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Numerical analysis of in-cylinder tumble flow structures – parametric 0D model development

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Abstract

Both in the automotive and in the motorcycle fields the requirement of step-by-step improvements for optimizing the engine cycle is still present. In particular the focus of the optimization process is to reduce the raw emissions and at the same time to not penalize the engine performance. In this research field the engine modeling is of great importance because the application field of the experimental measurements is very narrow, time-consuming and expensive. Hence the modeling technique is a wide used and a wide recognized instrument for helping in the design process. Another important function of the modeling is to provide the engine designers with the most important guidelines. The main focus is to fast provide designers with some fundamentals during the first designing stage which, if not the conclusive, is close to the final project.
The present paper deals with the development of a theoretic-interpretative 0D model which could highlight the most significant parameters in the engine design process and in particular in the determination of:

- The tumble velocity at IVC and its residual value at TDC;
- The squish velocity at TDC;
- Their mutual interaction.

These parameters are well recognized to be especially meaningful because they determine, at different times of the combustion process, the combustion velocity. The faster the combustion velocity, the lower the engine cycle-by-cycle variability.

Keywords: Small PFI Engine; Tumble Motion; Engine Geometrical Parameters; Engine Design; 0D Parametric Model;

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1. Introduction

The emission reduction process started involving only the passenger cars: it started with the Euro 1 standards in 1992 and it will move into the Euro 6 standards in 2014. For motorcycles and mopeds the European Standards were less stringent than for passenger cars in the past: recently the European Commission has decided to introduce Euro 5 Standards for both the motorcycles and the mopeds in 2020, with the same emission thresholds as passenger cars.

Starting from these considerations the necessity of further engine improvement for motorcycles and passenger cars is obvious. The modeling technique is a wide used and wide recognized instrument for helping in the design process.

The main focus of the present paper concerns the development of a theoretic-interpretative 0D model which could highlight the most significant parameters in the engine design process and in particular in the determination of:

1. The tumble velocity at IVC and its residual value at TDC;
2. The squish velocity at TDC;
3. Their mutual interaction.

They are especially meaningful because they determine, at different times of the combustion process, the combustion velocity. The faster the combustion velocity, the greater the indicated efficiency, the low the raw emissions and the fuel consumption. In particular the tumble motion is of great importance in determining the final level of turbulence around the ignition time in the combustion chamber and so it greatly affects the start of the combustion process. To get faster combustion process in SI engines, the primary strategies are:

1. To obtain a high turbulence level in the combustion chamber close to the ignition timing. This is done by the generation of a structured and coherent tumble motion during the suction phase. More details on it could be find in [1, 3]. In order to have a structured and coherent tumble vortex it is necessary to adopt the intake ducts oriented toward the exhaust side of the intake valve. The parameter able to sum up and numerically quantify these characteristics is the tumble ratio.
2. To promote a right oriented and proper time-tuned squish motion around TDC. The squish motion acts around TDC feeding the flame with the fresh mixture coming from the engine’s periphery. Its importance is related to both its directionality and its timing. More details on it could be find in [1].

Between the two main in-cylinder motions, the first to promote is the tumble motion because it directly determines the final level of turbulence close to TDC. The importance of the squish motion is limited to the time close to TDC, when the combustion process is already activated.

The present paper main focus is the development of a theoretic-interpretative model able to highlight the most significant geometric and fluid-dynamic parameters in determining the tumble velocity, the squish velocity and their possible interaction.

2. Literature summary on the tumble motion

The literature review collects different experience and evidence on the effectiveness of the tumble and the squish flows. In [1] the authors carried out an analysis of a small 3-valves PFI engine characterized by a small bore and a high stroke-to-bore ratio. They compared three geometries differing only in the squish area distribution and values.

They carried out a deep insight of the in-cylinder flow motion for highlighting the possible mutual influence between squish and tumble. Moreover it was highlighted that the squish motion interacts strongly with the tumble motion.

In [2] the authors varied some engine parameters (the intake duct angle, the intake valve lift, the piston shape, and the compression ratio) for extracting a ‘value scale’ based on their influence on the tumble motion .

Authors in [3] focused on the effect of the joint use of the tumble and the squish flows in promoting the in-cylinder turbulence for fast combustion purposes. The squish velocity to the tumble velocity ratio at TDC was used to evaluate the effectiveness of the joint use of the tumble and the squish flow.

In [4] authors used the numerical simulation to assess the influence of some intake duct geometrical parameters on the tumble motion generation during both the intake and the compression strokes to highlight the turbulence production process. In particular the authors tried to understand the importance of the geometrical duct parameters by means the use of the tumble torque parameter. This analysis was deepened in [5]: the tumble torque parameter was used for comparing in details the different results obtained in changing the intake duct parameters. Moreover a probability distribution function (pdf) of the turbulence intensity close to the spark plug was introduced.

The literature on tumble motion is quite wide: here only some references are reported. In particular the methodological approach on the tumble motion analysis could be based on the CFD simulations [6-15] or on the experimental results [16, 17]. As far as the squish motion is concerned, the squish flow control is a key technology for improving the knock limit in the spark ignition engines [18, 17]. A more detailed references analysis could be find in [2].

3. Zero-dimensional (0D) parametric model development

The current paper deals with the development of the 0D parametric model for predicting which engine parameter affects the tumble velocity value at IVC and at TDC, the squish velocity and the ratio of the squish velocity to the residual tumble velocity at TDC. Moreover the 0D model summarizes the effect of the ratio of the squish velocity to the residual tumble velocity at TDC depending on the engine class, i.e. the C/D ratio value.

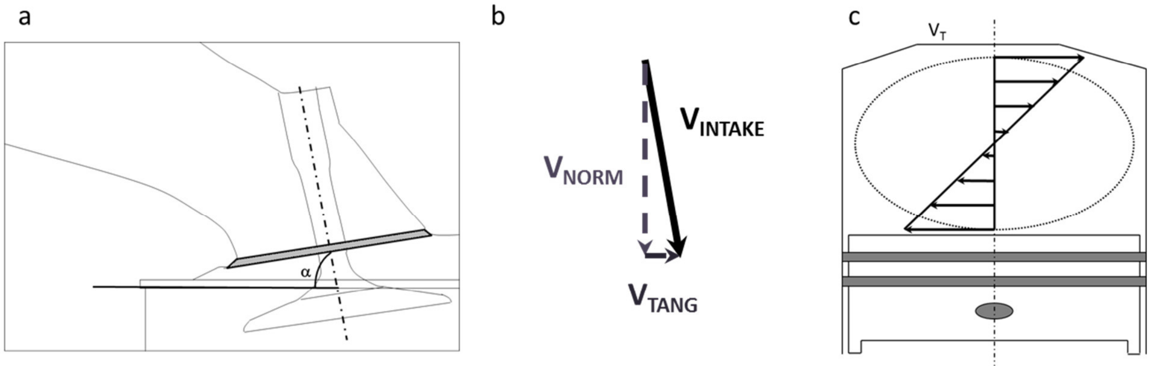

Fig. 1. (a) : Scheme of the engine chamber and of the intake manifold; (b) Intake velocity vector - Vectorial sum; (c) Tumble velocity vector distribution.

3.1. The tumble velocity at IVC

Under the hypothesis of steady flow conditions through the intake valve at IVC the expression of the tumble velocity V_T is derived. Applying the proportionality law of the volumetric flow rate through the valve intake area (in red in Fig. 1a):

$$V_P \cdot A_P = V_{INTAKE} \cdot A_V \quad (1)$$

where V_P is the mean piston speed, A_P is the piston area, V_{INTAKE} is the inlet velocity of the flux trough the inlet valve, A_V is the whole intake valve area, i.e. considering the intake valve number too. The intake velocity vector V_{INTAKE}, directed along the intake valve axis, is the vectorial sum of the axial component V_{AXIAL} and the tangent component V_{TANG}, as shown in Fig. 1b. In particular the axial component V_{AXIAL} is the term the inlet mass flow rate is due to, while the tangent component V_{TANG} is the dissipation term, i.e. the term generating the vortex inside the cylinder. So the last term is coincident with the tumble velocity V_T, which is the velocity tangent to the ideal single main tumble vortex having dimension comparable to the engine stroke: its distribution is depicted in Fig. 1c. The angle between the intake velocity and the tangential velocity is the angle α (Fig. 1a), so it is possible to write Eq. (1) as follows:

$$V_P \cdot A_P = \frac{V_T}{\cos(\alpha)} \cdot A_V \tag{2}$$

Some considerations can be derived from Eq. (2): the less the angle α, the less the axial component, the more the tangent component. Moreover the more the angle α, the more the engine volumetric efficiency. These considerations can be summed up introducing the parameter K_P proportional to the angle *α*. The Eq. (2) could be re-arranged as below reported, considering the tumble velocity V_T at IVC (V_{T_IVC}):

$$V_P \cdot A_P \propto \frac{V_{T\_IVC}}{K_P} \cdot A_V \tag{3}$$

The range of variation of the parameter K_P (Fig. 2) is linked to the admitted variation of the intake duct angle α (Fig. 1a). In particular the range of the angle α was chosen between 0° and 90°, whereas the typical automotive values range between 25° and 70°. For α=90°, the inlet velocity has only the axial component, so there is no theoretical contribution to the formation of the tumble structure and the inlet mass flow rate is maximized. For the chosen range of the angle α, the corresponding range for the parameter K_P becomes between 0.22 and 0.72, as visible in Fig. 2. The less the value of the angle α, the more the value of the parameter K_P, the more the tumble velocity value V_{T_IVC}.

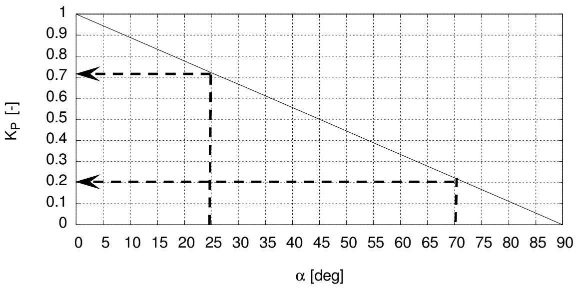

Fig. 2. Trend of K_P versus intake duct angle α.

At this point it is necessary to consider that only a fraction of the geometrical intake valve area is devoted to the generation of the tumble vortex: this concept is summed up into the parameter K_T. The more the parameter K_T, the more the engine volumetric efficiency. K_T represents the percentage intake valve area which contributes to the inlet mass flow rate. The final expression of Eq. (1), highlighting the value of V_{T_IVC}, becomes:

$$V_{T\_IVC} \propto \frac{V_P \cdot A_P}{A_V \cdot K_T} \cdot K_P \propto \frac{V_P \cdot D^2}{A_V \cdot K_T} \cdot K_P \tag{4}$$

where the piston area A_P was substituted by the squared bore D. Eq. (4) represents the final expression of the tumble velocity at IVC. It let to assess that the raise of the parameters V_P, K_P and the bore D increases the tumble velocity at IVC. On the other hand the raise of the parameters K_T and A_V decreases the tumble velocity at IVC. The intake valve area is writable as:

$$A_V^2 \propto d_V^2 \propto K_V^2 \cdot D^2 \tag{5}$$

where K_V is the ratio of the intake valve diameter to the engine bore. This is a typical engine design parameter. Based on the automotive data, it could range between 0.30 and 0.45. A too high value could be faced with head geometrical and constructive limits, while a too low value could penalize the cylinder air filling. Considering the equations above written and introducing the expression of the mean piston speed, the Eq. (5) could be then rewritten as follows:

$$V_{T\_IVC} \propto \frac{C \cdot \omega_{ENGINE}}{K_V^2 \cdot K_T} \cdot K_P \tag{6}$$

where C is the engine stroke and ω_{ENGINE} is the engine angular speed. Hence the more the engine stroke and the engine speed, the more the tumble velocity at IVC. In particular an increase of the engine stroke could led to a more available space for the incoming flow for closing on itself, as stated by the authors in [1]. A proper self-closed flow is able to generate a structured vortex inside the engine chamber. The above discussed expressions let to highlight that the most critical engine operating conditions are at low load (partial load) and at low engine speed.

3.2. The tumble velocity at TDC

For deriving the expression of the residual tumble velocity at TDC, the tumble model has to be explained here. In particular it is necessary to point out that at TDC the tumble vortex is not still present: in the present coverage the assumption of a still existing residual tumble vortex at TDC is made.

A rotating solid body of angular momentum J equal to the angular momentum of the instantaneous in-cylinder mass, evaluated with respect to an axis perpendicular to the cylinder axis and centered between the head and the instantaneous piston position, is considered. The expression of the angular momentum is:

$$J = I \cdot \omega \tag{7}$$

Where I is the fluid vortex inertia and ω is the angular velocity of the equivalent rotating solid body. Moreover the generation of only one main vortex, having dimensions comparable with the instantaneous piston stroke, is assumed (Fig. 2). During the suction phase the variation of the inertia angular momentum J is related to the variation of the inertia angular momentum due to the incoming mass flow rate and the torques, which in turn are due to both the internal friction T_W and the interaction with the piston face T_S, as follows:

$$\frac{dJ}{dt} = \left.\frac{dJ}{dt}\right|_{INLET} - T_W - T_S = \dot{m}_{AT} \cdot V_{TANG} \cdot r\left(\frac{C}{D}\right) \tag{8}$$

The variation of the inertia angular momentum due to the incoming mass flow rate was also expressed as a function of the inlet mass flow rate and the characteristic dimension r of the tumble vortex, function of the C/D ratio.

Neglecting the friction and the internal losses, and under the assumption that at IVC the inertia angular momentum is constant, the inertia angular momentum J due to the incoming mass flow rate could be rewritten as:

$$J_{INLET} = const = I \cdot \omega \tag{9}$$

So the inertia angular momentum J is constant at IVC. During the compression stroke the vortex deformation due to the reduction of the distance between the head and the piston induces a reduction of the fluid vortex inertia I and so an acceleration of the vortex rotational speed ω: this is called the spin-up phase. This causes an increase of the tangential component of the intake velocity and so of the tumble velocity (Fig. 3).

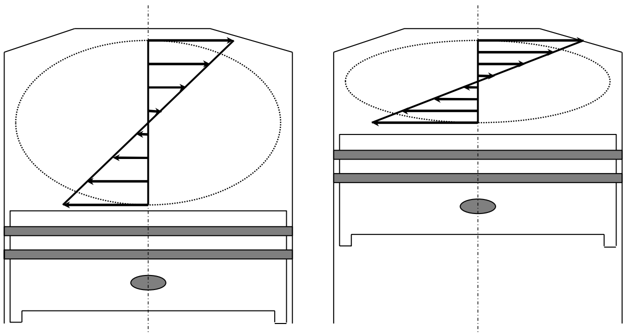

Fig. 3. Tumble velocity V_T distribution: (a) at IVC; (b) approaching TDC.

At the end of the compression stroke the tumble velocity component is dissipated and the turbulence is generated. The vortex angular speed equation is:

$$\omega \propto \frac{V_T}{r} \tag{10}$$

The characteristic dimension of the vortex r is called h in this coverage and it represents the half distance between the head and the instantaneous piston position. Under the above reported assumptions and combining Eq. (9) and Eq. (10), it is possible to write:

$$I_{IVC} \cdot \frac{V_{T\_IVC}}{h_{IVC}} \propto I_{TDC} \cdot \frac{V_{T\_TDC}}{h_{TDC}} \tag{11}$$

Moreover the fluid vortex inertia is defined as the fluid mass of the equivalent rotating solid body multiplied by the squared characteristic dimension of the vortex h. So substituting the fluid vortex inertia with the squared characteristic dimension of the vortex h and combining the Eq. (12) with the Eq. (11), it is possible to deduce the expression for the tumble velocity at TDC:

$$V_{T\_TDC} \propto \frac{I_{IVC}}{I_{TDC}} \cdot V_{T\_IVC} \cdot \frac{h_{TDC}}{h_{IVC}} \propto \frac{h^2{}_{IVC}}{h^2{}_{TDC}} \cdot V_{T\_IVC} \cdot \frac{h_{TDC}}{h_{IVC}} \propto V_{T\_IVC} \cdot \frac{h_{IVC}}{h_{TDC}} \tag{12}$$

Applying the scaling rules, the half distance between the head and the instantaneous piston position h at IVC can be assumed to be proportional to the half distance between the head and the instantaneous piston position at TDC and the C/D ratio. In fact the ratio of h_{IVC} to h_{TDC} lets assess the degree of deformation of the tumble vortex during the compression stroke: the larger is the C/D ratio, the larger is the ratio of h_{IVC} to h_{TDC}, the larger is the compression stroke and thus the degree of distortion of the tumble vortex (Fig.4). Increasing the compression stroke, there is an increase of the C/D ratio. So there is a proportional correlation between the C/D ratio and the ratio of h_{IVC} to h_{TDC}. Finally the Eq. (12) can be written as follows:

$$V_{T\_TDC} \propto \frac{V_P}{K_V^2 \cdot K_T} \cdot K_P \cdot \frac{C}{D} \tag{13}$$

This is the final expression of the residual tumble velocity at TDC. Its trend is the same of the tumble velocity at IVC.

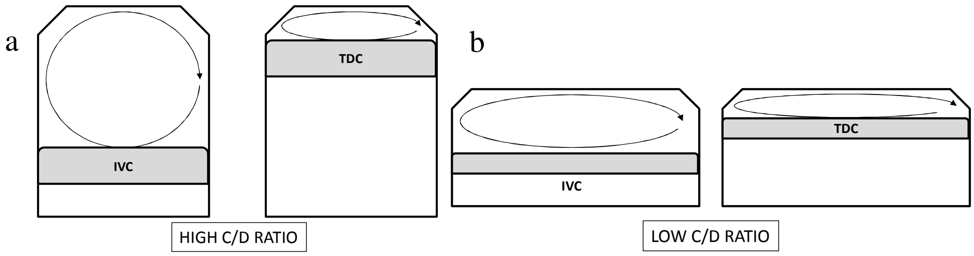

Fig. 4. Sketch of variation of the degree of distortion of the tumble vortex with C/D ratio: (a) high C/D ratio; (b) low C/D ratio.

3.3. The squish velocity at TDC

The squish motion importance is related to its efficiency in boosting the combustion process just developed. The squish motion acts around the TDC feeding the flame with the fresh mixture coming from the engine's periphery. Its importance is related to both its directionality and its timing. In fact the squish motion must be oriented toward the flame front (directionality) and the squish motion must feed the flame front when the piston is approaching the engine head, pushing the fresh mixture forward the combustion chamber middle (timing). If the squish flow is properly oriented towards the flame front and properly time-tuned, it allows to increase the fuel oxidation reaction rate just around the TDC.

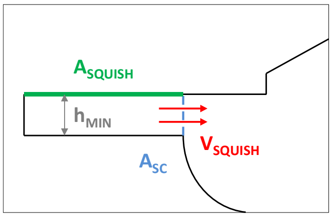

Fig. 5. Sketch of the squish motion geometry of the engine's head.

Looking at Fig. 5, where a sketch of the squish motion geometry is depicted, it is possible to apply the mass conservation law trough the curtain squish area A_{SC}:

$$V_{SQUISH} \cdot A_{SC} \propto A_{SQUISH} \cdot \frac{dh_S}{dt} \tag{14}$$

V_{SQUISH} is the squish velocity perpendicular to the curtain squish area, A_{SQUISH} is the plane squish area and h_S is the instantaneous squish height. The second term of Eq. (14) stands for the squish volume variation due to the piston motion: when the piston approaches the engine's head, due to the reduction of the squish volume, the fresh mixture trapped in the periphery of the combustion chamber is pushed forward the combustion chamber center. The term dh_S/dt represents the instantaneous piston speed and in stationary conditions it is proportional to the mean piston speed. The curtain squish area is proportional to the engine's bore and the instantaneous squish height. At TDC the instantaneous squish height is replaced by the minimum squish height h_{MIN}, which also is a measurable design parameter. Finally the squish area is proportional to the squared bore and a coefficient, called K_S, which represents the percentage squish area. So the expression of the squish velocity is as follows:

$$V_{SQUISH} \propto \frac{D^2 \cdot K_S}{D \cdot h_{MIN}} \cdot V_P \propto \frac{D \cdot K_S}{h_{MIN}} \cdot V_P \tag{15}$$

Looking at the final expression of the squish velocity in Eq. (15) some considerations come out. First of all the squish velocity is proportional to the mean piston speed, to the percentage squish area K_S and to the bore D. On the other hand the squish velocity is inversely proportional to the minimum squish height at TDC. These considerations confirm the results reported in [1].

3.4. Relation between the squish velocity and the tumble velocity at TDC

Finally in this paragraph the expression explaining the mutual relation between the squish motion and the tumble motion at TDC is derived. The relation is derived as the ratio between the Eq. (13) and the Eq. (15):

$$\frac{V_{SQUISH}}{V_{T\_TDC}} \propto \frac{K_S}{h_{MIN}} \cdot \frac{K_V^2 \cdot K_T}{K_P} \cdot \frac{D}{\left(\frac{C}{D}\right)} \tag{16}$$

The Eq. (16) summarizes the relation between the squish velocity and the residual tumble velocity at TDC by means some design parameters. The latter is of importance since it allows to evaluate the interference effect of the residual tumble velocity on the squish velocity. Moreover the Eq. (16) allows to estimate how the ratio of the squish velocity to the residual tumble velocity changes depending on the variation of some basic fluidynamic parameters for different classes of engines. For analyzing the influence of each single parameter on this ratio, it is necessary to keep constant all the parameter except the one under analysis, as done in [3].

In this coverage only some considerations about the relationship between the squish velocity to the tumble velocity ratio and the engine class, expressed in terms of the bore value, the stroke value and hence the C/D ratio value, are reported.

4. Parametric analysis of the squish velocity to the residual tumble velocity ratio at TDC

In the present paragraph a parametric analysis of the previous developed model is analyzed in detail, considering only the degree of magnitude. Eq. (16) sums up the most important parameters affecting the ratio of the squish velocity to the residual tumble velocity at TDC. The graphs reported below are referred to a non-specific engine having the characteristics summed up in Table 1. For performing the parametric analysis, the engine class (C/D ratio value), the bore and the stroke were respectively varied keeping constant all the other values of Table 1. The focus was to highlight the importance of the tumble motion and the squish motion depending on the engine class, using the Eq. (16).

Table 1. Engine parameter values

h_{MIN} [mm]	% squish area K_S	K_T	K_V	K_P
0.9	0.2	0.5	0.39	0.47

In Fig. 6a the trend of the ratio between the squish velocity and the tumble velocity versus the engine stroke is visible: the ratio decreases linearly with the engine stroke. The physical meaning of this trend was above discussed. It is related to the more space available for the tumble vortex closure by an increase of the engine stroke. In Fig. 6b is illustrated the trend of the ratio between the squish velocity and the tumble velocity versus the engine bore: their relationship is quadratic. It highlights the strong importance of the engine bore value for the squish motion effectiveness, confirming the previous results in [1]. In Fig. 7 the trend of the squish velocity to the tumble velocity ratio versus the C/D ratio is depicted: the higher the C/D ratio, the lower the importance of the squish motion and the higher the importance of the tumble motion. For explaining it in detail, a deep investigation on the influence of the C/D ratio is carried out. In particular an estimation of the variation of the squish velocity at TDC is performed. The squish velocity at TDC is calculated by Eq. (14) under stationary condition hypothesis.

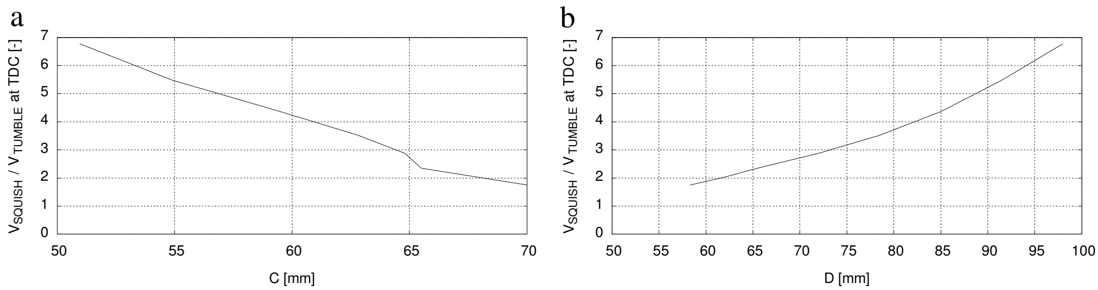

Fig. 6. Squish velocity to tumble velocity ratio trend versus the (a) stroke C; (b) Bore D.

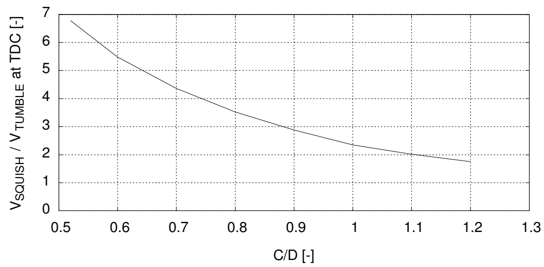

Fig. 7. Squish velocity to tumble velocity ratio trend versus the C/D ratio (engine class).

In Fig. 8 the ratio between the radius variation Δr and the cylinder area versus the C/D ratio for a percentage squish area of 25% (K_S=0.25) and 30% (Ks=0.30) respectively is plotted. The physical meaning of the ratio between Δr and the cylinder area is the space occupied by the plane squish area with respect to the engine bore value. For a fixed bore value, i.e. for a fixed C/D ratio value, the more the ratio between Δr and the cylinder area, the more the space occupied by the squish area and then the more the ‘level of intrusion’ of the squish motion geometry on the engine head. Fixing the value of the ratio between Δr and the cylinder area to 0.3 for example, this value is reached for an engine having the C/D ratio equal to 0.78 if the parameter K_S is 30%, or for an engine having the C/D ratio equal to 0.95 if the parameter K_S is only 25%. In fact increasing the C/D ratio value (and so decreasing the engine bore) the percentage space left available for the squish area location on the engine head gets less wide. Moreover the more the C/D ratio, the more the tumble motion effectiveness and then the less the squish velocity strength.

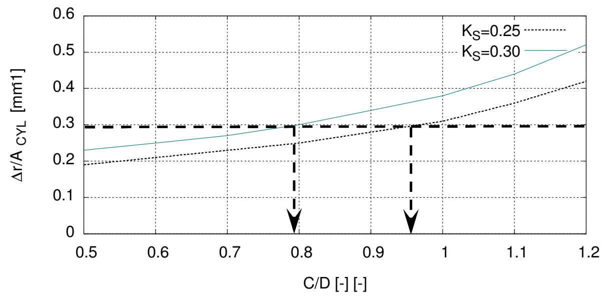

Fig. 8. Variation of the ratio between Δr and the cylinder area versus the C/D ratio.

5. Conclusions

The main focus of the present paper concerned the development of a theoretic-interpretative 0D model which could highlight the most significant parameters in the engine design process and in particular in the determination of: the tumble velocity at IVC and at TDC, the squish velocity at TDC and their mutual interaction. They are especially meaningful because they determine, at different times of the combustion process, the combustion velocity, and so they have a strong effect on the indicated efficiency, the raw emission levels and the fuel consumption.

The model seems to suggest that it could be a problem handling engines with a high stroke and a small bore (high C/D ratio), confirming the results in [1]. In particular the model emphasizes that would be some limits to the effective squish velocity for the engine at high C/D ratio value. The parameters most influencing the tumble motion have to be pursued at high C/D ratio values.

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