A VALVED TWO STROKE ENGINE AS A NEW POWER SOURCE

Wladyslaw Mitianiec

Cracow University of Technology
Faculty of Mechanical Engineering
Jana Pawla II 37 , 31-864 Cracow, Poland
tel.: +48 12 6283692, fax: +48 12 6283690
e-mail: wmitanie@usk.pk.edu.pl

Konrad Buczek

Cracow University of Technology
Faculty of Mechanical Engineering
Jana Pawla II 37 , 31-864 Cracow, Poland
e-mail: buczek@fev.com

Abstract

The present work defines main conception of modern, efficient and ecologic two-stroke engine. It indicates the scope of necessary modifications that have to be made in typical four-stroke engine to prepare it to work in two-stroke mode. Taking into account some limitations (especially in valve train design) authors performed several analyzes in order to check the possibility of proper scavenging process and obtain main engine characteristics. The paper contains description of simulation approach and selected mathematical models that was applied in carried over analyzes. The results of foregoing simulations, which were obtain taking advantage of GT-Power software should be considered as preliminary since several parameters have to be defined during research and 3D simulations. In order to expose advantages of modern two-stroke engine a simple comparison of it and its predecessor was performed taking into consideration BSFC maps. The objects of simulation were fluid flow, scavenging and combustion. This comparison indicates that proposed two-stroke engine achieve significantly less break specific fuel consumption (up to 27 g/kWh drop). Paper presents diagram of new type two-stroke engine with poppet valves, which is probably the most feasible layout of two-stroke engine, and GT-Power model of simulated single-cylinder engine based on four-stroke single cylinder engine.

Keywords*: transport, engine development, two-stroke engine, valve timing, pollutants*

1. Introduction

Four-stroke internal combustion engines have been highly modernized and improved during last years. Despite of their numerous disadvantages, it owes the market success to perfect combination of manufacturing cost, performance and ecological properties. Especially, recent increase of mechanical efficiency by downsizing kept the four-stroke engine development at a high level. Nevertheless, it seems that we are achieving the limit of four-stroke engine performance nowadays. High thermal and mechanical load or risk of engine knock presents the threshold in contemporary high efficient four-stroke engines.

Further reduction of brake specific fuel consumption (BMEP) of this type of engine is hardly possible without any innovative design changes. One of revolutionary approach to achieve this goal is to reduce two idle strokes: intake and exhaust in order to obtain significant reduction in mechanical losses. It simply means the return of two-stroke engine as a common power source. Two-stroke, but completely different than two-stroke that we already know. The new two-stroke engine will not produce high amount of hydrocarbons and obtain long time between overhauls thus dispose of the most important disadvantages of the predecessor. This can be achieved by application of common four-stroke engine structure with its intake and exhaust poppet valves and crankcase intended for lubricating purposes to the engine working in a two-stroke mode.

Above mentioned conception force to use well controlled air-charge system to ensure proper scavenge in all engine steady and transient operating states. Probably the most feasible layout of this system is shown in Fig. 1.

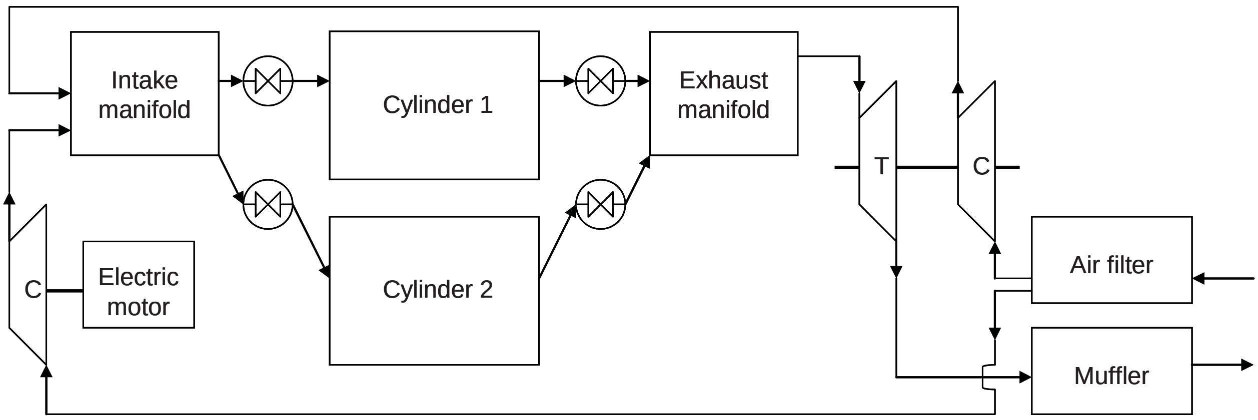

Fig. 1. Flow diagram of new type two-stroke engine with poppet valves

The turbocharger presents the main component of intake ducts. It ensures a required pressure in intake manifold for scavenging process in medium and high engine load (steady states). Additional compressor driven by an electric motor (or any other easy to control drive) is applied in order to perform scavenging at engine idle, low load and transient states. This compressor may consume some energy from crankshaft what slightly reduces overall engine efficiency in transient states but alternatively it can be for example supplied from hybrid systems what eliminates any additional losses.

In order to obtain theoretically continuous exhaust gas flow and engine smoothness only two cylinders may be applied, what gives similar effect as four-stroke four cylinder engine. This highly reduces engine friction losses and provides proper parameters for turbine work. Additionally, engine dimensions and overall mass are being lowered reducing vehicle fuel consumption and improving its performances.

Another difference between new two-stroke engine and its four-stroke predecessor occurs in a timing drive. Full period of valve actuation takes place during one rotation of crankshaft currently, what means that camshaft or camshafts have to rotate at the same rotational speed as crankshaft do. This may cause some problems with valve train elements acceleration, which rises proportionally to the square of rotational velocity using the same cam profiles. It obviously facilitates to obtain proper time areas in short cam angles, what takes place in the typical two-stroke engine. Nevertheless, it seems to be feasible to design a proper valve train system that can combine relatively low valve acceleration and sufficient time-area of valve lift.

To guarantee maximum fuel utilization, engine shown in Fig.1 has to be equipped with direct fuel injection system. It enables to scavenge the cylinder with fresh air without any fuel losses during this process and to form stratified or homogeneous mixture depending on engine operating conditions. In three main mixture formation systems: wall guided, air guided and spray guided, the last one looks like it would be the most suitable. First produces some amount of particulate matter by fuel coking on piston crowd surface and second requires high charge motion and produces additional number of hydrocarbons caused by increased wall-wetting. Spray guided system is free of above-mentioned disadvantages and enables proper mixture formation in wide range of engine speed.

Described above new type of two-stroke engine eliminates all main faults of the typical one, but also reduces some of its main advantages: simple structure and low manufacturing cost. The valved two-stroke engine would have been expensive nowadays, especially because of high cost of the turbocharger, electric charger and additional high voltage power supply for the electric charger motor as well. Nevertheless, taking into consideration wide application of TC's in passenger cars in the nearest future, the cost of foregoing parts should significantly drop, making the cost of modern two-stroke engine at acceptable level.

2. Simulation approach and applied mathematical models

2.1. General description of simulation approach

In order to investigate the feasibility of proper scavenging and obtain main engine characteristics, several analyses have been performed. All of them have been carried out in GT-Power software as one-dimensional simulations. The results of this type of simulations are preliminary and have to be supplemented with proper 3D analysis in order to obtain reliable model response. Authors are intended to perform them in the nearest future taking advantage of KIVA software and in the end, check the simulation results on the real modified four-stroke engine adapted to operate in two-stroke mode.

Nevertheless, one-dimensional simulation results give the direction of valve timing modifications and specify the range of required intake pressure that enables scavenge process. Analysed two-stroke engine was based on existing four-stroke single cylinder engine (Fig. 2). Because the charge system and its control wasn't the subject of research, pressure difference between intake and exhaust substituted their influence on gas exchange process. Therefore, single cylinder engine model could have been employed.

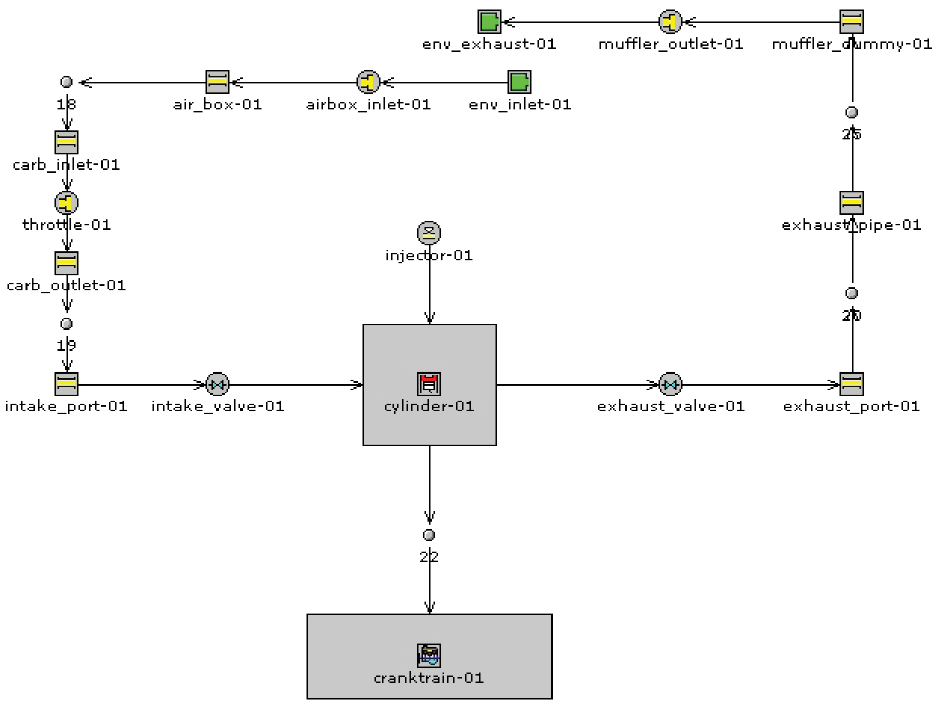

Fig. 2. GT-Power model of simulated single-cylinder engine

2.2. Fluid flow

Fluid flow modelling is crucial to simulate the scavenge process properly. In one-dimensional models, intake and exhaust ducts are split into sub-volumes. Then the continuity (1), momentum (2) and energy (3) equations are solved simultaneously for all elements.

$$\frac{dm}{dt} = \sum_{boundaries} \dot{m}\,, \tag{1}$$

$$\frac{d\dot{m}}{dt} = \frac{dp\,A + \sum\limits_{boundaries} (\dot{m}u) - 4C_f \dfrac{\rho u|u|}{2}\dfrac{dx\,A}{D} - C_p(\dfrac{1}{2}\rho u|u|)\,A}{dx}\,, \tag{2}$$

$$\frac{d(me)}{dt} = p\frac{dV}{dt} + \sum_{boundaries}(\dot{m}H) - h\,A_s(T_{fluid} - T_{wall})\,, \tag{3}$$

where:

$\dot{m}$ – boundary mass flux into volume,
m – mass of fluid in the volume,
V – volume,
p – pressure,
ρ – density,
A – flow area (cross-sectional),
A_s – heat transfer surface area,
e – total specific internal energy (internal energy plus kinetic energy) per unit mass,
H – total enthalpy,
h – heat transfer coefficient,
T_{fluid} – fluid temperature,
T_{wall} – wall temperature,
u – velocity at the boundary,
C_f – skin friction coefficient,
C_p – pressure loss coefficient,
D – equivalent diameter,
dx – length of mass element in the flow direction,
dp – pressure differential acting across dx.

Foregoing formulas represents conservation equations witch occurs in both space and time. There are two different approaches of solving this type of mathematical problems.

Explicit method returns the values of pressure, temperature etc. basing on the values of only the subvolume in question and its neighbours. The iteration is not necessary in this type of calculation (direct solution of each time step), but to provide numerical stability the Courant condition has to be satisfied:

$$\frac{\Delta t}{\Delta x}\left(|u| + c\right) \le 0.8\,m\,, \tag{4}$$

where:

Δt – time step,
Δx – minimum discretized element length,
u – fluid velocity,
c – speed of sound,
m – time step multiplier (≤ 1).

Explicit method is recommended for simulations of objects where pressure pulsation has to be taken into account, thus this method was used in all analyses performed by authors.

The implicit method obtains values of all sub volumes at the new time by solving system of algebraic equations which base on values from previous time step. In consequence of non-linearity of system of equations, the iterations are necessary until the solution is converged. This method provides great numerical stability (long time step can be applied) therefore its advantages are especially useful in transient analyses with long time period. Nevertheless, the implicit method is useless for modelling of fluid flow in the intake and exhaust ducts of internal combustion engines because of difficulties in modelling pressure waves.

2.3. Scavenging

In connection with fact, that one-dimensional analyzes are not able to simulate the in-cylinder flow directly during gas exchange period, the scavenging curve has to be defined. It describes the relation between cylinder residual ratio (ratio of the mass of burned gases in the cylinder to the mass of all gases in the cylinder) and the scavenging ratio (ratio of the mass of burned gases exiting the cylinder to the mass of all gases exiting the cylinder). This relation has to be defined either by a proper 3D analysis or an experimental research. Because of preliminary character of the performed analyzes, authors assumed specific scavenging function basing on their experience. This relation, typical two-stroke engine curve and perfect mixing curve are all shown in Fig. 3.

In the nearest future proper 3D simulations will be performed in KIVA, thus corrected scavenging curve will have been applied.

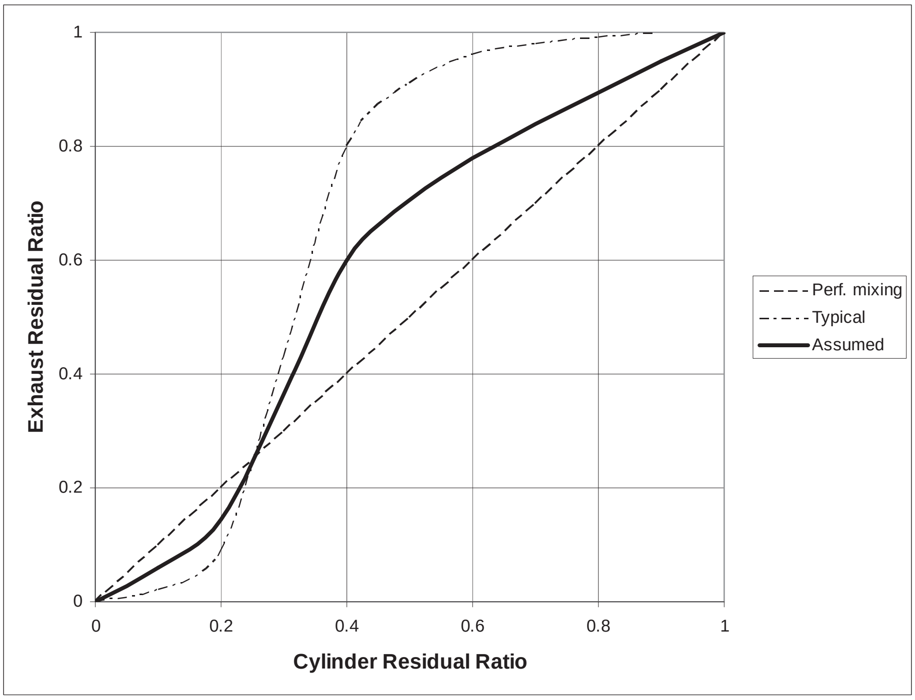

Fig. 3. Scavenging functions: typical two-stroke engine curve, perfect mixing and assumed by authors curves for two-stroke engine with poppet valves

2.4 Combustion

Two-zone combustion model was applied in all performed simulations. This means that the in-cylinder gases were divided into both unburned and burned zone. At the beginning of the process in this type of combustion model only unburned zone occurs, which includes new charge and residual gases (from previous cycle and EGR system – if exists). The burn rate defines the amount of mixture of air and fuel that is transferred form unburned to burned zone at each time step. Simultaneously the chemical composition of burned zone is calculated by formulas of equilibrium concentration of 11 products of combustion (N_2, O_2, H_2O, CO_2, CO, H_2, N, O, H, NO, OH) and then, following energy conservation equations are solved for unburned (5) and burned (6) zone separately:

$$\frac{d(m_u e_u)}{dt} = -p\frac{dV_u}{dt} - Q_u - \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right) + \frac{dm_{f,i}}{dt}h_{f,i}, \tag{5}$$

$$\frac{d(m_b e_b)}{dt} = -p\frac{dV_b}{dt} - Q_b - \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right), \tag{6}$$

where:

m_u/m_b – unburned/burned zone mass,
m_f – fuel mass,
m_a – air mass,
$m_{f,i}$ – injected fuel mass,
e_u/e_b – unburned/burned zone energy,
p – cylinder pressure,
V_u/V_b – unburned/burned zone volume,
Q_u/Q_b – unburned/burned zone heat transfer,
h_f – enthalpy of fuel mass,
h_a – enthalpy of air mass,
$h_{f,i}$ – enthalpy of injected fuel mass.

Described procedure enables to obtain temperatures of burned zone, unburned zone and cylinder pressure.

In all performed analysis, the Wiebe function approximated with sufficient accuracy the cumulative burn rate of typical SI engine in comparison with real engine combustion process. It circumscribes the burn rate as a derivative of cumulative burn rate. The function can be expressed as (7):

$$CBR(\theta) = (CE)[1 - e^{-(WC)(\theta - SOC)^{(E+1)}}], \tag{7}$$

where:
$CBR(\theta)$– cumulative burn rate as a function of crank angle,
CE – fraction of fuel burned (combustion efficiency),
WC – Wibe constant,
θ – instantaneous crank angle,
SOC – start of combustion,
E – Wibe exponent (def. 2).

Wibe constant and start of combustion are defined by (8), (9):

$$WC = \left[\frac{D}{BEC^{1/(E+1)} - BSC^{1/(E+1)}}\right]^{-(E+1)}, \tag{8}$$

$$SOC = AA - \frac{(D)(BMC)^{1/(E+1)}}{BEC^{1/(E+1)} - BSC^{1/(E+1)}}, \tag{9}$$

with input data:

AA – anchor angle,
D – duration,
BMC – burned midpoint constant,
BSC – burned start constant,
BEC – burned end constant.

Foregoing constants are determined by relations:

$$BMC = -\ln(1 - BM), \tag{10}$$

$$BSC = -\ln(1 - BS), \tag{11}$$

$$BEC = -\ln(1 - BE), \tag{12}$$

where:

BM – burned fuel percentage at anchor angle (def. at 50%),
BS – burned fuel percentage at duration start (def. at 10%),
BE – burned fuel percentage at duration end (def. at 90%).

The typical curves of cumulative burn rate and burn rate of SI engine are shown in Fig. 4. Several functions like in figure below have been used to model combustion process correctly in optimization analyzes performed by authors.

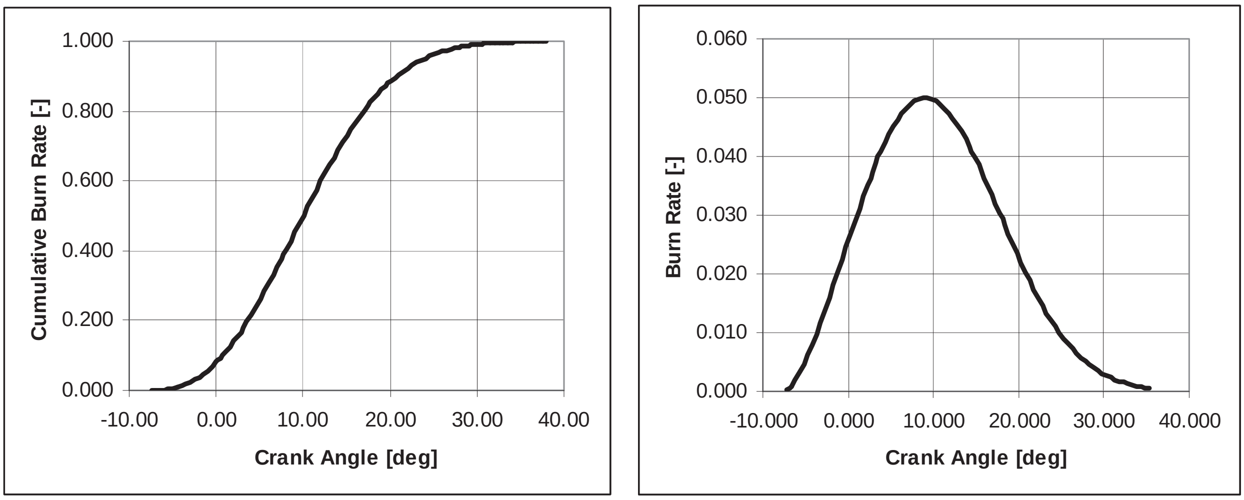

Fig. 4. Cumulative burn rate (left) and burn rate (right) of typical SI engine at full load and 4000 rpm engine speed

3. Simulation results

The objective of performed simulations was to investigate the feasibility of proper scavenging of the cylinder of two-stroke engine with poppet valves and obtain main engine characteristics. Carried over simulations have shown the crucial influence of valve timing on gas exchange process. It has been found that there are optimum of cam angles that enable to obtain both high BMEP and low BSFC but they can not occur simultaneously because of different required valve opening angles. Nevertheless, both goals can be easily realized with any common variable valve train system. The optimal valve lifts are shown in Fig. 5. All of them have been obtained using 120deg exhaust and 100deg intake cam angles.

In order to achieve high BMEP at assumed intake – exhaust pressure difference (1 bar) following valve timing angles are to be applied: EVO: 110 deg, IVO: 150 deg. Relatively long blow down angle (40 deg) lowers the in-cylinder pressure to minimum thus enables the intake air to get into cylinder without excessive backflow at the start of intake valve opening. The exhaust process starts early therefore transformation of in-cylinder energy to piston work is slightly limited what increases BSFC, but also enables to maximally scavenge cylinder and obtain high BMEP.

Different timing angles should be applied to achieve high engine efficiency. It requires elongated power stroke in comparison with compression stroke, thus both intake and exhaust valve opening angles have to be shift forward (EVO: 140 deg, IVO: 170 deg). In order to avoid excessive backflow during intake valve closing, it should be shifted less than exhaust valve opening, not to exceed 180 deg. It automatically decreases blow down angle and reduces BMEP at specific intake-exhaust pressure difference.

Several different simulations were performed to find foregoing results. The variable was: intake and exhaust cam angles, valve timing angles, engine speed and intake-exhaust pressure difference. It enabled to observe the engine air flow phenomenon at different conditions. Fig. 6 shows the mass flow rate through intake valve for three sets of valve timing angles.

The red line presents a backflow which occurs as a result of too short blow down angle (only 10 deg). In this situation the in-cylinder pressure is higher than instantaneous intake port pressure, thus the backflow take place at the beginning of intake valve opening. It is undesirable for both BMEP and BSFC.

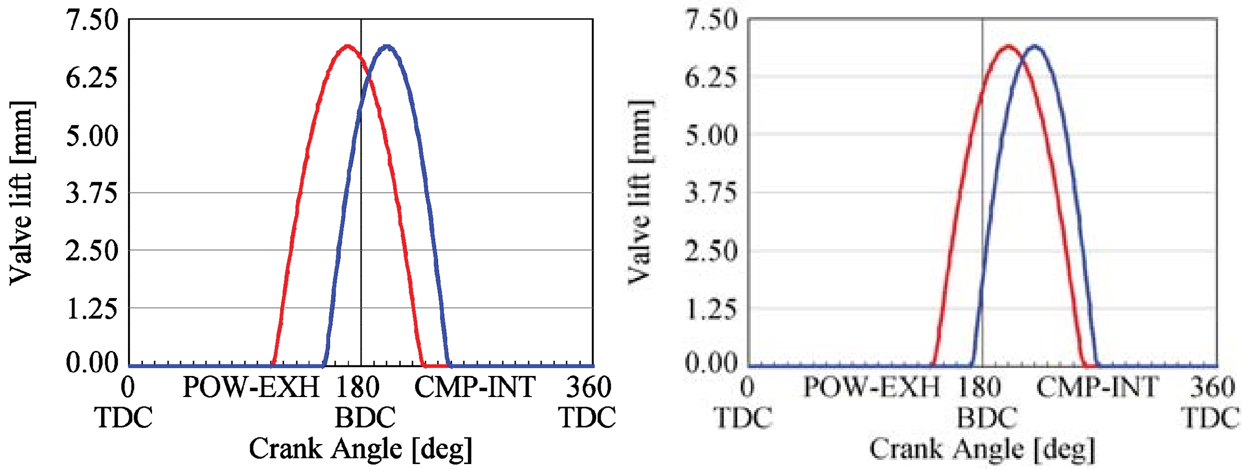

Fig. 5. Intake (blue line) and exhaust (red line) valve lift curves. Left scatter plot presents optimal timing for high BMEP and right for low BSFC

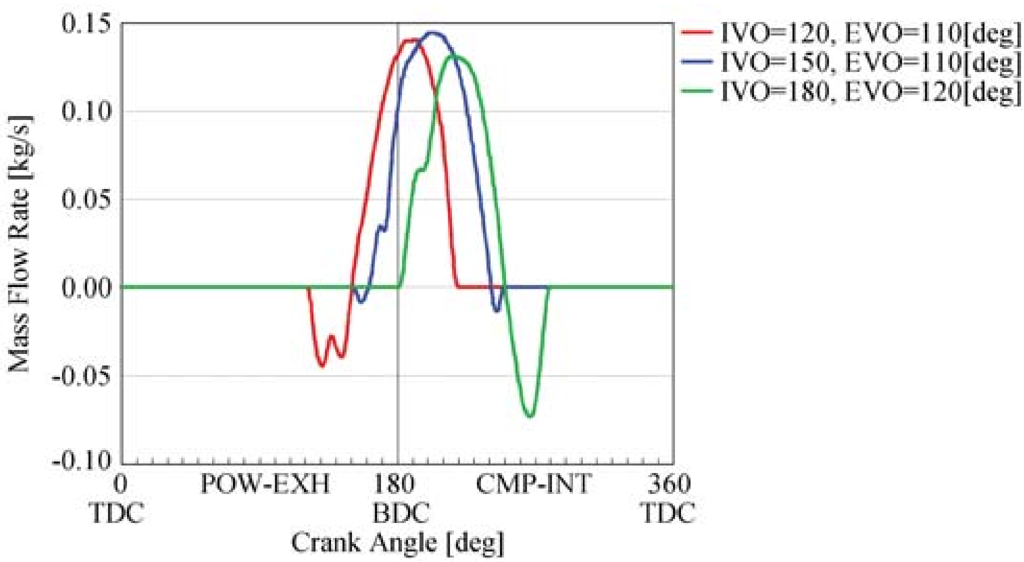

Fig. 6. Mass flow rate through intake valves for different valve timing angles

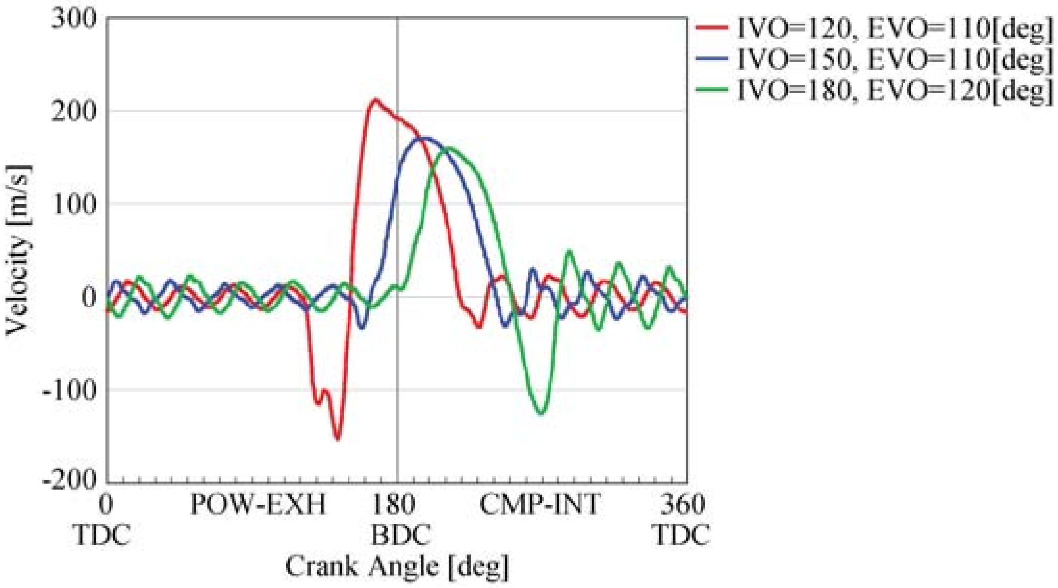

Fig. 7. Air flow velocities through intake port for different valve timing angles

The green line in Fig. 6 shows an effect of too late opening of the intake valve. There is no backflow at the beginning of valve opening (because of long blow down period) as it was in previously described situation but a significant backflow occurs at intake valve closing. It is caused by the fact that decreasing of volume above the piston occurs simultaneously with opened intake and exhaust valve. Because exhaust valve is almost closed then, the flow through it presents with high resistance. Therefore, in-cylinder pressure rises above the instantaneous intake port pressure and as a result significant backflow during intake valve closing.

The optimal valve timing angles (the same as in Fig. 5 – left) results in mass flow rate through the intake valve shown as a green line in Fig. 6. It reduces to minimum both backflows giving proper scavenging process and therefore also high BMEP.

Fig. 7 presents the air flow velocity through intake port of the same analysis that was shown in Fig. 6. The curves have generally similar shape, but velocity pulsation can be noticed during intake valve close period as a result of pressure wave occurrence.

In order to expose advantages of modification of typical four-stroke engine to work in two-stroke mode, BSFC maps of both four- and two-stroke engines have been prepared (Fig. 8). The empty zones of two-stroke engine map are caused by specific simulation approach – engine load was set by determination of pressure difference between air filter inlet end exhaust pipe outlet. The lowest value of this difference was specified as 0.4 bar and this was the reason of empty bottom left zone occurrence (Fig. 8 – left). The top right empty zone was generated in connection with top limit of above-mentioned pressure difference (2 bar) and the last empty zone by probability of engine knock.

It can be noticed that at the same BMEP (9 bar as a maximum of four-stroke engine) the two-stroke engine obtain ca. 20 g/kWh lower BSFC. This is caused especially by better transformation of in-cylinder energy to mechanical work (valve timing of right scatter plot of fig. 5 have been applied) and obvious reduction of necessary strokes. After taking into consideration entire two-stroke engine operating area, the drop of BSFC in comparison with four-stroke engine rises up to 27 g/kWh.

4. Conclusions

1. Combination of advantages of both two- and four-stroke engine is possible by application
2. of typical four-stroke engine structure to modern two-stroke engine.

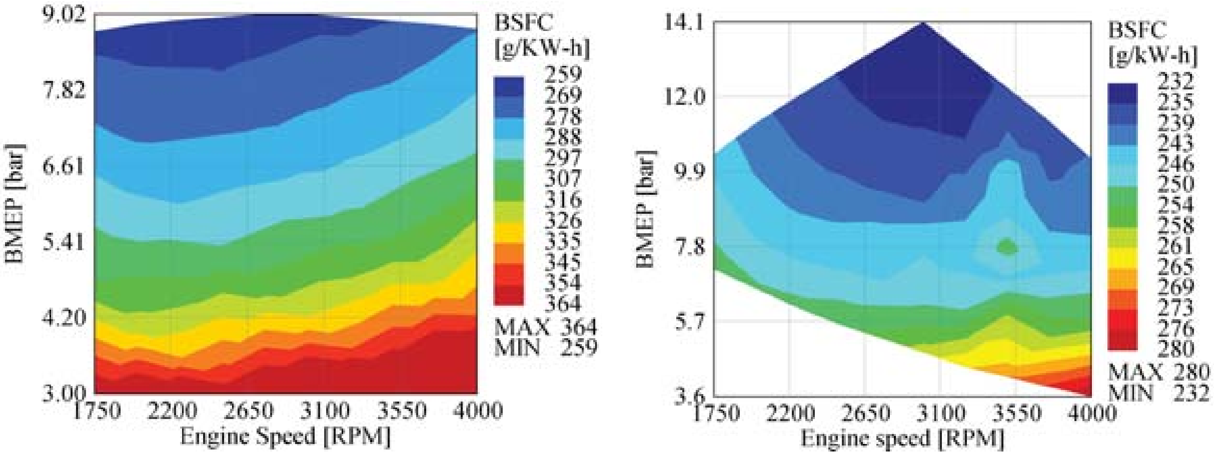

Fig. 8. BSFC map of four-stroke predecessor (left) and two-stroke engine with poppet valves (right)

3. Significant downsizing effect can be obtained by application of new type of two-stroke engine.
4. Application of two-stroke engine with poppet valves at simulated example gave 27 g/kWh reduction of BSFC in comparison with the four-stroke precursor.
5. There is necessity to apply complicated and expensive charge system to ensure scavenging process at steady and transient states.
6. Proper valve timing is crucial to engine performance and efficiency.
7. There is possibility to adjust engine load by variation of intake pressure (VTG, waste gate, additional charger etc) without throttle application.

References

[1] Basshuysen, R., Schaefer, F., *Internal Combustion Engine – Handbook*, SAE International, 2004.
[2] Blair, G. P., *Design and Simulation of Two-Stroke Engines*, SAE Inc., Warrendale 1996.
[3] *GT-Power User’s Manual*, Gamma Technologies, V.6.2, 2006.
[4] Heywood, J.B., *Internal Combustion Engines Fundamentals*, McGraw-Hill Inc., New York 1988.
[5] Mitianiec, W., Buczek, K., *A New Approach in Development of Two-Stroke Engines*, PTNSS-2009-SC-042, Bielsko-Biała 2009.
[6] Mitianiec, W., Buczek, K., *A New Design of Ecologic Two-Stroke Engine*. EAEC, 2009.
[7] Wajand, J. A., Wajand, J. T., *Tłokowe Silniki Spalinowe średnio- i szybkoobrotowe*, WNT, Warszawa 2005.