FORMATION OF FUEL MIXTURE IN A SI TWO-STROKE ENGINE WITH DIRECT FUEL INJECTION

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Abstract

Decreasing of fuel consumption and high hydrocarbon emission in a SI two-stroke engine applied in motorcycles, scooters and small power units is possible by change of fuelling system on direct fuel injection. The main problem of mixture formation is short time for fuel evaporation after injection which should begin after closing of the exhaust port. However well interaction between the scavenge air and fuel jet from the injector located in the cylinder head can induce the small size of fuel droplets and fast evaporation. The paper presents the computational results of fuel mixture formation and combustion process in direct fuel injection two-stroke engine Robin EC-12 with capacity 115 cm^3. The simulation was carried out by use KIVA code with assumption of initial parameters from the experiment of the carbureted engine. Earlier fuel injection influences on better evaporation and mixture formation with small stratification, which enables full mixture combustion. The paper shows phases of mixture formation, combustion and concentration of the combustion product in the cylinder. Spray guided high pressure direct injection system in SI two-stroke engine enables fulfill the restrictions of gas emission of one-wheeler vehicles and small power units. New modern two-stroke engines and environmental protection require new fuelling systems and one of the most promising methods is direct spray guided fuel injection. Many work were done in this subject, however still there is no better option for small power two-stroke engines.

Keywords*: combustion engines, crank drive*

1. Introduction

The exhaust emission of the conventional two-stroke engine has significant influence on environmental protection. In the classical carburettor two-stroke engine there is a considerable short circuiting of the air fuel mixture. In order to decrease hydrocarbon emission a direct fuel injection is applied in modern two-stroke engines. Location of the conventional automotive injectors should enable forming the mixture without any contact of liquid fuel with cylinder and piston walls. The overall gas flow in the two-stroke engine has a significant effect on the motion and evaporation of the fuel spray [7]. It was found that the most important parameters that strongly influence in cylinder droplet vaporisation process and spatial vapour distribution are: fluid flow pattern, injector location and injection timing and injection pressure. The evaluation of the scavenging process has an important role in the simulation process of direct fuel injection.

The CFD tools are preferably applied in simulation of mixture formation in the cylinder in a regard of costs. There is known the possibility of observations of vapour formation during injection by applying the Exciplex method used by Melton[6] or video technique for spray formation used by Ghandhi et al [3]. However the experiments concerned to the stationary condition or very low rotational speed. New computer technique enables to observe the mixture formation after fuel injection during whole engine work cycle. Until now much work has been done in the experimental investigation of atomisation of fuel injection, but numerical modelling of such injection is relatively rare, especially for two-stroke engines. Kuo and Reitz [5] were the first who used computational model of injection and fuel droplet distribution in the cylinder. The two-stroke engine scavenge process depends highly on the overlapping period of the exhaust port and transfer ports. The modelling in KIVA code of fuel spray and mixture formation together with gas motion during the scavenge and compression process was used.

2. Mathematical model of direct fuel injection

Multiphase flows are characterised by two or more fluids in motion relative to each other. The fluids will also usually have different physical properties - temperature, density, conductivity. The dispersed fluid is liquid fuel injected to the cylinder shortly before or after the exhaust port closure. The amount of fresh air with lower value of enthalpy in the cylinder than that of residual gas after scavenge process depends on the ports timing and rotational velocity.

The motion of different phases of charge in the cylinder can be described by general equation:

$$\frac{\partial(r_i \cdot \rho_i \cdot \varphi_i)}{\partial t} + \nabla(r_i \cdot \rho_i \cdot \mathbf{u}_i \cdot \varphi_i - r_i \cdot \Gamma_{\varphi i} \cdot \mathbf{grad}\varphi_i) = r_i \cdot S_{\varphi i}, \tag{1}$$

where:

ρ_i - density of phase i,

φ_i - properties of phase i as: specific enthalpy, specific momentum of gas motion, mass fractions of chemical components, turbulence energy etc.,

$\mathbf{u}_i$ - velocity vector of a phase i,

$\Gamma_{\varphi i}$ - diffusion coefficient of φ value in the phase i,

$S_{\varphi i}$ - source term of exchange φ_i,

∇ - Nable's operator.

In the modelling of the injection processes and gas motion the turbulence model of Chen-Kim (κ-ε) was used [4].

The basic conservation equations for modelling dispersed phase are: the mass transfer, the momentum and the thermal energy transfer equations Motion of fuel droplet with mass m_d and velocity $\mathbf{u}_d$ is expressed in a simple form as follows:

$$m_d \frac{d\mathbf{u}_d}{dt} = D_p(\mathbf{u} - \mathbf{u}_d) - V_d \nabla p, \tag{2}$$

where:

vector $\mathbf{u}$ - represents the charge velocity,

V_d - volume of fuel droplet,

∇p - pressure gradient of the continuous phase.

Drag force function D_p is calculated from the equation:

$$D_p = 0.5 \cdot C_D \cdot \rho_d \cdot A_d \cdot |\mathbf{u} - \mathbf{u}_d|, \tag{3}$$

where:

ρ_d - density of fuel droplet,

Ad - drag area of fuel droplet,

C_D - dimensionless drag coefficient given by:

$$C_D = \frac{24}{\mathrm{Re}}\left(1 + 0,15\,\mathrm{Re}^{0,687}\right) + \frac{0,42}{1 + 4,25 \cdot 10^4\ \mathrm{Re}^{-1,16}} \text{ forRe<300 000.} \tag{4}$$

Disappearing of liquid phase of fuel due to the evaporation can be expressed as follows:

$$\frac{dm_d}{dt} = -A_d \cdot K_g \cdot p_t \cdot \ln\frac{p_t - p_{v,\infty}}{p_t - p_{v,s}}, \tag{5}$$

where:

K_g - mass transfer coefficient,

p_t - total pressure,

$p_{v,\infty}$ - partial pressure in the droplets surroundings,

$p_{v,s}$ - vapour partial pressure at the droplet surface,

The formation of fuel jet including droplet breakup, wall interaction and droplet evaporation was fully described by Amsden [1] and author [7]. During evaporation of fuel heat transfer between continuous and dispersed phases takes place based on the temperature difference as the driving force.

$$m_d \frac{d}{dt}(c_{p,d}T_d) = -A_d \cdot H_d \cdot (T_d - T) + L_d \cdot \frac{dm_d}{dt}, \tag{6}$$

where:

$c_{p,d}$ - droplet specific heat,

L_d - latent heat of evaporation,

H_d - heat transfer coefficient between droplet and gas,

T_g - temperature of continuous phase,

A_d - surface area of liquid droplet per unit volume, which is given by:

$$A_d = \frac{6 \cdot r_2}{d_d \cdot V_{ol}}, \tag{7}$$

The heat transfer coefficient is computed from the local Nusselt number based on the convective coefficient *k*:

$$H_d = \frac{k \cdot Nu}{d_d}, \tag{8}$$

The droplet size varies throughout the domain, as the result of evaporation. The vapour phase behaves like the disperse phase 2, but without interphase mass transfer. Then changes in droplets size can be calculated from local volume fraction ratios:

$$\frac{d_d}{(d_d)_{in}} = \left(\frac{r_2}{r_s}\right)^{0,333}, \tag{9}$$

The value of r_s is the volume fraction of evaporated fuel and $(d_d)_{in}$ is the initial diameter of droplet.

3. Engine mesh generation

Tab. 1. Engine specification with direct fuel injection

Number of cylinder	1
Swept volume, cm^3	115
Bore, mm	54
Stroke, mm	50
Connecting rod length, mm	110
Compression ratio	8
Transfer ports close, °ABDC	57
Exhaust port close, °ABDC	77

The engine used for the present simulation is a single cylinder two-stroke gasoline engine. The piston top was flat and the combustion chamber consists of a hemisphere. Two transfer ports are symmetric about the vertical plane crossing the axis of the cylinder and the centre of exhaust port. The generation of mesh was done by pre-processor KIVA3V and simulation was done by using recompiled version of this programme. Modelling concerned the stationary two-stroke engine Robin EC12 from Heavy Fuji Industries. The mesh layers in z-axis changes during piston motion causing the deformation of cells in this direction. Therefore the appropriate relaxation values were needed to fill Courant conditions especially for velocity vectors by change of time step during calculations.

The specification of the modelling two-stroke engine is given in Tab. 1 and the engine mesh and geometry of location of the injector for the top injection is shown in Fig. 1. In simulation it was assumed that the injector was located in the cylinder head enabling the fuel injection before closing of the injector port by the piston.

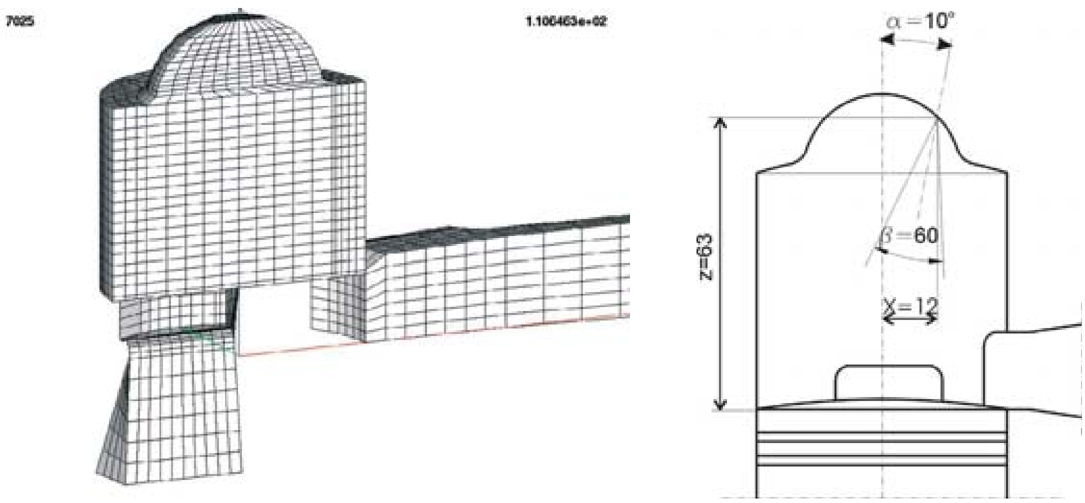


Fig. 1. Mesh of two-stroke engine Robin EC12 and geometry of injector at top position

The model enables change of the mesh during piston motion and simulation concerned the whole engine work cycle.

4. Boundary and initial conditions

The initial values were taken from experimental research and can be taken also from the numerical simulation based on zero-dimensional model. Both transfer ports had the same values of velocity and pressure but changeable in time. Variation of pressure in the crankcase and exhaust port is shown in Fig. 2.

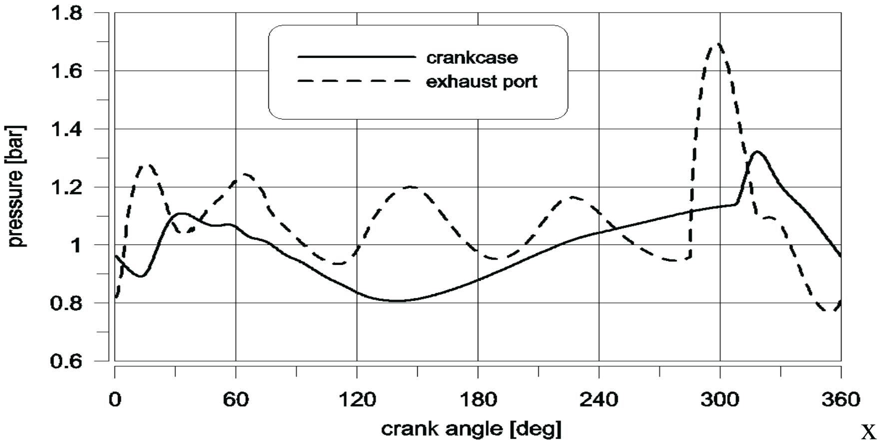


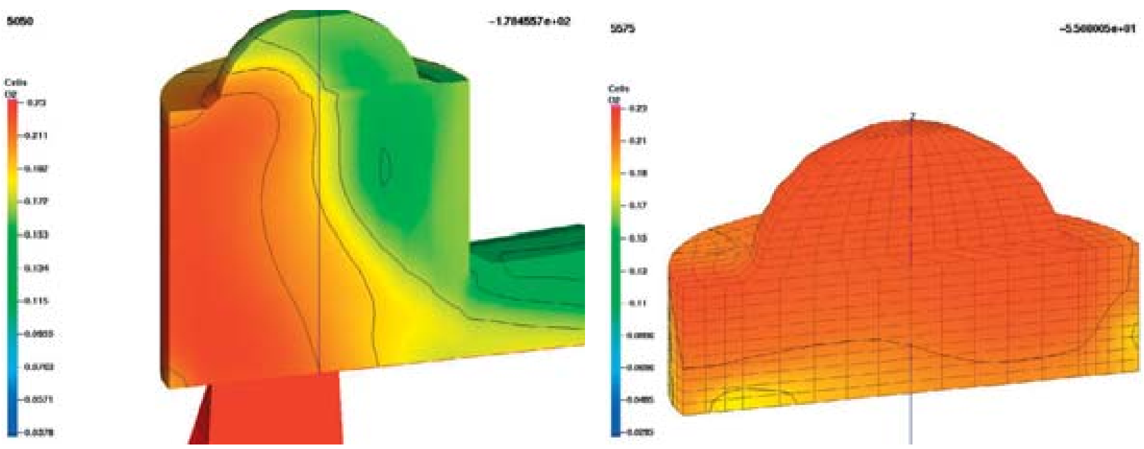
Fig. 2. Pressure in exhaust port and transfer port from experiment at 3000 rpm

The appropriate parameters were given in simulation programme as boundary conditions. The scalar factor for fresh charge (air) is 1.0 at the transfer port and zero for exhaust port. The cylinder walls were assumed to be smoothed and heat exchange with walls was taken into account. The k-ε turbulence model was considered and turbulence initial length scale was given as 0.001 m. The temperature of the air in transfer port was assumed as 300 K.

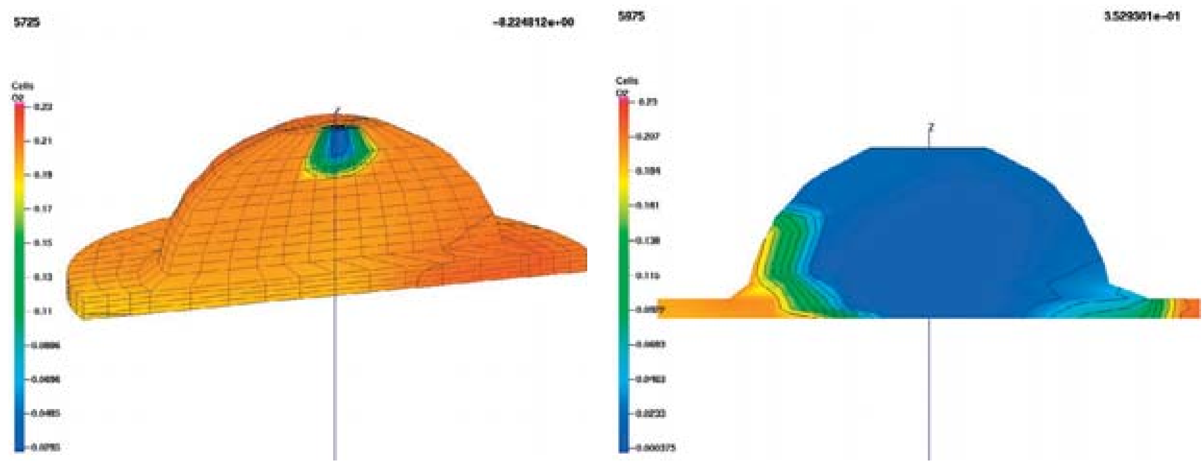
Simulations were carried out for whole work cycle in order to obtain the initial values for simulation of fuel injection, spray and mixture formation. Modelling of injection process was conducted only for one rotational speed 3000 rpm. In calculation equal initial values of fuel droplet 0.025 mm were assumed.

5. Calculation results

Computational results of the DFI model provide three-dimensional information of gas flow velocities, fresh charge fraction, pressure and density distribution, turbulence as well as fuel droplet position, size, flow direction and fuel vapour fraction. Combustion of the injected fuel depends on local air-fuel ratio and the scavenge process has significant effect on the air distribution in the cylinder. After scavenge process in the cylinder stay much residual exhaust gas particularly at high rotational speed. However, much exhaust gas causes bigger temperature of the charge which helps to evaporate of injected fuel. The mass ratios of oxygen in the cylinder are shown in Fig. 3 for different piston positions.



a) b)



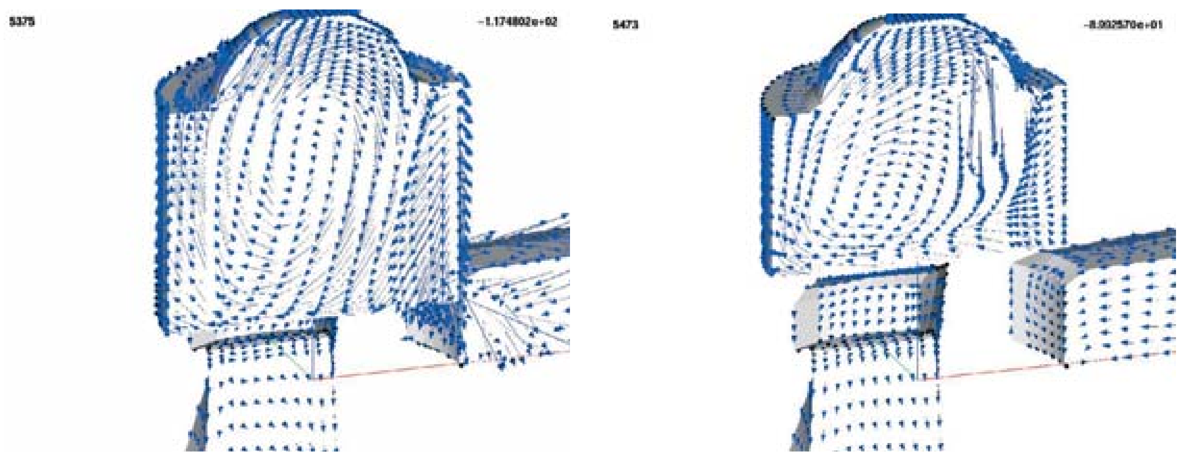
c) d)

Fig. 3. Mass concentration of oxygen in the cylinder at piston positions: a) BDC, b) 55 deg BTDC, c) 8 deg BTDC and d) TDC

Most of residual gas stays near the combustion chamber and because of higher temperature has an effect on quicker evaporation of fuel. At higher speed the start of injection should be earlier in order to evaporate the fuel in the same time as for lower speed. Distribution of oxygen at start of ignition is almost uniform in whole space of the combustion chamber.

The injection parameters were given as initial values: injection pressure - 20 bar, initial fuel velocity - 50 m/s and angle of injection cone - 60-70°. The scavenge process and piston motion take effect on the forming of the charge motion in clock-wise direction in the combustion chamber called as „tumble". The gas motion influences on a local fuel distribution during fuel injection. Velocity vectors of continuous phase of injection are shown in Fig. 4 for different piston positions. During the fuel injection process there is an interaction of the fuel and gas in the combustion chamber, that is seen in Fig. 4 a) and 4b). Fuel injection started after closing of the exhaust port. After fuel injection the charge in the combustion chamber is still in a "tumble" motion, which enables better propagation of the fuel spray and evaporation in the gaseous charge.

During top injection in the cylinder the strong tumble of the charge in the parallel planes to the z-axis takes place and turns the fuel spray directly to the piston and causes the break of the fuel droplets. In this case shortly before the ignition, the maximum of velocity reaches 30 m/s. Compressed clock-wise tumbling flow is formed, which may assist fuel droplet evaporation and also the combustion process



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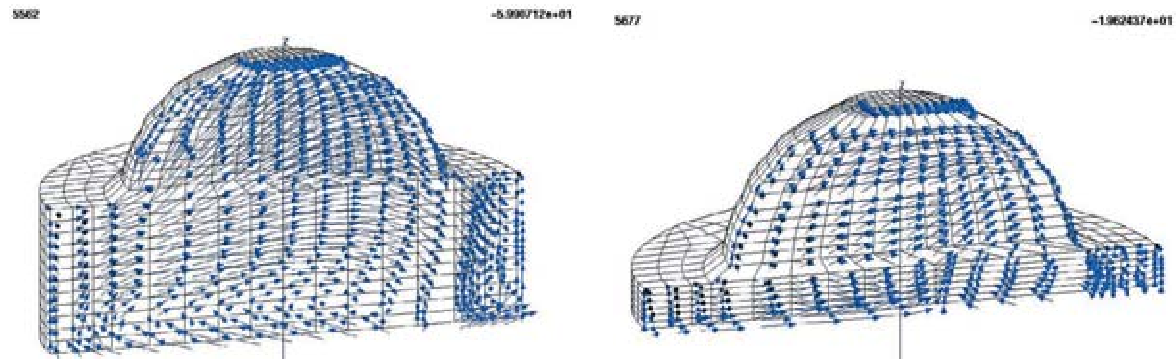
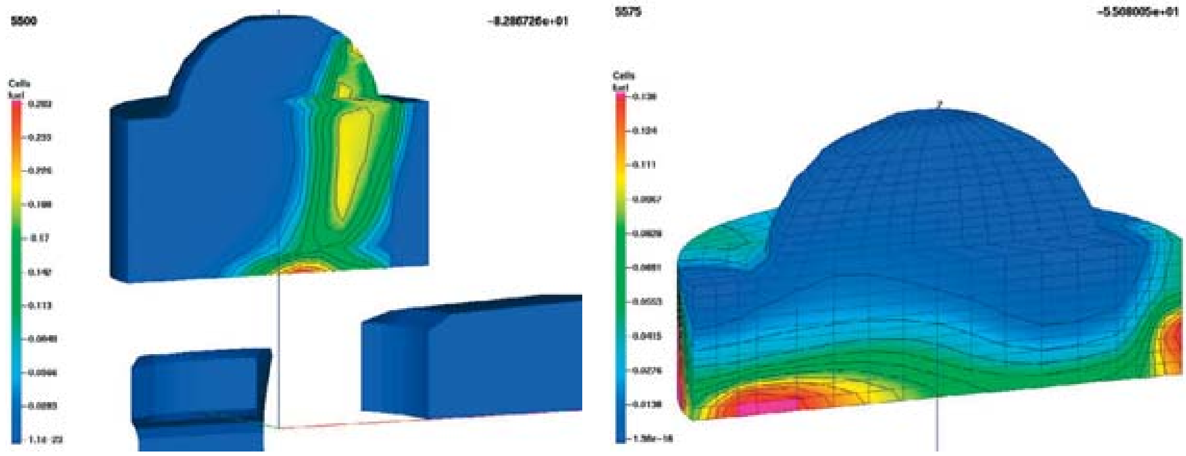


c) d)

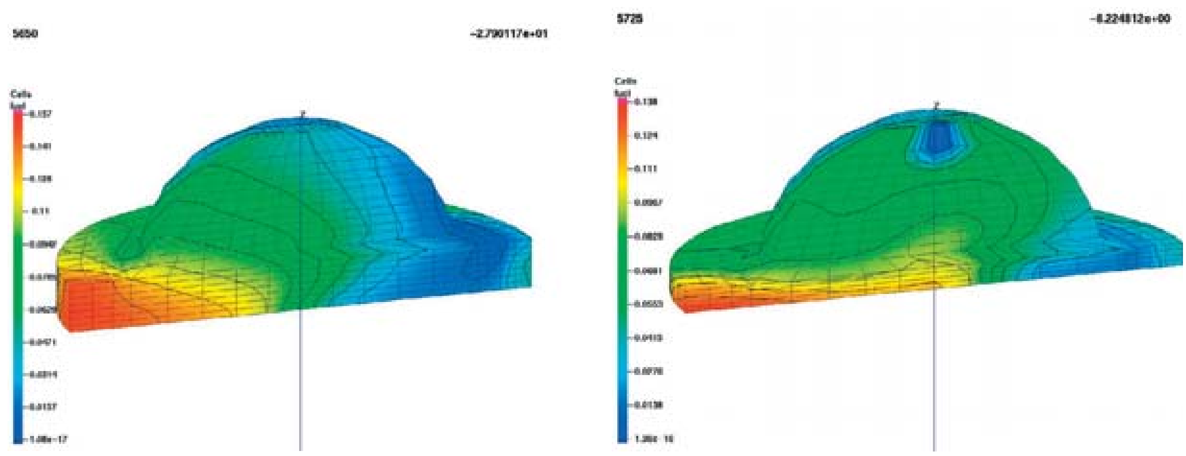
Fig. 4. Velocity vectors in cylinder at piston positions: a) 117 deg CA BTDC, b) 80 deg CA BTDC, c) 60 deg CA BTDC and d) 20 deg CA BTDC

The slides in Fig. 5 a) and 5 b) represent the sectional contour plots of the mass fuel gaseous phase concentration for piston positions at 83 deg CA BTDC and 55 deg CA BTDC, respectively. The bigger initial velocity of injected fuel causes a narrow spray in the case of top injection. The fuel vapours reach the piston crown and next the gas tumble causes their propagation in the chamber. Most of gaseous fuel phase is concentrated near piston and next as a result of squish effect this phase moves into the combustion chamber. The distribution of mass ratio of fuel vapours near TDC is shown in Fig. 6 a) and 6 b).



a) b)

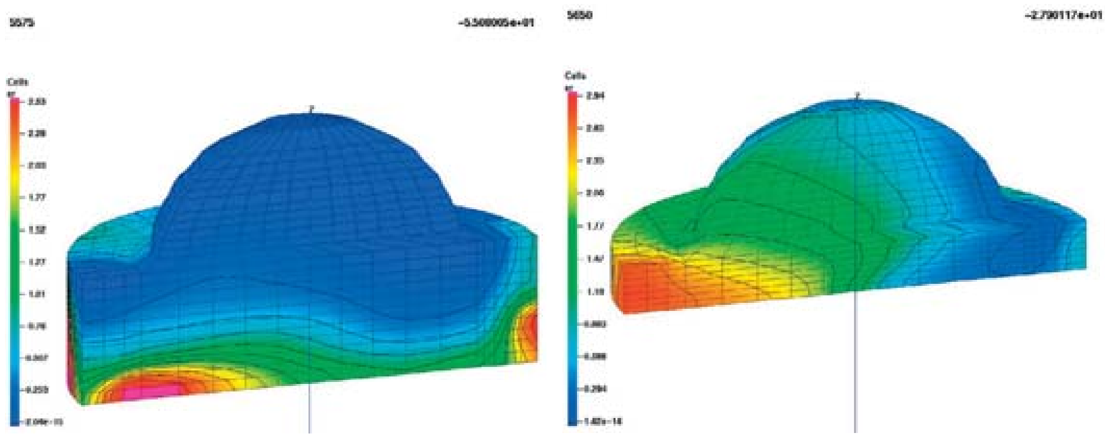
Fig. 5. Mass fraction of vapour phase at piston positions: a) 83 deg CA BTDC, b) 55 deg CA BTDC



a) b)

Fig. 6. Mass fraction of vapour phase at piston positions: a) 28 deg CA BTDC and b) 8 deg CA BTDC

Very important factor at direct fuel injection is the air-fuel ratio near the spark plug. At TFI case the mass concentration of vapours is higher at opposite side of the exhaust port and reaches air excess ratio near one ($\lambda = 1$). This phenomenon is caused by the gas motion that does not enable to propagate of the fuel into the chamber. When the combustion process begins the liquid phase quickly evaporates. Fig. 7 shows the distribution of fuel/air equivalence ratio at piston positions: 55 and 28 deg CA BTDC. These graphs show the charge stratification with possible ignition of the mixture in the region of the spark plug.



a) b)

Fig. 7. Fuel/air equivalence ratio at piston position: a) 55 deg CA BTDC, b) 28 deg CA BTDC

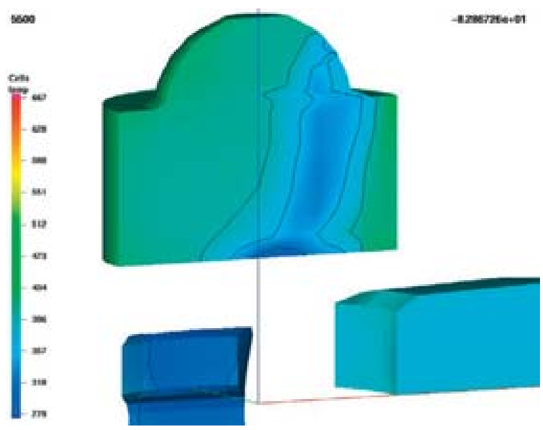


Fig. 8. Temperature in the cylinder at 83 deg CA BTDC

Temperature of gas phase changes during motion of fuel spray and evaporation and its distribution inside the cylinder is shown in Fig.8. Injected fuel lowers the temperature of continuous phase. Temperature distribution inside the cylinder determines the process of fuel evaporation. At top injection most of fuel evaporates quickly after leaving of the nozzle and fuel vapours reach the piston crown. Only small amount of liquid fuel stay on the piston. Fuel evaporation decreases the charge temperature about 100 K. Shortly before the ignition temperature inside the cylinder is about 750 K and ignition of the air-fuel mixture depends on a local excess air ratio. In two-stroke engines with direct fuel injection there is a short time for fuel evaporation. Total amount of injected fuel was assumed as 0.55 g/cycle and fuel evaporates fully at 60 deg CA BTDC (Fig. 9). For the assumed initial conditions and geometrical parameters the top fuel injection system (TFI) is optimistic for evaporation and full burning of the fuel.

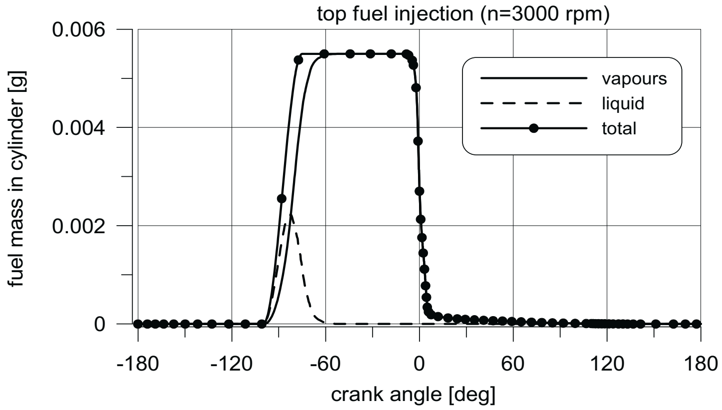


Fig. 9. Mass of liquid and vapours of fuel in cylinder at top injection

6. Conclusions

1) New modern two-stroke engines and environmental protection require new fuelling systems and one of the most promising methods is direct spray guided fuel injection. Many work were done in this subject, however still there is no better option for small power two-stroke engines.
2) High pressure direct fuel injection enables to form smaller fuel droplets in the spray and quick fuel evaporation as a result of interaction of charge and fuel,
3) The tumble motion of the gas is favourable for fuel droplets break-up, mixing and propagation inside the combustion chamber for top fuel injection,
4) The excess air ratio near the spark plug is enough for ignition,
5) With direct fuel injection the charge in the combustion chamber is stratified and the local air-fuel ratio changes at time, which is caused by gas tumble,
6) The location of injector is one of important factors influencing on the mixture formation.

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