ANALYSIS OF DIRECT FUEL INJECTION IN A SMALL POWER TWO-STROKE ENGINE

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Abstract

Small power two-stroke engines can be competitive in regard of work parameters and pollutants emission in relation to four-stroke engines at some modifications, particularly changing of fuelling system. The paper presents a modified air-assisted direct fuel injection in a two-stroke engine, which changes propagation of fuel jet in the combustion chamber. The main problem of mixture formation is short time for fuel evaporation after injection which should begin after closing of the exhaust port. The paper shows interaction between the scavenge air, additional air stream and fuel jet from the injector located in the cylinder head, which can induce the small size of fuel droplets and fast evaporation. 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 are presented. The simulation was carried out by use KIVA code with assumption of initial parameters from the experimental tests on the carburetted engine. Setting of the injector in a regard of the cylinder head decides of fuel motion and evaporation in the combustion chamber. Air-assisted fuel injection influences on higher mixing of fuel and better combustion than by using DFI system. The paper shows phases of mixture formation, combustion and concentration of the combustion product in the cylinder with additional air stream. Spray guided high pressure direct injection system in SI two-stroke engine enables fulfil the restrictions of gas emission of one-wheeler vehicles and small power units.

Key words: *transport, engine development, two-stroke engine, direct fuel injection*

1. Introduction

In the classical carburettor two-stroke engine there is a considerable short circuiting of the air fuel mixture. Direct fuel injection reduces hydrocarbon emission with proper design of the injection timing and the positioning of the injector. The overall gas flow in the two-stroke engine has a significant effect on the motion and evaporation of the fuel spray. It was found that the most important parameters that strongly influence in cylinder on droplet vaporisation process and spatial vapour distribution are: fluid flow pattern, injector location, injection timing and injection pressure. The scavenging process and squish effect play an important role in the simulation process of direct fuel injection. Unlike the four-stroke engine in which the gas exchange occurs in the exhaust stroke, the two-stroke engine scavenging depends highly on the overlapping period of the exhaust port and transfer ports. 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. The maximum reduction of HC emission can be reduced by air-assisted cylinder head fuel injection or by direct fuel injection with additional stream of air from low pressure compressor [3, 4]. This solution can minimize the size of droplets and their quicker evaporation. Each solution of direct injection system cannot enable full reduction of HC emissions, because of irregular value of air-fuel ratio, changing with engine load and speed. Fuel droplets after injection have certain value of kinetic energy depended on the mass and velocity, which is function of pressure difference in the rail and in the cylinder. The air in the cylinder after scavenge process during compression process is in “tumble” motion, which affects on deflection of the fuel spray. The air forces on the droplets and vapours and the spray penetration is quite different than during experimental test in the chamber in steady state. The additional air stream during injection process can force the fuel stream in the centre of the combustion chamber and there is a big probability that fuel droplets do not reach the cylinder walls and piston crown. The big effect on the behaviour of droplets has turbulence described by Chen and Kim[2], viscosity of fuel, surface tension, velocity, temperature and other parameters affected on collision, break-up, rebounding and evaporation[7].

2. Air-assisted fuel injection

For the cylinder head direct fuel injection (DFI) presented in paper [8, 9], after loop scavenge process, the air causes deflection of fuel spray and fuel vapours into the side of the exhaust port. Next "tumble" motion is affected on spreading of fuel near the spark plug. Direct fuel injection causes stratification of the mixture and the local air-fuel ratio depends on the start of injection, location of the injector and angle of the spray. This work is continuation of the previous achievements of authors by the change of the fuel spray penetration. In proposed solution an additional air stream from the compressor at lower pressure is led to the combustion chamber from the same side of the injector. The diagram of the injection system with air spray is shown in Fig.2.

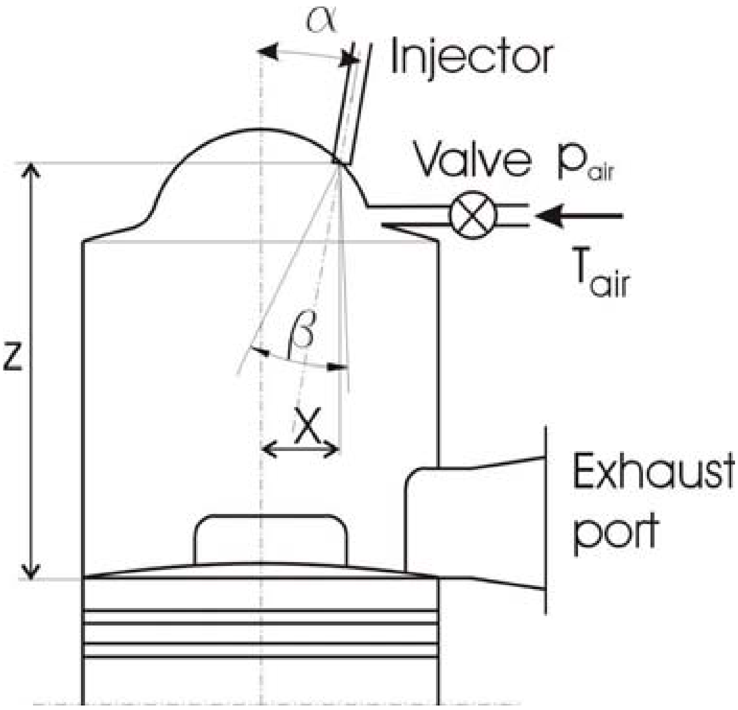


Fig. 1. Air-assisted fuel injection system in two-stroke engine

Injection of fuel is synchronized with injection of air in order to achieve an effect of braking-up the fuel stream and its deflection in the centre of the combustion chamber. The process of supplying of the air is controlled by the electronic valve in strictly defined period and the air is delivered from the piston compressor driven by the crankshaft with pressure above 5 bar. Electronic controlling of the air flow enables breaking-up of the fuel stream and its deflection. Smaller fuel droplets can quicker evaporate, which prevents of their splashing on the walls and forming a fuel film. Location of the injector is the same as in previous solution with angle β=20^{o} and x=12 mm. The standard industry two-stroke engine Robin EC12 with capacity 115 cc and with modified fuelling system was taken into consideration. Air-assisted injection system developed by Orbital [3] as one unit was applied in many engines enabling mixing of fuel air in axial direction. Proposed solution enables additionally deflection of the fuel stream by moving the droplets and fuel vapours into centre of the combustion chamber.

3. Spray motion

The effects of swirl on the fuel jet penetration should be considered that jet of fuel is deflected by pressure and viscous forces. Sinnamon, et al. [10], presented the model based on balance of mass continuity and momentum for a steady-state gaseous jet. The drag forces cause the deflection of jet. Motion and deformation of fuel spray in the combustion chamber as a result of gas motion (swirl or tumble) is shown in Fig.2.

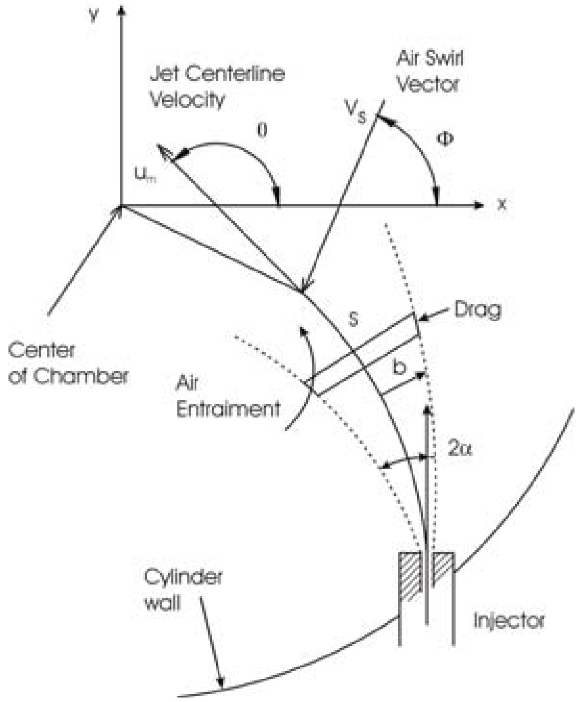


Fig. 2. Geometry of a spray model

The model is based on the following set of equations:

$$\frac{d}{ds}\left(\int_0^{r_0} \frac{m_f}{m} \rho \nu 2\pi r\, dr\right) = 0, \tag{1}$$

$$\frac{d}{ds}\left(\int_0^{r_0} \rho \nu 2\pi r\, dr\right) = \left[\rho(r=0)\rho_\infty\right]^{1/2} 2\pi r_0 \left[\alpha\left|\nu - V_t\right| + \lambda\left|V_n\right|\right] = \frac{d\dot{m}}{ds} \quad , \tag{2}$$

where:

- ν - velocity along the jet axis,
- θ -angle the spray forms with the horizontal (x) direction,
- V_t - air velocity component parallel to the jet axis,
- V_n - air velocity component normal to the jet axis,
- ρ_∞ - air density outside the jet,
- $\dot{m}$ - air entrainment rate,
- s - coordinate along the jet,
- α - adjustable axial entrainment parameter,
- λ - adjustable normal entrainment parameter.

Equation (1) shows that the fuel flow rate at any cross section of the jet is constant, whereas equation (2) expresses the conservation of the total mass of air and fuel. Conservation of linear momentum in the horizontal and vertical directions is written in the following form:

$$\frac{d}{ds}\left(\cos\theta \int_0^{r_0} \rho \upsilon^2 2\pi r\, dr\right) = \nu \cos\phi \frac{d\dot{m}}{ds} + F_d \cos\phi\, , \tag{3}$$

$$\frac{d}{ds}\left(\sin\theta\int_0^{r_0}\rho\upsilon^2 2\pi r\,dr\right)=\nu\cos\phi\frac{d\dot{m}}{ds}+F_d\sin\phi\,, \tag{4}$$

where:

$$\nu-V_t=\left(\nu_{\max}-V_t\right)\left[1-\left(\frac{r}{r_0}\right)^{1,5}\right]^2,$$

where: F_d - drag force.

The first term on the right-hand sides of eq.(3) and (4) accounts for the jet momentum changes due to entrainment. Local concentration in the jet is:

$$\frac{c}{c_{\max}}=1-\left(\frac{r}{r_0}\right)^{1,5}. \tag{5}$$

Local density is calculated as follows:

$$\rho=\frac{\rho_\infty}{1-\frac{c}{c_{\max}}\left(1-\frac{\rho_{\max}}{\rho_f}\right)}, \tag{6}$$

where: ρ_f is density of fuel.

The equations (1)-(6) yield ordinary differential equations for $\nu_{\max}$, c_{max}, r_0 and θ. Later these values can be inserted into equations (5) and (6) in order to obtain the velocity and concentration distributions in the fuel jet.

4. Calculation model

Calculation of mixture formation of two-stroke engine was carried out in CFD program Kiva3v written in Fortran language. Creation of CFD model demands the prescription concerning the computational grid structure, fluid properties, boundary conditions and numerical solution procedures. Kiva3v program was created for simulation thermodynamic processes for piston engines in National Laboratories in Los Alamos by group leaded by Aamsden[1]. Unfortunately not all of parameters can be defined with confidence as some are found from experimental tests or from arbitrary choices. The special iterative procedures are used in order to obtain stability and numerical convergence of finding unknown variables in differential equation systems. The program takes into account the mathematical model described above for finding trajectory of fuel droplets and fuel vapours. The air fuel ratio in region of the spark plug should be in the range of burning of gasoline during ignition. Mesh of the engine for spatial simulation of engine thermodynamic parameters is shown in Fig. 3.

The air was delivered to the inlet under pressure 6 bar and temperature 350 K. Beginning of fuel injection was assumed at 100 deg CA BTDC and lasted 55 deg CA with total mass 0.005 g/cycle. On the other hand the air was delivered to the combustion chamber at 98 deg CA BTDC and the air valve closed at 50 deg CA BTDC.

5. Spatial distribution of parameters

Simulation of engine parameters was carried out by modification of the source code of the program Kiva by adding additional procedure for the air injection at given period. Calculations of whole thermodynamic processes in the direct fuel injection two-stroke engine were carried out for different engine rotational speeds. In the paper one show only the chosen results of fuel stream propagation, temperature distribution and influence of charge velocity caused by scavenge process and additional stream of the air on the fuel jet. Result of CFD simulation concerns to engine work at 3000 rpm with one nozzle injector and assumed 1500 droplet parcels. The initial parameters (pressure and temperature) in the crankcase and transfer ports were taken from experiment on the real carburetted engine. At outlet port it was assumed the ambient pressure without reverse flow of gas.Propagation of fuel vapours after injection is shown in Fig. 4 a) and b) for piston position at 83 deg and 71 deg CA BTDC, respectively. In comparison to the work [8] and [9] the fuel jet is deflected in the centre of the combustion chamber.

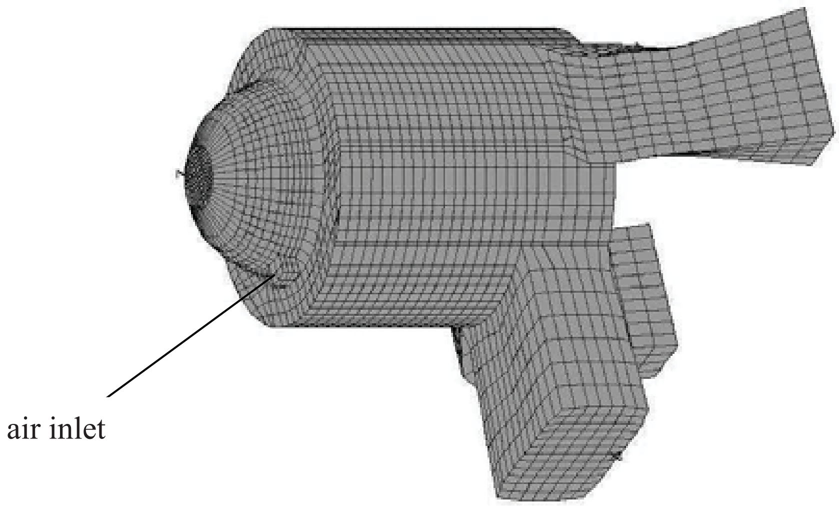
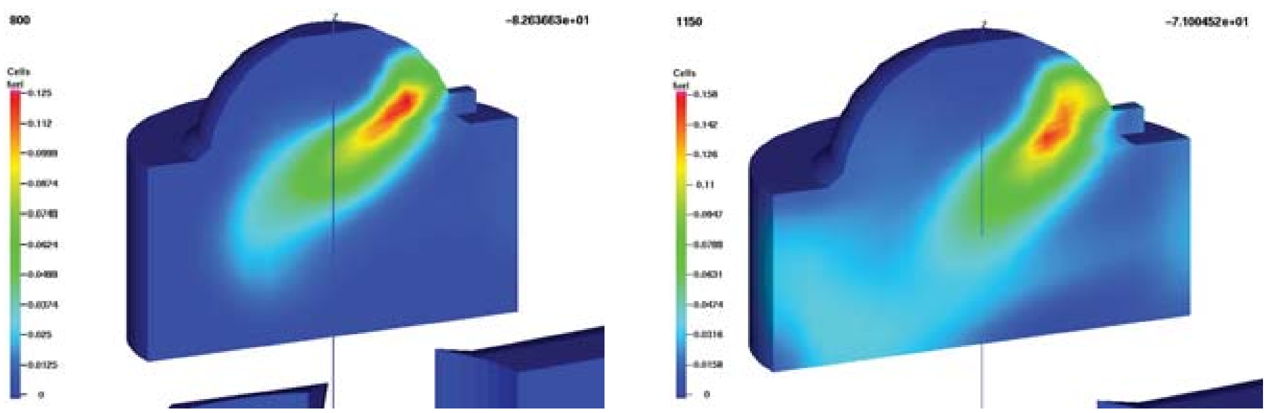


Fig. 3. Computational mesh of two-stroke engine Robin EC12

Further development of fuel stream in the combustion chamber at different crank angle positions of piston is shown in Fig. 5 at a) 63 deg CA BTDC; b) 25 deg CA BTDC; d) 21 deg CA BTDC, respectively. Before ignition higher concentration of fuel vapours takes place near the wall on the opposite side of the injector. At the end of combustion process a small amount of fuel vapours stays on this side in the squish region.



a) b)

Fig. 4. Mass ratio of fuel vapours at a) 83 deg CA BTDC and b) 71 deg CA BTDC

Velocity of the air stream depends on pressure difference in the compressor and cylinder. Despite the small pressure difference in the air inlet velocity of gas can reach value of the sound speed. Absolute velocities in the cylinder are shown in Fig.6 at piston positions 83 and 21 deg BTDC and one can see a big velocity change at the volume connected to the air inlet.

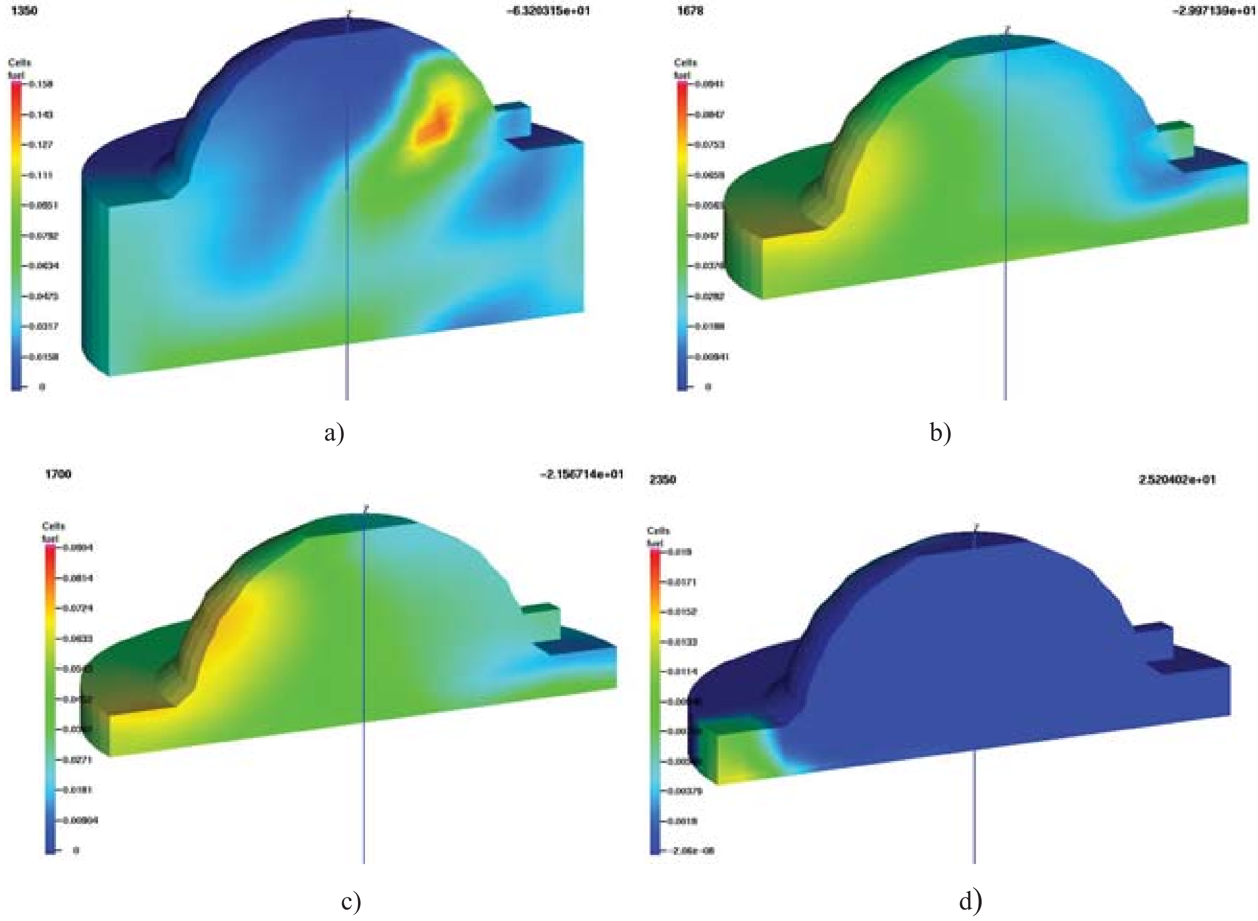


Fig. 5. Mass ratio of fuel vapours at a) 63 deg CA BTDC; b) 25 deg CA BTDC; d) 21 deg CA BTDC and d) 25 deg ATDC

Additional air with low temperature causes a decrease of the charge temperature, however near the spark plug temperature changes a little in comparison with previous solution. Temperature distribution in the cylinder for two piston positions 83 deg CA BTDC and 63 deg CA BTDC, respectively, is shown in Fig. 7 a and 7b.

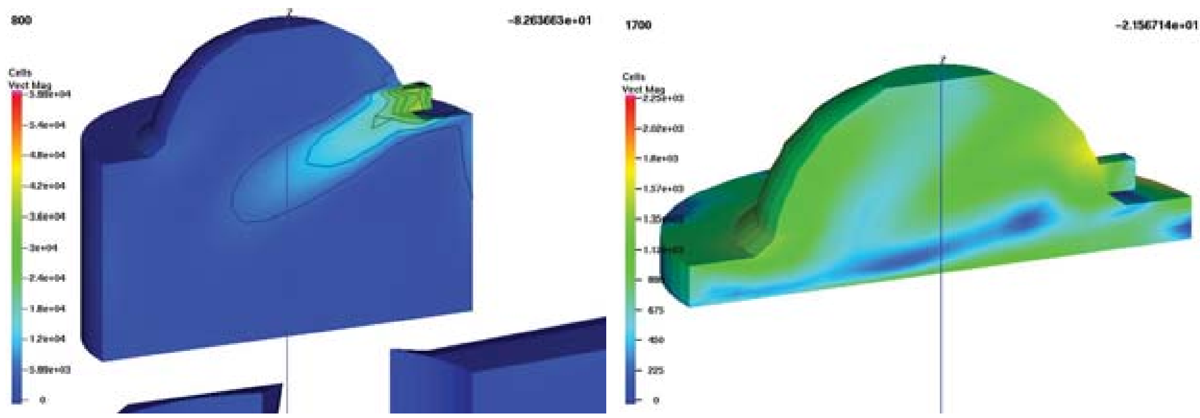
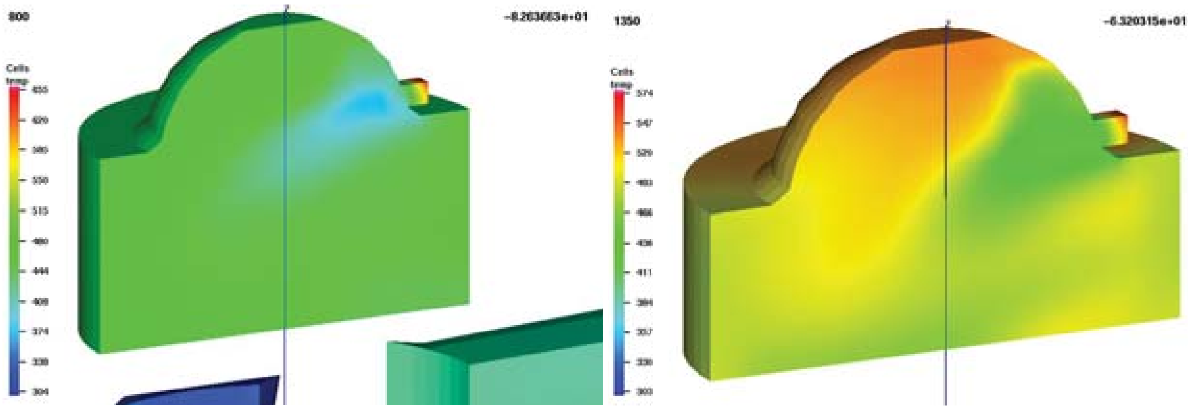


Fig. 6. Absolute velocities of charge [cm/s] at piston position 83 and 21 deg CA BTDC

6. Mean engine parameters

The engine with air assisted fuel injection proves high cylinder pressure (47 bar) and also high value of the maximum temperature (2400 K). Additional air leans the mixture and there is possibility to burn the whole fuel dose, because mixing of the air with fuel causes a higher access of fuel droplets to the air. Variation of cylinder pressure and temperature in a function of crank angle is presented in Fig.8. During scavenge process temperature decreases to 400 K.



a) b)

Fig. 7. Temperature distribution at piston position: a) 83 deg CA BTDC and b) 63 deg CA BTDC

The proposed air-assisted fuel injection system enables an increase of the mass charge (mostly air) after the exhaust port close, when a big amount of fresh air escapes to this port. Air injection causes the total mass increase from 0.088 g to 0.105 g, which means about 19%. Variation of total charge mass of the charge in the cylinder is shown in Fig.9. The results are taken only from one simulation and therefore the start value are not equalled the end value.

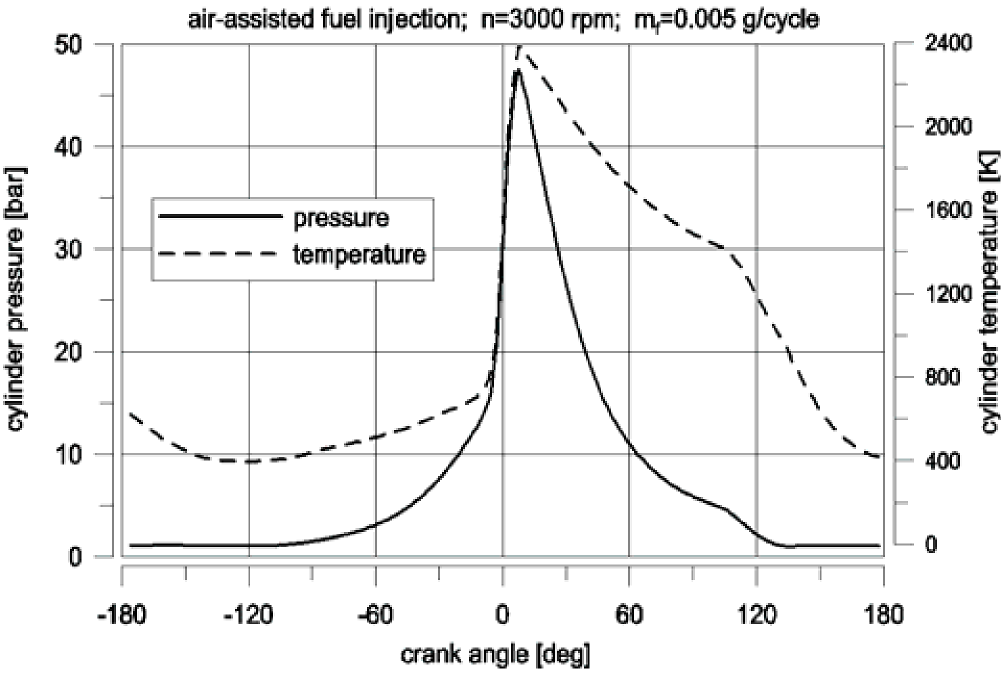


Fig. 8. Pressure and temperature in cylinder with air-assisted fuel injection

For modified direct fuel injection system with additional air stream a well mixing of fuel droplets with air is occurred. This phenomenon influences on better fuel evaporation than in the cylinder with DFI system. Fig. 10 shows mass of evaporated fuel and liquid fuel during whole engine cycle. Only small part of liquid fuel stays in the cylinder for a very small period about 40 deg CA rotation.

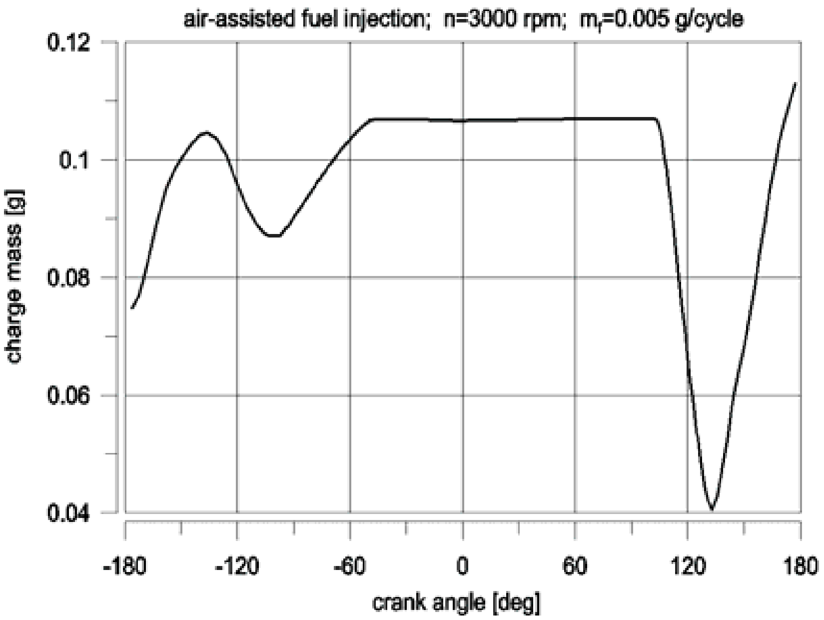


Fig. 9. Change of charge mass in cylinder with air-assisted fuel injection

The air stream with high velocity causes higher turbulence in the combustion chamber and lower turbulence length scale in comparison with engine equipped with DFI system. Higher turbulence influences on the quicker break-up of droplets and mixing and directly also on the combustion process. Comparison of kinetic turbulence energy for engine with the air-assisted system and DFI system is presented in Fig.11. One can see higher energy in the cylinder after opening of air valve and in whole cycle kinetic turbulence energy is higher than in engine with DFI system.

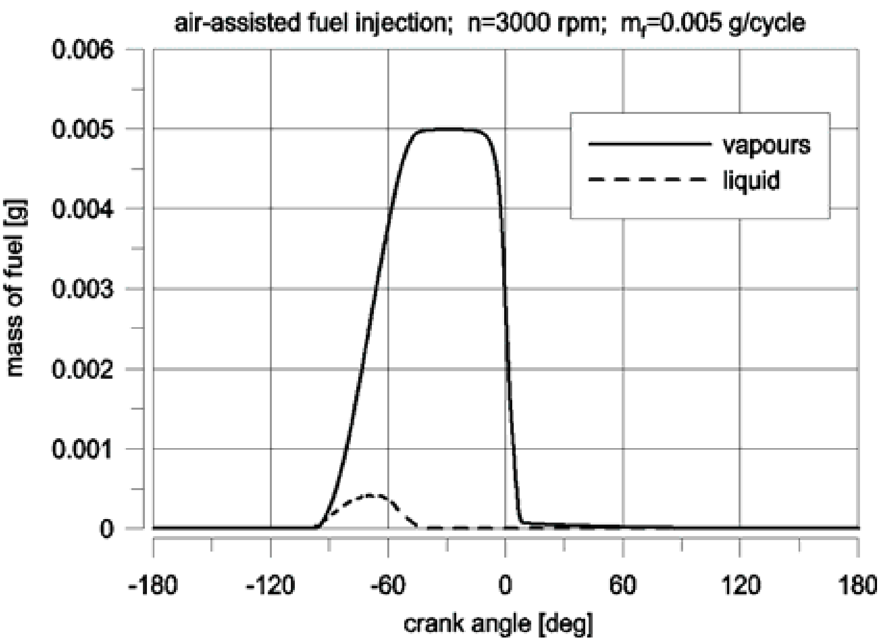


Fig. 10. Variation of liquid and vapour mass of fuel in cylinder with air-assisted fuel injection

Theoretical study of air-assisted fuel injection shows also a big decrement of hydrocarbons level in the cylinder after opening of the exhaust port (near 0-level) and low value of carbon monoxide, which is presented in Fig.12. This means that additional stream of the air can help to fully oxidize the fuel during combustion process.

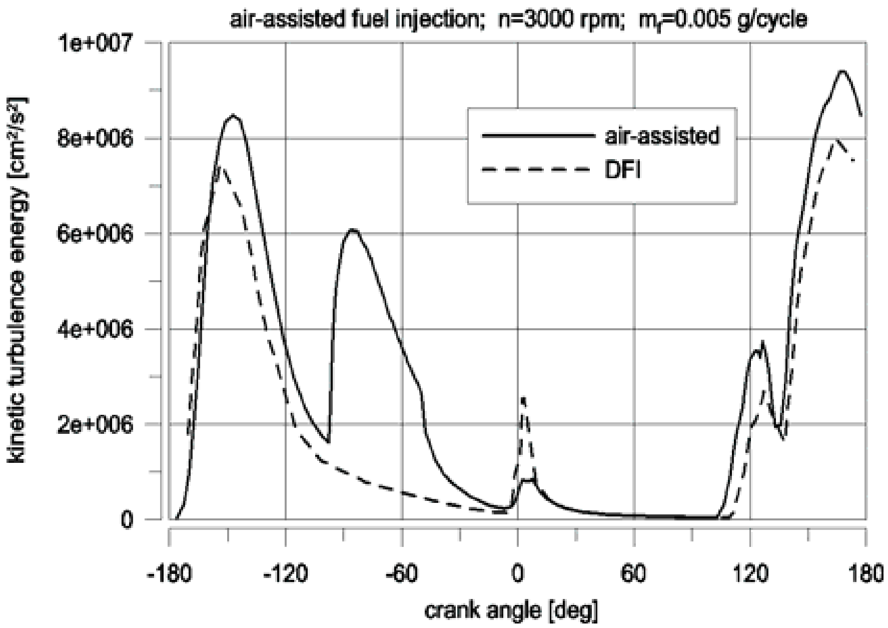


Fig. 11. Kinetic energy of turbulence in cylinder for air-assisted fuel injection and direct fuel injection (DFI)

In conventional two-stroke engines emission of NOx is very low as a result of the residual gases after previous cycle and EGR ratio sometimes reaches value 0.3. For the presented solution NO reaches in the cylinder high level about 4000 ppm, however in the exhaust port maximum mass concentration does not exceed 1000 ppm during engine cycle. Variation of CO_2 and NO in the cylinder is shown in Fig. 13.

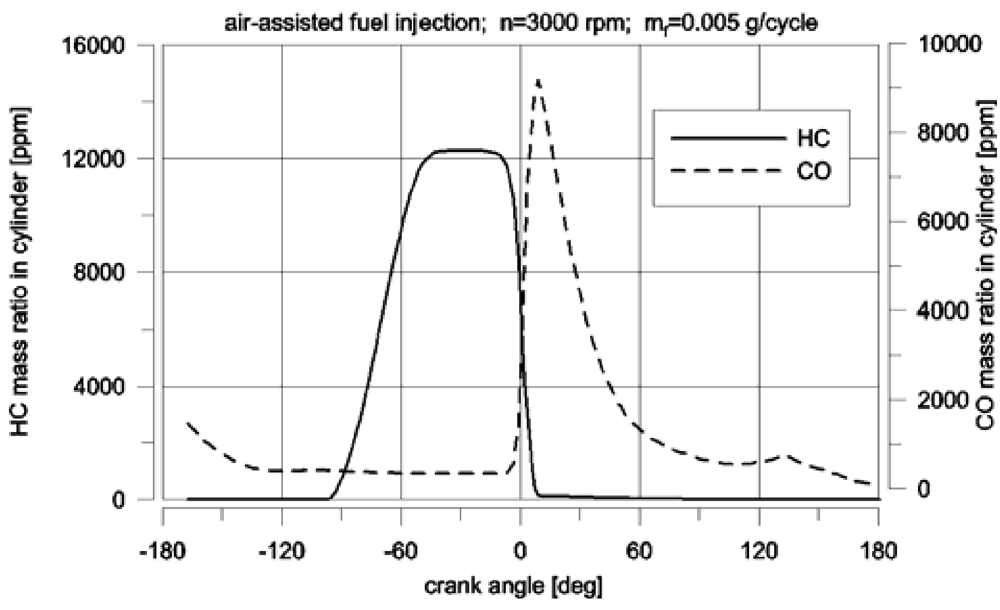


Fig. 12. Variation of mass fraction of HC and CO in cylinder

7. Conclusions and remarks

A new version of air-assisted direct fuel injection with side interaction of the air on the fuel spray is presented in the paper. For modern two-stroke engines, which have to fulfil environmental protections, the proposed fuelling system is one of the most promising solutions. This system can be classified as direct spray guided fuel injection. On the basis of carried out analysis by using CFD the following conclusions can be drawn:

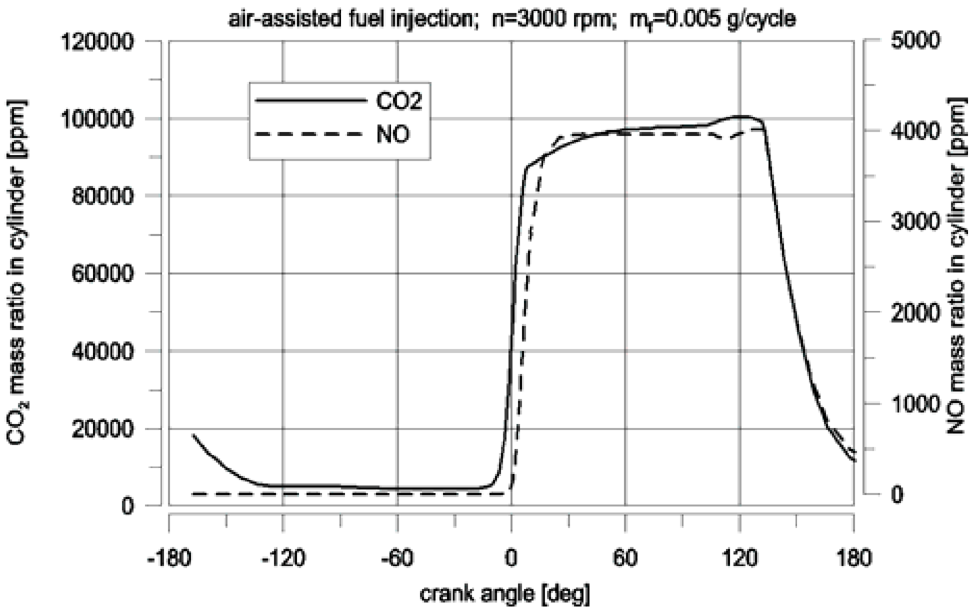


Fig. 11. Variation of mass fraction of CO_2 and NO in cylinder

1. High pressure direct fuel injection enables to form smaller fuel droplets in the spray and quick fuel evaporation as a result of interaction of gas charge and fuel.
2. The additional air spray during direct fuel injection enables better break-up of fuel droplets, better mixing and better contact of fuel vapour with the air, which influences on better evaporation and combustion.
3. The excess air ratio near the spark plug is enough for ignition at assumed start of injection 100 deg CA BTDC.
4. With air-assisted direct fuel injection the charge in the combustion chamber has higher turbulence intensity caused by the air stream with high velocity.
5. Additional air compensates the escape charge during scavenge process and enables almost full burning of fuel. Almost 19% of air is additionally delivered to the combustion chamber.
6. Mass ratios of toxic components in the combustion products have small value and it is very promising to obtain such results during experiments on the laboratory stand.
7. Fuel injection and air injection should take place in the same time.

The location of injector and the air inlet is one of important factors influencing on the mixture formation in two-stroke engine. Evolution of two-stroke engine is still up-to-date and for small applications this type of engine has still a big potential.

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