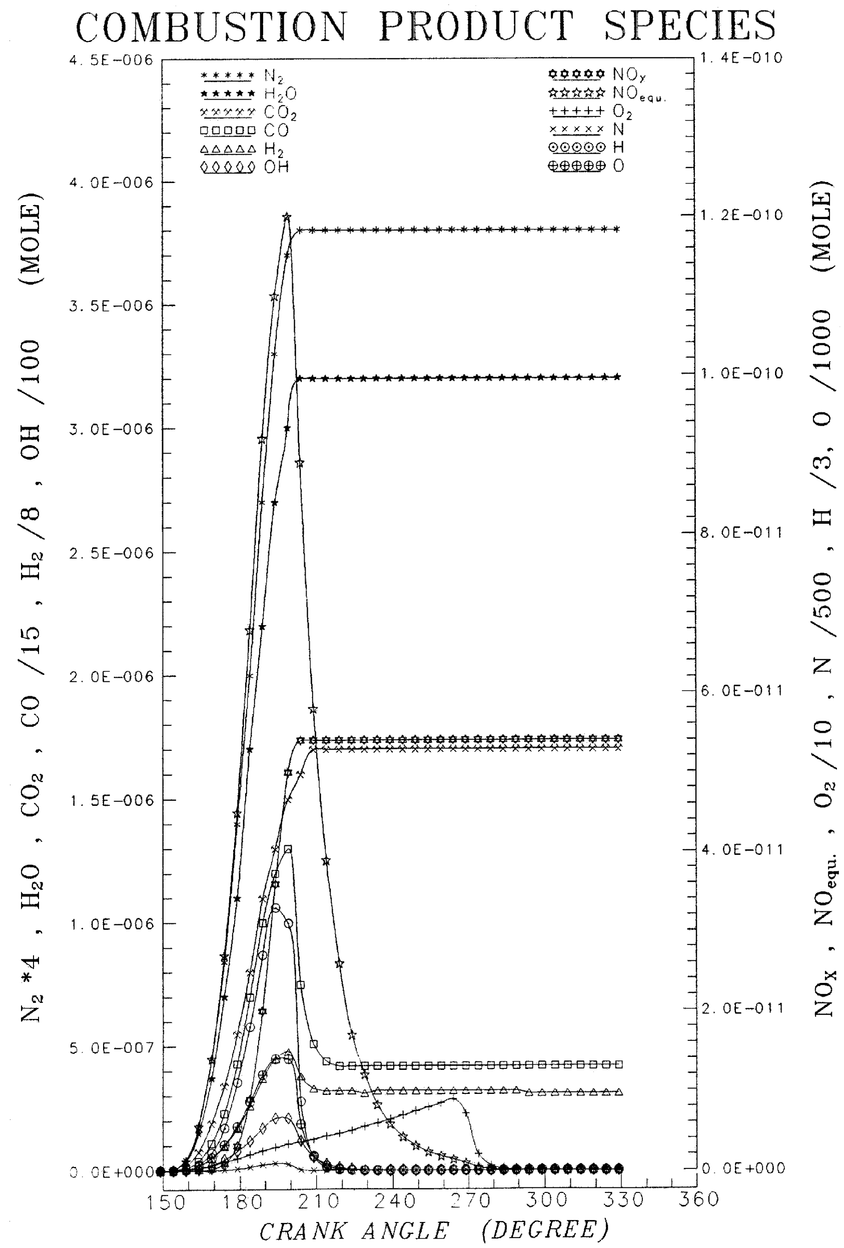

Fig. 7. Variation of concentration of combustion products with crank angle. Fuel mass ratio: 90% gasoline +10% hydrogen; equivalence ratio: 0.8; compression ratio: 7.5; engine speed: 1500 rpm; spark advance: 35 BTDC; turbulence flame factor: 3.7.

lower flame temperature. Therefore, in general, the leaner operation allowed by hydrogen blending with gasoline will reduce NOx emission.

3.3. *Effect of hydrogen blending on performance and emission of the engine fueled with lean mixture*

The operation of gasoline engine with an equivalence ratio less than stoichiometric is accompanied by many problems, such as the cylinder-to-cylinder mixture composition variation, cyclic pressure and mixture composition variation, and misfiring.

Figure 9 presents the effect of the amount of the added hydrogen, (from 0 to 20%, by mass), on the performance and emission of the spark ignition engine, working at the optimum spark timing for best torque, 7.5 compression ratio, 1500 rpm engine speed and 0.8 equivalence ratio.

The performance and emission of the engine, for each

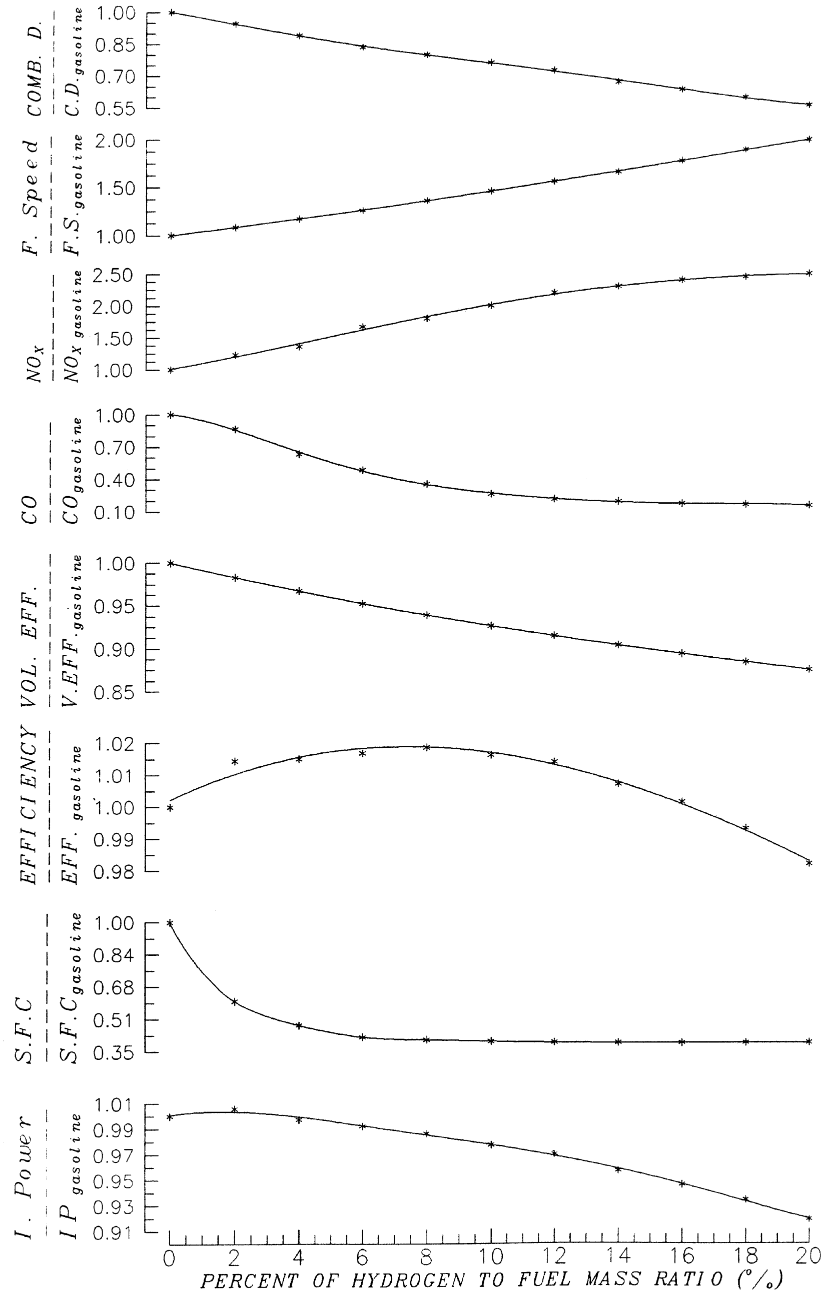

Fig. 8. Effect of hydrogen blending on the performance and emission of SI engine. Equivalence ratio: 1.0; compression ratio: 7.5; engine speed: 1500 rpm; optimum spark advance; turbulence flame factor: 3.7.

state of hydrogen added at 0.8 equivalence ratio, is presented as a ratio to performance and emission of the engine when fueled with stoichiometric mixture of gasoline and air.

In general the results show that a comparatively small amount of hydrogen mixed with gasoline and air produces a combustible mixture which can be burned in a conventional SI engine at an equivalence ratio below the stoichiometric limit without any problem. The resulting lean combustion produces low flame temperature and leads directly to lower heat transfer to the walls, higher engine efficiency, higher volumetric efficiency, lower specific fuel consumption, and lower concentrations of CO and NOx.

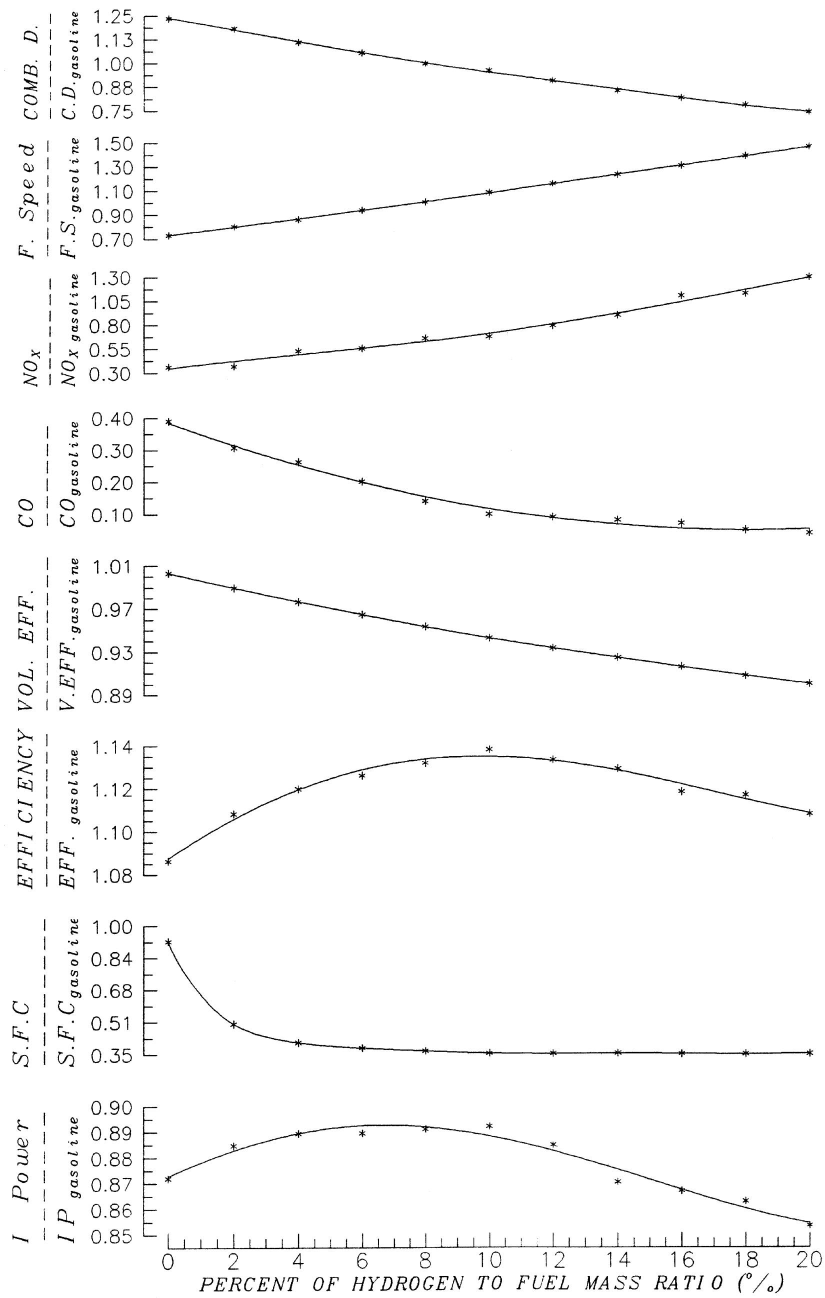

Fig. 9. Effect of hydrogen blending on the performance and emission of SI engine. Equivalence ratio: 0.8; compression ratio: 7.5; engine speed: 1500 rpm; optimum spark advance; turbulence flame factor: 3.7.

Figure 9 shows 10% blending by mass of hydrogen with lean mixture operation (i.e. $\phi = 0.8$) causes a 30% reduction in NOx emission, 85% reduction in CO emission, 60% reduction in specific fuel consumption, and an increase in the thermal efficiency by 14% compared with that of a stoichiometric pure gasoline–air mixture. However, this produces a reduction of 10.5% in the power, which is less than the reduction in power when a lean mixture of pure gasoline–air ($\phi = 0.8$) is used (13%).

4. Conclusion

The following conclusions were 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 saving a considerable part of the available oil and save our environment from toxic pollutant.
2. The hydrogen added to gasoline engine acts as a burning promoter, and expands the range of combustibility of the fuel–air mixture and hence a leaner mixture can be burnt.
3. It 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.
4. The thermal efficiency of the engine is increased until a hydrogel–fuel mass ratio of 8% for stoichiometric mixture and 10% for 0.8 equivalence ratio.
5. The concentration of CO is reduced and the concentration of NOx is increased due to hydrogen blending.
6. The specific fuel consumption is also reduced.
7. The peak pressure and temperature increase, and the pressure diagram gets closer to the ideal diagram.
8. The blending of hydrogen increases the heat release rate.
9. The power is increased until hydrogen–fuel mass ratio of 2% for stoichiometric mixture and 10% for 0.8 equivalence ratio.
10. The exhaust temperature is reduced, as does the crevice flow energy.
11. The volumetric efficiency is reduced.
12. The NOx emission can be reduced by operating the engine with lean mixture.

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