Numerical approach of the effects of gasoline-hydrogen blends on the cyclic variability in spark ignition engines

S.D Martinez Boggio[a], P.L. Curto Risso[b], A. Medina[c] , and A. Calvo Hernández[d]

[a] Department of Mechanical Design, University of La República, Montevideo, Uruguay, smartinezb@fing.edu.uy
[b] Department of Applied Thermodynamics, University of La República, Montevideo, Uruguay, pcurto@fing.edu.uy
[c] Departamento de Física Aplicada, Universidad de Salamanca, Salamanca, Spain, amd385@usal.es
[d] Departamento de Física Aplicada and IUFFYM, Universidad de Salamanca, Salamanca, Spain, anca@usal.es

Abstract:
The present work analyzes the cyclic variability in spark ignition engines fueled by different gasoline-hydrogen blends by means of quasi-dimensional computer simulations. The cyclic variability is a well-known phenomenon in this kind of engines, especially when operating at lean conditions. It can affect engine stability and efficiency to a great extent. Among its main causes is the development of turbulent combustion in each cycle, which is associated with turbulent flow during admission. On the other hand, hydrogen is a fuel known by its elevated laminar flame speed which is, in principle, interesting to increase the rate of combustion, and so decrease cyclic variability effects. Our group has developed a quasi-dimensional numerical simulation that incorporates cycle-to-cycle variability. In the present work the effect of hydrogen addition is considered in the model. Wrinkled flame effects are taken into account by introducing a linear factor that depends on the amount of hydrogen in the fuel, affecting the flame area development. The model is validated by direct comparison with experimental results. Studying the coefficient of variation of power output time series, it is concluded that for lean mixtures, when the percentage of hydrogen increases, the cyclic variability decreases up to a minimum value which depends on the fuel-air equivalence ratio. Minimum values of cyclic variability amplitude are observed at fuel ratios near 0.9 and 70% percentage of hydrogen in fuel by volume.

Keywords:
Hydrogen gasoline blends, spark ignition engine, cycle-to-cycle variability, engine simulations.

1. Introduction

The improvement on efficiency records as well as the reduction of pollutant emissions are key objectives in the development of spark ignition engines. From the viewpoint of fuels, another basic issue is the search for substitutive, or at least, supplementary fuels to avoid the problems associated to fossil fuels: shortage, increasing production difficulties, and pollution concerns. One of the promising renewable alternatives or supplements is hydrogen. It can be used in fuel cells to generate power in electric engines or directly in combustion engines. The first still have high production costs and need specific infrastructures along roads, although achieve a large power output at zero emissions rate. The latter provides a more flexible use because standard spark ignition engines do not need extensive changes to work with gasoline-hydrogen blends, allowing governments to progressive infrastructure investments.

Engines fueled with pure hydrogen, apart from safety considerations, lead to worse power output because of its reduced energy density by unit volume. Moreover, lead to increased NO_x emissions

due to the high flame temperature of hydrogen. Nevertheless, considered as an additive in gasoline-hydrogen blends, hydrogen improves combustion and reduces cycle-to-cycle variability because of its high laminar flame speed, which shortens combustion times. Among the drawbacks of using hydrogen added to gasoline it is necessary to mention the energy consumption required for its production, the difficulties of its storage, and the modifications needed on standard spark ignition engines fueled with gasoline. A review on the hydrogen-fueled internal combustion engines is due to Verhelst et al. [1].

Our research group has developed during the last years a quasi-dimensional simulation model capable to reproduce and analyze the cyclic variability observed in real engines. From a 0-dimensional scheme the model incorporates the evolution of the flame front radius and two control volumes or zones (for unburned and burned gases) during combustion. The evolution of the masses is calculated from an eddy-burning model, by solving a set of coupled differential equations in parallel with the thermodynamic differential equations. Its accuracy to reproduce the main engine records as well as variability effects has been proved for engines fueled with pure gasoline [2, 3, 4] and also for gasoline-ethanol blends [5]. Among its main advantages with respect to other simulation techniques it is easy to implement and has a reduced computational cost. This enables on one side to check the influence of a wide variety of parameters (and so to perform optimization studies) and on the other side to obtain long time series that allows an accurate analysis of cyclic variability. All the variables involved have a clear physical or chemical meaning, so it is relatively easy to analyze the consequences of physical and chemical issues in real engines. The model deals with valves overlapping, heat transfer from the cylinder to the coolant, the effects of residual gases from previous cycles, and specific turbulent flow models and turbulent combustion in the chamber.

The main objective of this work is to present a version of the model adequate to simulate the behaviour of cyclic variability for engines fueled with gasoline-hydrogen blends for several hydrogen concentrations and fuel-air equivalence ratios. The results given by the model will be compared with the experimental ones in the literature with validation purposes. Moreover, cycle-to-cycle variability of power output time series will be analyzed in detail. Previous experimental and CFD simulation studies have analyzed the performance of spark ignition engines by blending gasoline with hydrogen, and also have obtained time series for the cyclic variability of parameters like IMEP, engine speed, etc., for several hundreds of cycles [6-7]. A key advantage of quasi-dimensional simulations, associated to their reduced computing requirements, is their capability to obtain long time series, of about several thousands of cycles. This long time series are susceptible of a detailed study from the perspective of non-linear dynamics analysis.

2. Numerical model

The model developed by our group during the last years [8-10] is based on the first principle of thermodynamics for open systems. The control volume considered is delimited by the cylinder inner wall. The working fluid is an adiabatic mixture of unburned (u) and burned (b) gases except during combustion, where a two-zone scheme is followed. All gases are considered as ideal with temperature dependent specific heats. The adiabatic constant for the burned gases is calculated from the equilibrium composition at any time instant. Enthalpy changes are only associated to temperature changes except during combustion, where species redistribution according to chemical reactions is taken into account [8-12]. Details about the model can be found in [10].

A system of differential equations for pressure and temperature inside the combustion chamber is built, after taking particular values for the initial conditions. These equations are valid during the evolution of the system (including the overlapping period where intake and exhaust valves are simultaneously open) except for combustion. For all the stages of the cycle particular models are required. In the combustion time lapse two different control volumes are considered. They are separated by the flame front. The initial temperature of burned gases is taken as the adiabatic flame temperature at constant pressure. Moreover, the pressure is considered as uniform during all this period [11-12]. In the next section we shall deepen in the details of the combustion model.

The model is applicable to stationary (constant rotation speed) or non-stationary conditions. In the first case, from the mechanical viewpoint the only relationship required is between the volume of the combustion chamber at any given instant and the crankshaft angle.

The mass flow rates through the valves are quantified by means of the standard equations for the isentropic flow of a compressible fluid through an orifice [12]. For friction components the empirical correlation by Barnes-Moss is considered, including both linear and quadratic terms of the engine speed [13]. The heat transfer between the gas inside the cylinder and the cylinder inner wall is modelled with the Eichelberg correlation [2,14].

2.1. Combustion model and effects of hydrogen on flame wrinkling

To simulate combustion we use the model proposed by Blizard and Keck [15-16] and afterwards improved by Beretta [17]. Combustion is considered as turbulent. During the flame propagation the flame front is approximately spherical but not all the mass inside is burned, but there exist vortices or eddies of unburned gases. They have a characteristic length, l_t. The set of coupled differential equations that give the time evolution of the total mass inside the flame front, m_e, (this is the total mass inside the flame front, unburned and burned gases) and the mass of burned gases, m_b, is written as:

$$\dot{m}_b = F_w A_f \rho_u S_L + \frac{(m_e - m_b)}{\tau_b} \tag{1}$$

$$\dot{m}_e = F_w A_f \rho_u \left[u_t\left(1 - e^{-t/\tau_b}\right) + S_L\right] \tag{2}$$

In Eq. (2), t represents the time elapsed from the beginning of combustion, A_f, is the area of the spherical flame front, calculated from its radius, and this in turn from the volume of gases inside the flame front [11], V_f. This volume is given by $V_f = V - (m - m_e)/\rho_u$ where V is the total cylinder volume, ρ_u, the density of unburned gases and m, the total mass in the cylinder. The parameter, F_w , represents the flame front wrinkling, associated to the addition of hydrogen [18]. We shall detail this point hereafter. The first term at the right of Eq. (1) represents the laminar propagation of the flame front, and the rightmost the combustion of the fresh mixture inside the flame front. The time parameter $\tau_b = l_t/S_L$, is the characteristic time required for the combustion of a vortex of characteristic length, l_t, at an speed S_L, which is the laminar flame speed. Moreover, τ_b also represents the time the flame front needs to develop into a turbulent flame from the initial conditions: laminar flame and spherical symmetry [12]. Equation (2) describes the total mass inside the flame front rate that evolves at a velocity $u_t + S_L$ ($\tau \gg \tau_b$), where u_t is the characteristic speed of the mixture when crossing the flame front. During combustion these equations are coupled to the thermodynamic ones.

The dependence on temperature and pressure of the laminar flame speed is calculated from the correlation by Blizard and Keck [15-16],

$$S_L = S_{LO}\left(T_u/T_{ref}\right)^{\alpha}\left(p/p_{ref}\right)^{\beta}(1 - 2.1y_r) \tag{3}$$

where T_u is the temperature of unburned gases in the chamber, p is the pressure, y_r is the mole fraction of residual gases from the previous cycle and T_{ref}, and p_{ref}, reference values. For isooctane-hydrogen blends the reference laminar speed, S_{L0}, and the exponents α, and β where taken from Ji *et al.* [18], who considers mixtures up to 10% hydrogen in the incoming fresh air.

The flame surface wrinkling in spark ignition engines is always associated to turbulence. The degree of wrinkling depends on the combustion reaction rate and on combustion duration. Ji *et al.* [6] suggested that due to the high thermal and mass diffusivities of hydrogen, its addition to a fuel could reduce the flame thickness. This would mean that hydrogen-enriched flames are more sensitive to local turbulence and thus, experience more wrinkling. Moreover, the larger flame speed

after hydrogen blending could expand the local reaction rate, which also contributes to the generation of flame surface [19].

In this work, wrinkling effects are taken into account in the flame front area. In Eqs. (1) and (2), the flame front area is multiplied by a factor F_w that fits the difference between a smooth spherical flame front and a wrinkled one. From the data obtained by Ji *et al*. [6] through CFD simulations in the same conditions considered here, we built a correlation that reads as:

$$F_w = 0.41737 y_{H2} + 1 \tag{4}$$

In this equation, y_{H2}, represents the hydrogen in the fuel blend per unit volume. This parameter adequately fits the evolution of burned gases for blends up to 80% hydrogen per volume in the blend. Details of combustion conditions for which the correlation holds can be checked in [6-7]. The increase in flame wrinkling together with the increase of the laminar speed when hydrogen is added to the fuel provokes an acceleration of the flame front propagation, so reducing the combustion time.

2.2. Chemical reactions

For the chemical reaction associated to combustion, chemical equilibrium conditions with dissociation were considered for a mixture of isooctane, hydrogen, dry air, and previous cycle reaction products. We use a method presented by Ferguson [20] to solve chemical equilibrium for ten species, but considering residual gases. The corresponding equation reads as follows:

$$\begin{aligned}\frac{(1-y_r)}{1+\varepsilon\emptyset}[\varepsilon\emptyset\{(1-y_{H2})C_8H_{18}+y_{H2}H_2\}+0.21O_2+0.79N_2] \\ +y_r[y'_1CO_2+y'_2H_2O+y'_3N_2+y'_4O_2+y'_5CO+y'_6H_2+y'_7H+y'_8O \\ +y'_9OH+y'_{10}NO] \\ \rightarrow\ v_1CO_2+v_2H_2O+v_3N_2+v_4O_2+v_5CO+v_6H_2+v_7H+v_8O+v_9OH \\ +v_{10}NO\end{aligned} \tag{5}$$

In Eq. (5), $\varepsilon\emptyset$ is the amount of fuel per mole of dry air, with $\varepsilon = \frac{0.21}{(1-y_{H2})12.5+y_{H2}0.5}$ and $\emptyset$ the fuel-air equivalence ratio. The mole fraction of residual gases coming from the previous combustion event is denoted by y_r. The mole fraction composition of reactants in the fresh mixture is denoted by y'_i for each specie i. In the right hand of the equation, for each chemical specie, the unit of the multiplying coefficients, v_i, are the corresponding moles of each specie i, divided by the moles of reactants. These coefficients are obtained by stating a set of algebraic equations arising from the conservation of the number of atoms of each specie [10].

3. Validation of the simulation

The simulation scheme previously described was validated by comparing with the experimental results by Ji *et al.* [7] for isooctane-hydrogen blends. Table 1 summarizes engine parameters.

Table 1. Engine specifications

Engine Type	In-line, 4 stroke
Engine Manufacturer	Beijing Hyundai Motors
Bore [mm]	77.4
Stroke [mm]	85
Connecting Rod [mm]	141
Displacement [l]	1.599
Compression Ratio	10:1
Number per valve	4 per cylinder
IVO/°CA BTDC	45
IVC/°CA BTDC	135
EVO/°CA ATDC	135
EVC/°CA ATDC	45
Rated Torque [N.m/rpm]	143/4500
Rated Power [KW/rpm]	82/6000

The engine speed for simulations was taken as 1400 rpm and the spark advance 22°CA BTDC, the same values that in experiments [7].

Model validation was based upon the comparison with the experimental curves of the simulated pressure the cylinder, as well as the burned gas mass fraction during combustion. If the evolution of pressure with the crank angle is similar to experimental results, it can be stated that the indicated power output is alike in both approaches. Similarly, if the computed evolution of the burned gases mass fraction is almost identical that the one shown by the experiments, it could be assured that both techniques lead to the same rate of energy release and so efficiencies will be similar.

Figure 1(a) displays the evolution with the crank angle, CA, of the mole fraction of burned gases, x_b, and Fig. 1(b), the evolution of the in-cylinder pressure. The plots show that simulation results are globally in good accordance with the experimental ones by Ji *et al.* [7]. For the pressure the relative deviations are always below 5% and more that 90% of the computed points have a deviation under 2%. Deviations for the burned gases mass fraction are even smaller.

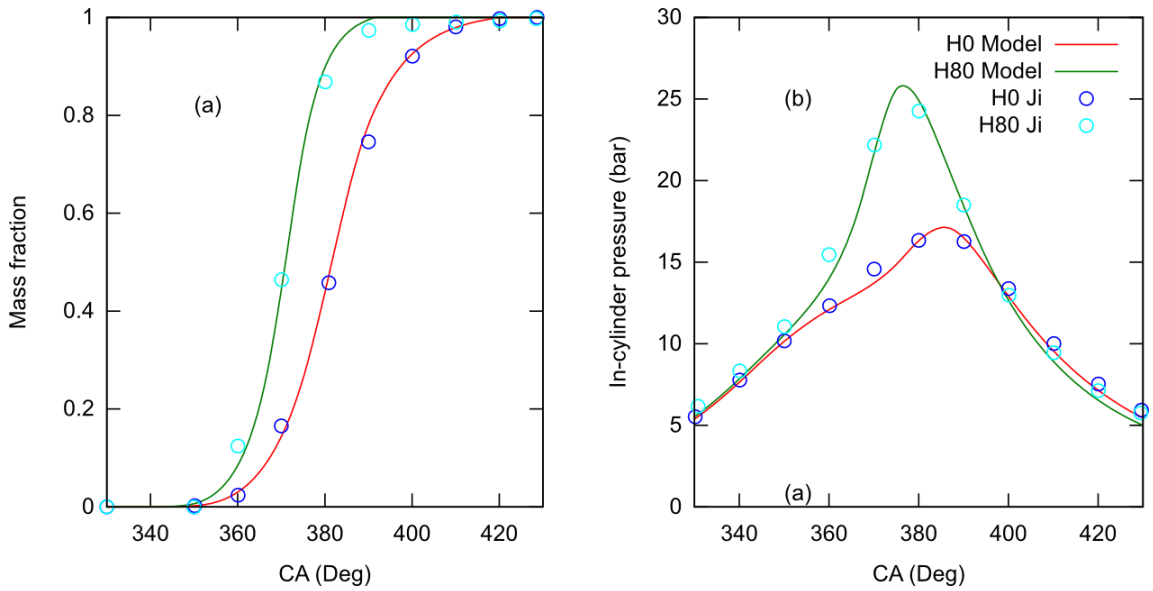

Fig. 1. Experimental (dots) [7] and simulated curves (solid) for the evolution of the burned gases mass fraction (a) and the pressure inside the cylinder (b) for two blends: pure isooctane (H0) and 80% hydrogen in fuel by volume (H80).

Because of the high flame speed of hydrogen (about five times that of isooctane) and its higher flame temperature, for the blend with an appreciable amount of hydrogen (H80, 80% hydrogen in volume), the curve corresponding to the burned gases mass fraction increases much rapidly that for pure isooctane (H0) in the fastest combustion region. This provokes a higher pressure peak. Thus, the mixture burns in a shorter time period and in a zone close to the Top Dead Center (TDC) that corresponds to 360º in our computations. As a consequence the cycle evolution is closer to an ideal constant volume combustion, which guarantees a better engine performance.

After reaching its maximum value, the pressure decays faster in the case H80 than in the case of pure isooctane, associated to a shorter combustion duration when isooctane is blended with hydrogen (as seen in Fig. 1 (a)). This reduces losses during the exhaust process because less fuel is burned during this stage.

Summarizing the validation process, our quasi-dimensional computer simulation is capable to reproduce the main engine indicators and its evolution during engine operation. The validation was performed for blends up to 80% hydrogen in fuel by volume, which represents a quite wide interval of hydrogen concentration. So, our numerical model presumably is a good tool for the analysis of spark ignition engines fueled with isooctane-hydrogen blends.

4. Cyclic variability on the power output

Once we have presented the validation of our simulation model, the objective turns to show and analyze the cycle-to-cycle fluctuations that simulations predict for the engine when fueled with isooctane-hydrogen blends.

Cyclic variability in spark ignition engines affects driving comfort, increases pollutant emissions and reduces the average thermal efficiency of the engine. For this kind of engines it is assumed that the main physical ingredients of variability are the following: turbulences during gases motion in the cylinder as well as during combustion, homogeneity of the mixture in the surrounding area of the spark plug, and the memory effects associated to the presence in the chamber of residual gases from the prior cycle [21]. Variability is more evident for poor mixtures and low engine speeds.

In our model, the laminar flame speed connects combustion dynamics and the mole fraction of residual gases from the preceding cycle, y_r (see Eq. (3)). The essential parameters that in the assumed combustion model affect the development of the flame front are: the characteristic length of the entrained eddies, l_t, the characeristic speed, u_t, at which the mixture crosses the flame front (that is related to the turbulence intensity) and the location of the ignition kernel.

Previous studies from our group [5, 8] showed that the consideration of stochastic fluctuations at least on l_t or u_t is essential to reproduce the phenomenology of cyclic variability that experiments reveal. The turbulence intensity (related to u_t) directly influences the flame speed, that is a key factor in cyclic variability. Any increase in flame speed tends to diminish the importance of cyclic variability. Thus, any action that could contribute to increase the turbulence intensity in the cylinder is convenient to reduce variability. But usually this also provokes a reduction in volumetric efficiency and engine performance at high speeds. In consequence, it is interesting to identify other mechanisms that could increase flame speed without affecting cycle performance at high speed[22]. One possible mechanism is to modify the fuel in the engine. In this context hydrogen is a good candidate because of its high diffusivity and combustion speed. As commented in the Introduction, it seems preferably to use it as an additive to isooctane, thus fueling the engine with isooctane-hydrogen blends.

In order to incorporate variability in our simulation model, a random component on l_t at the beginning of combustion (in each cycle) is introduced and thereafter it evolves depending on the ratio of the densities of the burned and unburned gas mixture (see [10] for details). It is important to note that the empirical correlations by Beretta [17] for l_t and u_t, depend on the mean inlet gas speed and on the maximum valve lift, not on the chemical composition of the mixture. So, we assume the random component for l_t identical that for pure isooctane. We adjust the random component by

comparing the coefficient of variation of the IMEP as obtained in our model with the experimental data presented by Ji *et al.* [23] for H0 and H80[1].
When making computations of variability we take the same parameters that in the validation section. Although the validation presented in Sec. 3 was performed for blends up to 80% hydrogen per volume of the incoming fuel, it was observed that qualitatively the behaviour is correct when comparing with the results by Ji *et al.* [7], up to 100% hydrogen, H100. Thus, we explore all the range, from pure isooctane, H0, to pure hydrogen, H100. In the interval not presented in the validation, the results were extrapolated by using the same factor, F_w, from Eq. (4).
In figure 2 the aspect of the time series of power output for 1000 cycles are shown. Three different hydrogen concentrations and three values of the fuel-air ratio, $\emptyset$, were taken.

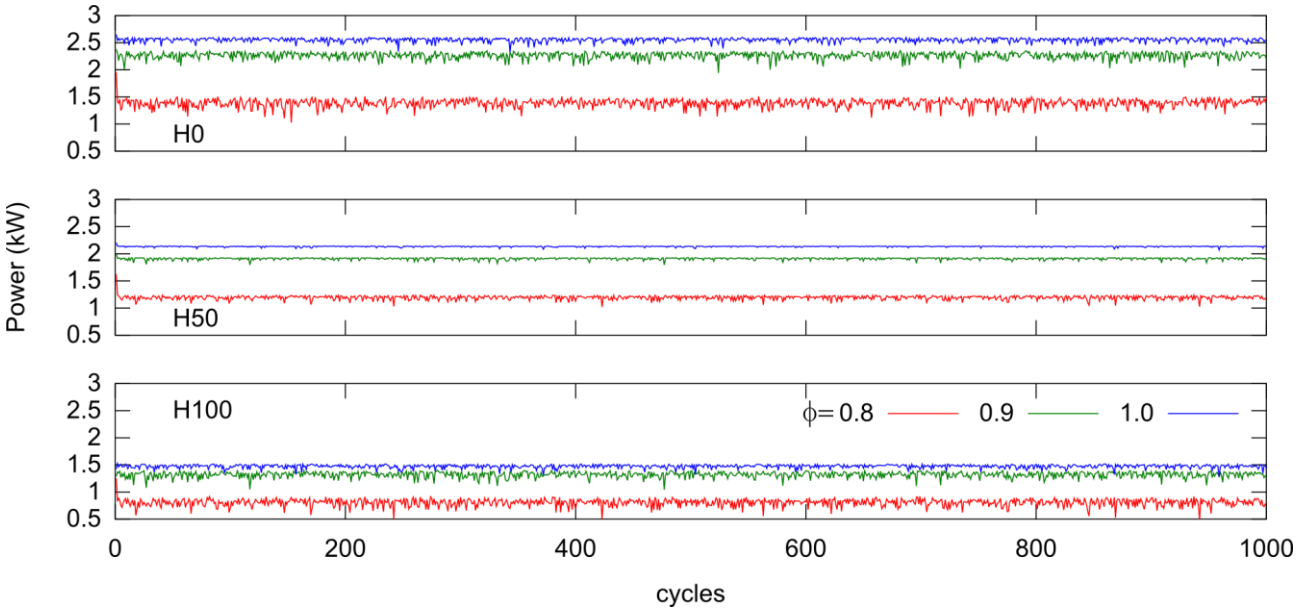

Fig. 2. Cycle-to-cycle variability of the power output obtained from the simulation model for three concentrations in volume of hydrogen (H0, H50, and H100) and different fuel-air ratios ($\emptyset = 0.8, 0.9, and\ 1.0$).

The figure qualitatively shows that variability amplitude is minimum for the blend H50 and any of the considered fuel-air ratios. For all the hydrogen concentrations, the same tendency is clear: higher fuel-air ratio provokes an increase in the average power output and a decrease in variability amplitude. Hereafter we statistically analyze the time series of power output. Four indicators are calculated: the mean value μ, the coefficient of variation COV, defined as $COV = \sigma/\mu$ where σ is the standard deviation, the kurtosis K and the skewness S. The calculation of these parameters constitutes a basic preliminary step in order to analyze the dynamics of non-linear complex time series [10].

Figure 3 displays the evolution with the concentration of hydrogen per fuel volume of the mean value, μ, of the power output. As abovementioned, the power output decreases with the fuel-air ratio. The power output has its maximum average value for pure isooctane, then, when hydrogen is added, suffers a decrease for hydrogen concentrations up to 20%. Afterwards, average power remains almost constant when increasing H_2 between 20% and about 80 %. Finally, another decreasing step is observed when hydrogen concentration reaches 100%. The plateau for the lowest value of fuel-air ratio has a slightly positive slope. The energy density of hydrogen on volume basis is much lower than that of gasoline, so hydrogen powered engines suffer a weak power output.

[1] Notice that Ji [7] considers a percentage of hydrogen in the fresh air mixture. We use a percentage of hydrogen in the mixture of isooctane-hydrogen, therefore H3 defined by Ji corresponds to about 79.6 % of hydrogen in the mixture of fuel (approx. H80 in this work).

Compared with the pure hydrogen fueled engine, using a small amount of hydrogen as an additive to hydrocarbons fueled engines keep the positive properties of isooctane avoiding harmful power loses.

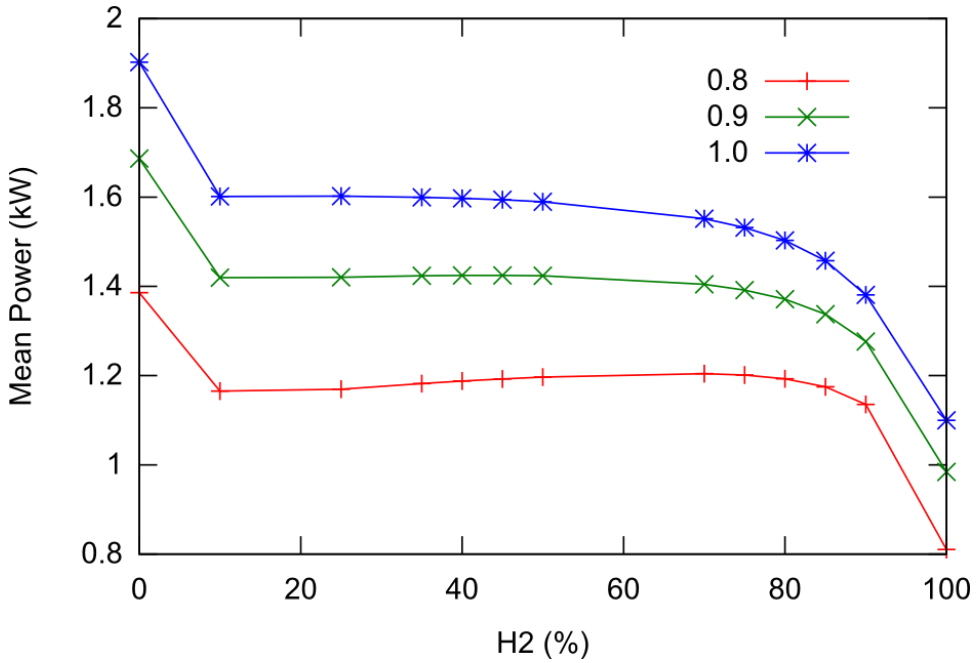

Fig. 3. Evolution of the mean value, μ, of the power output as a function of the volume percentage of hydrogen in the blend for several fixed values of the fuel-air equivalence ratio $\emptyset = 0.8$, 0.9, *and* 1.0.

The coefficient of variation, COV, gives a straightforward measure of the amplitude of the variations of the time series with respect to the mean value. In Fig. 4 it is shown the COV of all the time series presented before. For any fuel-air ratio the COV presents a minimum. Its location depends on the particular value of $\emptyset$. For lower fuel-air ratios, the minimum is located at higher concentratios: 90% for $\emptyset = 0.8$. Nevertheless, for rich mixtures this effect is opposite: for $\emptyset = 1.0$, the minimum is observed for the blend H50. Moreover, minima are more flat as the mixture is richer. In summary, poorer is the mixture higher hydrogen concentrations are required to decrease variability amplitude in the power output. It is also interesting to recall attention about one fact: there is an interval of concentrations (between 65% and 90%) where the COV is higher for $\emptyset = 1.0$. than for $\emptyset = 0.9$. In all the other range of hydrogen percentages the amplitude of irregularities is higher for poorer mixtures. In consequence, the incorporation of the adequate percentage of hydrogen in the fuel allows running the engine with poor mixtures without provoking undesirable high cycle-to-cycle variability. This in turn improves engine efficiency because thermal losses decrease and reduces fuel consumption.

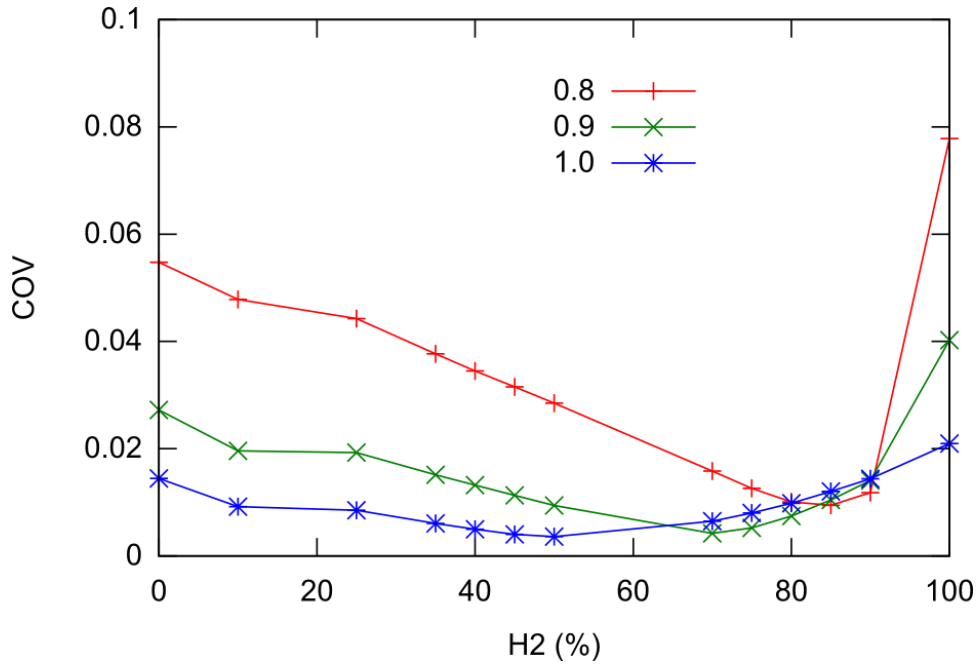

Fig. 4. Evolution of the coefficient of variation, COV*, of the power output as a function of the volume percentage of hydrogen in the blend for different fuel-air ratios (*$\emptyset = 0.8,\ 0.9,$ *and* 1.0*).*

The kurtosis, *K*, is a measure of how peaked is the data relative to a normal distribution: values of *K* over 3 indicate a peaked distribution and below 3 a flat one. In other words it measures the existence of particular values far away from the mean of the distribution [10]. In Fig. 5 it is presented the shape of *K* as a function of the hydrogen concentration in volume for three different fuel-air ratios. For $\emptyset = 0.8$, *K* has very large values for H_2 concentrations between approximately 70% and 90%. Particularly, its highest value is around 85% hydrogen. As observed from the zoom in the figure, for the other values of $\emptyset$, there is also a maximum for K, but with numerical values well below that for $\emptyset = 0.8$.

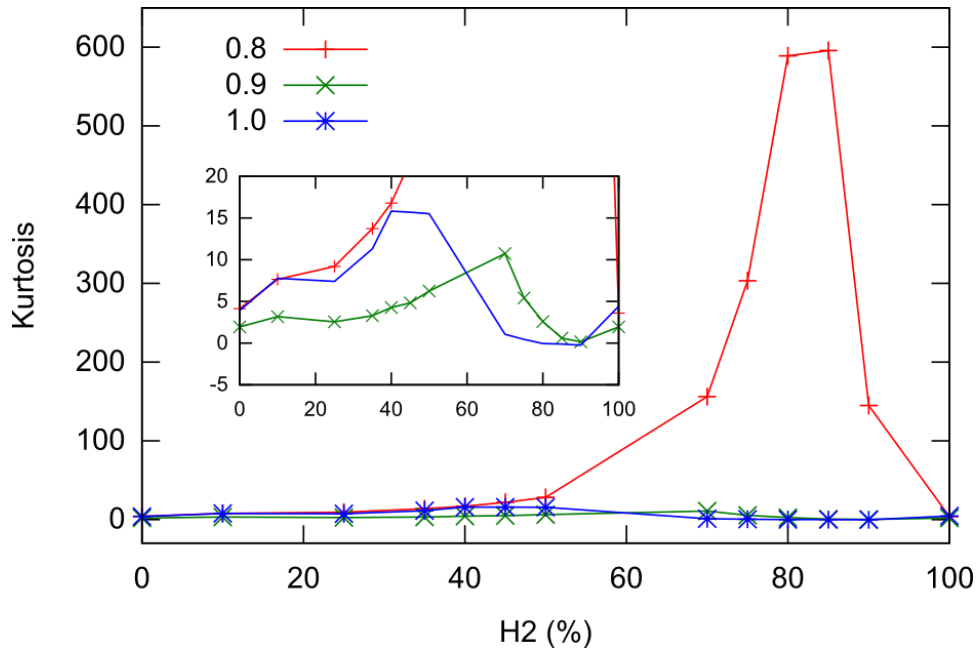

Fig. 5. Evolution of the kurtosis, K, of the power output as a function of the volume percentage of hydrogen in the blend for different fuel-air ratios ($\emptyset = 0.8,\ 0.9,$ *and* 1.0*).*

The skewness S of a time series is a measurement of the lack of symmetry of the pattern [10]. Negative values of S represent the existence of left asymmetric tails longer than the right. In the case considered, values of S are always negative for $\emptyset$ above 0.8 (see the inset in Fig. 6). Nevertheless for $\emptyset = 0.8$ is quite different: S is negative for low hydrogen concentrations, but increases to reach a high positive value around 90% hydrogen in volume and then rapidly decreases, in such a way that for H100, S is again negative.

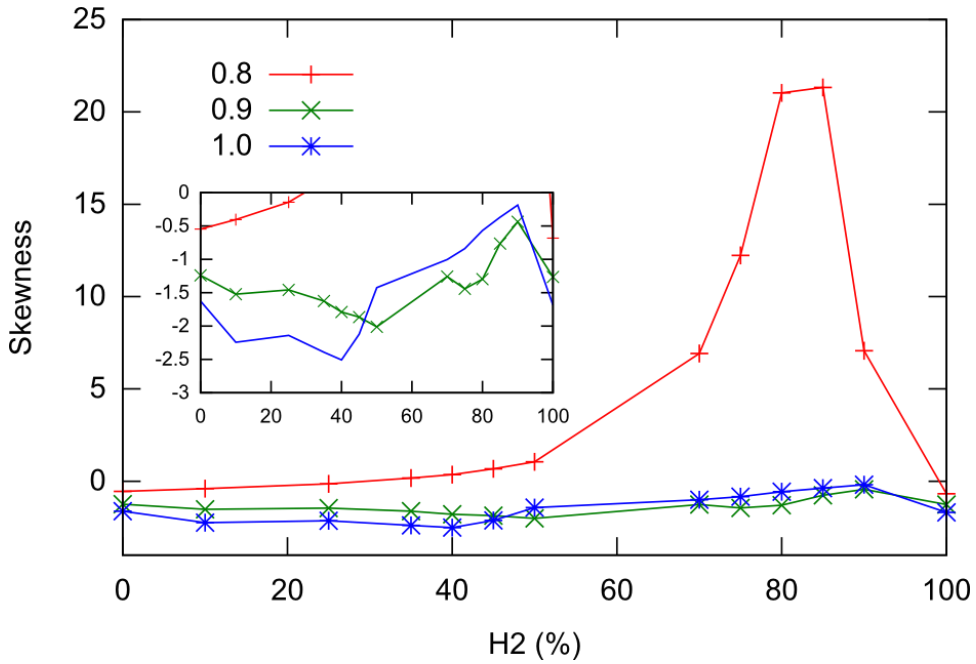

Fig. 6. Evolution of the skewness S of the power output as a function of the volume percentage of hydrogen in the blend for different fuel-air ratios ($\emptyset = 0.8,\ 0.9, and\ 1.0$).

4. Conclusions

We have developed a quasi-dimensional simulation in order to analyze the variability on the power output of an engine fueled with gasoline-hydrogen blends. The starting point is a previous model validated for pure isooctane and for isooctane-ethanol blends [5]. The model showed its capability to reproduce the main experimental characteristics of cycle-to-cycle variations. In this work we modified the chemistry of the combustion and the laminar flame speed. Moreover, the effect of flame wrinkling caused by hydrogen addition was taken into account by introducing a parameter that is linear on the hydrogen concentration per fuel volume. It is obtained from the numerical results by *Ji et al*. [6], relating the areas of smooth and wrinkled flame fronts.

We validated our model by recovering the experimental results by Ji [7] for the evolution of in-cylinder pressure and burned mass gas fraction. Results show that the model reproduces with accuracy experimental results up to 80% hydrogen per volume in the mixture.

To incorporate cycle-to-cycle fluctuations we introduce a random component on the size of the unburned gas eddies that entrain into the flame front during its development. This random component was adjusted by comparing experimental and simulation results, particularly the COV of IMEP. Within this scheme the computed time series of the power output for several values of the fuel-air ratio of the blend and for several hydrogen concentrations were presented and analyzed.

Results show that the average power output reaches its maximum value for pure isooctane, and decreases when increasing quantities of hydrogen are added to the mixture, although curves show a plateau between approximately 25% and 70-80% of hydrogen per volume. The mean power output also decreases as the fuel-air ratio of the mixture becomes lower. The coefficient of variation, COV, is globally higher for poor mixtures and displays a minimum for each fuel-air ratio. That minimum

is located towards higher hydrogen concentrations as the mixture becomes poorer. The largest COV values are those for an engine fueled with pure hydrogen. This is a peculiarity of the power output for hydrogen blends, since results from the literature [4-5, 24-25], for instance for the cyclic variability of heat release for gasoline-ethanol blends, show that the amplitude of cyclic variability monotonically decrease for increasing ethanol concentration, independently of the fuel-air ratio.
In summary, this work shows the capability of quasi-dimensional computer simulations in order to reproduce and analyze cycle-to-cycle variations in real spark ignition engines when the fuel is a gasoline-hydrogen blend. Because of the evident interest to use alternative fuel blends more work along this line is desirable.

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