OPTIMIZATION OF THE RICH MIXTURE INJECTION SYSTEM IN TWO-STROKE ENGINES

Wladyslaw Mitianiec

Krakow University of Technology,
Al. Jana Pawla II 37, 31-864 Krakow, Poland
tel: +48 12 6283692; Fax: +48 12 6283690; e-mail: wmitanie@usk.pk.edu.pl

Abstract

Two-stroke engines used in small applications, in order to fulfill the emission requirements, should be equipped with a direct injection system or pneumatic rich mixture injection system. The FAST system in the Piaggio engines is based on the two-throat carburetor. Till now the Piaggio company has not published the physical processes taking place in such engines. The author has carried out the deep study of the process of rich mixture injection by an additional compressor and the results of calculations are presented in the paper. The simulation was done by using of the own computer program based on the mathematical model taking into account the unsteady gas flow of semi-perfect gas, the changed parameters of engine geometry and controlled parameters. The paper includes description of the fuelling system, mathematical model, calculation algorithm, results of calculations for different controlled parameters and comparison with other systems. The FAST computer program enables to define the characteristics of the cylinder charge, engine power, fuel consumption and exhaust gas composition. The simple mechanical fuelling system without electronic participation enables to apply it for the engines driven the small vehicles. However the drive of the compressor by tooth belt from the engine crankshaft decreases the total engine power about 20%.

Keywords: *Transport, two-stroke engine, fuelling*

1. Introduction

The Piaggio company in the beginning of 21st century led on the market the new motorbike engine of capacity 50 cm^3 with the compressed injection of rich air-fuel mixture called FAST [6]. The engine is equipped with an additional piston compressor driven by a tooth belt from the engine crankshaft. The rich mixture formed in a two-throat carburetor is delivered to the compressor. During the compressed cycle the compressor piston squeeze out the mixture through the automatic valve to the main cylinder at the beginning of the compression stroke. The principles of working of the FAST engine were described earlier by Nuti [8] and the author [5]. The power on the motorbike rear wheel driven by 50 cm^3 two-stroke FAST engine and the hydrocarbon exhaust emission is shown in the Fig.1 and the relative air-fuel ratio in Fig.2, respectively.

The authors of the FAST system have not introduced any details about influence of the geometrical and control parameters of the fuelling system on the engine work performance. In this paper is shown the study of behaviour of the engine with FAST system at the change of chosen geometrical and control parameters. The scheme of the fuelling of the combustion chamber with the rich mixture from the compressor is shown in Fig.3.

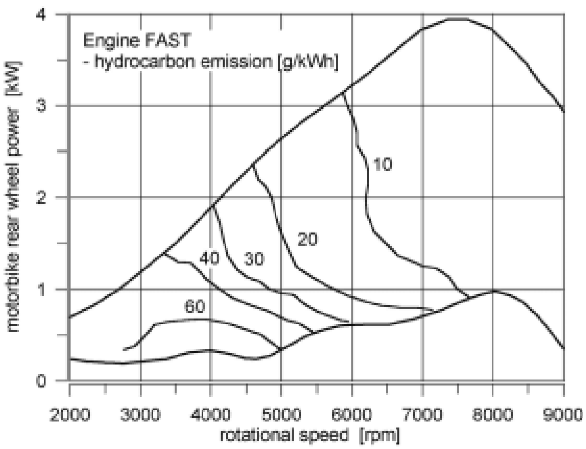

Fig.1. Performance of HC emission of the motorbike 50 cm^3 Piaggio engine with FAST system [8]

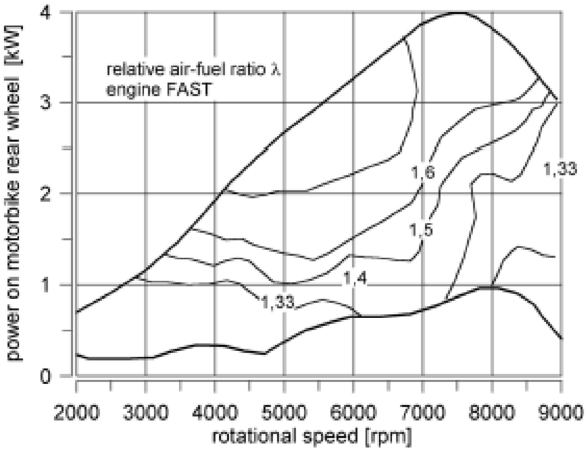

Fig.2. Relative air–fuel ratio on the power characteristic of the motorbike 50 cm^3 Piaggio engine with FAST system [8]

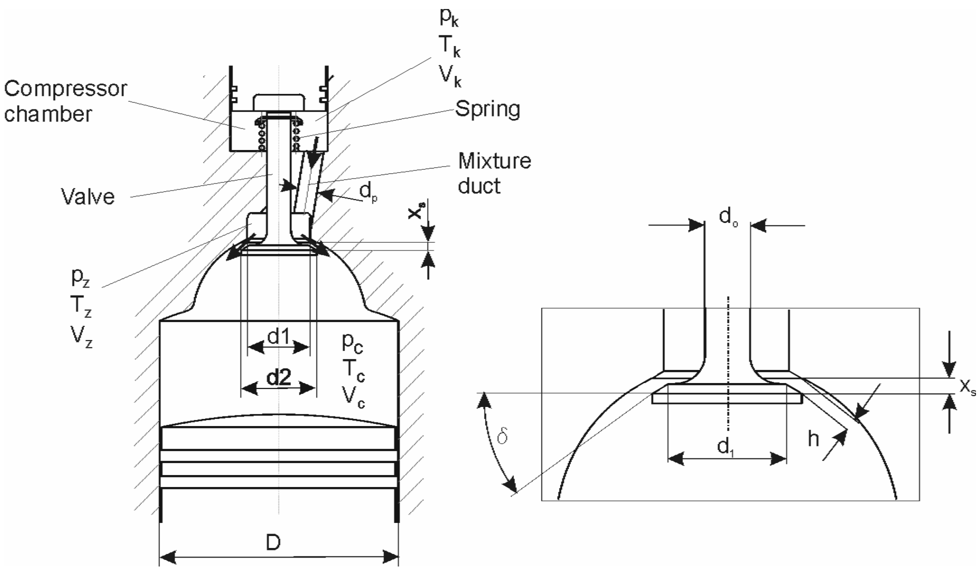

Fig.3. Scheme of the rich mixture flow from the compressor to the cylinder

Angular position of the compressor crankshaft is moved earlier of an angle φ in relation to the engine crankshaft, so the rich mixture formed in the compressor should be squeezed into the cylinder in the compression process. The rich mixture can be delivered to the compressor with relative air-fuel ratio λ_k or the fuel can be injected to the cylinder compressor forming the mixture also with composition λ_k. The flow of rich mixture follows as a result of pressure difference between the compressor and the cylinder after the opening of the valve head tensioned by the spring with deflection constant *k*.

2. Physical model of fuelling system

The crankshaft of the piston compressor is driven from the engine crankshaft with the same rotational speed. The swept volume of the compressor amounts V_{ks} at a given bore D_k with the compression ratio ε_k. In each cylinder of the engine and compressor the change of the pressure p_c occurs according to the formula [7]:

$$\mathrm{d}p_c = \left(\frac{k_c - 1}{V}\right)\left(\mathrm{d}Q_s + \mathrm{d}Q_{\mathrm{ch}} - \frac{\kappa_c}{\kappa_c - 1} p_c \mathrm{d}V + \frac{\kappa_d}{\kappa_d - 1} R_d T_d \mathrm{d}m_d - \frac{\kappa_w}{\kappa_w - 1} R_w T_w \mathrm{d}m_w\right), \tag{1}$$

where κ is the ratio of specific heats, *V* is a volume, *R* is an individual gas constant, m is the mass, *T* – temperature and *Q* is the heat. In the compressor the compression ratio can be almost the same as in the main cylinder so the increased pressure in the compressor is bigger than in the engine cylinder during squeezing of the rich mixture. The force acting on the valve from the side of the compressor amounts:

$$F_k = \frac{\pi}{4}(d_1^2 p_z + d_0^2 p_k). \tag{2}$$

The thrust force from the side of the cylinder amounts:

$$F_c = \frac{\pi}{4} d_2^2 p_c. \tag{3}$$

The spring force acting on the valve can be calculated as follows:

$$F_x = k_c(x_0 + x_s), \tag{4}$$

where: k_c is the spring constant, x_0 is the initial spring deflection at valve closing, x_s is the work deflection of the spring (valve lift) and p_z represents the pressure in the chamber above the valve head.

The inertia force of the valve with mass m_z amounts:

$$B = m_z \cdot \ddot{x}_s. \tag{5}$$

On the equilibrium of forces acting on the poppet valve the lift is calculated on the basis of differential equation:

$$m_z \ddot{x}_s + k_c x_s = F_k - F_c - k_c x_0. \tag{6}$$

Using the Runge-Kutta method of 4th order the valve lift can be calculated at assumption of constant values of pressure in the chamber and cylinder for a given time step of the calculation. The flow area of the mixture depends on the angle of the valve stem δ (Fig.3):

$$A_z = \pi d_1 x_s \cos\delta \, . \tag{7}$$

The mixture flow between the compressor and the space above the valve head at omitting of unsteady gas motion in the short duct of diameter d_p can be described by equations of subsonic or sonic flow

$$\frac{\mathrm{d}m}{\mathrm{d}t} = \beta F_r \frac{p_d}{\sqrt{T_{u,0}}} \sqrt{\frac{2\kappa}{\kappa-1}\left[\left(\frac{p_d}{p_{u,0}}\right)^{\frac{2}{\kappa}} - \left(\frac{p_d}{p_{u,0}}\right)^{\frac{\kappa+1}{\kappa}}\right]} = \beta F_r \frac{p_d}{\sqrt{T_{u,0}}} \psi_2 \, , \tag{8}$$

$$\frac{\mathrm{d}m}{\mathrm{d}t} = \beta \cdot F_r \cdot \frac{p_d}{\sqrt{T_{u,0}}} \cdot \sqrt{\frac{\kappa}{R} \cdot \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}} = \beta \cdot F_r \cdot \frac{p_d}{\sqrt{T_{u,0}}} \psi_2 \, , \tag{9}$$

where F_r is the area of the connecting duct and $p_d = p_k$ and $p_{u,0} = p_z$. On the contrary the mass flow rate is signified as dm_z / dt and temperature in the space under the valve head with volume V_z amounts T_z. The outflow of the charge from this space to the cylinder through the area A_z can be described also by equations (8, 9), while the pressure are signed as $p_d = p_z$ and $p_{u,0} = p_c$, respectively and the mass flow rate as dm_c / dt. The loss flow coefficient is defined by Kastner [4] formula:

$$\beta = 1 - 1{,}5 \frac{x_s}{d_1} \, . \tag{10}$$

The change of mass in the volume V_z amounts:

$$dm_z = dm_k - dm_c \, . \tag{11}$$

The internal energy balance in this space is defined as follows:

$$\mathrm{d}U = i_k \mathrm{d}m_z - i_z \mathrm{d}m_c \, . \tag{12}$$

On the basis of the energy equation the pressure change in the volume V_z is found similarly as for the compressor chamber (1) without additional terms:

$$\mathrm{d}p_z = \frac{R_z}{V_z \cdot c_v} \cdot \left(\left(c_p\right)_k \cdot T_k \cdot \mathrm{d}m_z - \left(c_p\right)_z \cdot T_z \cdot \mathrm{d}m_c \right). \tag{13}$$

If the charge is treated as semi-perfect gas then specific heat is obtained for given temperature on the basis of the empirical data included in the JANAF tables [3]. The charge flow between the cylinder of the compressor and of the engine is described the equations (8)(9)(12) and gas state equation in the space V_z for a given position of the crankshafts. The system of non-linear equations combined the unknown values of pressure p_z, temperature T_z and mass m_z is solved by Newton-Raphson method.

In the production version the compressor is fed by a mixture, which is formed in the carburettor with the relative air-fuel ratio λ_k. It was assumed, that the charge in the compressor and the space under the valve head is homogeneous and all fuel is in a vapour form. Then the change of the squeezed fuel to the cylinder amounts:

$$\mathrm{d}m_p = \frac{\mathrm{d}m_c}{\theta \cdot \lambda_g} \cdot \tag{14}$$

Using the formulas (8) and (9) the total fuel mass squeezed into the cylinder during the opening of the valve is calculated as follows:

$$m_p = \int_0^Z \beta \cdot F_r \cdot \frac{p_d}{\sqrt{T_{u,0}}} \psi_i \frac{1}{\theta \cdot \lambda_g} \mathrm{d}t \ , \tag{15}$$

where i is the index of expression on the subsonic flow (i = 1) or sonic flow (i = 2), θ is the stoichiometric constant of fuel (for gasoline $\theta = 14.95$). The relative air-fuel ratio in the engine before the ignition amounts:

$$\lambda = \frac{m_a}{\theta \cdot m_p} \ , \tag{16}$$

where m_a is the air mass in the engine before the ignition.

3. Computer program and calculation data

Table 1. Geometrical parameters of Piaggio engine

Cylinder bore D	52 mm
Stroke S	58 mm
Connecting rod length L	110 mm
Engine compression ratio ε	9.0
Crankcase compression ratio	1,34
Beginning of opening of exhaust port	105° CA ATDC
Beginning of opening of transfer port	123° CA ATDC
Beginning of opening of inlet port	65° CA BTDC
Amount of transfer ports	5
Width of exhaust port	32 mm
Width of main transfer ports	22 mm
Width of inlet port	33 mm
Diameter of inlet pipe	20 mm
Length of inlet port	105 mm
Muffler capacity	2,5 l
Length of exhaust pipe	1480 mm

In order to realize the calculation of the work cycle of 2-stroke engine with the rich mixture pneumatic injection according to proposed mathematical model the program FAST was prepared on a PC computer. The calculations concerned to the 2-stroke engine Piaggio with swept volume 125 cm^3 with the compressor of swept volume 25,7 cm^3 (0,2 V_s). The main geometrical parameters of the engine are given in Table 1. The computer program takes into account unsteady gas flow in the inlet, transfer and exhaust systems. It reads geometrical, boundary, initial and control data from a text file and creates 1-dimensional mesh all pipes, and then match the numbering of the mesh nodes. The calculations begin from the lowest to highest declared rotational speed with the equal step. Time step of calculations Δt is automatically chosen in dependence of the speed of motion of pressure waves in the elementary cell with a given length Δx, to fulfil the Courant condition [2]. The calculation results are written in a text format for further processing in other useful programs. The calculations were carried out during several working cycles for each rotational speed until the following condition was fulfilled:

$$\frac{(L_i)_j - (L_i)_{j-1}}{(L_i)_j} < \delta\ , \tag{17}$$

where L_i is the indicated work, j is the index of the next working cycle and δ is a given acceptable calculation error (usually $\delta = 0{,}005$).

The program up-to date calculates gas concentration and temperature in the whole engine system and on that basis it calculates the specific heats, gas constants and gas heat conductivity to the walls. In the program the empirical formulas of mechanical losses and ignition delay given by Blair [1] were used. This program calculates the mean indicated and effective pressures, the specific fuel consumption, the delivery and scavenges ratios, the total relative air-fuel ratio and residual gas after scavenge period, as well exhaust losses of fuel (HC emission).

4. Simulation tests of fast system

The study of fuelling of two-stroke engine by rich mixture using FAST system was done on the engine with increased swept volume 125 cm^3. The correct fuelling of such engine can be done by the proper selection of the following parameters (see Fig.3):

- swept volume of the compressor and its compression ratio ε_k,
- dimensions of valves d_0, d_1, d_2 and the angle δ,
- shift angle α_c of the cranks of the compressor and the engine,
- hole diameter d_p,
- space volume over the valve V_z,
- spring constant k_s,
- mixture composition formed in the carburettor λ_g.

5. Calculation results

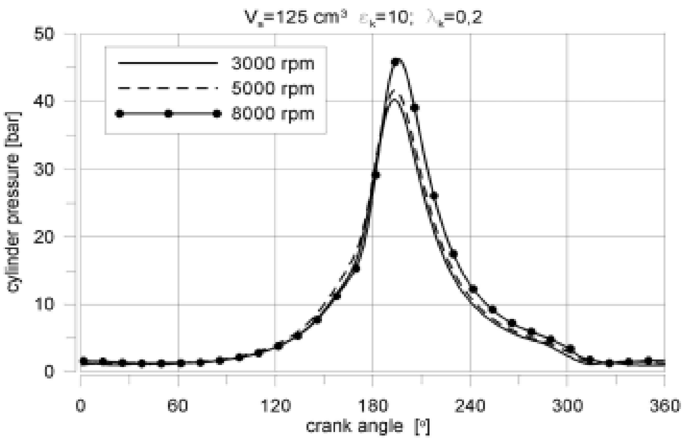

Fig.4. Cylinder pressure at several rotational speeds

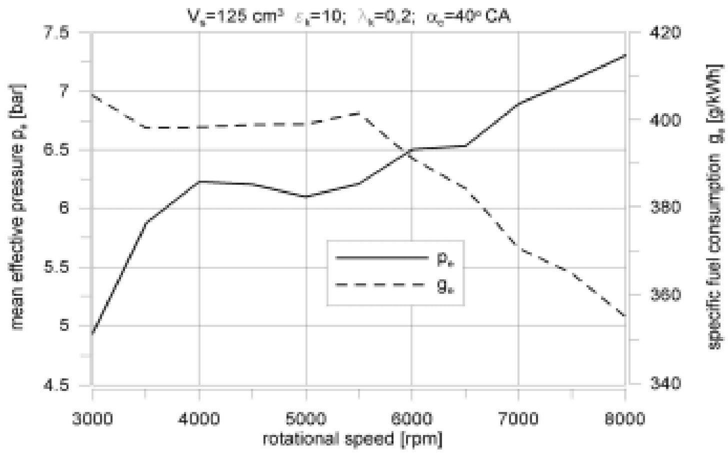

Fig.5. Mean effective pressure and specific fuel consumption in the engine with the compression ratio ε_k=10 in the compressor

Hypothetical pressure variations in the cylinder for several rotational speeds at constant values of the compression ratio of the compressor $\varepsilon_k = 10$, relative fuel ratio of the rich mixture $\lambda_k = 0{,}2$ and the shift angle of the compressor crank in relation to the engine crank $\alpha_c = 40\,^\circ$ CA is shown in Fig.4. With increased rotational speed there are also observed the increase of pressure maximum in the cylinder, as well the charge temperature. For highest speeds the temperature of the exhaust gases also increases during the opening of the outflow port.

Using the FAST program the calculations of the mean effective pressure and specific fuel consumption of the engine with the compression ratio of the compressor ε_k=10 and the shift angle of the crankshafts α_o=40^o CA for the range of rotational speeds 3000-8000 rpm were carried out. The engine characteristics presented in Fig.5 show an increase of p_e and reduction of g_e with increasing of the rotational speed, however the power consumption by the compressor was not taken into account. The engine fuelling from the compressor was simulated at assumption of very rich mixture with λ_k=0,2. The increase of the compression ratio in the compressor is demonstrated by the increase of the engine effective power in whole range of engine speeds (Fig.6). The real engine power is decreased by power needed for driving of the compressor. The pressure variation in the compressor depends on the spring constant and the backpressure in the engine cylinder. The pressure decrease in the compressor follows as a result of the valve opening during the piston motion towards TDC. The pressure variation in the compressor for ε_k=10 in a small extend depends on the rotational speed, which is shown in Fig.7 in a function of crank angle.

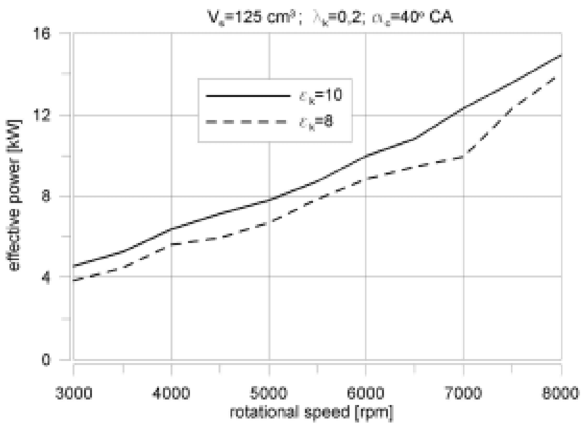

Fig.6. Engine power for two different compression ratios of the compressor

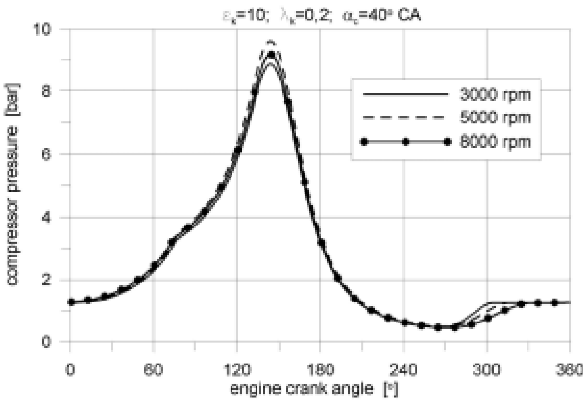

Fig.7. Pressure in the compressor at different rotational speeds and ε_k=10

The temperature of the rich mixture changes also independently of the rotational speed and reaches maximal value 450 K. The change of the compression ratio of the compressor causes the change of the relative air-fuel ratio in the main cylinder at the same carburettor adjustment. A decrease of the coefficient λ in the cylinder follows with an increase of the compression ratio of the compressor. Variation of λ in a function of rotational speed for ε_k=8 and ε_k=10 is shown in Fig.8. However the change of this coefficient in the cylinder is shown in Fig.9 at n=3000, 5000 and 6500 rpm.

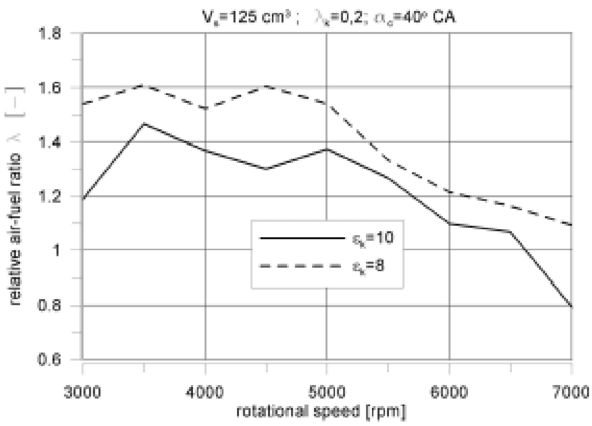

Fig.8. Comparison of relative air-fuel ratios in the engine at different compression ratios ε_k

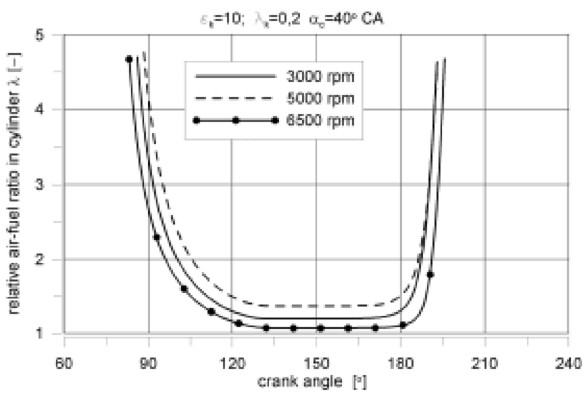

Fig.9. Comparison of relative air-fuel ratio in the cylinder at different rotational speeds and ε_k=10

The air-fuel ratio depends exactly on the adjustment of the carburettor ($\lambda_k \approx 0{,}2$) and the compression ratio of the compressor. The fuel mass delivered to the cylinder decreases with lower compression ratio, which is shown in Fig.10 at rotational speed 5000 rpm. The opening of the automatic valve depend on the initial spring tension, spring constant and pressure difference in two gaseous medium. The valve lift in a function of the crankshaft angle for three rotational speeds and ε_k=10 is shown in Fig.11. The change of the compression ratio in the compressor influences on the valve lift in a small value. This phenomenon takes place as a result of the small change of the pressure in the compressor. The maximal value of the valve lift amount 1.6 mm at 8000 rpm.

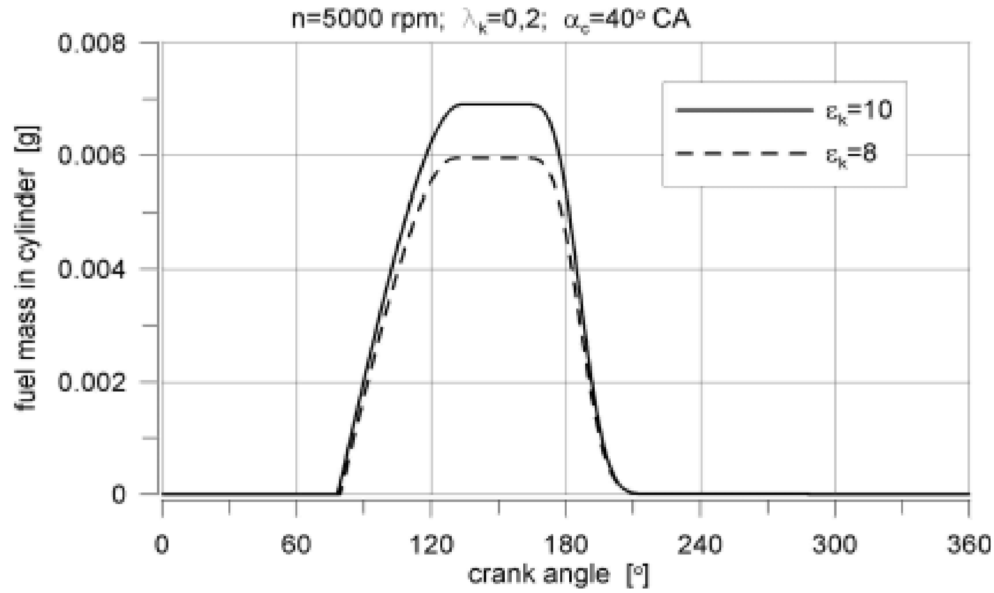

Fig.10. Fuel mass in the cylinder at 5000 rpm and for two compression ratios in the compressor

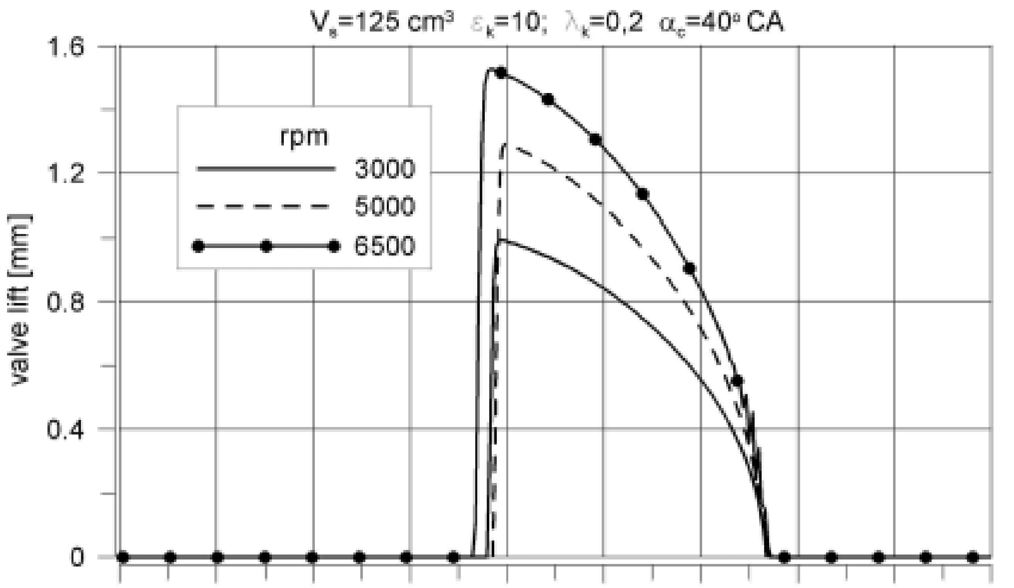

Fig.11. Valve lift at different rotational speeds and ε_k=10

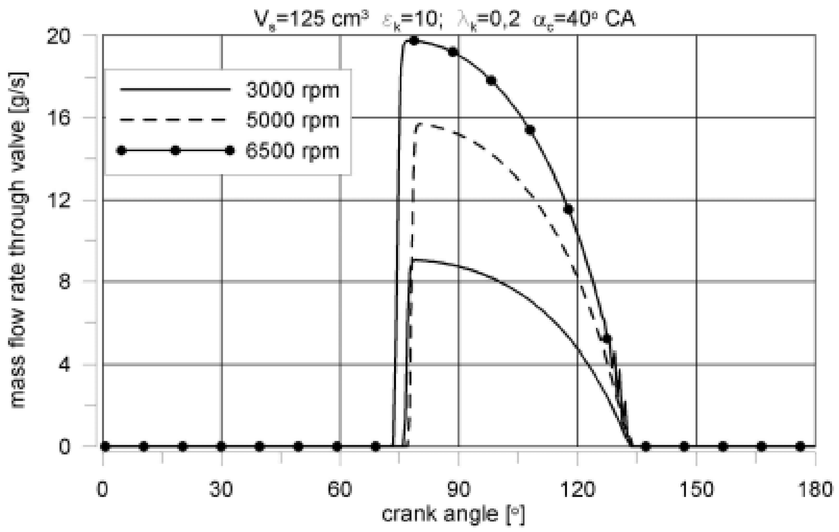

Fig.12. Mass flow rate of gas through the automatic valve at different rotational speeds

The flow velocity of the mixture through the minimal area of the valve opening is depended on the engine rotational speed and valve lift ratio. The flow velocity reaches maximal value about 160 m/s for higher rotational speeds. The mass flow rate in a function of crank rotational angle at several rotational speeds is shown in Fig.12.

6. Formation of nitrogen oxides

Piaggio company has not given any concentration of NO in exhaust gases of two-stroke engine with FAST system. The author carried out the simulation of formation of nitrogen oxides in the two-stroke engine using the Kauffman definition [9] (see Fig.13). The maximal calculated value of the NO mass concentration amounted 750 ppm. Nitrogen oxides form after the main combustion process when the charge temperature reaches maximal value about 2500 K. The beginning of a strong increase of NO concentration in the cylinder charge follows about 20° CA ATDC. During expansion process the NO concentration in the cylinder remains on the same level.

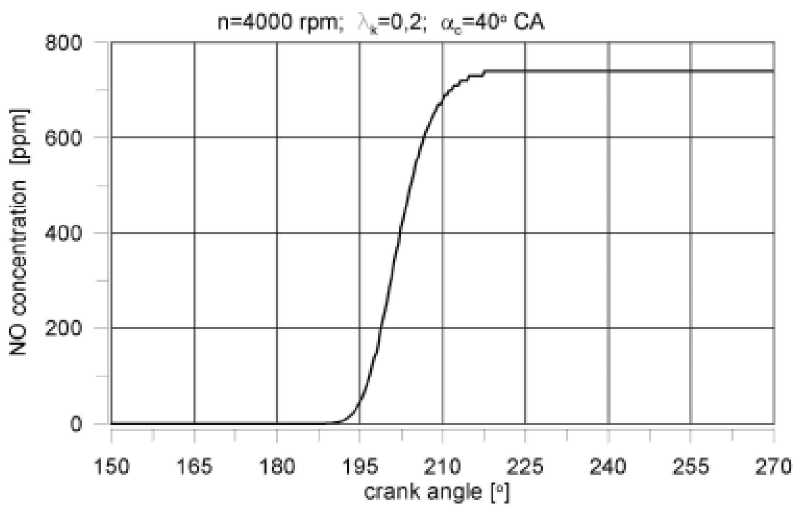

Fig. 13. Formation of NO in the cylinder at 4000 rpm

7. Conclusions

The compressed rich mixture fuelling system worked out by the Piaggio company depends strictly on the selection of:

1. compression ratio of the compressor,
2. shift angle of the compressor crankshaft and the engine crankshaft,
3. spring characteristic,
4. swept volume of the compressor,
5. adjustment of the carburettor feeding the compressor,
6. engine rotational speed.

The computer program FAST enables to define the characteristics of the cylinder charge, engine power, fuel consumption and exhaust gas composition. The calculation results are a supplement of the experimental test carried out by the Piaggio. The simple mechanical fuelling system without electronic participation enables to apply it for the engines driven the small vehicles. However the drive of the compressor by tooth belt from the engine crankshaft decreases the total engine power about 20%, which was theoretically determined.

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