GAS EXCHANGE IN VALVED TWO-STROKE SI ENGINE

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Abstract

The paper describes the work of high speed charged spark ignition overhead poppet valve two-stroke engine, which enables to achieve higher total efficiency and exhaust gas emission comparable to four-stroke engines. The work of such engines is possible by proper choice of valve timings, geometrical parameters of inlet, outlet ducts and charge pressure. The engine has to be equipped with direct fuel injection system enabling lower emission of pollutants. The work is based on theoretical considerations performed in GT-Power in previous authors' research and carried out in CFD code (KIVA 3V) for different engine configurations. The initial results included in the paper show influence of inlet port geometry and charge pressure on engine scavenging process. Additionally, optimum fuel spray injector position was considered in order to obtain proper fuel vaporization and avoid significant wall–wetting. The simulation results show that the nitrogen oxides are considerably reduced in comparison to four-stroke engines because of higher internal exhaust gas recirculation. The innovation of this proposal is applying of poppet intake and exhaust valves with turbocharging in the two-stroke engine and obtaining a significant downsizing effect. The conclusion shows the possibilities of proper gas exchange process in this type of two-stroke engine and thus, the feasibility of its application as a power unit for transportation means with higher total efficiency than traditional engines with possible change of engine work in two modes: two- and four-stroke cycles.

Keywords: *transport, engine development, two-stroke engine, boosting*

1. Introduction

The two-stroke engine still presents the competitor of four-stroke power unit especially nowadays and in the near future, when highly complicated boost systems containing turbocharger and additional easy to control compressor are becoming more and more popular and cost reasonable. Using of these systems in modern two-stroke engine for scavenging purposes via intake and exhaust poppet valves will remove all typical disadvantages of two-stroke engines like high hydrocarbon emission and short time between overhauls. It can be achieved by use of crankcase for lubricating purposes like in typical four-stroke engine rather than for scavenging.

More exact considerations about general idea of valved two-stoke engine are involved in [6], [7] and [8]. There were based on theoretical analyses and GT-Power simulations. The results show that optimum timing angles for intake and exhaust valves are in range:

- IVO: 150 – 170 deg ATDC,
- IVC: ca. 100 deg after IVO,
- EVO: 110 – 140 deg ATDC,
- EVC: ca. 125 deg after EVO,

with charge overpressure required for proper scavenging between 0.8 - 2 bar in comparison with exhaust manifold pressure.

The processes and phenomenon that occurs in cylinder of internal combustion engine presents highly complicated problem especially because of its multidisciplinarity. It includes 3D gas flows, chemical reactions, heat transfers and during direct injection also spray problems. This makes it unfeasible to simulate the in-cylinder processes with proper accuracy in a different approach than by use of CFD codes. Thus, authors applied mentioned above results of 1D analysis in the 3D simulations in order to check scavenging course and exhaust emission.

There are many COD codes that enable to calculate flows with sprays and chemical reactions in three-dimensional space. One of them is KIVA 3V [3] and it is assumed to be the best software for simulating processes that occurs in internal combustion engine cylinder, probably because it was created in order to solve directly engine problems. Thus, many constants and lows that were incorporated in calculations base on empirical research focused on engine aspects. Also, KIVA 3V [3] is provided as a source code in FORTRAN, so the user can incorporate his own subroutines and change the constant values.

2. The theoretical base of performed CFD calculations

The general equations that were solved by KIVA-II [1] code during calculations are transport equations of mass of each species (1), momentum (2) and internal energy (3):

$$\frac{\partial \rho_m}{\partial t} + \vec{\nabla} \cdot (\rho_m \vec{u}) = \vec{\nabla}\left[\rho D \vec{\nabla}\left(\frac{\rho_m}{\rho}\right)\right] + \dot{\rho}_m^c + \dot{\rho}^s \delta_{m1}, \tag{1}$$

$$\frac{\partial (\rho \vec{u})}{\partial t} + \vec{\nabla} \cdot (\rho \vec{u}\vec{u}) = -\frac{1}{a^2}\vec{\nabla} p - A_0 \vec{\nabla}(\frac{2}{3}\rho k) + \vec{\nabla}\overline{\sigma} + \vec{F}^S + \rho \vec{g}\,, \tag{2}$$

$$\frac{\partial (\rho e)}{\partial t} + \vec{\nabla}(\rho \vec{u} e) = -p\vec{\nabla} \cdot \vec{u} + (1 - A_0)\overline{\sigma} : \vec{\nabla}\vec{u} - \vec{\nabla}\vec{J} + A_0 \rho \varepsilon + \dot{Q}^c + \dot{Q}^s\,. \tag{3}$$

The fluid was assumed to be an ideal gas mixture and thus the state relations of ideal gas mixture were used. During calculating flows in turbulent state, two additional transport equations were solved:

$$\frac{\partial (\rho k)}{\partial t} + \vec{\nabla}(\rho \vec{u} k) = -\frac{2}{3}\rho k \vec{\nabla} \cdot \vec{u} + \vec{\sigma} : \vec{\nabla}\vec{u} + \left[\left(\frac{\mu}{\mathrm{Pr}_k}\right)\vec{\nabla} k\right] - \rho\varepsilon + \dot{W}^s\,, \tag{4}$$

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \vec{\nabla}(\rho \vec{u} \varepsilon) = -\left(\frac{2}{3}c_{\varepsilon_1} - c_{\varepsilon_3}\right)\rho\varepsilon\vec{\nabla} \cdot \vec{u} + \vec{\nabla} \cdot \left[\left(\frac{\mu}{\mathrm{Pr}_\varepsilon}\right)\vec{\nabla}\varepsilon\right] + \frac{\varepsilon}{k}\left[c_{\varepsilon_1}\vec{\sigma} : \vec{\nabla}\vec{u} - c_{\varepsilon_2}\rho\varepsilon + c_s \dot{W}^s\right]. \tag{5}$$

Chemical reactions which were calculated by the code were divided into two classes: kinetically proceeding and in equilibrium state. The kinetic reactions are described by Arrhenius formulae.

The spray phenomena present high difficulty in mathematical modelling. It has to take into account such variables as drop position, velocity, size, temperature and deformation from ideal shape (spherical). Moreover, drop oscillations, breakup, collisions and coalescence are important for good convergence of calculations with real process of spray. To manage with such a complicated problem, droplet probability function that describes the number of droplets per unit volume was introduced in form:

$$f(\vec{x}, \vec{v}, r, T_d, y, \dot{y}, t)\, d\vec{v}\, dr\, dT_d\, dy\, d\dot{y}\,, \tag{6}$$

which depends on:

$\vec{x}$ - drop position vector,

$\vec{v}$ - drop velocity vector,

r - equilibrium radius of drop (radius of drop of the same volume but spherical shape),

T_d - drop temperature (assumed constant through drop),

y - distortion from sphericity of drop,

$\dot{y}$ - rate of distortion of sphericity changes,

t - time.

The spray process that is described by the time evolution of function f was determined by spray equation in form:

$$\frac{\partial f}{\partial t}+\vec{\nabla}_{\vec{X}}(f\,\vec{v})+\vec{\nabla}_{\vec{V}}(f\,\vec{F})+\frac{\partial}{\partial r}(f\,R)+\frac{\partial}{\partial T_d}(f\,\dot{T}_d)+\frac{\partial}{\partial y}(f\,\dot{y})+\frac{\partial}{\partial \dot{y}}(f\,\ddot{y})=\dot{f}_{coll}+\dot{f}_{bu}, \qquad (7)$$

where:

$\vec{F}$ - droplet acceleration,
R - rate of ideal radius change,
$\dot{T}_d$ - rate of temperature change,
$\dot{f}_{coll}$ - source term due to droplet collisions,
$\dot{f}_{bu}$ - source term due to droplet breakups.

The source term due to droplet collisions was defined as:

$$\begin{aligned}\dot{f}_{coll}=\frac{1}{2}\iint & f(\vec{x},\vec{v}_1,r_1,T_{d_1},y_1,\dot{y}_1,t)\,f(\vec{x},\vec{v}_2,r_2,T_{d_2},y_2,\dot{y}_2,t)\,\pi(r_1+r_2)^2\left|\vec{v}_1-\vec{v}_2\right|\cdot\\ & \cdot[\sigma(\vec{v},r,T_d,y,\dot{y},\vec{v}_1,r_1,T_{d_1},y_1,\dot{y}_1,\vec{v}_2,r_2,T_{d_2},y_2,\dot{y}_2)-\\ & -\delta(\vec{v}-\vec{v}_1)\delta(r-r_1)\delta(T_d-T_{d_1})\delta(y-y_1)\delta(\dot{y}-\dot{y}_1)-\\ & -\delta(\vec{v}-\vec{v}_2)\delta(r-r_2)\delta(T_d-T_{d_2})\delta(y-y_2)\delta(\dot{y}-\dot{y}_2)]\\ & d\vec{v}_1\,dr_1\,dT_{d_1}\,dy_1\,d\dot{y}_1\,d\vec{v}_2\,dr_2\,dT_{d_2}\,dy_2\,d\dot{y}_2,\end{aligned} \qquad (8)$$

where σ presents collision transition probability function. The source term due to droplet collisions was expressed by:

$$\dot{f}_{bu}=\int f(\vec{x},\vec{v}_1,r_1,T_{d_1},1,\dot{y}_1,t)\,\dot{y}_1B(\vec{v},r,T_d,y,\dot{y},\vec{v}_1,r_1,T_{d_1},\dot{y}_1,\vec{x},t)\,d\vec{v}_1\,dr_1\,dT_{d_1}\,d\dot{y}_1 \qquad (9)$$

that contains B – breakup transition probability function.

3. Simulation approach and results

The analyses performed by authors have been carried out by use of geometry of Toyota 2SZ-FE engine (1.3 dm^3, VVT-i) equipped with direct injection, with standard and modified (top entry) intake port. Three simulations have been calculated with main parameters included in Tab. 1.

Tab. 1. Main parameters of performed simulations

Sim. No.	Valve timing		Injection parameters				Ignition parameters				Pressures	
1	IVO	150 deg	INJ_start	230 deg	INJ_r	8 mm	IGN_start	345 deg	IGN_x	-7 mm	INT_press	1.8 bar
	IVC	250 deg	INJ_durat.	60 deg	INJ_theta	0 deg	IGN_durat.	10 deg	IGN_y	0 mm	EXH_press	0.99 bar
	EVO	110 deg	INJ_mass	0.020 g	INJ_z	84 mm			IGN_z	86 mm		
	EVC	235 deg			tilt_xy	0 deg						
					tilt_xz	15 deg						
2	IVO	170 deg	INJ_start	260 deg	INJ_r	31.28 mm	IGN_start	340 deg	IGN_x	0 mm	INT_press	2 bar
	IVC	270 deg	INJ_durat.	50 deg	INJ_theta	180 deg	IGN_durat.	10 deg	IGN_y	0 mm	EXH_press	0.99 bar
	EVO	130 deg	INJ_mass	0.026 g	INJ_z	80 mm			IGN_z	88.5 mm		
	EVC	240 deg			tilt_xy	0 deg						
					tilt_xz	75 deg						
3	IVO	150 deg	INJ_start	230 deg	INJ_r	8 mm	IGN_start	342 deg	IGN_x	-7 mm	INT_press	1.8 bar
	IVC	250 deg	INJ_durat.	50 deg	INJ_theta	0 deg	IGN_durat.	10 deg	IGN_y	0 mm	EXH_press	0.99 bar
	EVO	110 deg	INJ_mass	0.022 g	INJ_z	84 mm			IGN_z	86 mm		
	EVC	235 deg			tilt_xy	0 deg						
					tilt_xz	-15 deg						

The simulation No. 1 base on standard intake port and injector position on top of combustion chamber near the spark plug, tilted from vertical of 15 deg in plane normal to crankshaft axis. The scavenging process is presented in Fig. 1. The four-stroke engine intake port cause significant short-circuiting especially easy to notice at the beginning of scavenging. Thus, it is difficult to remove burned gases from cylinder via exhaust ducts. Described situation results in high internal EGR and connection with this, low amount of nitrogen oxides produced – 1500 ppm without exhaust gases after-treatment. Vertical position of injector generates some problems with the fuel spray and vaporization process. Late beginning of fuel spray caused by end of scavenging force it to take place during the end of compression. Thus, significant wall-wetting occurs in connection with approaching piston crowd – Fig. 2.

In the simulation No. 2, the intake pressure and valve timing angles have been increased in order to observe the scavenging process in this conditions. Additionally, fuel injector position has been changed to the side of cylinder liner. The results are in Fig. 3 and 4. Comparing to sim.1, the short-circuiting becomes much bigger. The intake air goes to the exhaust valve not only via shortest possible way, but also through the "back" of exhaust valve (between exhaust valve and cylinder liner). Despite this, bigger amount of oxygen have been trapped in the cylinder – Fig. 7. Although 30 deg delay of spray in comparison with sim.2 have been applied, side position of injector produced smaller wall-wetting.

Last simulation have been performed on geometry with modified intake port but valve timing angles and intake pressure the same as in simulation No. 1. The solution patented by Ricardo – top entry intake port [9] produce so-called reverse tumble, which enables to obtain U-loop scavenging. Simulation results (Fig. 5) shows, that scavenging of this intake system is much better than in simulation No. 1 (compare oxygen content of in-cylinder gases for simulation No. 1 and 3 in Fig. 7). Simulation No. 2 on this plot is not comparable because of higher intake pressure used. Regarding fuel injection in simulation No. 1, the reverse tumble in connection with up-stream directed vertical spray gives very good result of fuel vaporization – Fig. 8. It enables to inject fuel with higher pressure without significant wall-wetting, enables better mixing and lower residual unburned fuel occurs during exhaust period – Fig. 6.

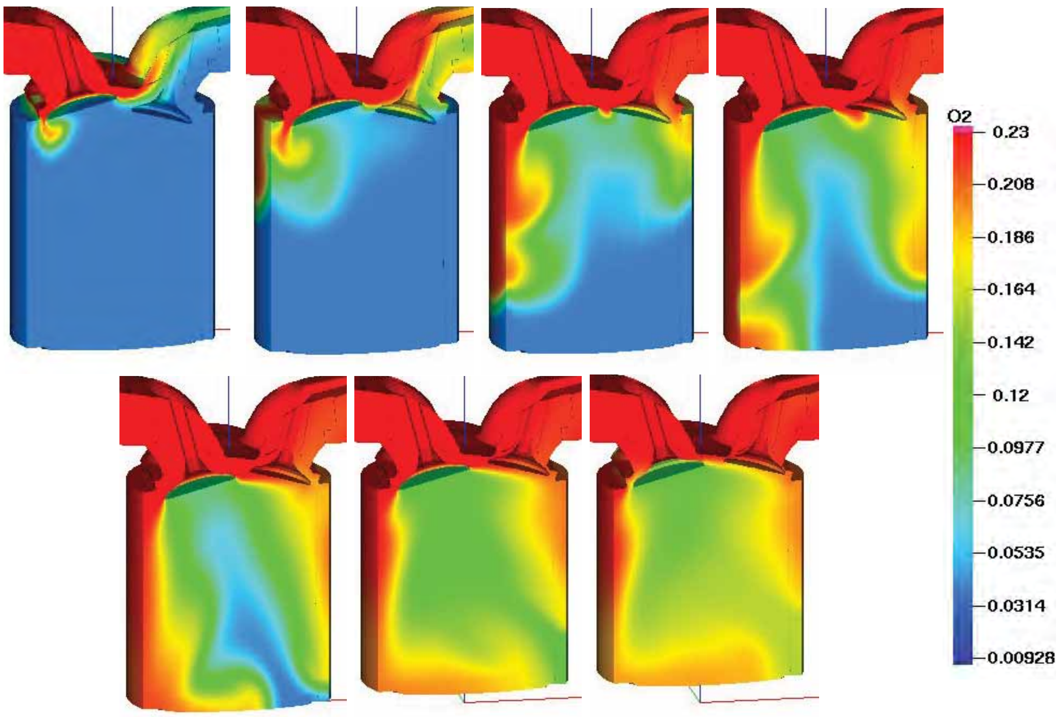

Fig. 1. The gas exchange process in simulation No. 1. Crank angles: 169, 180, 191, 200, 210, 226, 234 deg

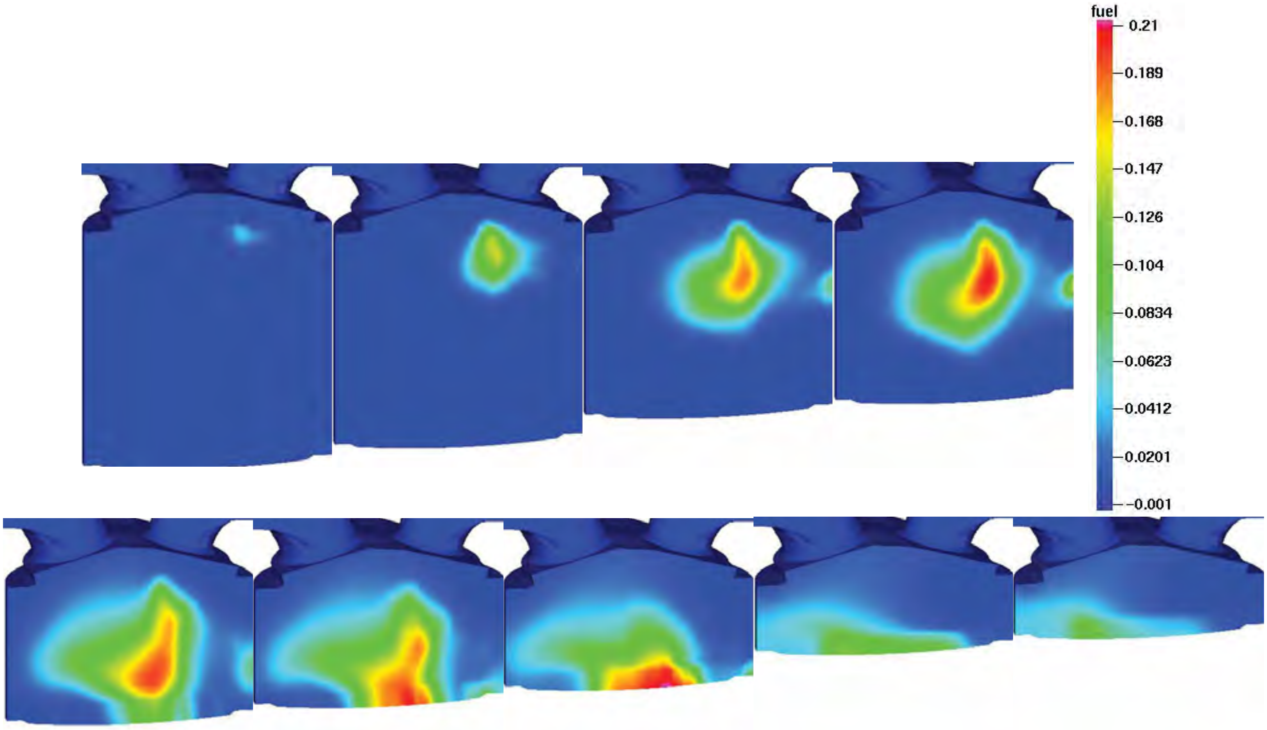

Fig. 2. Simulation No.1 – fuel spray course. Crank angles: 236, 249, 263, 270, 284, 292, 298, 313 and 320 deg

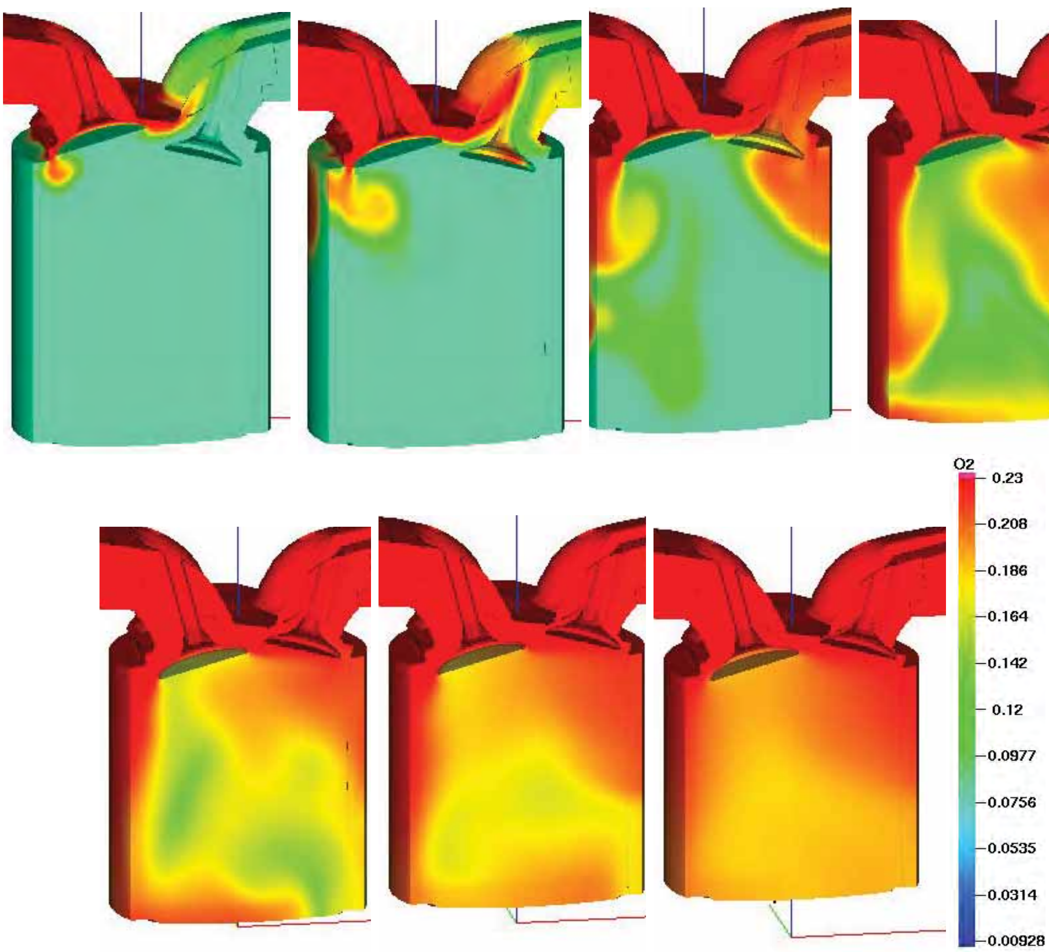

Fig. 3. The scavenging in simulation No. 2. Crank angles: 180, 191, 200, 210, 222, 230,245

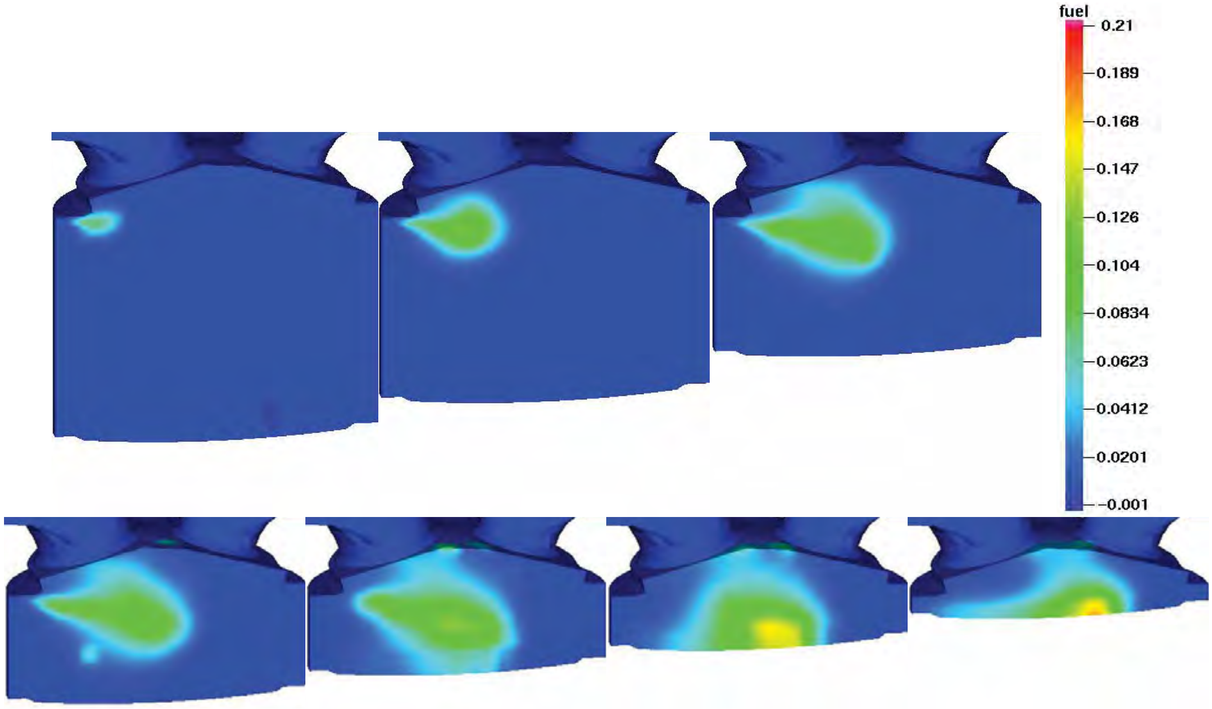

Fig. 4. Simulation No.2 – the course of fuel injection. Crank angles: 268, 280, 295, 302, 312, 324, 344

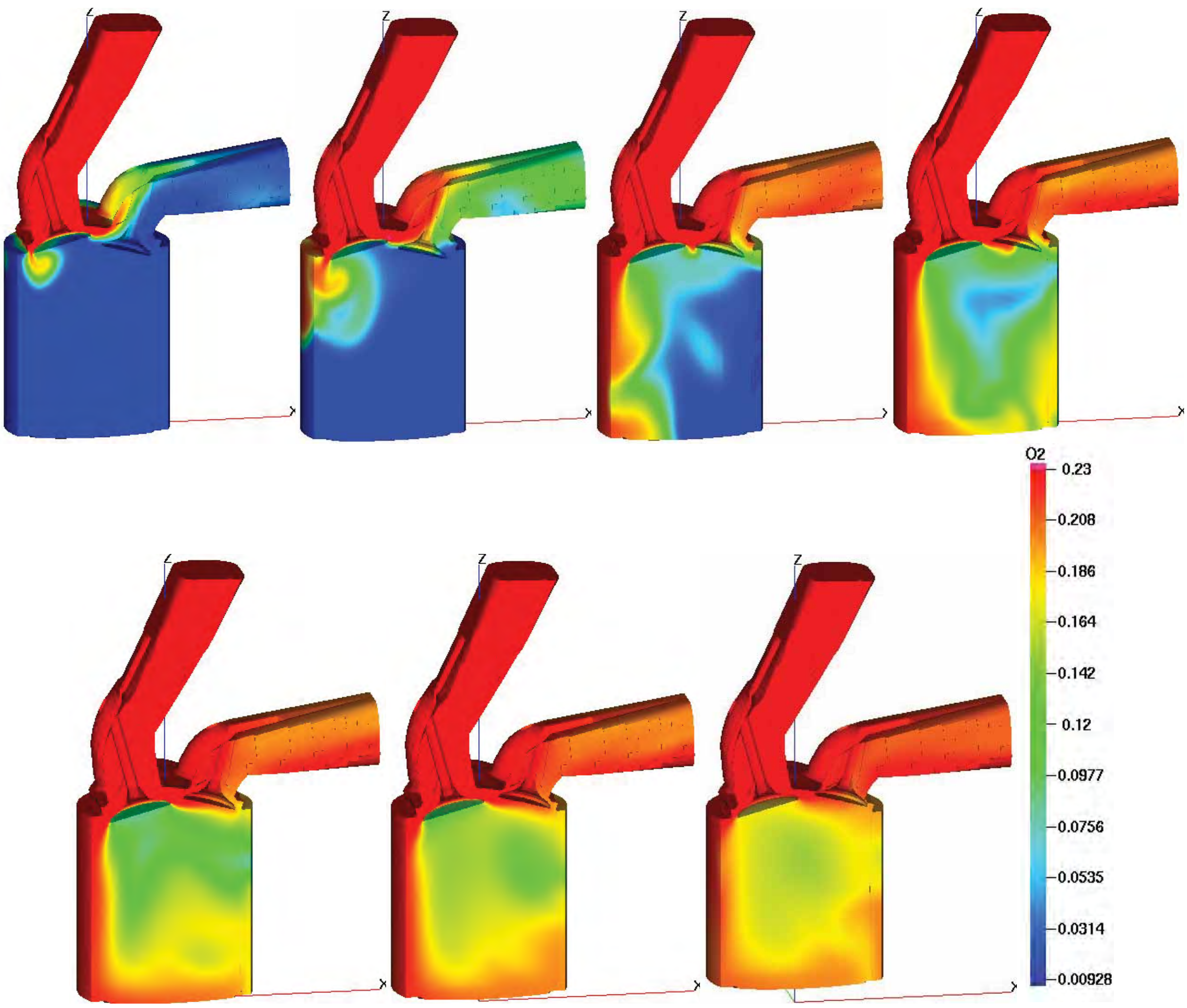

Fig. 5. The gas exchange process – simulation No. 3. Crank angles: 169, 179, 190, 200, 212, 220, 230 deg

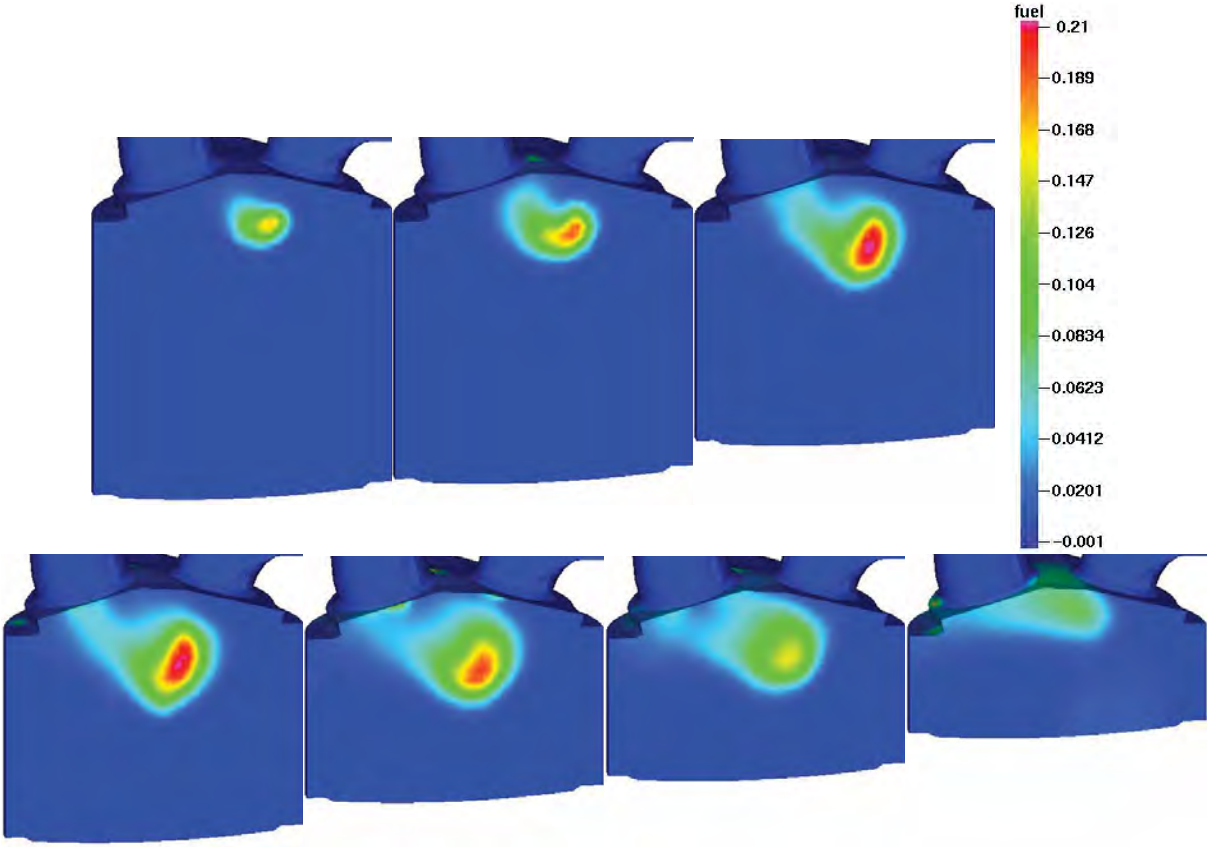

Fig. 6. The course of fuel injection of simulation No. 3. Crank angles: 232, 241, 248, 262, 270, 284, 292, 306

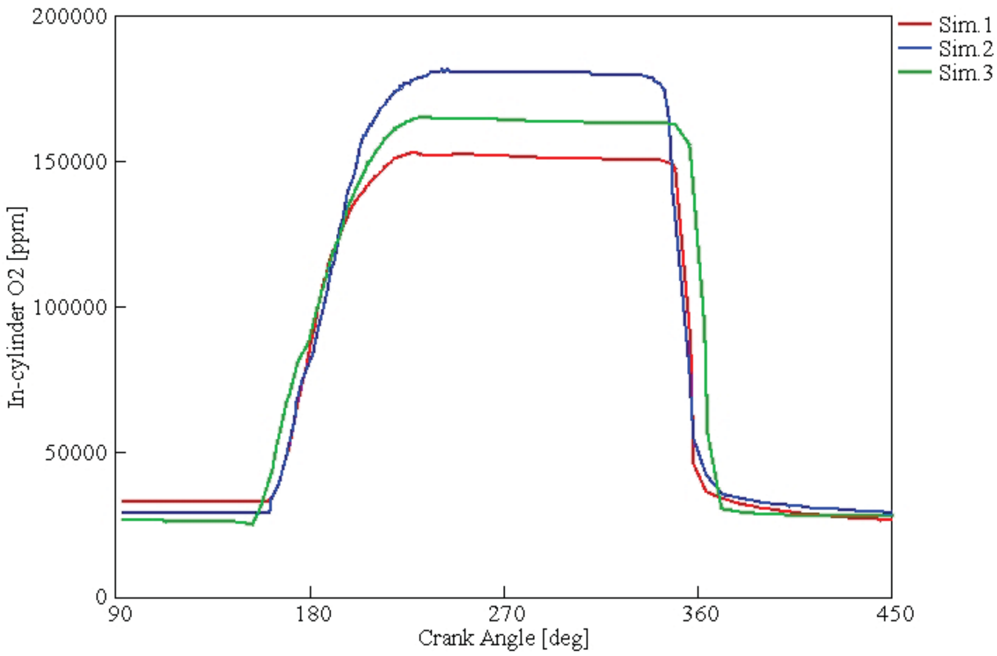

Fig. 7. The in-cylinder contents of O_2 for all performed simulations

The arrangement of inlet ports in the valved two-stroke engine influences significantly on volumetric efficiency as is shown in Fig. 7. The standard configuration of Toyota 2SZ-FE is the worst of the analyzed case, because it decreases amount of oxygen and thus amount of air about 15% in comparison to the second case and about 7% in comparison to the third case. Therefore amount of injected fuel in the second case could be bigger (Fig. 8) for the same air-fuel ratio.

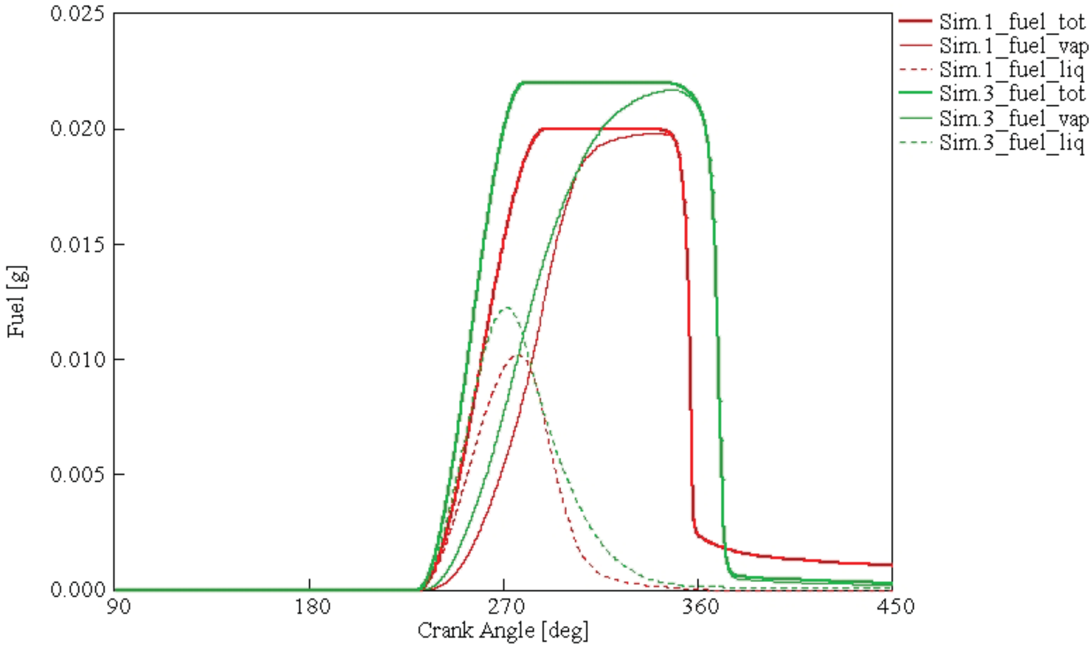

Fig. 8. The fuel spray process: total injected mass, liquid mass and fuel vapours of simulation No. 1 and 3

5. Conclusions

Poppet valve two-stroke engine force some significant changes in intake (and eventually exhaust) ducts commonly used in four-stroke engines because of short-circuiting problems during scavenging. Additionally, required in connection with risk of fuel losses during scavenging process direct injection impose the need of strict determination of spray parameters such as injection position and direction, spray cone shape, fuel pressure and injection timing. Presented results show that top entry ports present good performance of scavenging and mixture preparation process. Nevertheless, authors would like to check also other arrangements of cylinder head elements that seem to improve scavenging course.

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