CALCULATION OF GAS VELOCITY INSIDE THE CYLINDER OF SPARK IGNITION ENGINES

G. MARTIN, N. APOSTOLESCU*

A fost realizat un studiu teoretic şi experimental, şi s-a determinat viteza de referinţă a gazelor din camera de ardere a motorului. Această viteză a fost calculată cu ajutorul codului CFD KIVA, se bazează pe energia cinetică a gazelor din camera de ardere, şi reprezintă o viteză de referinţă destinată utilizării în relaţiile de tip Woschni. S-au realizat comparaţii între această viteză de referinţă, şi vitezele de referinţă dezvoltate de Huber-Woschni şi Hohenberg. Testele experimentale s-au realizat pe două motoare cu caracteristici diferite de curgere, în condiţii de funcţionare independentă (pentru un motor), şi în condiţii de antrenare (pentru ambele motoare). Pentru pregătirea calculului, s-au realizat cu KIVA două reţele de discretizare 3D, utilizând elemente hexaedrice.

A theoretical and experimental study was carried out to compute the mean gas velocity inside the combustion chamber. This velocity was computed with a CFD code, KIVA, as a reference velocity to be used in Woschni type equations, and is based on the flow field kinetic energy of the in-cylinder gases. Comparisons were made between this reference velocity, and the Huber-Woschni and Hohenberg reference velocities. For experimental purposes, two engines were considered, with different fluid motion, and the tests were made for fired conditions (for one engine), as well as for motored conditions (for both engines). For computing purposes, two 3D meshes were made in KIVA, using hexahedron elements.

Keywords: reference velocity, computing mesh

Introduction

One of the main difficulties encountered by many authors is to define the reference velocity which controls the heat transfer by convection in the combustion chamber of IC engines. Various solutions were proposed, based on the model for heat transfer resulting from Reynolds analogy:

$$Nu = C\,\mathrm{Re}^{b} \tag{1}$$

with the exponent b ranging from 0.7 [1], [2], to 0.8 [3].

* Prof. PhD, Dept. of Mechanical Engineering, University POLITEHNICA of Bucharest, Romania

Beginning with Nusselt, and then Brilling, Eichelberg and Van Tyen, different functions of the mean piston speed were proposed to estimate the gas velocity to be used in the heat transfer coefficient formulas.

The mean piston speed term proposed by Nusselt was:

$$w_r = 1 + 1{,}24 \cdot \overline{w}_{mp} \tag{2}$$

and by Brilling

$$w_r = 3.5 + 0.185 \cdot \overline{w}_{mp} \tag{3}$$

Van Tyen advanced the slightly modified term:

$$w_r = 3.22 + 0.864 \cdot \overline{w}_{mp} \tag{4}$$

while Eichelberg:

$$w_r = \sqrt[3]{\overline{w}_{mp}} \tag{5}$$

Henein [4] proposed another approach of the reference velocity, a more representative. Based on Eichelberg studies, he considered a theoretical velocity based on the squish, swirl and tumble motion of the gases in the cylinder, and experimentally validated his work on an one cylinder engine.

Also based on Eichelberg results, Pflaum [5] modified the coefficients for the mean piston speed. Knight proposed a reference velocity based on the mean kinetic energy for the gas mass unity, computed in a Diesel direct injection engine:

$$w_r = \sqrt{\frac{E_{cm}}{2 \cdot m}} \tag{6}$$

where E_{cm} is the mean kinetic energy of the in-cylinder gases, and it was computed based on the flow characteristics of the gases at the intake and exhaust. He considered that all the kinetic energy is gradually transformed into turbulent energy; the model takes into account the flow in the intake and exhaust areas, and also in the pre-chamber exit area.

The to day widely used formula was proposed by Woschni in 1967 [3].

$$w_r = \left[C_1 \cdot \overline{w}_{mp} + C_2 \cdot \frac{V_S \cdot T_x}{p_x V_x} \cdot (p - p_0) \right] \tag{7}$$

C_1 = swirl constant

C_2 = combustion constant

For the gas exchange period: $C_1 = 6{,}18; \quad C_2 = 0$

For the compression period: $C_1 = 2{,}28; \quad C_2 = 0$

For the combustion and expansion period:

$$C_1 = 2{,}28; \quad C_2 = 3{,}24 \cdot 10^{-3} \text{ [m/sK]}$$

The instantaneous pressure p and the displacement volume V_S are considered in [kPa] and m^3 respectively; p_x, V_x, T_x are the gas pressure, volume and temperature at the "x" moment (e.g. IVC).

Apostolescu and Grünwald [6] proposed the expression:

$$w_r = C_1 \cdot \left(\sum_{k=1}^{n} w'^2_k \right)^{0.5} \tag{8}$$

where C_1 is a coefficient and w'_k are turbulence intensities. A similar approach was proposed by Davis and Borgnakke [7].

Woschni made succesive improvements to his formula by proposing a swirl term [8] for engines with swirl and high speed engines, and finally in 1990 by introducing special corrections [9] for low load and motored conditions. The final form, named here Huber-Woschni, takes into account the swirl and squish, and also the engine load:

$$w_{rHW} = C_1 v_{gas} \tag{9}$$

where

$$C_1 = 2.28 + 0.308 \frac{w_p}{\overline{w}_{mp}} \tag{10}$$

$$v_{gas} = \overline{w}_{mp} \left[1 + 2 \left(\frac{V_C}{V_i} \right)^2 imep^{-0.2} \right] \tag{11}$$

$$\text{If } C_2 \frac{V_S \cdot T_x}{p_x \cdot V_x} (p - p_0) \geq 2 C_1 \cdot \overline{w}_{mp} \cdot \left(\frac{V_C}{V_i} \right)^2 \cdot imep^{-0.2} \tag{12}$$

then

$$v_{gas} = \overline{w}_{mp} + \frac{C_2}{C_1} \cdot \frac{V_S \cdot T_x}{p_x V_x} \cdot (p - p_0) \tag{13}$$

V_c = compression volume [m^3]

V_i = cylinder volume [m^3]

Hohenberg [10], [11] modified Woschni's equation and proposed the formula

$$w_{rHO}^{0,8} = const \cdot p^{0,2} \cdot T^{0,1} \left(\overline{w}_{mp} + C_2 \right)^{0.8} \tag{14}$$

where C_2 = 1.4 and expresses the combustion effects on the turbulence.

LeFeuvre *et al.* [12] and Dent and Suliaman [13] considered the flow upon the flat-plate, and made experimental measurements. The proposed equation is:

$$w_r = r\omega_g \tag{15}$$

where ω_g is solid-body angular velocity of the charge, and r is radius from the cylinder axis to the measurement point.

Poulos and Heywood [14] proposed a reference velocity based on the instantaneous piston speed $v_p(t)$, the mean flow kinetic energy E_c and the turbulent kinetic energy

$$w_r = \sqrt{2 \cdot E_c + 2 \cdot k + \left(\frac{v_p(t)}{2}\right)^2} \tag{16}$$

Morel and Keribar [15] defined the reference velocity as a function of u_x, u_y and k

$$w_r = \left(U_x^2 + U_y^2 + 2 \cdot k\right)^{\frac{1}{2}} \tag{17}$$

where u_x, u_y are the two velocity components, being outside the boundary layer, and parallel with the surface.

A similar approach was proposed by Schubert *et al.* [16], where u_x, u_y were considered the axial, tangential or radial velocity, depending of the chosen surface. The axial velocity was modeled as half of the mean piston speed, the radial velocity on the base of the continuity equation, and the tangential velocity on the base of the momentum conservation equation.

Payri *et al.* [17] analyzed recently the heat transfer using CFD methods, with a modified Woschni based equation. The reference velocity proposed by Woschni was modified by the means of the swirl constant C_1:

$$C_1 = C_{w1} + C_{w2}\frac{v_u(\alpha)}{\overline{w}_{mp}} \tag{18}$$

where

$$v_u(\alpha) = x^p(\alpha) \cdot v_{u,TDC} \tag{19}$$

C_{w1}, C_{w2} are constants

$v_u(\alpha)$ is the tangential velocity [m/s]

$v_{u,TDC}$ is the tangential velocity at TDC [m/s]

$x^p(\alpha)$ is a trigonometric swirl function

The reference velocity proposed by the authors in this work is based on the flow field calculations. It is spatially averaged, and its instantaneously value is defined as:

$$w_{rAM} = \sqrt{\frac{2E + 2k}{m}} \qquad (20)$$

where E is the mean kinetic energy and considers all three velocity components.

1. The computing mesh

The experimental work was made on two spark ignition engines, with different fluid motion: DACIA, a small engine without swirl, and D2156, a large displacement one with swirl generation during induction. The engines were studied on a test bed: DACIA was experimentally studied under motoring conditions, and D2156 under motoring and fired conditions. The indicated mean pressure, fuel and air consumptions, exhaust composition, temperatures were obtained and recorded.

Both combustion chambers were meshed. As the combustion chamber is asymmetric for both engines, D2156 and DACIA, a three dimensional (3D) computational mesh was used. The mesh is made up of hexaedrons cells.

For D2156 and DACIA, the model meshes has a maximum of 44000 nodes, respectively 66300 nodes at BDC, with the cell height from 2 to 5 mm, depending on the geometrical features.

In Fig. 1 is represented the mesh for D2156 engine, while in Fig. 2 the mesh for DACIA engine. For both engines were meshed all the flow regions, the combustion chamber, the intake and exhaust ports and runners, as well as the intake and exhaust valves. The valves lift movements were taken into account.

The D2156 model was meshed at BDC, with both valves in opened position (Fig. 1 a,b). The main difficulties encountered for meshing were related to the valve guides (fig 1.c), the intersection zones between the piston cup mesh and the bottom cylinder mesh (Fig. 1 b), which represents a clone of the top cylinder mesh with specific design for valves (Fig. 1 c). In Fig. 1 c) is represented a view from the top of the cylinder; only the intake valve guide was modeled, because the exhaust valve guide have a special shape which is opposed to obtaining an optimized mesh. The valves were modeled with flat faces, for the same reasons (Fig.1 b). Special shape for the intake runner was modeled to generate the swirl at the intake phase (Fig 1. c). No inversed or non convex cells were obtained.

The DACIA model was meshed at BDC, with both valves in closed position (Fig. 2 a,b). The main difficulties for meshing were the generation of the wedge chamber and the tilting process of the assembly valves-chamber. The horizontal section (Fig. 2 d) by the cylinder shows the shape of the combustion chamber mesh. In DACIA case, both valves have their own shape of the meshed faces. In Fig. 2 b), it can be seen the mesh of the intake valve with profile. To avoid the distortion of the elements, the valve guides for DACIA were not modeled (fig 2. c). Nevertheless, the valve guides for this engine are not so intruded in the valve ports as in D2156 case. Due to tilting process, 22 non convex cells were obtained, but with a low influence for the final results.

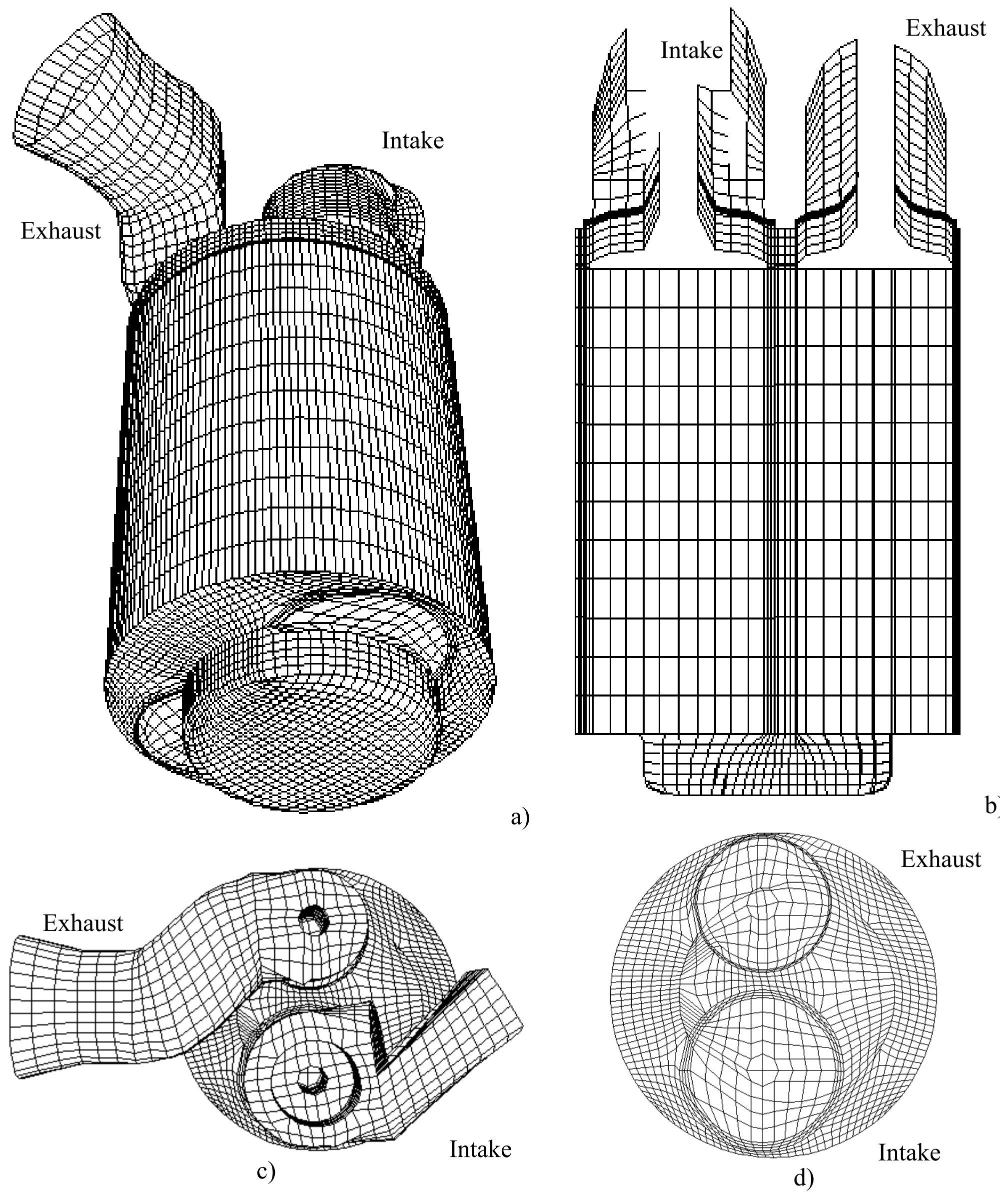

Fig.1. The computing mesh for D2156 engine, at BDC, 44000 nodes

a) view from the bottom of the combustion chamber

b) section by the cylinder axis

c) view from the top of the cylinder

d) horizontal section by the cylinder

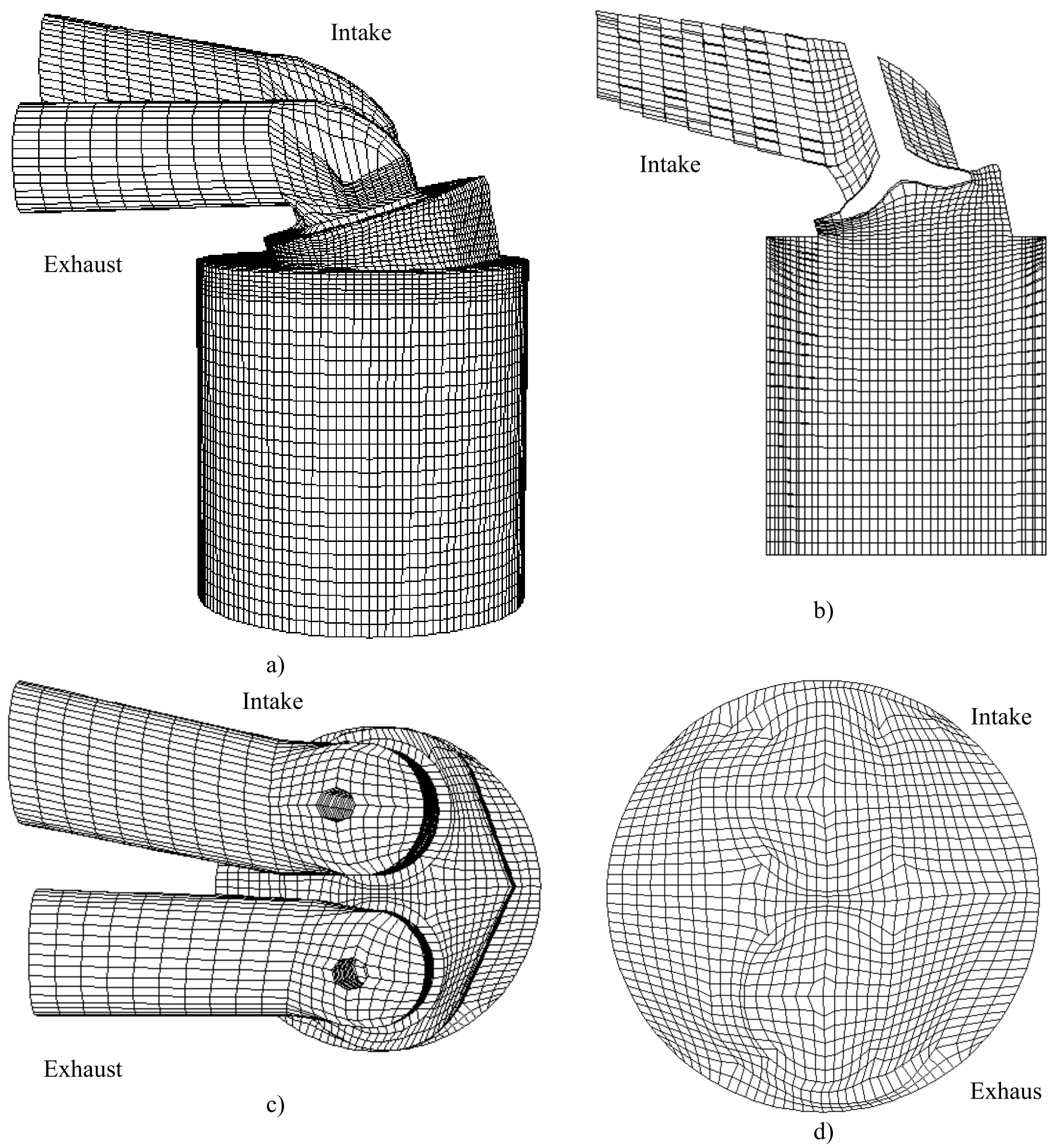

Fig.2. The computing mesh for DACIA engine, at BDC, 66300 nodes
a) lateral view from of the combustion chamber
b) section by the cylinder axis
c) view from the top of the cylinder
d) horizontal section by the cylinder

The grids are dynamically changing when the valves are moving. In the case of the piston, the vertices planes are activating or deactivating, depending on the direction of the piston movement.

2. The equations

The code KIVA solves the finite-difference approximations of the standard flow and heat transfer equations in correlation with a k-ε turbulence model as follows:

- the conservation of mass equation:

$$\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial}{\partial x_i}\left(\bar{\rho} \cdot \bar{u}_i\right) = S_m \tag{21}$$

- the conservation of momentum equation:

$$\frac{\partial}{\partial t}\left(\bar{\rho} \cdot \bar{u}_i\right) + \frac{\partial}{\partial x_j}\left(\bar{\rho} \cdot \bar{u}_i \cdot \bar{u}_j + \bar{\rho}\overline{u'_i u'_j} - \bar{\tau}_{ij}\right) + \frac{\partial p}{\partial x_i} - \bar{\rho} g \frac{x_i}{|x|} = 0 \tag{22}$$

- the conservation of thermal energy equation:

$$\frac{\partial}{\partial t}\left(\bar{\rho} h\right) + \frac{\partial}{\partial x_j}\left(\bar{\rho} \cdot \bar{u}_j \cdot \bar{h} + \bar{\rho}\overline{u'_j h'}\right) - \frac{\partial}{\partial x_j}\left(\frac{\bar{\lambda}}{c_p} \frac{\partial \bar{h}}{\partial x_j}\right) - \frac{\partial}{\partial x_j}\left(\tau_{ij} u_i\right) - \frac{\partial \bar{p}}{\partial t} - S_h = 0 \tag{23}$$

- the passive scalar equation:

$$\frac{\partial}{\partial t}\left(\bar{\rho} f\right) + \frac{\partial}{\partial x_j}\left(\bar{\rho} \cdot \bar{u}_j \cdot \bar{f} + \bar{\rho}\overline{u'_j f'}\right) - \frac{\partial}{\partial x_j}\left(D \frac{\partial \bar{f}}{\partial x_j}\right) = 0 \tag{24}$$

where

$$f = \frac{m_1}{m_1 + m_2} \tag{25}$$

m_1 = the mass of in-cylinder hot gases
m_2 = the mass of the fresh air

- the turbulent kinetic energy equation:

$$\frac{\partial}{\partial t}(\overline{\rho}k)+\frac{\partial}{\partial x_j}\left(\overline{\rho u}_j k\right)+\frac{\partial}{\partial x_j}\left[\overline{\overline{\rho}u'_j\left(k+\frac{p'}{\overline{\rho}}\right)}\right]+$$

$$+\overline{\rho}\overline{u'_i u'_j}\cdot\frac{\partial\overline{u}_j}{\partial x_i}+\mu\left(\frac{\overline{\partial u'_j}}{\partial x_i}\right)^2-\frac{\partial}{\partial x_j}\left(\mu\frac{\partial k}{\partial x_j}\right)=0 \tag{26}$$

- the dissipation equation:

$$\frac{\partial}{\partial t}(\overline{\rho}\varepsilon)+\frac{\partial}{\partial x_j}\left(\overline{\rho}\cdot\overline{u}_j\varepsilon\right)+\frac{\partial}{\partial x_j}\left(\overline{\rho}\cdot\overline{u'_j\cdot\varepsilon}\right)+2\mu\left\{\frac{\partial\overline{u}_j}{\partial x_l}\cdot\overline{\frac{\partial u'_i}{\partial x_l}\cdot\frac{\partial u'_i}{\partial x_j}}+\frac{\partial\overline{u}_i}{\partial x_j}\overline{\left(\frac{\partial u'_i}{\partial x_l}\cdot\frac{\partial u'_i}{\partial x_l}\right)}\right\}+$$

$$+2\mu\left\{\overline{\frac{\partial u'_i}{\partial x_l}\cdot\frac{\partial u'_j}{\partial x_l}\cdot\frac{\partial u'_i}{\partial x_j}}+\frac{\mu}{\rho}\overline{\left(\frac{\partial^2 u'_i}{\partial x_j\cdot\partial x_l}\right)^2}\right\}-2\overline{\rho}\varepsilon\frac{\partial\overline{u'_j}}{\partial x_j}-\overline{\rho}\varepsilon\frac{\partial\overline{u}_l}{\partial x_l}+$$

$$+2\mu\frac{\partial^2 u_i}{\partial x_l\partial x_j}\overline{u'_j\frac{\partial u'_i}{\partial x_l}}-S_\mu=0 \tag{27}$$

These equations are discretized in space and time, where the spatial diferencing is based on the ALE (arbitrary Langragian-Eulerian) method.

Calculations start at TDC (gas exchange phase, 0 CAD), and end at 720 CAD. The experimental data (pressure and temperature in the inlet and exhaust ports, and in the cylinder at 0 CAD, the equivalence ratio ϕ, the electrical discharge duration and energy) were used as input data for KIVA. Also, boundary conditions were assigned, as the walls surface temperature of the combustion chamber. Based on the experimental data, a mean averaged temperature was assigned for every surface, like the surface of cylinder, piston crown, cylinder head, intake valve and exhaust valve.

The reference velocity proposed by the authors (20) was computed by KIVA. This velocity was integrated in the code.

3. Results and discussion

Calculations were made with the KIVA code for both engines: for the D2156 engine in motored condition, at low speed, and at full charge at different speeds, for DACIA engine only at motored condition, at low and high speeds. The flow field was computed also in the runners, even with the both valves closed; so, the flow field is never frozen in the runners.

The flow field was analyzed at 360 CAD. For this angle, the combustion chambers meshes are shown in Fig. 3 for the D2156 engine and Fig. 4 for the DACIA engine. In Fig. 5 and 7 are represented the flow field for D2156 in motored condition at 150 rpm. It can be seen that in the vertical plane, the flow is dominated by the tumble, and in transversal plane by the swirl. The squish is very low due to the high distance between the piston crown and the head at TDC (9.18 mm). A high intensity organized flow is present in the combustion chamber of D2156 engine. The same projections are made for DACIA engine in Fig. 6 and 8, 150 rpm. The motion in the cylinder is dominated by the squish, with no representative tumble or swirl motion. Comparatively, the velocities in the D2156 engine are higher compared with DACIA, even by a factor of 4.

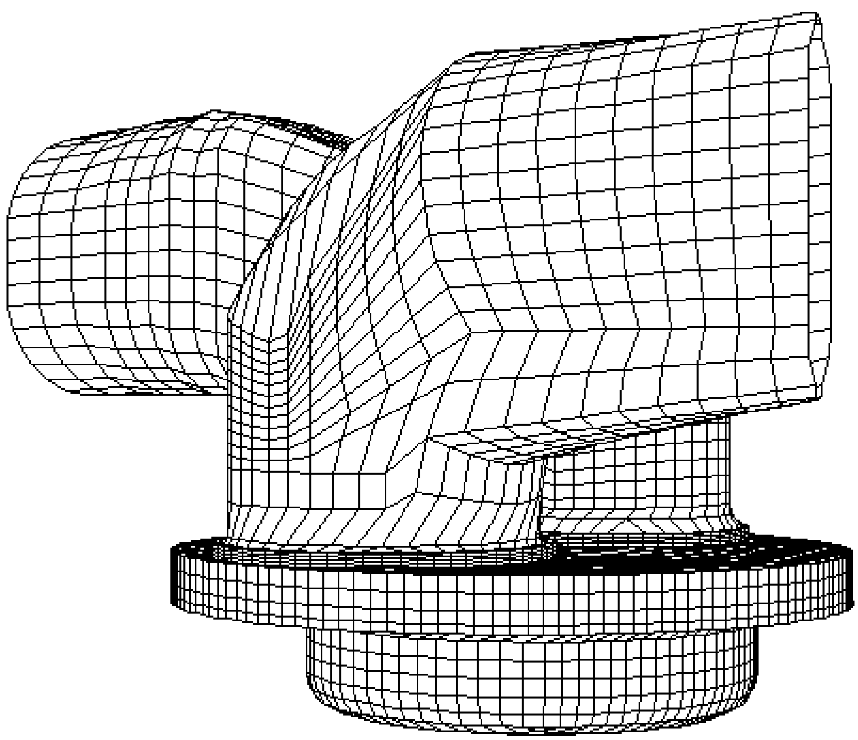

Fig. 3. D2156 combustion chamber mesh at 360 CAD

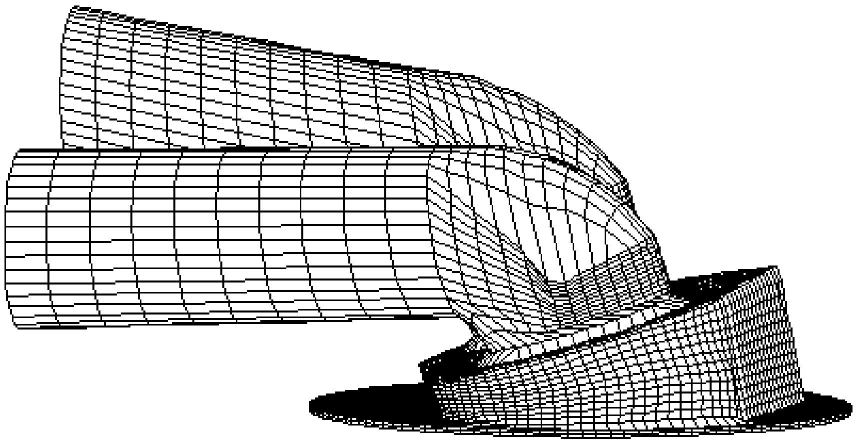

Fig. 4. DACIA combustion chamber mesh at 360 CAD

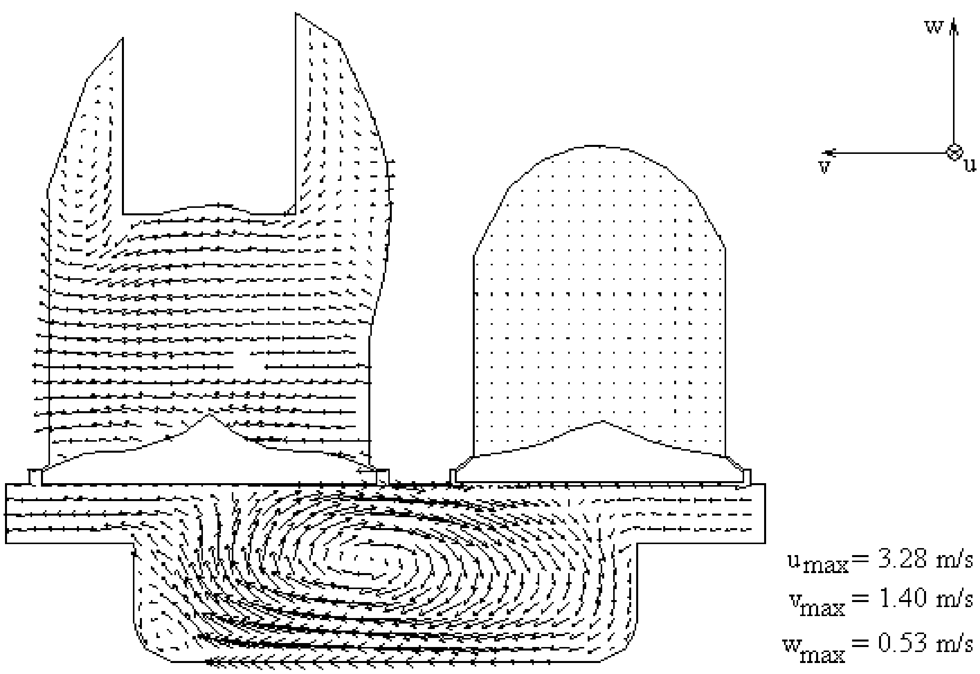

Fig. 5. Velocity vectors projections at 360 CAD, on the longitudinal plane; D2156 engine, 150 rpm, motored

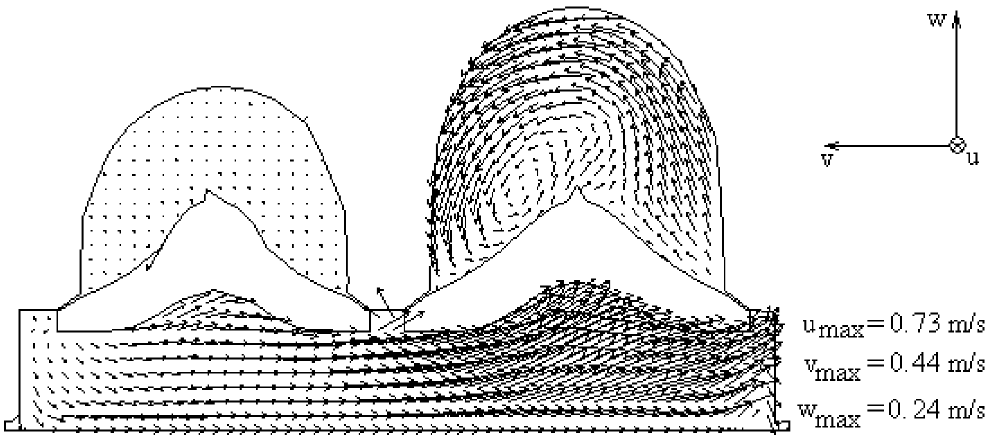

Fig. 6. Velocity vectors projections at 360 CAD, on the longitudinal plane; DACIA engine, 200 rpm, motored

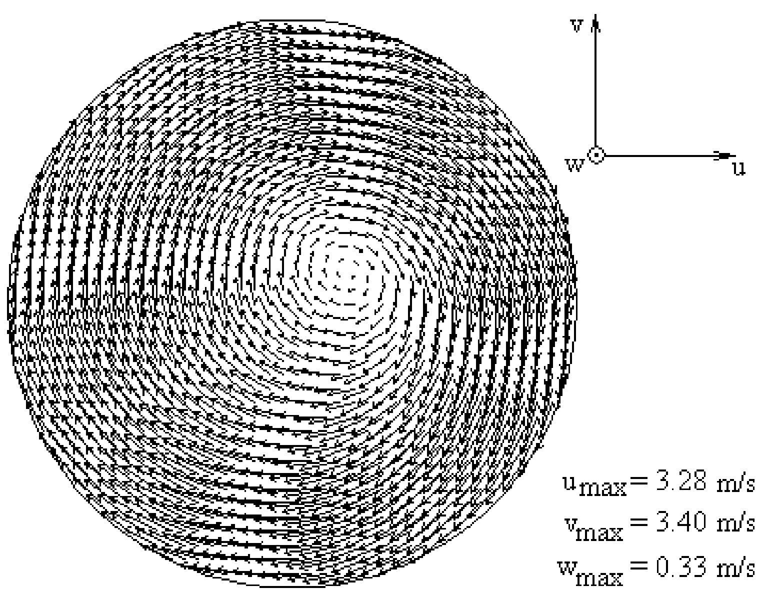

Fig. 7. Velocity vectors projections at 360 CAD, on the transversal plane, in the upper part of the cylinder, D2156 engine, 150 rpm, motored

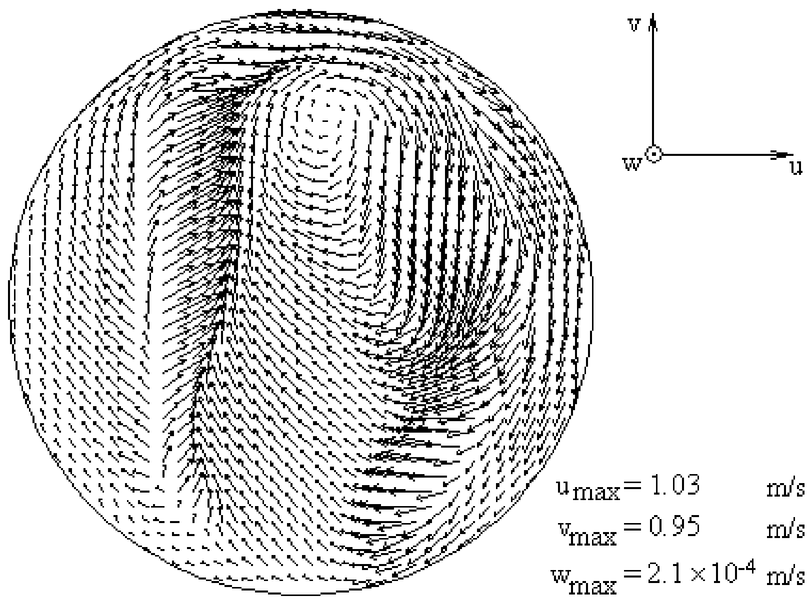

Fig. 8. Velocity vectors projections at 360 CAD, on the transversal plane, in the upper part of the cylinder, DACIA engine, 200 rpm, motored

For cyclic intervals, three different reference velocities were comparatively analyzed. The reference velocity proposed by the authors w_{rAM} (20) was provided by KIVA. An in-house computer software program was developed to compute the Hohenberg and Huber-Woschni reference velocities, w_{rHO} and w_{rHW}. The reference velocity calculated with the Hohenberg's formula w_{rHO} have the same value of the velocity at IVC, by a conveniently chosen value of the proportionality constant.

The resulting reference (AM) velocities were represented for D2156 engine at fired operation, 900 rpm (Fig. 9), 2100 rpm (Fig. 10), and at motored condition, 150 rpm (Fig.11) and compared with those calculated with Huber-Woschni (HW) and Hohenberg (HO) formulas. For DACIA engine, the velocities were represented for motoring condition, at 200 rpm (Fig. 12), and 2000 rpm (Fig. 13).

The common observation is that for all the cases, the velocity based on the calculation of kinetic energy w_{rAM} shows a decreasing trend with a slow rate, corresponding to the natural decay process of the kinetic energy. The combustion process has no significant effect on the reference velocity although the calculations take into consideration the nonhomogeneities generated by combustion.

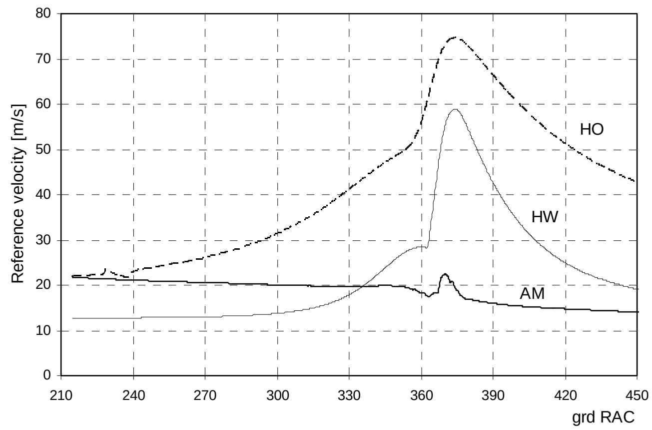

Fig. 9. Comparison of reference velocity calculated with the flow field kinetic energy (AM), the Woschni's (HW) and Hohenberg's (HO) formulas: D2156 engine, fired at 900 rpm

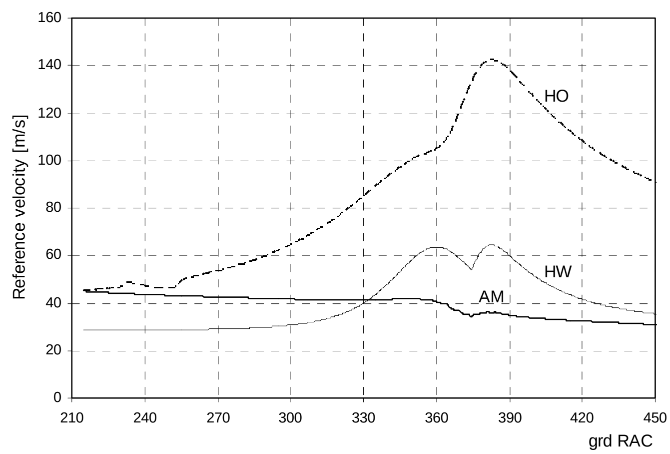

Fig. 10. Comparison of reference velocity calculated with the flow field kinetic energy (AM), the Woschni's (HW) and Hohenberg's (HO) formulas: D2156 engine, fired at 2100 rpm

For D2156, the decreasing trend is less markedly than for the engine DACIA. Both Woschni's and Hohenberg's formulas have an increasing trend in compression, in opposition with the natural effect of the kinetic energy decay. For D2156 engine, for fired operating condition (Fig. 9 and 10), the Hohenberg's reference velocity w_{rHO} have excessively high values. While the engine speed increases, it seems that the Huber-Woschni w_{rHW} reference velocity trend is to be closer with w_{rAM}.. The discontinuity in Woschni's velocity is generated by the effect of pressure rapid rise due to combustion. For D2156 engine, but motored condition at 150 rpm (Fig. 11), still Hohenberg's reference velocity has the higher values.

For DACIA engine, motored at 200 and 2000 rpm (Fig. 12, 13), the Huber-Woschni reference velocity w_{rHW} is too high. An explanation is that the Huber-Woschni's formula was obtained on a Diesel engine, with swirl motion, while DACIA is an engine without swirl. Also Hohenberg's velocity w_{rHO} presents high values, but lower than Huber-Woschni's.

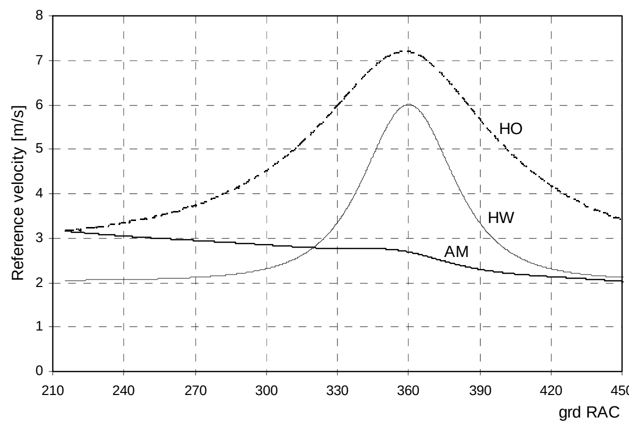

Fig. 11. Comparison of reference velocity calculated with the flow field kinetic energy (AM), the Woschni's (HW) and Hohenberg's (HO) formulas: D2156 engine, motored at 150 rpm

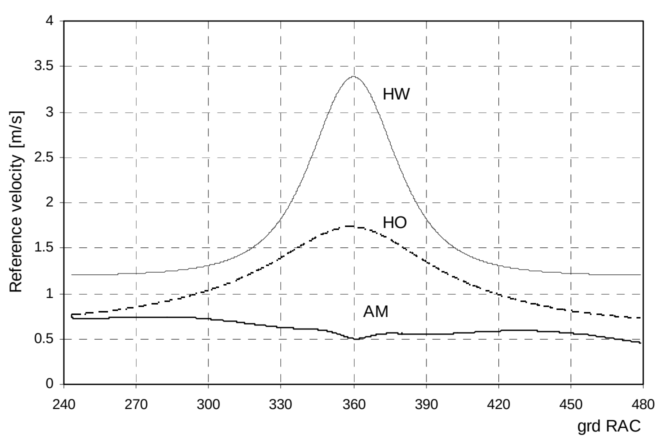

Fig. 12. Comparison of reference velocity calculated with the flow field kinetic energy (AM), the Woschni's (HW) and Hohenberg's (HO) formulas: DACIA engine, motored at 200 rpm

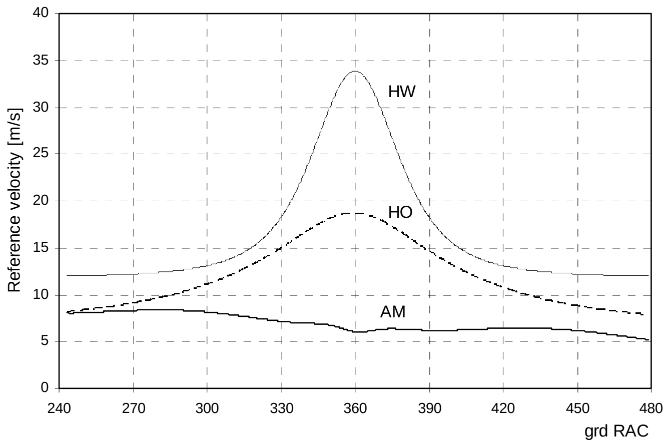

Fig. 13. Comparison of reference velocity calculated with the flow field kinetic energy (AM), the Woschni's (HW) and Hohenberg's (HO) formulas: DACIA engine, motored at 2000 rpm

Conclusions

1. The D2156 engine has a strong organized motion of the mixture, while DACIA engine hasn't an organised flow motion and the velocities are lower than D2156's, even by a factor of 4.
2. The velocity based on the calculation of kinetic energy w_{rAM} shows a decreasing trend with a slow rate, corresponding to the natural decay process of the kinetic energy, while the trend of the Huber-Woschni and Hohenberg reference velocities is to increase in the compression period.
3. The reference velocity, based on the global kinetic energy, is not significantly influenced by combustion.
4. The Hohenberg's reference velocity provide high values, for motored or fired opreation. For engines without swirl, in motored condition, the Woschni-Huber equation for reference velocity provides excessively high values.

ACKNOWLEDGEMENTS

The helpful discussions and the experimental data provided by Prof. Radu Chiriac are gratefully acknowledged.

The authors wish also to thank Mr. Chr. Aïcardi and Mr. C. Buchou from RENAULT for the support with KIVA II.

LIST OF NOTATIONS AND ABBREVIATIONS

α	crank angle degree
E	mean kinetic energy
imep	mean indicated pressure
k	turbulent kinetic energy
m	gas mass
Nu	Nusselt number
p	pressure
p_0	pressure in motored conditions
ρ	density of gas
Re	Reynolds number
S	source terms
T	gas temperature; mass averaged gas temperature
u, v, w	gas velocity components
V	volume
V_S	displacement volume
$\overline{w}_{mp}$	mean piston speed
w_r	reference velocity
w_{rAM}	reference velocity based on the mean kinetic energy
w_{rHW}	Huber-Woschni reference velocity
w_{rHO}	Hohenberg reference velocity
w_p	circumferential speed
$x_{i,j,k}$	Cartesian coordinates

Abbreviations

BDC	bottom dead center
CAD	crank angle degrees
CFD	computational fluid dynamics
IC	internal combustion
IVC	intake valve closing
rpm	rotations per minute
TDC	top dead center

R E F E R E N C E S

1. *G. Sitkei*, “Beitrag zur Theorie des Wärmeüberganges im Motor”, in Konstruktion, **vol. 14**, no. 2, 1962, pp. 67-72
2. *W. Annand*, “Heat transfer in the cylinders of reciprocating IC engines”, in Proceedings of the Inst. of Mech. Engrs, London, **vol. 177**, no. 36, 1963, pp. 973-995
3. *G. Woschni*, A universally applicable equation for the instantaneous heat transfer coefficient in the internal combustion engine, SAE Paper no. 670931, 1967
4. *N. A. Henein*, Instantaneous heat transfer rates and coefficients between the gas and combustion chamber of a diesel engine, SAE Paper no. 650096, 1965
5. *W. Pflaum*, “Warmeübergang bei Dieselmaschienen mit und ohne Aufladung”, in MTZ 22, 1961, pp. 70-74
6. *N. Apostolescu, B. Grünwald*, Contribuții la teoria transferului de căldură din cilindrul motorului cu ardere internă, Constructia de Masini, 1967
7. *G. C. Davis, C. Borgnakke*, The effect of in-cylinder flow processes (swirl, squish and turbulence) on engine efficiency – model predictions, SAE Paper no. 820045, 1982
8. *G. Woschni*, Die Berechnung der Wandverluste und der Thermischen Belastung der Bauteile von Dieselmotoren, MTZ 31, **vol. 12**, 1970, pp. 491-499
9. *K. Huber, G. Woschni, K. Zellinger*, Investigations on Heat Transfer in Internal Combustion Engines under Low Load and Motoring Conditions, SAE Paper no. 905018, 1990
10. *G. F. Hohenberg*, Advanced approaches for heat transfer calculations, SAE Paper no. 790825, 1979
11. *G. Hohenberg*, “Berechung des gasseitigen Wärmeüberganges in Dieselmotoren”, MTZ Mototechnische Zeitsehrift 41, **vol. 718**, 1980, pp. 321-326
12. *T. LeFeuvre, P. Myers and O. Uyehara*, Experimental instantaneous heat fluxes in a diesel engine and their correlation, SAE Paper no. 690464, 1969
13. *J. C. Dent and S. J. Suliaman*, Convective and radiative heat transfer in a high swirl direct injection diesel engine, SAE Paper no. 770407, 1977
14. *S. G. Poulos and J. B. Heywood*, The effect of chamber geometry on spark ignited engine combustion, SAE Paper no. 830334, 1983
15. *T. Morel and R. Keribar*, A model for predicting spatially and time resolved convective heat transfer in bowl-in-piston combustion chambers, SAE. Paper no. 850204, 1985
16. *C. Schubert, A. Wimmer, F. Chmela*, Advanced Heat Transfer Model for C.I. Engines, SAE Paper no. 2005-01-0695, 2005
17. *F. Payri, X. Margot, A. Gil, J. Martin*, Computational study of Heat Transfer to the Walls of a DI Diesel Engine, SAE Paper no. 2005-01-0210, 2005