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Modeling of in-cylinder pressure oscillations under knocking conditions: A general approach based on the damped wave equation

Alessandro di Gaeta *, Veniero Giglio, Giuseppe Police, Natale Rispoli

Istituto Motori, National Research Council, 80125 Napoli, Italy

HIGHLIGHTS

- ▶ We model in-cylinder pressure oscillations due to knocking combustion.
- ▶ A novel and general approach is used in modeling pressure oscillations.
- ▶ The model derives from explicit integration of wave equation with a dissipation term.
- ▶ A generic spatial distribution of end-gas at knock onset can be assumed by the model.
- ▶ Time and frequency features of experimental pressure oscillations are well reproduced.

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ABSTRACT

The modeling of the in-cylinder pressure oscillations under knocking conditions is tackled in this work. High frequency pressure oscillations are modeled by the explicit integration of a partial differential wave equation augmented with a time-dependent dissipation term. The general solution of such equation is determined by the Fourier method of separation of variables whereas the integration constants are obtained from the boundary and initial conditions. The integration space is a cylindrical acoustic cavity whose volume is that of the combustion chamber evaluated at the knock onset. The domain of integration is assumed to be formed by a finite set of small volumes having the shape of annulus sectors. This approach involves that knock region can assume more realistic shape of the kernels where abnormal combustion initiates. The initial conditions are evaluated by means of a two-zone thermodynamic model applied to low-pass filtered experimental pressure cycles. The damping coefficient and the knock region are model parameters to be assigned or identified experimentally by means of a proper least-squares optimization process. Experimental data obtained on a direct injection spark ignition engine, operating under knocking conditions at different speeds, have been used to validate the model both in time and frequency domains.

1. Introduction

Knock is the word used in automotive field to name a sharp metallic noise appearing in spark ignition engines as a consequence of an abnormal combustion. In normal conditions the whole air/fuel charge is progressively burned by the propagation through the cylinder of the flame front initiated by the spark, and energy is released likewise. According to a widely accepted theory [9], if the induction time for the auto-ignition of the unburned gases ahead of the flame front is lower than the time required for the whole charge to burn by flame propagation, a local rapid combustion occurs with a sudden energy release triggering the knock phenomenon. Such a theory well explains knock appearing at low engine speed with medium–high loads. Again according to this theory, high speed operation should be knock free since it was found that the increase of engine speed leads to an increase of turbulence intensity, enhancing flame propagation, and to deceleration effects on pre-flame reactions. Actually this is not so, and other theories are proposed [16] to explain the more destructive knock occurring at high engine speed, over 3000 rpm.

A deepening of theories on this abnormal combustion is out of the scope of this work. However, whatever is the genesis of the mechanisms leading to knock, this is triggered by rapid combustion and energy release unevenly distributed in the combustion chamber. The consequent time and space pressure gradients excite the acoustic resonance modes of the chamber, resulting in damped pressure oscillation around the mean cylinder pressure, and in engine block vibration [1].

* Corresponding author.
E-mail addresses: a.digaeta@im.cnr.it (A. di Gaeta), v.giglio@im.cnr.it (V. Giglio), g.police@im.cnr.it (G. Police), n.rispoli@im.cnr.it (N. Rispoli).

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The acoustic wave theory was used for the first time in [6] to describe steady pressure oscillations inside a cylindrical fixed cavity. The author basically set up an analytical method to identify the resonant modes inside the engine cylinder, assumed as a fixed volume, deriving the general solution of the wave equation with a two dimensional approach. This was justified by the following observations: knock appears when piston is near Top Dead Centre (TDC) position, where its velocity is very low (and accordingly volume variation); at gas temperatures during combustion sound speed has an order of magnitude of 1000 m/s, and since at TDC the axial dimension is by far lower than the radial one, axial steady waves, propagating much faster than radial ones, can be neglected with respect to steady radial waves. With these considerations, near TDC the integration space can be assumed as cylindrical with top and bottom flat ends, and vibrations along the cylinder axis are neglected. As the piston moves downward, resonant frequencies change because sound speed decreases and waves propagating along the cylinder axis direction are no more negligible.

To tackle this problem a 3D FEM (Finite Element Method) approach was followed by several researchers to describe the resonance phenomena occurring during the piston stroke. In this regard, we can mention the work [10] where the influence of cylinder head shape was studied, and the work [20] where the effect of piston position on the resonances of two different cylinder heads featured by two and four valves, respectively, was investigated. In the latter work the authors found that, for a given combustion chamber, the various vibration modes exhibit different dependencies of frequency versus piston position (or crank angle). However the frequency profiles of the main vibration modes of the two considered combustion chambers were overall similar, suggesting that cylinder head shape is not a factor of first importance on resonance features. The research work of [20] was extended in [4] where, for the first time, transient simulation was performed to describe the explosive features of knock.

These investigations on acoustics phenomena, albeit very interesting, are not usable in SI engine design and optimization oriented to knock prevention. Actually, simulation of combustion processes leading to knock is a challenging task, requiring 3D CFD codes with detailed and validated models for pre-flame chemistry and turbulent flame propagation. To calculate knock related pressure oscillations in combustion chamber, very fine discretization of the calculation domain is needed that can lead to prohibitive computational loads (some examples of this approach are given in [1,5]).

Detailed chemistry can be included in 0-D simulation models, allowing to estimate the instant when knock occurs and the corresponding amount of unburned mixture ready to burn. However, the only presence of detailed chemistry does not enable 0-D models to reproduce the typical deadening pressure oscillations found in experiments with knock phenomena. In this regard, in [2] an analytical formulation was proposed based on experimental pressure traces in knocking conditions, modifying the traditional approach [6] to describe such deadening behavior.

In the present work a different approach is proposed. Pressure oscillation are simulated thanks to the explicit integration of a Partial Differential Wave Equation (PDWE) augmented with a damping term [21]. A general solution of such equation is found applying the Fourier method of separation of variables. Integration is performed in a cylindrical space domain corresponding to the acoustic volume of the combustion chamber fixed at knock onset. The initial conditions required for the integration have been derived from a linear relation between the time derivative of pressure and expansion velocity of burned gas. The damping factor in the PDWE together with the size and distribution of unburned gas volume, are the model parameters to be identified experimentally.

A number of experimental in-cylinder knocking pressure cycles were low pass filtered, removing oscillating high frequency components, and processed by a two-zone thermodynamic model estimating unburned and burned gases mean temperatures, crank angle of knock onset, unburned gas volume at knock onset and expansion velocity of burned gas. These data allowed the calculation of the initial value for the time derivative of oscillating pressure. A least-squares parameter identification process, using as reference the high frequency pressure oscillations and based on genetic algorithms, was set up to identify the damping factor and, assuming a radial and polar discretization of the cylinder volume into 48×20 annulus segments with basis on piston top surface, a two dimensional distribution of unburned mixture at knock onset. The model quite well reproduces high frequency pressure oscillations, and the resulting unburned volume distribution at knock onset is suggestive of possible locations of auto-ignitable kernels.

The paper is outlined as follows. The mathematical model of the pressure oscillations is described in Section 2. A numerical analysis of the spatial distribution of the vibration modes in terms of amplitude and frequency is presented in Section 2. A physics-based formulation of the initial condition is given in Section 4. The experimental setup is described in Section 5. Model validation results are finally presented in Section 6.

2. Derivation of the pressure oscillation model

The three-dimensional wave motion in a gas is governed by a set of nonlinear Partial Differential Equations (PDEs) descending from the mass, energy and momentum balance equations. In this work we employ the *method of small perturbations* to linearize such a set of PDEs since the following hypothesis can be assumed valid in our problem, that is:

- the process is isentropic (absence of internal losses) and without heat transfer;
- the pressure disturbance is very small;
- the velocities and their derivatives are small compared to the speed of sound in the undisturbed gas.

Under these assumptions any *perturbation* or *disturbance*, denoted with the symbols u, v, w, p, ρ and a, is considered linearly superimposed to the pertinent flow property, that is:

$$\begin{aligned}
&\tilde{u} = \bar{u} + u, \quad (x - \text{direction gas velocity})\\
&\tilde{v} = \bar{v} + v, \quad (y - \text{direction gas velocity})\\
&\tilde{w} = \bar{w} + w, \quad (z - \text{direction gas velocity})\\
&\tilde{p} = \bar{p} + p, \quad (\text{pressure})\\
&\tilde{\rho} = \bar{\rho} + \rho, \quad (\text{density})\\
&\tilde{a} = \bar{a} + a, \quad (\text{speed of sound}),
\end{aligned} \tag{1}$$

where *tilde* and *bar* symbols denote the total and average (*undisturbed*) values of each flow property, respectively. The following set of four linearized PDEs are then considered:

$$\bar{\rho}\bar{a}^2\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right) + \frac{\partial p}{\partial t} = 0, \tag{2a}$$

$$\bar{\rho}\frac{\partial u}{\partial t} + \frac{\partial p}{\partial x} = 0, \tag{2b}$$

$$\bar{\rho}\frac{\partial v}{\partial t} + \frac{\partial p}{\partial y} = 0, \tag{2c}$$

$$\bar{\rho}\frac{\partial w}{\partial t} + \frac{\partial p}{\partial z} = 0, \tag{2d}$$

where according to the previous assumptions, $\bar{u}$, $\bar{v}$, $\bar{w}$ are assumed to be zero, u, v and w are very small, and p, ρ and a are much smaller than the corresponding undisturbed values $\bar{p}$, $\bar{\rho}$ and $\bar{a}$. The reader is