ASPECTS OF COMBUSTION PROCESS IN A HIGH PRESSURE DIRECT FUEL INJECTION TWO-STROKE ENGINE

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Abstract

The paper presents combustion process in loop scavenging two-stroke engine with high pressure direct injection of gasoline. A new method of determination of heat release rate on the base of experimental test in a two-stroke engine with direct gasoline injection is also presented. The target of the work was achieving of information about implementation of approximated Viebe function in mathematical models and simulation computer programs of a combustion process in spark ignition two-stroke engine with direct fuel injection due to the lack of such data in the literature. The paper show propagation of the flame in the combustion chamber depending on charge flow after scavenge process. The paper presents several examples of heat release and approximate function, for which variable coefficients of Viebe function were determined for certain rotational speeds at wide openning throttle. Pressure measurements were carried out on the engine Robin EC12 wit capacity 115 cm^3. Determination of heat release and burned fuel rate was calculated by means of own computer program. The paper also presents the method of determination of variable specific heat coefficients in the function of crankshaft rotation, which are also used in modeling of the engine processes.

Keywords: *transport, combustion engines, heat release*

1. Problems of combustion process in two-stroke engines

Heat release in internal combustion engines is one of the most important factors influencing on engine work parameters and exhaust gas emission. Very fast combustion process leads to rapid increase of cylinder pressure and temperature and is the main reason of a big NOx emission and knocking. The main problem of conventional loop scavenged two-stroke spark ignition engines is a big hydrocarbon emission and high specific fuel consumption, which is caused by flow of a part fuel mixture to the exhaust port during scavenge process. This internal exhaust gas recirculation in a two-stroke engine influences on lower NOx emission in comparison to four stroke engine. The flow of fresh fuel-air mixture from the transfer ports causes that most of burnt gases occupy the space near the exhaust port. Usually the space near the spark plug is fulfilled by fresh mixture and there is no problem with the ignition of the charge. The new fuelling system of two-stroke engine was worked out by the authors in Cracow University of Technology in order to eliminate their basic faults and imperfections. The high pressure direct fuel injection system was applied to the industrial small capacity (115 cm^3) two-stroke engine Robin EC12 from Fuji Heavy Industries. Simulation analysis and experimental work indicated that direct fuel injection decreases both specific fuel consumption and hydrocarbon emission. Simulation of combustion process in 0-dimensional model requires a verified model of heat release. One cannot find such model for two-stroke engine with direct fuel injection in the world’s literature. Therefore much work was concerned to find an approximation Vibe function for full loads and rotational speed of tested engine.

2. Two-stroke engines with direct fuel injection

Only one industrial application of direct fuel injection in outboards high power multi-cylinder two-stroke engines from Mercury Marine (in marine boats) is known in the world The main advantages of applying of such engines are compactness, high unit power, lightness and possibility of working in different positions. There are many small two-wheel and three wheel vehicles driven by carburetted two-stroke engines in the world. Limitation of exhaust gas emission and fuel consumption of such engines is a main object of recent research works concerning such engines. The goal of the work was to define the injector position in the combustion chamber with regard of the air-fuel mixture motion and orientation of fuel spray and determination of these factors on the combustion process and exhaust gas emission. Such relations could be simply done by CFD simulations by different injector orientation in the combustion chamber. For that case KIVA3V program was used as common tools for solving combustion and fuel injection problems in internal combustion engines. The fundamental goal of the work was determination of following aspects:

- influence of the injector location in respect to possible ignition of the air-fuel mixture and influence of gas motion on mixture distribution by using CFD simulation,
- decreasing of specific fuel consumption and pollutants emission,
- determination of heat release rate on experimental pressure measurements and Vibe function for zero-dimensional models of two-stroke engines with direct fuel injection, because such functions are not met in literature,
- influence of fuel injection pressure on working parameters and exhaust gas emission.

3. Simulation of combustion process in two-stroke engine

Analysis of combustion process in the Robin two-stroke engine was carried out in CFD program Kiva3v. Charge motion in the combustion chamber influences on flame propagation. Development of burning zone for four piston positions is shown in Fig. 1.

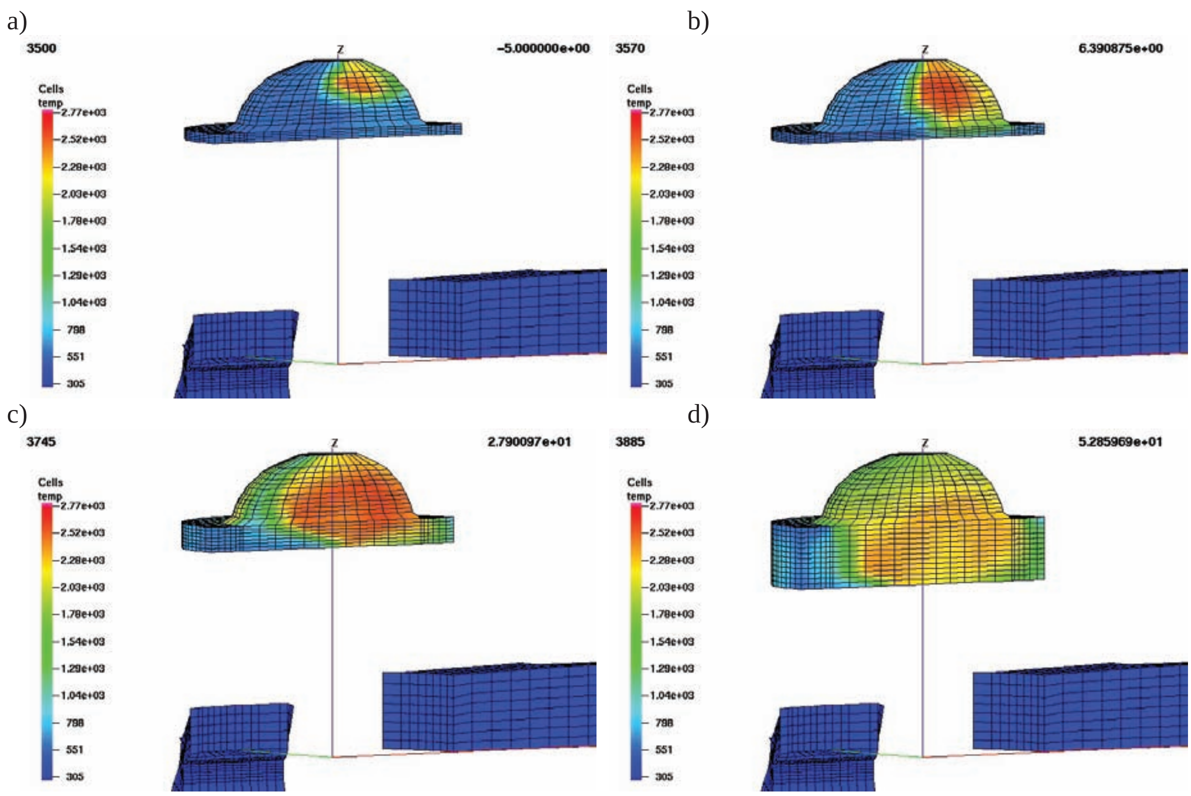


Fig. 1. Distribution of temperature in DFI two-stroke engine at 3000 rpm and at piston position: a) 5 deg CA BTDC, b) 6.3 deg CA ATDC, c) 27.9 deg CA ATDC and d) 53 deg CA ATDC

The flame is not spherical as in homogenous charge because is distorted by gas movement and stratification of the charge. Beginning of fuel injection occurred at 90 deg CA BTDC and lasted only 700 μs under pressure 50 bar. Such high pressure influences on high break-up of droplets and their lower size, which causes quicker evaporation of fuel. Variation of volumetric fraction of fuel in gaseous state for full engine cycle at 3000 rpm is shown in Fig. 2. During combustion process the whole fuel is evaporated. Because of stratification combustion process lasted until opening of the exhaust port. Combustion of air-fuel mixture is extended in comparison to homogenous charge.

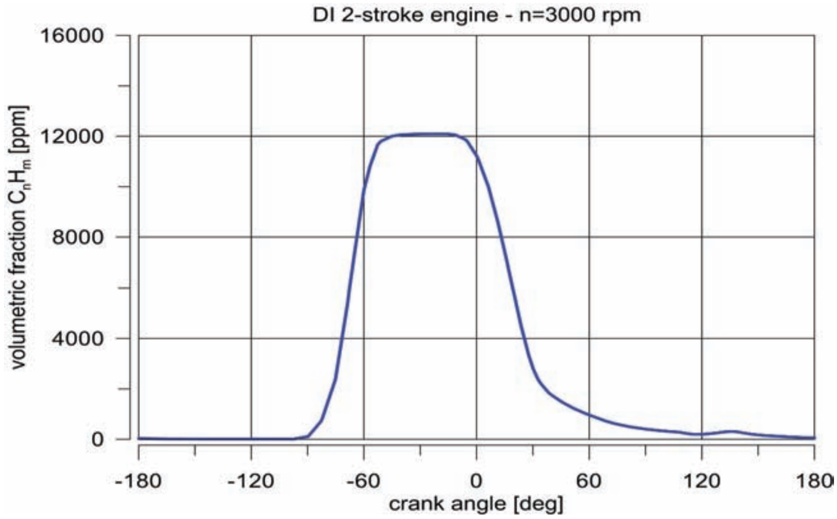


Fig. 2. Volumetric fraction of gaseous fuel (C_nH_m) in cylinder of 2-stroke engine DFI 125 cc at WOT and 3000 rpm

Simulation of work of whole engine system was carried out in GT-Power program for wide range of rotational speeds (1500-4500 rpm) and wide opening throttle (WOT).Combustion process was modelled by using of Viebe function with the same assumed parameters *a* and *m*. Variation of cylinder pressure for chosen engine speeds is shown in Fig. 3. The highest maximal pressure occurs at 2500 rpm (maximum of torque) and lowest at 4000 rpm. However, pressure maximum occurs at the same crank angle (about 20 deg CA ATDC).

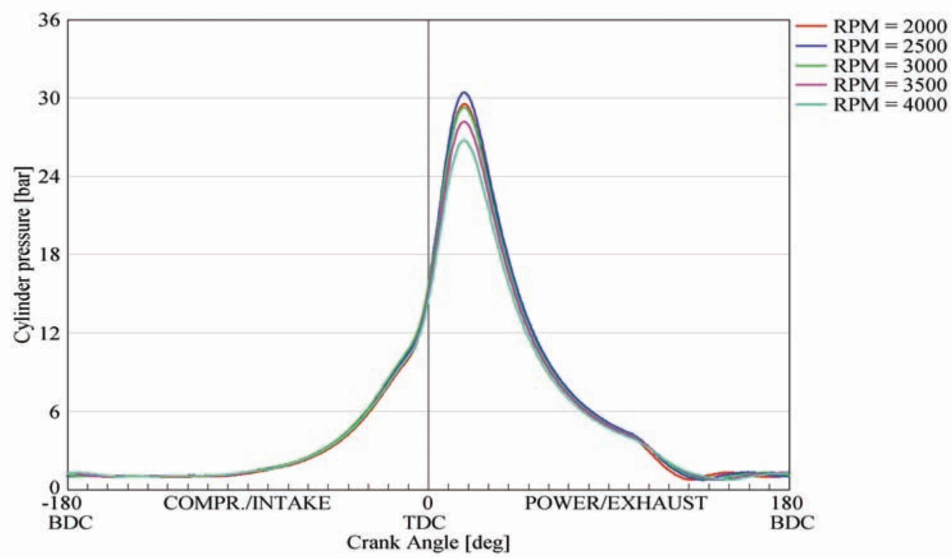


Fig. 3. Courses of cylinder pressure at different rotational speeds

On the other hand higher rotational speed influences on lower heat transfer to the wall, which causes higher cylinder temperature at higher engine rotational speeds. However, maximum of temperature does not exceed 2500 K. Courses of cylinder temperature for chosen rotational speeds are shown in Fig. 4. Higher temperature during scavenge process influences on better vaporization of injected fuel.

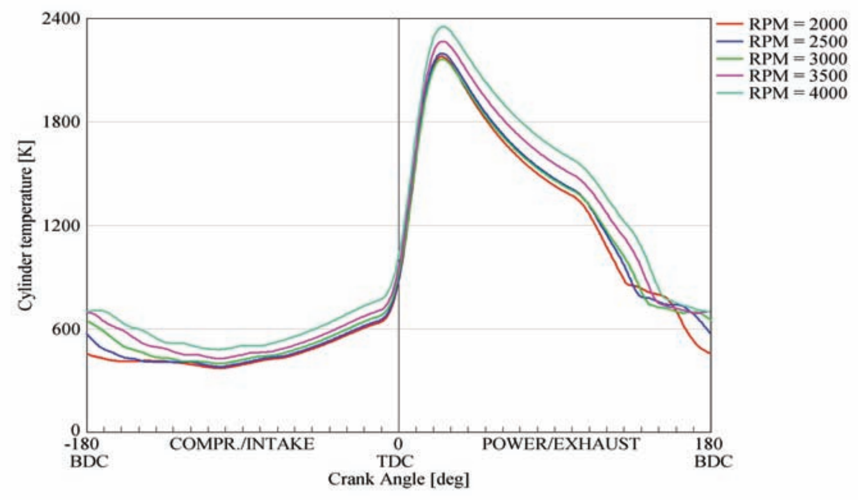


Fig. 4. Courses of cylinder temperature at different rotational speeds

4. Mathematical model of heat release in DFI engine

Thermodynamic state of combustion engines is presented widely in literature e.g. by Heywood [9], Blair [4] and Mitianiec [16, 17]. Dynamics of fuel injection process is still under development and was a subject of the work of Feath [7], Ghandi [8], Ikeda [12] or Melton [15]. Combustion processes and charge turbulence are described e.g. by Chen and Kim [5], Spalding [20] or Higelin [11]. The engine cylinder after closing of transfer and exhaust ports is a close thermodynamic system. The First Law of Thermodynamics for such closed system states that heat delivered to the charge $\mathrm{d}Q_R$ increases internal energy $\mathrm{d}U$ and quite amount of energy has been lost through heat transfer $\mathrm{d}Q_c$ to the cylinder walls and coolant at the same time and work done by the piston $\mathrm{d}W$:

$$\mathrm{d}Q_R - \mathrm{d}Q_c - dQ_{vap} = \mathrm{d}U + \mathrm{d}W\,. \tag{1}$$

The term dQ_{vap} can be neglected because in spark ignition engines almost whole fuel is in vapour state during combustion process. Increment of internal energy in the close system is a function of increment of temperature.

$$dU = mc_v dT\,. \tag{2}$$

The specific heat at constant volume c_v is a function of temperature and also of the gas properties. During combustion process temperature and properties of the charge change rapidly. Specific heat capacity c_v is not a constant value and should be evaluating every time during calculation. After differentiation of the state equation of close system at assumption of constant mass of charge m and gas constant R:

$$dT = \frac{pdV + Vdp}{mR}\,. \tag{3}$$

Change of internal energy in the cylinder is described by equation:

$$dU = \frac{pdV - Vdp}{k-1}\,, \tag{4}$$

where k is ratio of heat capacities. After substitution of work and internal energy to the main formula (1) one obtains:

$$\mathrm{d}Q_R - \mathrm{d}Q_c = \frac{pdV + Vdp}{k-1} + pdV\,. \tag{5}$$

If the combustion process had not occurred, then the compression or expansion process would have continued in a normal fashion. The polytrophic process is taking place with relationship defined by:

$$\frac{dp}{p} + np\frac{dV}{V} = 0. \tag{6}$$

In imaginary process the non-heat addition dQ_R is zero and the First Law of Thermodynamics could be rewritten to calculate the heat loss dQ_c [4]:

$$-dQ_c = p_1\left\{\frac{V_2\left(V_1/V_2\right)^n - V_1}{k-1} + \frac{(V_2 - V_1)\left(\left(V_1/V_2\right)^n + 1\right)}{2}\right\}. \tag{7}$$

On the assumption that the heat loss is the same and continues during the combustion process the following equation on the heat release rate (HRR) can be given:

$$dQ_R = \left\{p_2 - p_1\left(\frac{V_1}{V_2}\right)^n\right\}\left(\frac{V_2}{k-1} + \frac{V_2 - V_1}{2}\right). \tag{8}$$

Specific heat at constant volume c_v is a function of temperature T and can be determined on JANAF Tables [13] and is expressed as:

$$c_p = A_1 + A_2 z + A_3 z^2 + A_3 z^3 + A_4 z^4 + \frac{A_5}{z^2}, \tag{9}$$

where $z = T/100$ and coefficients of the polynomial amount as follows:

A_1=-0.917 , A_2=9.775 , A_3=-7.682 , A_4=2.466 , A_5=0.022.

In order to determine temperature in the cylinder during combustion process, the mass of charge should be known. In the calculations the one took this value from CFD calculations of the whole engine work cycle. The gas constant R changes during combustion, however it is close to the gas constant of the air. On these simple assumptions the gas temperature can be calculated. Polytrophic exponent n is obtained from equation (9) by measured pressure values during whole combustion period Δa_b from the state 1 and state 2 and calculated volumes at these states. Total released heat Q_t can be determined by integration of heat release rate at combustion period and the current burned mass ratio x_z is calculated as follows:

$$x_z = \frac{\int_1^a Q_R da}{Q_t}, \tag{10}$$

where index 1 signifies state 1 as the beginning of combustion and state 2 signifies the end of combustion.

In the engine theory of combustion engines the Vibe [9] function is used for determination of combustion process based on approximation of measured pressure on real engines. This function is commonly used in literature [4, 12] and research work for analysis of combustion process in piston engines. Vibe function in most cases takes the following form:

$$x_z = 1 - \exp\left(a\left(\frac{a - a_1}{\Delta a_b}\right)^m\right). \tag{11}$$

The angle α is the current angle of crankshaft position, α_1 is an angle of crankshaft position, when combustion process begins, on the contrary, $\Delta\alpha_b$ is total angle of crankshaft rotation during combustion process. The coefficients of a and m are determined by continuous change their values until the shape of Vibe function reaches measured fuel mass burned ratio. Vibe function is regarded as the best approximation function for determination of ratio of burned fuel. It is needed in 0-dimensional model used in computer program simulations for analysis of whole engine system. Determination of coefficients in Vibe function for verification of combustion model applied in own ICE models and GT-Power software, used in this work, was needed**.**

5. Measurement results

The engine was tested on the dynamometer stand with experimental high pressure direct injection system and apparatus for measurement of exhaust gas emission. Engine was tested for different loads at chosen rotational speeds with modification of injection parameters such as injection time, angle of injection advance in relation to TDC and injection pressure. It was found and optimal value of injection pressure between 50 and 60 bar for applied automotive injector. One of the main task was to measure cylinder pressure for different loads and rotational speeds in order to determine heat release rate and mass burned ratio. On the base of such experiment one can find an approximation Vibe function. For that reason the piezo-optical sensor from Optrand company was used, which was directly mounted in the wall of cylinder head. All electric signals from sensors were transformed by amplifiers to the computer in the function of crank angle (encoder) and variations of pressure can be performed on the graphs. Variation of pressure in the cylinder reflects combustion process. Higher increment of pressure is caused by faster fuel burning. Fig. 5 presents pressure traces in the cylinder of two-stroke engine with direct fuel injection at 50% throttle opening for engine speed 3300 rpm. It is observed inequality of the pressure maximum from cycle to cycle.

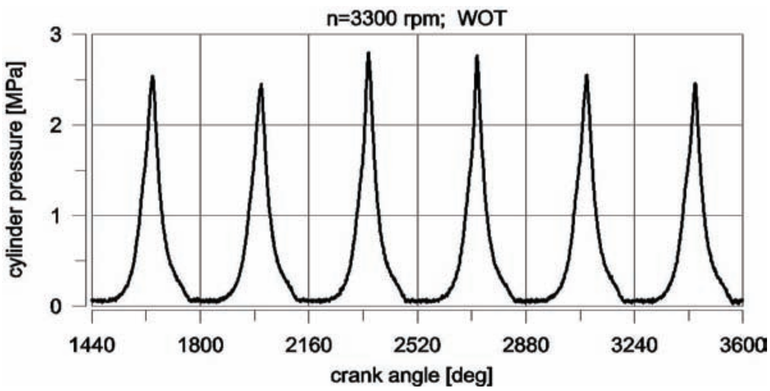


Fig. 5. Inequality of pressure in engine with direct fuel injection at 3300 rpm 50% WOT

The maximum pressure in the cylinder changes from 2.3 to 2.68 MPa. It is caused by existing of the rest of exhaust gases and not complete cleaning of the cylinder by the fresh air during the scavenge process. The rest of exhaust gases stays in the cylinder after closing of the ports as a result of incomplete scavenging by fresh air, when both exhaust and transfer ports are opened. Strong dynamic effect of pressure waves in the exhaust system takes place in two-stroke engines, which stops the outflow of the gases. Therefore the injected fuel does not form a proper mixture composition, which enables good combustion process. Mean temperature of the charge during combustion was calculated on the base of pressure measurement and general formula of gas state.

When the transfer ports start to open the pressure in the cylinder is higher than in the crankcase. The exhaust gases from the cylinder flow to the transfer ports causing a rapid increase of pressure in the crankcase. Figure 6 presents variation of pressure of the cylinder charge and temperature, which was calculated on the base of gas state equation in the range of combustion process. For considered rotational speed 3400 rpm the maximum of temperature takes place about 30 deg ATDC and insignificantly exceeds 2000 K. Low value of the maximal temperature is caused by longer time of combustion process and lower heat released rate. Combustion process is retarded by big amount of exhaust gases in the fuel mixture.

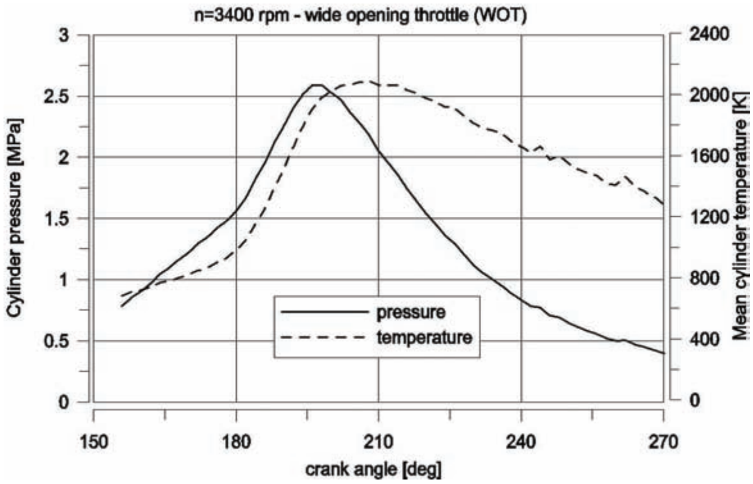


Fig. 6. Cylinder temperature and pressure during combustion process at 3400 rpm

Validation of applied mathematical model was one of the tasks in the work. Many factors influence on pressure variation both in the cylinder and crankcase particularly non-steady gas flow in all cylinder ducts and pipes, charge exchange through the ports or heat exchange with walls. One carried out several calculations of pressure changes in these spaces by using GT-Power software by applying of Vibe function of fuel burning ratio with coefficients obtained from experiment. Fig. 7 shows comparison of pressure changes in the cylinder and crankcase at full load and rotational speed 2750 rpm. One can notice quite good correlation between measured pressure and calculated pressure.

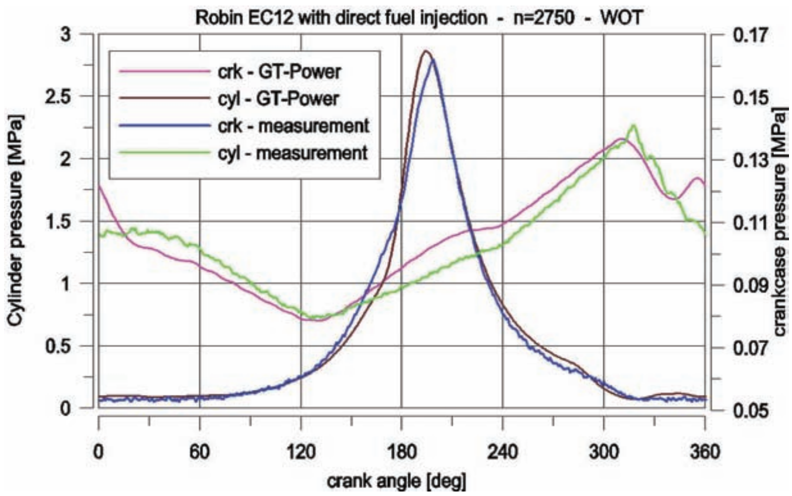
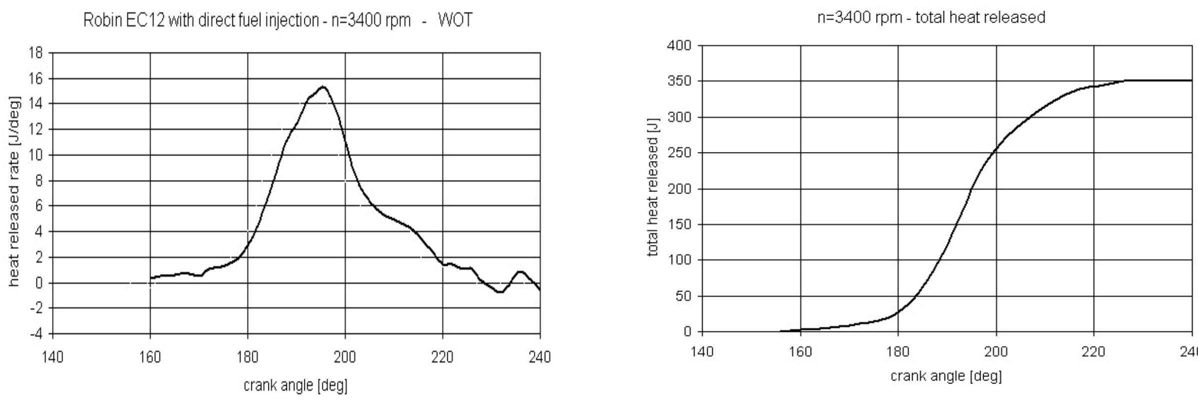


Fig. 7. Comparison of calculated and measured pressure in cylinder and crankcase

6. Calculation of heat release

On the base of observation of many pressure traces achieved from the indicated measurements it was not found any abnormal combustion process. In a support of the pressure indicating diagram and formulas presented in the section 3 the heat released rate and total heat released were calculated. One example of heat released rate (HRR) and total heat released in a function of crank angle for two-stroke engine with direct fuel injection is presented in Fig. 8 for engine rotational speed 3400 rpm. Total heat released shows the energy given to the charge. Increment of charge energy is a function of consumed fuel during combustion process. Variation of total heat released is shown in Fig. 8b for the same engine rotational speed. The burning period amounts 60 deg of crankshaft rotation and maximum of heat release rate amounts 16 J/deg and total heat released reaches value 330 J. Combustion process in the DFI two-stroke engine runs with non-monotonically heat released rate and maximal value of HRR takes place at maximum of pressure (15 deg ATDC).



a) b)

Fig. 8. a) Heat release rate at 3400 rpm and b) total heat released at 3400 rpm and full load

More rapid combustion process takes place after TDC and lasts about 20° CA and then the rest of fuel burns much slower, which is caused by increasing of the cylinder volume and lower turbulence of charge. Variation of fuel mass burned ratio is shown in Fig. 10a. Variations of volume specific heat and specific heat ratio are shown in Fig. 10b for the same rotational speed. In this engine the changeable HRR is caused by burning of non-homogenous mixture with different local air excess coefficient λ. In Fig. 10b the indexes 1 and 2 represent the beginning and the end of combustion process.

a) b)

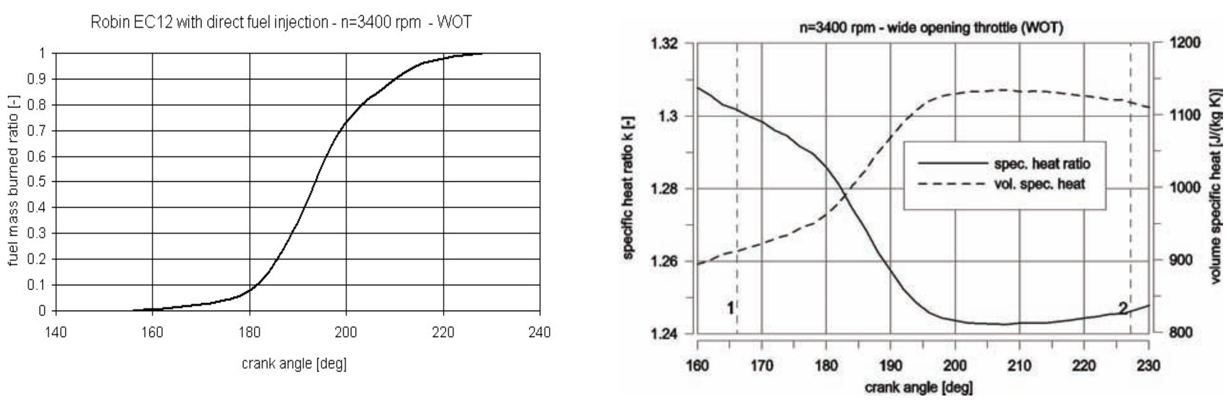


Fig. 10. a) Variation of fuel mass burned ratio at 3400 rpm, b) variations of specific heat at constant volume and specific heat ratio during combustion process at 3400 rpm

7. Vibe function

For the simulation of combustion process the Vibe function is widely used and in some computer programs the parameters *a* and *m* are needed. Approximation functions of the combustion process for direct fuel injection two-stroke engines are not met in the literature. Equations (10) and (11) enable to obtain the approximated Vibe function for different loads and rotational speeds. It is not possible to find the exact Vibe function for the real combustion process. Two examples of mass burned rate and approximated Vibe function are presented in Fig. 10 and 11 at 2750 rpm and 3400 rpm, respectively.

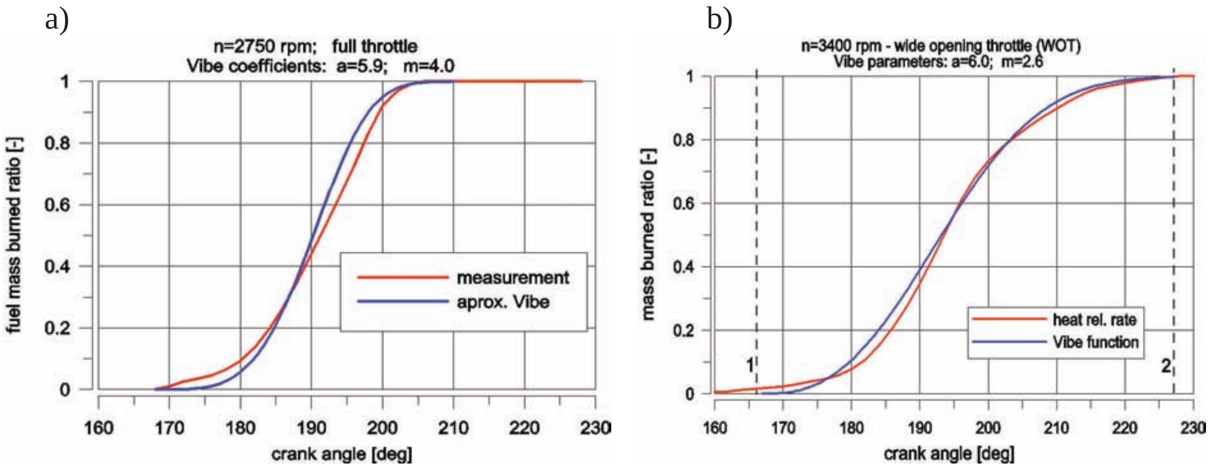


Fig. 10. Fuel mass burned ratio and approximated Vibe function in combustion process at a) 2750 rpm, b) 3400 rpm

At lower rotational speed 2750 rpm the burning period amounted to 36 deg CA (2.18 ms) and is shorter than at higher rotational speed 3400 rpm, where the burning period amounted to 55 deg CA (2.69 ms). The selected coefficients in Vibe function amount as follows:

n=2750 rpm, a=5.9, m=4.0,
n=3400 rpm, a=6.0, m=2.6.

At lower rotational speeds of DFI two-stroke engine the burning process is quicker than at higher speeds, where that process is more softly. For carburetted version the measured brake mean effective pressure is slightly higher than for DFI system in lower rotational speeds. For the sake of applied commercial injector tests of DFI engine were carried out in smaller engine range of rotational speed (2200 – 4200 rpm). One can state that coefficients in Viebe function are depended on engine load and rotational speed. However the first *a* coefficient can receive value about 6.0 independently of rotational speed. The second parameter *m* should decrease with rotational speed because of lengthen of combustion process accordance to crankshaft rotation and should be in the range of 2.2 to 4.0 for this engine.

8. Conclusions

The injector applied in tested two-stroke industrial engine Robin EC12 was the high pressure standard automotive injector applied in DFI four-stroke engines, which was not suitable for small capacity two-stroke engine. However, on the carried out calculations and experimental work some conclusions can be presented.

1. On the base of pressure measurements in the cylinder the unknown parameters of the approximate Vibe function were matched for some rotational speeds. This approximation function is needed in the most 0-dimensional models of combustion process in the piston engines. The time of fuel combustion increases with rotational speed.
2. Combustion process is determined by engine rotational speed, injected fuel mass, injection timing and charge turbulence in the combustion chamber. Knowing of the real combustion time of given fuel dose and the parameters of Vibe function the heat release rate can be calculated in the computer program with 0-dimensional combustion model.
3. For DFI engine injection process takes place after closing of the exhaust ports and only unburned fuel is delivered to the outflow system. Direct fuel injection does not assure higher engine power, however the engine has lower bsfc. Propagation of the fuel jet in DFI engine depends strongly on the gas motion in the combustion chamber, which is caused by the scavenge process and squeezing effect.
4. The modified engine almost ten times reduces hydrocarbons emission as an effect of elimination of escaping of fuel mixture during the scavenge process. Volumetric concentration

of HC changes with rotational speeds and can reach values below 100 ppm. The HC emission for λ=1 for the tested engine HC emission reaches 2 g/kWh. Volumetric ratio of NOx does not exceed 300 ppm at whole range of engine load.

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