Reducing the Cycle Variability in a Spark Ignition Simulated Engine

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ABSTRACT

The present work aims to study the optimal parameter configuration for different operating and design variables of a spark ignition engine with the goal of minimizing the cyclic variability (CV) of its efficiency time series. We make use of a quasi-dimensional numerical simulation of a mono-cylindrical spark ignition engine. The CV is modelled by incorporating a stochastic component in the characteristic length and the velocity of the turbulent combustion model.

The focus of this work is to reduce CV through the reduction of the coefficient of variation, considering five different parameters, related to the crankshaft angle, that have incidence in CV: the spark advance, the intake valve opening angle, the intake valve closing angle, the exhaust valve opening angle, and the exhaust valve closing angle. A Random Search method was used for sampling the search space of the different parameters, considering a discretization angle of one degree. The results show that a significant reduction in the coefficient of variation can be obtained by appropriately choosing the parameter values for the different operating scenarios of the spark ignition engine. The experimental evaluation also shows that the two most relevant parameters with a greater incidence in reducing the coefficient of variation are the spark advance and the intake opening valve angle.

NOMENCLATURE

l_t	[m]	Characteristic length
u_t	[m/s]	Characteristic velocity
S_l	[m/s]	Laminar flame speed
$S_{l,0}$	[m/s]	Laminar flame speed at reference condition
ρ	[kg/m^3]	Density
A_f	[m^2]	Flame front area
$\dot{m}$	[kg/s]	Time derivative of the mass
m_e	[kg]	Mass within the flame front
τ	[s]	Time from the beginning of combustion
τ_b	[s]	Characteristic time
T	[K]	Temperature
P	[Pa]	Pressure inside the combustion chamber
T_{ref}	[K]	Reference temperature
p_{ref}	[Pa]	Reference pressure
ϕ	[-]	Fuel air equivalence ratio
y_r	[-]	Volume fraction of residual gases
y_i	[-]	Mole fraction of reactant i in the fresh mixture
ν_i	[moli/molr]	Moles of specie i divided by the moles of reactants

Subscripts

U	Unburned gases
B	Burned gases

INTRODUCTION

In order to obtain high performance and low emissions, the engine manufacturers tend to set the operating range of the combustion near its stability limit. In this operating condition a phenomenon known as the cyclic variability (CV) is observed. The CV reduces the potential of operation at lean mixtures and therefore restrict the performance improvement.

The CV experienced by spark ignition engines is a fundamental combustion problem which is affected by many engine and operating variables like fuel properties, mixture composition near spark plug, charge homogeneity, ignition, in-cylinder charge, exhaust dilution, etc. Specially important seems to be the aerodynamics of the mixture just prior to ignition because it determines both the flame kernel growth in the initial phase and the variations in the flame propagation, thus affecting the burning rate [1].

In analyzing these kinds of fluctuation, many authors have reported extensive work by looking at different variables such as maximum pressure or heat release by means of different statistical methods including return maps [2], recurrence plots, correlation coarse-grained entropy [2,3], and sample entropy [2]. Daw et al. [4] proposed a discrete engine model that explains how both stochastic and deterministic features can be observed in a spark-ignited internal combustion engine, and they reproduced experimental observations.

Many authors have studied different mechanisms to reduce the cyclical variability and increase the engine performance. Kyaw and Watson [5] uses an ASTM CFR engine assisted by hydrogen jet ignition to achieve up to 70% maximum torque with NO, emissions close to ambient levels, and reducing the coefficients of variation of peak cylinder pressure and specific work per engine cycle by 50 to 80% from those normally achieved with this engine. This allows increased thermal efficiency at ultra lean operation, at equivalence ratios of around 0.5, and about two numbers increase in the highest useful compression ratio. Among other results, Fortea et al. [6] reported a study of a Twin Spark ignition system to improve combustion stability reached for part load conditions and they found a sensible reduction of cycle-by-cycle variability of indicated mean effective pressure. Sjeric et al. [7] introducing high pressure exhaust gas recirculation in supercharged engine for control of engine limiting factors such as knock, turbine inlet temperature and cyclic variability, allowing to reach stoichiometric mixtures when usually rich mixtures are used.

They found an improvement of fuel consumption of 8.7%, 11.2% and 1.5%, for low, medium and high speed.

This work aims to reduce the cyclical variability from a passive strategy, acting solely in the configuration of the engine valves and spark advance.

On the other hand, the study of CV in internal combustion engines has certain difficulties. From an experimental point of view is needed to acquire and to store large volumes of information in very small sample times. Additionally, from the point of view of numerical simulation, Computational Fluid Dynamics models require very high time to process a single cycle, making it impossible to reach very long time series. For these reasons, the best suited model to analyze this phenomenon is the quasi-dimensional, which it allows to implement large time series with low computational cost [1].

NUMERICAL MODEL

In the present work a quasi-dimensional numerical simulation of a mono-cylindrical spark ignition engine is used to calculate engine performance parameters. The model was previously presented and validated by our group in [8-10]. It is considered a two-zone model for combustion discerning between unburned (u) and burned (b) gases that are separated by a thin adiabatic flame front. A set of ordinary differential equations for pressure, temperature and masses is solved by a Runge-Kutta method. The particularities of each stage during engine evolution are taken into account.

In the course of combustion, the model considers that during flame propagation not all the mass within the flame front is burned, but there exist unburned eddies of typical length l_t. The coupled system of equations for the masses of the burned gas mixture m_b and the total mass within the flame front m_e (unburned eddies plus burned gas) reads as follows:

$$\dot{m}_b = \rho_u A_f S_l + \frac{m_e - m_b}{\tau_b} \tag{1}$$

$$\dot{m}_e = \rho_u A_f [u_t + S_l] \tag{2}$$

where u_t is the characteristic velocity at which unburned gases pass through the flame front, ρ_u is the unburned gas density, and A_f is the flame front area. $\tau_b = l_t / S_l$ is the characteristic time for the combustion of the entrained eddies and S_l is the laminar flame speed, which is determined from its value at reference pressure and temperature conditions [11-15]. Its numerical value is calculated from the correlation by Blizard and Keck [14,15],

$$S_l = S_{l,0}\left(\frac{T_u}{T_{ref}}\right)^{\alpha}\left(\frac{p}{p_{ref}}\right)^{\beta}\left(1-2.1y_r\right) \tag{3}$$

The reference laminar flame speed, $S_{l,0}$, and the exponents, α and β are were taken from Gülder [12]. 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 within the flame front. Equation (2) describes the evolution of the total mass inside the flame front (burned and unburned).

WORKING FLUID

In this study we have selected as fuel iso-octane C_8H_{18} (reference fuel for Otto's engines) and the following chemical reaction:

$$\frac{(1-y_r)}{1+\varepsilon\,\phi}\left[\varepsilon\,\phi\,\{C_8H_{18}\} + 0.21O_2 + 0.79N_2\right] + y_r\left[y'_1CO_2 + y'_2H_2O + y'_3N_2 + y'_4O_2 + y'_5CO + y'_6H_2\right] \rightarrow v_1CO_2 + v_2H_2O + v_3N_2 + v_4O_2 + v_5CO + v_6H_2 \tag{4}$$

In Eq. (4), $\varepsilon\,\phi$ is the amount of fuel per mole of dry air, with $\varepsilon = \frac{0.21}{12.5}$ and ϕ the fuel-air equivalence ratio. The mole fraction of residual gases coming from the previous combustion event is denoted by y_r. 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.

We make use of the subroutine developed by Ferguson [16] to solve combustion and calculate exhaust composition (but including residual gases among the reactants). Our model does not consider traces of C_8H_{18} in combustion products, but in the energy release we actually take into account combustible elements as CO, H, or H_2. The thermodynamic properties of all involved chemical species are obtained from the constant pressure specific heats, considered as 7-parameter temperature polynomials [17].

ACCOUNTING FOR CYCLIC VARIATION

In our model the term y_r in S_l connects combustion dynamics from the preceding cycle to the present one. Within the framework of the combustion model we are considering, the essential parameter determining the development of the flame front are: l_t and u_t. To incorporate variability, the model sets l_t randomly, at the beginning of combustion [1] and thereafter it evolves depending on the ratio of the densities of unburned gases and a reference state [18]. l_t is taken as a log-normal distribution variable, obtained by fitting experimental results by Beretta [18] and u_t is calculated by a connection parameter of densities ratio [18].

PROBLEM FORMULATION

The present work aims to find an optimal configuration of five different parameters that have incidence in the cycle variability: the spark advance (SA), the intake valve opening angle (IVO), the intake valve closing angle (IVC), the exhaust valve opening angle (EVO), and the exhaust valve closing angle (EVC). Our goal is to reduce the cycle variability through the reduction of the coefficient of variation defined as the ratio between the standard deviation and mean of the time series, $COV = \sigma/\mu$. From now on we use RCOV to refer to the coefficient of variation of the efficiency.

Table 1 presents the domain of case-study variables and the intervals considered in this work.

Table 1 Considered intervalsfor the decision variables

Decision variable	Description	Value range
IVO	Intake valve opening (degrees)	$-50 \leq \text{IVO} \leq -10$
IVC	Intake valve closing (degrees)	$210 \leq \text{IVC} \leq 250$
EVO	Exhaust valve opening (degrees)	$480 \leq \text{EVO} \leq 500$
EVC	Exhaust valve closing (degrees)	$735 \leq \text{EVC} \leq 750$
SA	Spark advance (in degrees)	$\text{IVC} \leq \text{SA} \leq 360$

We work with a discretization of one degree for every parameter that we have taken into consideration, which we consider realistic for a spark ignition engine. Even with this discretization, the search space (i.e. the number of possible combinations of parameter values) is really large (between 62 and 85 million of different combinations of values). Additionally, it should be taken into account that the runtime of a single execution of the numerical model takes about 600 seconds in the execution platform of our experiments.

For this reason, we use a stochastic search method that instead of performing an exhaustive search of the search space, only performs a sampling of the search space. In particular, we use a Random Search (RS) method that chooses randomly with a discrete uniform distribution between the values of the discretization for each of the parameters independently.

EXPERIMENTAL SETUP

Combustion in spark-ignition engines in lean burn conditions increases the cycle-to-cycle variations. This work surveys three lean conditions at three fuel-air equivalence ratios: 0.8, 0.85, and 0.9. The numerical values for the cylinder geometry are taken from Beretta et al. [18] for a fixed engine speed of 109 rad/s. Details on some running parameters of the computations can be found in [8].

Before selecting time series length, we perform a fast test to account for the evolution of COV as a function of the number of cycles. We conclude that COV monotonically decreases with the number of cycles up to an asymptotic limit which is reached not before 500 simulation cycles. This fact remarks the necessity to analyze long time series in order to recover representative information from the time series. Therefore the present work uses time series of 500 cycles. For each different fuel-air equivalence ratio, the RS method that randomly generates 5,000 sets of parameter values.

The execution platform is a PC with a Six Core Intel Xeon E5-2620 v2 processor at 2.10 GHz with 128 GB RAM using Linux O.S. The numerical model and the RS method were implemented in Fortran. All the executions were run as single-threaded applications.

COMPARISON WITH PREVIOUS WORKS

In a previous work [18], the parameters were configured using the following values: IVO -50.0º, IVC 214.0º, EVO 490.0º, EVC 750.0º and SA 330.0º. The numerical model was executed using those values for each fuel-air equivalence ratio, in order to have a reference value of the coefficient of variation of the efficiency (RCOV). This value is reported as previous work RCOV.

The RS method was executed generating 5,000 configurations of the parameter values for each equivalence ratio, which were then used for running the numerical method. The best value obtained of RCOV in the executions is reported as Best RS RCOV. We also analyze the average of the RCOV value obtained by the 1% best configurations, i.e. the 50 executions that obtained the lower values of RCOV. This value is reported as Best 1% RS Avg. RCOV. Table 2 presents a comparison of the three different RCOV values. The table also reports the percentage of improvement over RCOV value obtained with the configuration of the previous work.

The results show that a reduction of at least a 10% can be obtained by appropriately choosing the parameter values for the different operating scenarios. The methodology used for obtaining the best configurations is quite simple and it just consists in randomly sampling the search space of the different parameters values.

Table 2. Comparison of initial and best RCOV values.

ϕ	Prev. work RCOV	Best RS RCOV	Best 1% RS Avg. RCOV
0.80	3.20×10^{-2}	1.86×10^{-2} (-41.88%)	1.95×10^{-2} (-39.06%)
0.85	1.78×10^{-2}	1.60×10^{-2} (-10.11%)	1.65×10^{-2} (-7.30%)
0.90	1.54×10^{-2}	1.34×10^{-2} (-12.99%)	1.42×10^{-2} (-7.79%)

The results also show that, even when considering the 1% of the best configurations, the RCOV values obtained are more than a 7% better than the configuration of the previous work. In consequence, the methodology used in this work is robust since it allows to define several configurations with a reduced RCOV value. As mentioned before, when the values of the fuel-air equivalence ratio increase, RCOV decrease. Specifically, for $\phi = 0.8$, RCOV of the previous work is almost double that for $\phi = 0.85$. Additionally, and closely related with this, low values for equivalence ratio increases the sensibility for different parameters, i.e., for poor equivalence ratios optimization approaches, such as the one presented in this work, can offer major improvements. This can be confirm with the results in Table 2, where the improvement for equivalence ratio 0.80 is around 40%, 4 times the results obtained in other cases.

Table 3 presents the configuration leading to the best RS RCOV value for each equivalence ratio. It should be noted that while the parameter SA has similar values than the one used in the previous work, the rest of the parameters are quite different, specially the parameter IVO.

Table 3 Best configuration obtained by RS for each equivalence ratio. All angles are expressed in degrees.

ϕ	IVO	IVC	EVO	EVC	SA
0.80	-12.0	211.0	480.0	738.0	328.0
0.85	-11.0	227.0	500.0	736.0	334.0
0.90	-14.0	225.0	494.0	739.0	337.0

This maybe because the engine is originally optimized for best efficiency and volumetric efficiency, adjusting the spark advance and valve timing respectively. Since in this study the objective function is the coefficient of variation of efficiency, not necessarily efficiency, both set of values do not agree.

INCIDENCE OF THE PARAMETER VALUES

Let us now analyze the incidence of the different parameter values in the best RCOV values obtained. To this end, we analyze the parameter values of the 1% best configurations according to the RCOV value and the parameter values of the 5,000 configurations, which follows a discrete uniform distribution. If the values of a parameter do not influence the RCOV value, it is reasonable that the parameter values of the 1% best configurations should also follow a discrete uniform distribution. That is to say that, since the parameter has no impact, there should not be any bias in the values of the parameter. On the other hand, if there is a bias in the distribution of the values of the parameter, it shows that some values of the parameter produce better results and therefore the RCOV value is more sensitive to the parameter.

Since the configurations and the distribution of the parameter values are randomly generated, a statistical test is used to assess the significance of the difference of the distributions of the parameter values. The Kolmogorov-Smirnov (KS) test for discrete distributions [19] is used to check if the values of a parameter of the 1% best configurations and the values of the same parameter of the 5,000 configurations are drawn from the same discrete distribution. In particular, we have used the discrete KS test implementation of the `dgof` package in R. All the statistical tests are performed with a confidence level of 99%.

Table 4 presents the p-values of the discrete Kolmogorov-Smirnov test for each parameter between the distribution of the 5,000 configurations and the 1% best configurations. The values in bold indicate a p-value lower than 1.0×10^{-2}.

Table 4. P-values of the discrete Kolmogorov-Smirnov test

Parameter	ϕ = 0.80	ϕ = 0.85	ϕ = 0.90
IVO	$\mathbf{1.131\times10^{-5}}$	$\mathbf{7.962\times10^{-4}}$	$\mathbf{2.145\times10^{-7}}$
IVC	$\mathbf{5.399\times10^{-3}}$	4.463×10^{-2}	$\mathbf{8.941\times10^{-3}}$
EVO	1.899×10^{-2}	6.589×10^{-1}	5.071×10^{-1}
EVC	2.390×10^{-2}	1.867×10^{-2}	$\mathbf{9.726\times10^{-4}}$
SA	$\mathbf{< 2.200 \times10^{-16}}$	$\mathbf{< 2.200 \times10^{-16}}$	$\mathbf{< 2.200 \times10^{-16}}$

The Kolmogorov-Smirnov test shows that there are significant differences between the distributions, at all equivalence ratios, for the parameters SA and IVO. Additionally, there are also significant differences in two scenarios for the parameter IVC and in one of the scenarios for the parameter EVC. It should be noted that the p-values obtained for SA and IVO are very small, which is a strong evidence against the hypothesis that the parameter values are drawn from the same distribution.

Since the values of SA and IVO have a different distribution for the 1% best configurations than for the whole set of configurations, these parameters have a greater incidence in obtaining small RCOV values than the rest of parameters studied. This is physically consistent since the intake process and spark advance are directly related with combustion duration and therefore how the fuel energy is released during the cycle and affects the power output and efficiency by changing the indicator diagram. For the rest of the parameters, there is no statistical evidence that the values of the parameters for the 1% best configurations and for the whole set of configurations are drawn from different distributions. In other words, there is no better alternative to randomly select the value of the parameters IVC, EVO, and EVC with a uniform distribution. Table 5 presents the minimum, the maximum and quartile values for SA and IVO and the different fuel air equivalence ratios. The first quartile (Q1), second quartile (Q2) and third quartile (Q3) splits the data in 25%-75%, 50%-50% and 75%-25%, respectively. It can be noted that the distribution of the values of IVO for the 1% best configurations is skewed to the right. On the other hand, the values of SA for the 1% best configurations are concentrated in only a part of the range of values of the parameter. In order to gain insight on the values of both parameters that are used in the best configurations, we analyze the histograms of the parameters IVO and SA.

Figure 1 shows the histogram of the parameter IVO for the 1% best configurations with ϕ = 0.80. As it was previously stated, the values of IVO are skewed to the right and there is a high frequency of values concentrated around the value -15.0. Figure 2 shows the histogram of the parameter IVO for the 1% best configurations with ϕ = 0.85. The histogram shows a similar tendency than the previous one with the distribution skewed to the right. A high frequency of values are concentrated between -25.0 and -10.0.

Table 5. Minimum, quartiles, and maximum parameter values

		Min	Q1	Q2	Q3	Max
0.80	All	-50.0	-40.00	-30.0	-20.0	-10.0
IVO	Best 1%	-40.0	-26.75	-19.0	-14.0	-10.0
0.85	All	-50.0	-40.00	-30.0	-20.0	-10.0
IVO	Best 1%	-50.0	-29.75	-22.0	-17.0	-11.0
0.90	All	-50.0	-40.00	-30.0	-20.0	-10.0
IVO	Best 1%	-43.0	-26.00	-20.0	-16.0	-10.0
0.80	All	210.0	264.00	295.0	328.0	360.0
SA	Best 1%	325.0	326.00	326.5	328.0	330.0
0.85	All	210.0	264.00	295.0	328.0	360.0
SA	Best 1%	329.0	330.00	331.0	333.0	334.0
0.90	All	210.0	264.00	295.0	328.0	360.0
SA	Best 1%	333.0	334.25	335.0	336.0	338.0

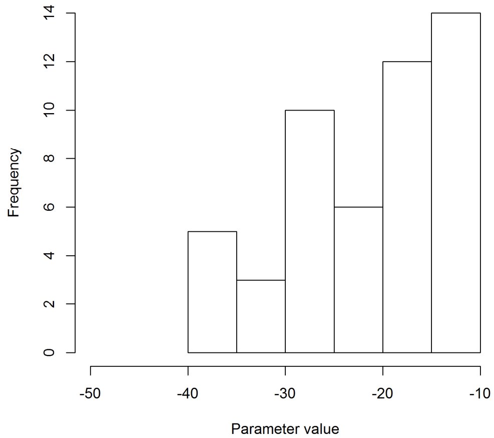

Figure 1. Histogram for the parameter IVO with ϕ = 0.80.

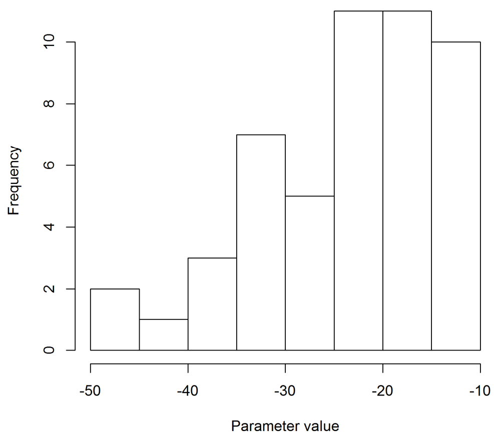

Figure 2. Histogram for the parameter IVO with ϕ = 0.85.

Figure 3 shows the histogram of the parameter IVO for the 1% best configurations with ϕ= 0.90. The histogram shows a similar tendency than the two previous ones. There is a high frequency of values concentrated between -30.0 and -10.0.

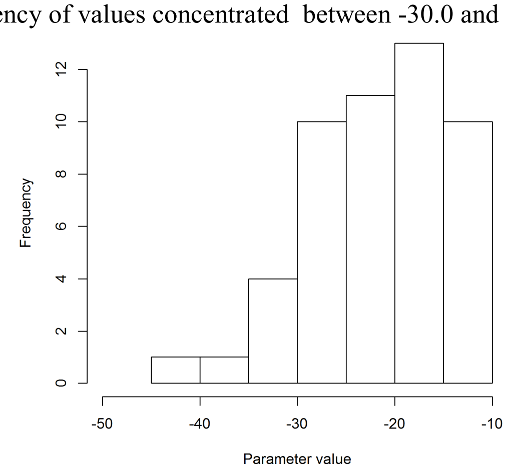

Figure 3. Histogram for the parameter IVO with ϕ = 0.90.

It is interesting to note that when equivalence ratio increases, the value of IVO that correspond a better solution of RCOV tends to decrease.

Figure 4, 5, and 6 show the histograms of the parameter SA for the 1% best configurations with ϕ = 0.80, 0.85 and 0.90. The values are concentrated in the range 325-330, 329-334, 333-338 for ϕ = 0.80, 0.85 and 0.90, respectively.

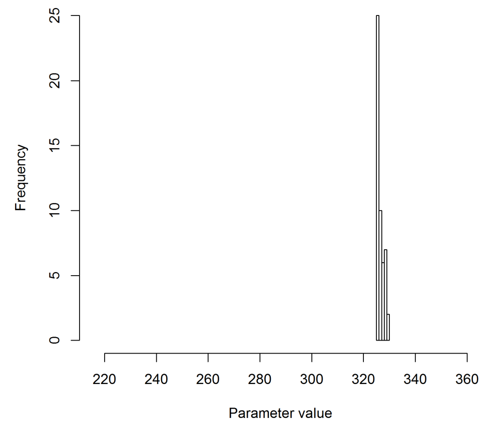

Figure 4. Histogram for the parameter SA with ϕ = 0.80.

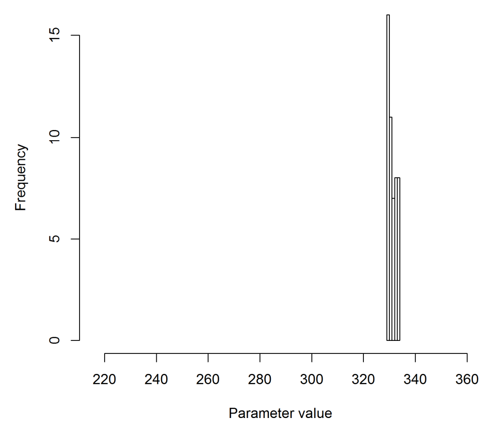

Figure 5. Histogram for the parameter SA with ϕ = 0.85.

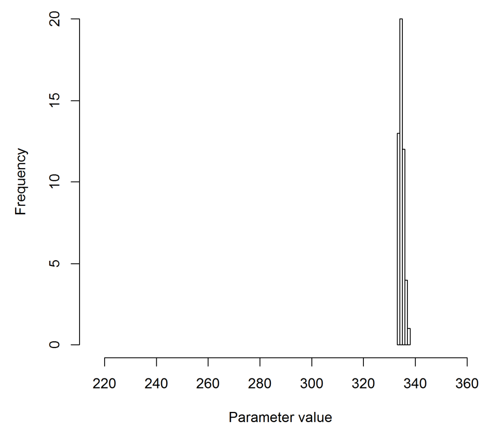

Figure 6. Histogram for the parameter SA with ϕ = 0.90.

Spark advance has a strong influence on combustion duration and is extremely sensitive to its changes. Therefore, it is expected that the values of SA that produce a better solution of RCOV tends to be in a small interval. However, it can be seen a small difference between the maximums, as equivalence ratio increases the spark advance angle increases slightly.

CONCLUSION

In this work an optimization analysis of an internal combustion spark ignition engine was performed. The objective was to find an optimal configuration in order to reduce the cyclic variability (in terms of the coefficient of variation) of efficiency time series (RCOV), considering five different parameters that have incidence in the cycle variability: the spark advance (SA), the intake valve opening angle (IVO), the intake valve closing angle (IVC), the exhaust valve opening angle (EVO), and the exhaust valve closing angle (EVC).

First we identify that IVO and SA have a greater incidence in obtaining small RCOV, than the rest of parameters.

The experimental evaluation shows that the better improvements are obtained for lower equivalence ratios, particularly for $\phi = 0.8$. The best result of RCOV reduces up to 42%. For higher values of equivalence ratio the improvements are lower, but significant, for $\phi = 0.85$ the RCOV reduces up to 10% and for $\phi = 0.9$ the RCOV reduces up to 13%.

The histograms for IVO show that when equivalence ratio increases, the value of IVO that corresponds to a better solution of RCOV tends to decrease. In the same manner, SA values for better results of RCOV, slightly increase when equivalence ratio increases. This shows that in future work is necessary to address a multi-objective analysis including these effects.

As a final comment, this work shows that it is possible to considerably reduce the cyclic variability, just by an adequate engine design.

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