

Combination of In-Cylinder Pressure Signal Analysis and CFD Simulation for Knock Detection Purposes

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ABSTRACT

A detailed analysis of knocking events can help improving engine performance and diagnosis strategies. The paper aim is a better understanding of the phenomena involved in knocking combustions through the combination of CFD and signals analysis tools.

CFD simulations have been used in order to reproduce knock effect on the in-cylinder pressure trace. In fact, the in-cylinder pressure signal holds information about waves propagation and heat losses: for the sake of the diagnosis it is important to relate knock severity to knock indexes values. For this purpose, a CFD model has been implemented, able to predict the combustion evolution with respect to Spark Advance, from non-knocking up to heavy knocking conditions.

The CFD model validation phase is crucial for a correct representation of both regular and knocking combustions: the operation has been carried out by means of an accurate statistical analysis of experimental in-cylinder pressure data.

The simulation results allow relating the combustion characteristics to their effect on the in-cylinder pressure signal. It is then possible to highlight critical issues regarding typical knock detection methodologies, while disclosing novel approaches. One of the main results is the validation of knock detection strategies based on the low-frequency content of the pressure signal. These strategies can be used together with standard high-frequency based techniques in order to improve the detection robustness.

INTRODUCTION

Knocking combustions limit Spark Ignition engines efficiency, preventing the achievement of the optimal combustion phase: the knock phenomenon is intrinsically cause of inefficiencies, and, in addition, it forces to keep the combustion with a safe – but often inefficient – retarded phase. A precise cycle-by-cycle knock monitoring strategy is required to improve the performance without increasing the risk of causing damages to the engine. Knock detection can be performed based on several types of signal [1-8]: in-cylinder pressure, ion current, accelerometer, light radiation, acoustic emissions. The first one is often used as a reference for the calibration of detection strategies based on other sensors. The diagnosis is usually carried out by means of in-cylinder pressure signals high frequency components [9-13]. The typical high-frequency knock signature is related to the combustion chamber resonant frequencies, excited by the rapid pressure rise. The pressure wave amplitude is used to define knock indexes: the drawback of such indexes is that they are based on observations referring to a narrow spatial domain. While for standard combustions the value measured at the sensor location can be considered as representative of the whole chamber pressure value, as knock happens, huge non-homogeneities take place preventing the extension of the local data to the global domain. The pressure wave arises where knock happens, but then it evolves throughout a non-homogeneous medium, interacting with the complex chamber boundary. As a consequence, the measured pressure wave amplitude has a local meaning and cannot be considered as the maximum

amplitude of the propagating wave. The consideration is confirmed by experimental evidence, as reported in [14]. All the indexes directly based on the amplitude of the pressure oscillations suffer from the measured amplitude sensitivity to the relative position between sensor and knock location. Therefore, these indexes have statistical consistency, but they are not deterministic in sensing knock intensity. Moreover, their values depend on other factors, such as operating conditions [15-16] and sensor type [14]. Possible solutions to the problem of relating the index values to an objective assessment of knock intensity have been proposed [16-17], but they do not compensate for the intrinsic lack of information related with the local nature of the pressure oscillations under knocking conditions.

An alternative approach is to look at the combustion characteristics in terms of burning rate: the Rate of Heat Release (ROHR) is usually evaluated based on the in-cylinder pressure signal, applying the first law of thermodynamics [18].

$$ROHR = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta_c} + \frac{1}{\gamma-1} V \frac{dp}{d\theta_c} \quad \text{Eq. 1}$$

The evaluation of the burning rate, however, becomes unreliable as the pressure wave propagates in the chamber: in this case, as already explained, the measured pressure value P does not represent the condition of the entire combustion chamber volume V . In [19-20] indexes based on the evaluation of ROHR are defined: the first methodology requires high sampling rate and sensor bandwidth, the second can hardly be applied in Real-Time. In [21] a novel approach has been proposed, based on the thermal losses increase during knock. The methodology, based on the analysis of the Cumulated Heat Release (CHR), seems to be promising, due to the direct relationship between the index value and the cause of knock-related engine damages [22-23]: a deep insight in the combustion and heat exchange processes by means of CFD simulations are of help in verifying the index reliability and improving its definition. CFD analysis can be used in order to simulate knock effects on the in-cylinder pressure trace. If the CFD model is able to predict the combustion evolution with respect to Spark Advance, from non-knocking up to heavy knocking conditions, it can be used to relate the actual combustion mode information to the measured signal characteristics. A deep insight on knock effects can be achieved by forcing different knock intensity levels, thus allowing a parametrical analysis of the phenomenon.

KNOCK EFFECT ON HEAT LOSSES

Heat transfer affects engine performance, efficiency and emissions. The higher the heat transfer, the lower the temperature and pressure inside the cylinder: lower pressure in the expansion stroke implies a reduction of the work transferred to the piston, i.e., a lower Indicated Mean Effective Pressure (IMEP). An increase in spark advance allows more fuel to be burned before Top Dead

Centre (TDC), leading to higher peak pressure and temperature in the chamber. In the range of spark timing with complete combustion and no knocking condition, the heat transferred to the walls can rise considerably, as SA is increased.

The heat flux to the walls under normal condition in Spark Ignition (SI) engines is dominated by convection. Heywood [18] estimates the ratio of radiative to convective heat flux in peak temperature condition to be in the range of 5%. Many experimental works demonstrate that the knock phenomenon can be associated to a strong peak heat flux increase. Three processes interact to potentially enhance the knocking heat transfer rate:

- Increase in convective velocities caused by the rapid expansion of the mixture mass involved in auto-ignition.
- Strong pressure transients in presence of reactions.
- Transient density variations which locally influence the heat release near the wall.

For heavy knock condition Lu et Al [24] showed that for pressure oscillations above 0.5 MPa the peak heat flux increases almost linearly with pressure amplitude. Other works ([25],[26],[27]) show that during knocking cycles high Peak Heat Flux (PHF) values can be measured, but the correlation coefficients between peak heat values and knock intensity indexes are low ($R = 0.26\text{--}0.47$) for light knock, higher ($R = 0.53\text{--}0.76$) for heavy knock, and strongly depend on the sensor position ([27]). Grandin et Al. [28], [29], [30], closely relate the peak heat flux to the increase of charge motion as a result of auto-ignition. They showed that for the case of heaviest knocking condition the peak heat flux is 280% of that in non-knocking conditions. Ezekoye et Al [25] made both a computational and analytical investigation of the heat transfer processes at the flame-wall interface during knock event. The analysis of the peak heat flux was conducted in three different locations: the burned gas zone, the zone of flame reactions and the unburned zone. They concluded that the pressure transients have a moderate effect on heat transfer processes in wall locations where heat-release effects are negligible (unburned zone) and modify the heat transfer rates in locations in which the flame has quenched (burned gas zone). The most important effect responsible for the magnitude of heat flux values appears to be the absolute pressure. In other words, knock indexes based on in-cylinder pressure amplitude are well related to peak heat flux. The results for different locations and timing clearly show that the peak heat flux in knocking condition is strongly increased all over the chamber. This final consideration allows extending the results on the increase of local peak heat flux all over the engine chamber.

The knock onset is generally located few degrees after TDC: in this condition the loss of efficiency due to knock occurrence is maximum, since the mean pressure decrease caused by heat transfer propagates throughout the whole expansion stroke. However, this aspect

have to be taken into account to assess the global effect of knock on performance: the balance between the gain of a volumetric combustion at TDC because of auto-ignition (thermal efficiency) and the energy lost at the wall usually leads to a reduction of the overall combustion efficiency (and then power output) for heavy knocking conditions. It has to be reminded that, knock being a stochastic phenomenon, a single knocking cycle is not representative of the engine running condition: a given SA could lead to some low-efficiency cycles due to knock, but the other cycles could benefit of an increased combustion speed, leading to a better combustion phasing (i.e., higher IMEP).

EXPERIMENTAL SETUP

To highlight the effect of SA on the combustion development, tests have been carried out on the test bench in steady state conditions, on a FIAT Fire 1.2 litres 4 cylinders PFI gasoline engine. Engine speed was kept constant by means of the eddy-current brake controller; a brushless motor driving the throttle was feedback controlled, based on the intake manifold pressure, to set engine load. Intake manifold pressure and temperature, Air to Fuel Ratio (AFR), coolant and oil temperatures have been maintained constant during the tests. Each test, counting 1000 engine cycles, has been executed holding a different SA value, up to and over audible knock. In-cylinder pressure was measured by means of Kistler 6117BCD15 measuring spark plugs and 5064B21 charge amplifiers. Angular position tracking has been carried out by means of a Sensor Instrument FIA-F optical sensor, coupled to a crankshaft-mounted measurement disk with 120 markers per revolution. The in-cylinder pressure signals have been sampled @ 100kHz, while the angular reference signal (encoder) has been detected by means of a timer-counter digital channel, with a 20 MHz clock. All the signals have been sampled using a Wavebook 516 Data Acquisition System.

The in-cylinder pressure signals have been low-pass filtered by means of an anti-aliasing analog filter set @ 30 kHz: the filter delay has been compensated in order to avoid referencing errors. A 3 kHz zero-delay low-pass 4th order Butterworth digital filter has been used for IMEP and net Cumulative Heat Release (CH_{NET}) calculations, in order to eliminate the combustion chamber resonance effect. This allows isolating the mean combustion chamber conditions contribution to CH_{NET} and indicated work trends, while rejecting the components related to local pressure oscillations. A 5 kHz zero-delay high-pass 4th order Butterworth digital filter has been applied to the pressure signal for the evaluation of knock indexes.

DESCRIPTION OF CFD NUMERICAL MODEL

The CFD simulation of the combustion process is nowadays a standard tool for the optimization of engine performance and emissions. The complexity of the

phenomena involved requires advanced numerical models to be developed and validated against experimental data in actual engine configurations.

The combustion simulations have been carried out by using the KIVA-3 [31] code customized by the University of Bologna [32-33]. The calculations are performed with the use of standard k-ε turbulence model and standard wall function. A Quasi Second Order Upwind Scheme is chosen for the derivative discretisation. The code runs an improved rezoning algorithm in order to accomplish with complex geometry combustion chambers.

The law of the wall is modified according to Han and Reitz [34] in order to improve the prediction of heat exchange with the wall: this energy loss plays a key role both in the overall energy balance efficiency and in the prediction of local evolution of mixture temperature near the wall.

The ignition process strongly influences the combustion development and it is very hard to simulate and calibrate: any inaccuracy introduced at this early stage will be amplified in further combustion evolution. The flame kernel modelling is quite complex because it involves the simultaneous treatment of physics of spark discharge and plasma formation, chemical kinetics, flow and turbulence interactions. Many of these processes occur at characteristic length scales which are much smaller than usual cell size adopted in CFD calculations. The ignition model used is based on a Lagrangian approach developed by Bianchi and Falfari [31]. A lump model for the electrical circuit of the spark plug is used to compute break down and glow energy. At the end of the very first plasma expansion, a spherical kernel is deposited inside the gas flow at spark plug location. The sphere surface is discretized by a set of triangular elements which move radially. Turbulence effects and thermodynamic properties of the air-fuel mixture are accounted for. A comprehensive validation versus turbulence, air index and mean flow convection has been carried out in [31]. The Lagrangian approach has proved to strongly reduce the grid sensitivity: this is particularly important for engine simulations in which complex geometry and grid movement do not often allow an easy control of the cell dimension.

The combustion model is based on the ECFM (Extended Coherent Flamelet Model, Duclos [35]) and is implemented in the form as proposed by Colins et Al. [36].

The transport equation of the mean flame front surface density is in the form:

$$\frac{\partial \bar{\Sigma}}{\partial t} + \frac{\partial \bar{u}_i \bar{\Sigma}}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\left(\frac{\mu_t}{Sc_t} + \frac{\mu}{Sc} \right) \cdot \frac{\partial \bar{\Sigma}}{\partial x_i} \right) + (P_1 + P_2 + P_3) \cdot \bar{\Sigma} - D$$

Eq. 2

where the different source and sink terms in the transport equations can be described as follows:

- $P_1 = \alpha K_t$ represents the flame surface production by turbulent stretch, where K_t is the turbulent flame stretch and α is a model constant;

- $P_2 = \frac{2}{3} \frac{\partial \bar{u}_i}{\partial x_i}$ is the production by the mean flow stretch;
- $P_2 = \frac{2}{3} \bar{S}_l \frac{1-\bar{c}}{\bar{c}}$ accounts for the effects of thermal expansion and curvature;
- $D = \beta \bar{S}_l \frac{\bar{\Sigma}^2}{1-\bar{c}}$ is the destruction, where β is a model constant.
- $\bar{c}$ is the mean progress variable and $\bar{S}_l$ the mean laminar flame speed evaluated based on mean local equivalence ratio and physical properties in the unburned mixture. Model constant β is kept fixed and equal to 1, while α is the only parameter which has been modified in the calibration of the combustion simulations.

The transport equation of fresh gas enthalpy is written in the form:

$$\frac{\partial \bar{\rho} h_u}{\partial t} + \frac{\partial \bar{\rho} \bar{u}_i h_u}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\left(\frac{\mu_t}{Sc_t} + \frac{\mu}{Sc} \right) \cdot \frac{\partial h_u}{\partial x_i} \right) + \frac{\bar{p}}{\rho_0} \cdot \frac{\partial \bar{p}}{\partial t} + \bar{p} \cdot \frac{\varepsilon}{k} \quad \text{Eq. 3}$$

Thus unburned gas temperature is evaluated at every time step by the thermodynamic relation:

$$dh_u = c_p(T_u) dT_u \quad \text{Eq. 4}$$

and c_p dependence from temperature is evaluated by the Janaf tables embedded in the KIVA3 code.

A one step chemistry is solved on the reaction front while the post-flame chemistry is based on Mentjies and Morgan equilibrium schemes [37]. A conditional averaging of flow properties is carried out for evaluating the thermodynamic unknown properties of fresh and burnt gases. A specific library of the fuel is used to consider its actual composition and properties (i.e., atoms composition, heat of formation, laminar flame speed correlation, etc.)

The ECFM allows a more accurate estimation of unburned thermodynamic conditions: this improves the determination of laminar flame speed used in normal combustion on the flame front but also supports the implementation of both empirical or reduced kinetics auto-ignition models for knock detection.

The choice of the knock model to implement is influenced by the needs of the present research: the aim is to provide useful information on the relationship between a knock event and the local pressure signal evolution in a given location in the chamber. The calculation time associated to realistic models based on the resolution of several chemical equilibria is not justified for this purpose. The reduced chemical kinetic mechanism based on the Shell model proposed by Eckert et Al [38] helps simplifying the complexity of the chemistry and is an interesting tool for the prediction of knock event without tuning parameters.

The knock model AnB as proposed by Lafossas [39] is an empirical model based on the imposition of the auto-ignition delay θ :

$$\theta = A \left(\frac{IO}{100} \right)^{3.4017} P^{-n} e^{\frac{B}{T}} \quad \text{Eq. 5}$$

where IO is the fuel octane number, P the pressure expressed in bar, and T the temperature expressed in Kelvin. The thermodynamic proprieties refer to the unburned gas. The auto-ignition delay is a parameter which is relatively easy to obtain for a given mixture of fuel and air: even if not available it is possible to change the three parameters A, n and B, in order to correctly reproduce the knock onset in engine configurations. The availability of these free variables for the calibration of knock event has taken to implement this model in KIVA in the same form as presented by Lafossas [39].

The model uses a two step chemistry for the simulation of auto-ignition: in the first step a precursor of the auto-ignition is calculated and then, when it reaches a critical concentration, the knock combustion is forced.

The fuel consumption rate due to auto-ignition is calculated as follows:

$$\frac{dY_{Fu}}{dt} = Y_{Fu} A_k \quad \text{with} \quad A_k = 10^4 e^{\frac{3500}{T_g}} \quad \text{Eq. 6}$$

where Y_{Fu} is the fuel mass fraction and T_g is the local temperature of burned gas expressed in Kelvin.

The chemical kinetics during auto-ignition delay are not linear: the model proposed by Lafossas doesn't use a simple time integration for the auto-ignition delay time, but calculates the appearance of the precursor with an exponential form:

$$\frac{dY_p}{dt} = Y_{Fu} F(\theta) \quad \text{Eq. 7}$$

where

$$F(\theta) = \sqrt{\frac{\delta^2 \theta^2 + 4(1-\delta\theta) \frac{Y_p}{Y_{TFu}}}{\theta}} \quad \text{Eq. 8}$$

δ is a constant and Y_{TFu} is the fuel tracer mass fraction.

The criteria of knocking combustion is satisfied when the precursor concentration locally reaches the fuel tracer mass fraction value.

Normal combustion development and knock events are closely related to local mixture composition. The AnB model allows correcting the definition of the auto-ignition delay based on the information of air to fuel ratio and EGR mass fraction. The influence of fuel composition is globally included in the definition of the octane number, letting to the tuning parameters A, n and B the role of fitting the real experimental conditions.

The effect of air to fuel ratio on knocking tendency is accounted for by correcting the IO number defined in eq. 5

$$IO_{ef} = IO \cdot e^{(0.1 \cdot (\Phi - 1)^2)} \quad \text{Eq. 9}$$

where Φ represents fuel/air ratio.

The residual gas content is accounted for by lowering the pressure term in eq. 5 by a factor of $1/(1+Y_r)$, where Y_r is the egr mass fraction.

The coupling between the auto-ignition and the ECFM combustion still represents an open issue. In the present implementation the main indication of Lafossas has been followed:

$$\Sigma = |\nabla \bar{c}| \quad \text{Eq. 10}$$

However in all the performed engine simulations, this imposition seems to have a small effect on fuel