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Ewaluation of influence of changes of camshaft's phases on filling the cylinder of the engine with self-ignition

Abstract: n the article, the evaluation of influence of changes of camshaft's phases of the engine with self-ignition on filling the cylinder of the engine with self-ignition has been introduced. The evaluation has been done based on the results of simulating tests of the engine's approaching system. For the purpose of the simulating tests the mathematical model of the engine and the elaborated by the author its numerical solution based on the McCormack's method have been used. In conducted simulating tests the changes of lifting of the approaching valve depending on the angle of crankshaft's rotation took place. The tests' results have been presented in the form of profile of an air mass changeability left in the cylinder after the approaching valve was closed as a function of the crankshaft's rotational speed of the engine for different angles of opening and closing of the approaching valve.

Key words: *phases of camshaft, filling, numerical simulation, verification of the model*

Ocena wpływu zmian faz rozrządu na napełnienie cylindra silnika o zapłonie samoczynnym

Streszczenie: W artykule przedstawiono ocenę wpływu zmian faz rozrządu silnika o zapłonie samoczynnym na napełnienie cylindra silnika o zapłonie samoczynnym. Oceny dokonywano na podstawie wyników badań symulacyjnych układu dolotowego silnika. Do badań symulacyjnych wykorzystano matematyczny model silnika i opracowane przez autora numeryczne jego rozwiązanie oparte na metodzie McCormacka. W przeprowadzanych badaniach symulacyjnych dokonywano zmian wzniosu zaworu dolotowego w zależności od kąta obrotu wału korbowego. Wyniki badań przedstawiono w formie charakterystyk zmienności masy powietrza pozostającej w cylindrze po zamknięciu zaworu dolotowego w funkcji prędkości obrotowej wału korbowego silnika dla różnych kątów otwarcia i zamknięcia zaworu dolotowego.

Słowa kluczowe: *fazy rozrządu, napełnienie, symulacja numeryczna, weryfikacja modelu*

1. Introduction

The piston internal-combustion engine, depending on type, operates according to the adequate thermal circulation. These circulation during the engine's operation repeat continuously. In each thermal circulation the processes of charge exchange, compression and decompression take place. The charge exchange in the piston internal-combustion engine depends on the removal from the engine's cylinder remains after the combustion process in the previous thermal circulation and to provide the cylinder with fresh charge. The process of providing the fresh charge usually is called as the filling process. The filling process is triggered off by the piston movement from the upper dead position (GMP) to the bottom dead position (DMP). During that time, in the cylinder the underpressure is generated, thanks to which the flow of charge from the surroundings to the engine's cylinder is possible. It should be mentioned here, that the charge's flow is accompanied by a number of usually unfavourable effects which cause smaller filling of the cylinder with fresh charge, than it would appear from the cubic capacity. The unfavourable effects which decrease the filling of the cylinder are: stifling the flow in the narrowings of the approach system, heating up charge in the approaching system, and the flow resistance in the engine's approaching system [3,4,6].

During exchange of charge in the cylinder, the processes of filling by fresh charge and take off of exhaust fumes take place, which period of duration did not fully agree with the strokes of filling and the take off of the exhaust fumes [10]

2. Model of the engine's approaching system

For ensuring sufficient effectiveness of the filling process, by providing the adequate mass of charge (air) to the engine's cylinders, without necessity of applying additional recharging devices (compressors), the geometrical parameters of the charge exchange system and the parameters of the camshaft system should be suitably selected.

In the system of charge exchange, the greatest importance from the increase of filling point of view has the engine's approaching system. However, in the camshaft system, the phases of camshaft have the greatest importance. The most significant parameters of the approaching system are:

- length of the approaching line,
- cross section surface area of the approaching line,
- shape of the approaching line.

The parameters of the approaching system can be determined as a result of the engine test bench. Such tests are carried out by using the method of successive approximation, that's why their execution is so laborious and expensive. Therefore, it seems to be appropriate to use the methods of modelling the course of the dynamic effects in the process of charge exchange [8].

The air flow in the approaching system can be described by the use of basic equations, which were formulated for the general model of liquid and resulting from tree basic principles of mechanics [6,8]:

- the principle of mass preservation,

$$\frac{\partial \rho}{\partial t} + div(\rho\, u) = 0 \qquad (2.1)$$

- the principle of speed and moment of speed preservation

$$\rho \frac{du}{dt} = \rho\, F + divS \qquad (2.2)$$

- the principle of energy preservation

$$\rho \frac{d}{dt}\left(T\, C_v + \frac{u^2}{2} \right) = \rho\, F\, u + \rho\, q + div(\Gamma\, grad\, T) + div(S\, u) \qquad (2.3)$$

Making a number of simplifying assumptions, the equations (4.1), (4.2), (4.3) can be written down as:

$$\begin{aligned} \frac{\partial \rho}{\partial t} &= -u \frac{\partial \rho}{\partial x} - \rho \frac{\partial u}{\partial x} \\ \frac{\partial u}{\partial t} &= -\frac{1}{\rho}\frac{\partial \rho}{\partial x} - u \frac{\partial u}{\partial x} - u \cdot k_t \\ \frac{\partial p}{\partial t} &= u^2 \cdot k_t \cdot \rho(\kappa - 1) - u \frac{\partial p}{\partial x} - \kappa \cdot p \frac{\partial u}{\partial x} \end{aligned} \qquad (2.4)$$

During the process of filling the cylinder with air, the physical occurrences are divided into two groups. In the first group, the springy flow through the approaching line described by the equations (4.1), (4.2), (4.3) are considered. The second group relates to the thermal occurrences in the cylinder. Analysing the air flow through the approaching system, the occurrences in the cylinder are described with the use of the zero-dimension models which skip the movement of the charge within the cylinder's space. Because of that, the principle of conservation of momentum and the moment of momentum is not taken under consideration. Also the equations of mass preservation and energy conservation can be presented in simplified form:

$$\frac{d}{dt} m = \frac{d}{dt}(m_d - m_w) \qquad (2.5)$$

$$E_d = \Delta U + E_w \qquad (2.6)$$

The MacCormack's method [1,2,4,7] has been used to find solution to a system of equations. The MacCormack's method is particularly useful for finding solutions to non-viscous flows of gases. The earlier attempt of finding the numerical solution of the problem with the use of straight (lines) method didn't give desirable results because of high instability.

Solution of the partial differential equations in this method take place in two steps. During the first step, the estimated value (predictor) of the function is calculated in the next time moment $\bar{f}(n+1)$, replacing the partial differential within the x domain by the the differential quotient in forward. During the second step *the corrector* calculates the value of the function f(n+1) using the results from predictor and replacing differential within the x domain by the differential quotient in back. The functioning of the method is presented by equations (2.7):

$$\bar{\rho}_i^{n+1} = \rho_i^n - u_i^n \frac{\Delta t}{\Delta x}\left(\rho_{i+1}^n - \rho_i^n\right) - \rho_i^n \frac{\Delta t}{\Delta x}\left(u_{i+1}^n - u_i^n\right)$$

$$\rho_i^{n+1} = \rho_i^{n+\frac{1}{2}} - u_i^{n+\frac{1}{2}} \frac{\Delta t}{2\Delta x}\left(\bar{\rho}_i^{n+1} - \bar{\rho}_{i-1}^{n+1}\right) - \rho_i^{n+\frac{1}{2}} \frac{\Delta t}{2\Delta x}\left(\bar{u}_i^{n+1} - \bar{u}_{i-1}^{n+1}\right)$$

$$\rho_i^{n+\frac{1}{2}} = \frac{\rho_i^n + \bar{\rho}_i^{n+1}}{2} \qquad (2.7)$$

$$\bar{\rho}^{n+1} = \rho^n - u^n \cdot \vec{D}\left(\rho^n \cdot \Delta t\right) - \rho^n \cdot \vec{D}\left(u^n \cdot \Delta t\right)$$

$$\rho^{n+\frac{1}{2}} = \frac{\left(\rho^n + \bar{\rho}^{n+1}\right)}{2}$$

The above equations can be presented in the vectorial form (2.8)

$$\rho^{n+1} = \rho^{n+\frac{1}{2}} - u^{n+\frac{1}{2}} \cdot \overleftarrow{D}\left(\bar{\rho}^{n+1} \cdot \frac{\Delta t}{2}\right) - \rho^{n+\frac{1}{2}} \cdot \overleftarrow{D}\left(\bar{u}^{n+1} \cdot \frac{\Delta t}{2}\right) \qquad (2.8)$$

3. Results of the simulation tests

For determining the influence of constructional parameters of the approaching system, the simulation program, elaborated by the author, and operated in the Matlab environment based on the MacCormack's method has been used.

Basic advantage of the simulation program is to determine the most profitable geometrical parameters of the approaching system for new designed or modernised engine or improving the construction of the approaching system of existing engine in order to improve its operational parameters or adapting the engine's operational profile to the conditions of its work.

To make a decision based on the analysis of results of the simulation tests reduces costs and time indispensable for determining the most profitable, in given conditions, parameters of the approaching system.

Taking advantage of calculation methods for finding solution of problems related to filling the cylinder of combustion engine, the experimental tests confirm the pertinence of taken decision.

The worked out program makes possible to determine:

- air mass left in the engine's cylinder after finishing the filling process,
- pressure in any given point of the approaching line,
- pressure in cylinder,
- temperature in cylinder.

The parameters m_c, p_c, and T_c are presented in the form of charts of changeability of a given parameter as a function of the angle of the crankshaft rotation. Additionally, it is possible to visualise the course of pressure change p_d in the approaching line along its length with plotted phases of opening and closing of the approaching valve.

In order to make the simulation, it was necessary to introduce the constant datas which characterised given engine and the variable datas which made possible the evaluation of the constructional parameters of the approaching system.

As constant datas for simulation the datas regarding the engine, surroundings parameters and basic thermodynamics datas were adopted /taken, accepted/. The engine with self-ignition, SW-680 served as an object of the simulation tests.

In table 3.1 are presented the lifts of the approaching valve as a function of the rotation angle of the factory made crankshaft (variant A), subsequent, to factory made, closing of the approaching valve (variant B), earlier, to factory made, closing the approaching valve (variant C) as well as earlier opening and earlier closing of the approaching valve (variant D).

Table 3.1. Setting up of lifts of the approaching valve in the process of filling
Tabela 3.1. Zestawienie wzniosów zaworu dolotowego w procesie napełniania

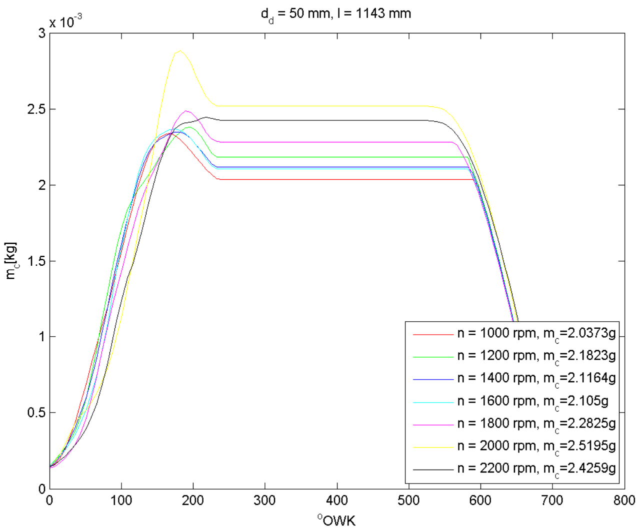
	Kąt obrotu wału korbowego									
Wariant	690°	0° GMP	30°	60°	90°	120°	150°	180° DMP	210°	240°
A	0	0,4	2,8	6,0	7,5	9,0	9,5	7,0	3,0	0
B	0	0,4	2,8	6,0	7,5	9,0	9,5	7,5	4,0	2,0
C	0	0,4	2,8	6,0	7,5	9,0	9,5	4,0	0	0
D	0,4	2,8	6,0	7,5	9,0	9,5	7,0	3,0	0	0

In Fig. 3.1 the characteristic of changeability of air mass as a function of the angle rotation of the crankshaft for the full engine cycle has been presented. The characteristic has been made for the factory made camshaft.

In the initial phase, from 0º to a little bit above 200º OWK the air mass flowing in to the engine's cylinder increases. During this time the piston moves from GMP do DMP. It is well-known, that closing of the approaching valve usually takes place after the piston passes DMP. Therefore within the range from DMP until the moment of complete closing of the approaching valve, the piston follows in the GMP direction pushing out from the cylinder a part of the delivered charge.

After the approaching valve has been closed, the air mass doesn't change, obviously with the assumption of passing over the blowthrough through the cylinder's leakage as well as with the assumption of lack of processes of fuel injection and combustion. In Fig. 5.1. that period is presented by straight line within the range from a little above 200º OWK to about 600º OWK, where the start of opening of the approaching valve takes place, after which the mass in the cylinder decreases, and then the cycle repeats.

Fig.3.1. Mass of air in the cylinder after closing the approaching valve as a function of the rotation



angle of the crankshaft of the factory made camshaft system (variant A)
Rys. 3.1. Masa powietrza w cylindrze po zamknięciu zaworu dolotowego w funkcji kąta obrotu wału korbowego w wykonaniu fabrycznym układu rozrządu (wariant A)

From Fig. 3.1. appears, that because of accepting such an angle of closing the approaching vale, a part of the charge is removed back to the approaching line. Obviously, it's a loss, because the coefficient of filling is getting lower and the lowering of power and the lowering of engine's torque follow. In presented case, the mass of air remaining in the cylinder after closing the approaching valve (horizontal line) covers the range from 2,1g to 2,5g.

In Fig.3.2. The course of air mass remained in the cylinder after closing the approaching valve later than it took place in factory implementation has been shown.

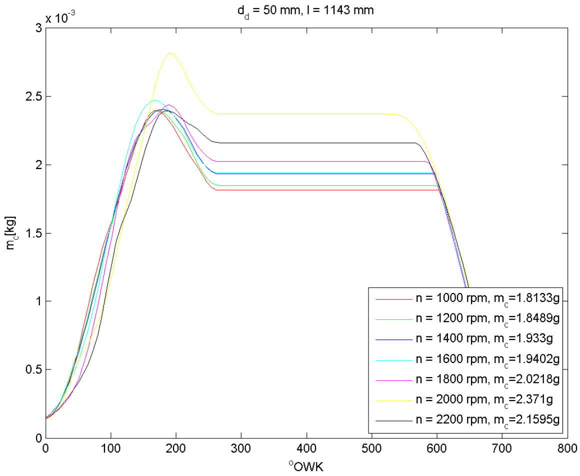


Fig. 3.2. The mass of air in the cylinder after later, than in factory implementation, closing of the approaching valve as a function of the rotational angle of the crankshaft (variant B)
Rys. 3.2. Masa powietrza w cylindrze po późniejszym, niż w wykonaniu fabrycznym, zamknięciu zaworu dolotowego w funkcji kąta obrotu wału korbowego (wariant B)

As it is shown in Fig. 3.2. the mass' loss is bigger than in case of factory implementation of the camshaft system. Obviously it's understood, because the approaching valve is open although from about 60º OWK last the stroke of compression and the piston pushing out operation. Beside, the obvious loss of charge's mass also we are dealing with the loss of compression's pressure. In this case the mass of air remaining in the cylinder after closing the approaching valve are within the range from 1,8g do 2,3g.

In Fig. 3.3. the course of air's mass in the cylinder in case of earlier, than in real engine closing of the approaching valve has been presented. It can be seen, that in this case the loss of air's mass in the cylinder is comparatively small. Hence the mass of air remaining in the cylinder, in this case, is the biggest from investigated so far variants of solutions of the camshaft system, reaching the values between 2.1g and 2,5g.

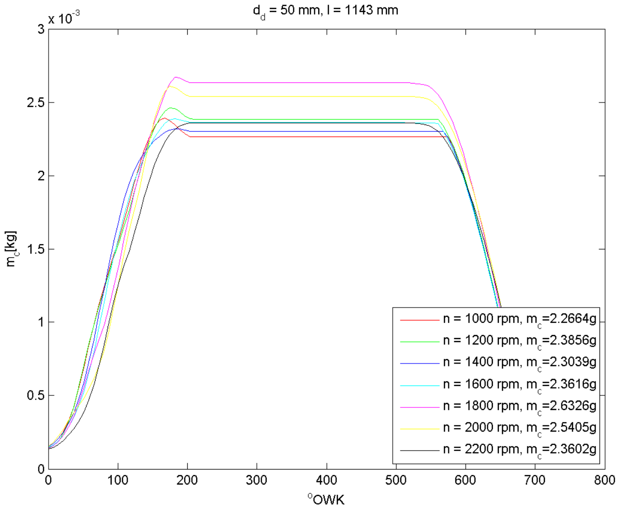


Fig. 3.3. The mass of air in the cylinder after earlier, than in factory implementation, closing the approaching valve as a function of the rotational angle of the crankshaft (variant C)
Rys. 3.3. Masa powietrza w cylindrze po wcześniejszym, niż w wykonaniu fabrycznym, zamknięciu zaworu dolotowego w funkcji kąta obrotu wału korbowego (wariant C)

The last simulation has been presented in Fig. 3.4., it concerns the camshaft system in which the approaching valve opens earlier and closes sooner compare to factory version (variant D), while the opening angle of the approaching valve doesn't change. As it is seen in Fig. 3.4. the loss of air's mass, resulting from the pushing out operation of the piston in this case doesn't take place and the masses of air remaining in the cylinder after closing the approaching valve (horizontal line) are covered by the range from 2,2g to 2,5g, that is they are imperceptibly smaller than in variant C.

Certainly, another simulation can be conducted, which would improve achieved results, however in the author's opinion the goal in the form of analysis of influence of camshaft's phases on filling has been reached.

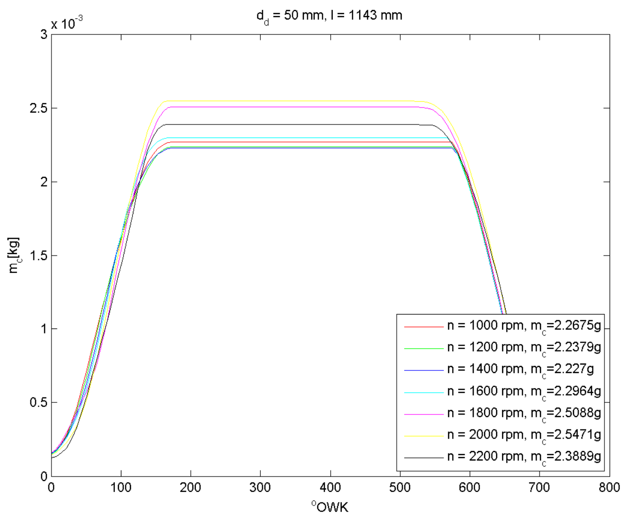


Fig. 3.4. The mass of air in the cylinder after earlier opening and earlier, than in factory implementation, closing of the approaching valve as a function of the rotational angle of the crankshaft (variant D)

4. Evaluation of compatibility of model and experimental tests

In the simulation tests the values of the air's mass in the cylinder after closing the approaching valve have been obtained, however, the effect of the engine test bench was to determine the value of rotational moment or the engine's effective power. Hence, in the author's opinion, it was intentional to calculate the power of the engine based on the values of air's mass in the cylinder received as a result of simulation and comparing them with the engine's power determined based on the experimental tests.

The power of the engine has been calculated based on relation [5]:

$$N_e = \frac{W_d \cdot \rho_{pow} \cdot V_s \cdot i \cdot n}{L_t \cdot \tau} \cdot \frac{\eta_i}{\lambda} \cdot \eta_m \cdot \eta_v \qquad (4.1)$$

The values of the filling factor /coefficient/ and the factor of the air excess λ received from the simulation using the relation:

$$\eta_v = \frac{m_{sym}}{V_s \cdot \rho_{pow}} \qquad (4.2)$$

$$\lambda = \frac{m_{sym}}{q_{pal} \cdot L_t} \qquad (4.3)$$

The mass of the fuel injected to the cylinder during one cycle of work calculated based on the consumption of specific mass of the fuel taken by the engine during the time measured on the engine test bench, based on the relation:

$$q_{pal} = \frac{20 \cdot m_{pal}}{t \cdot n} \qquad (4.4)$$

The constant "20" in the relation (4.4) came from converting the rotational speed from 1/min to 1/s, a number of cylinders and the number of injections due to one engine cycle.

The values of the engine's power calculated based on the simulation and the engine's power measured during the engine test bench, for considered length of the approaching lines with the line's diameter equal to 0,07 m and the distribution of powers' difference, expressed in percentages have been introduced in table 4.1.

Table 4.1. Setting up of the engine's power calculated based on the simulation, measured during the tests as well as the difference expressed in percentages.

Tabela 4.1. Zestawienie mocy silnika obliczonej na podstawie symulacji, zmierzonej w trakcie badań oraz różnica wyrażona w procentach

	0,1143 m		
n	sym	pomiar	ΔN [%]
1000	73,6	72,8	1,1
1200	88,1	92,0	-4,2
1400	101,7	107,5	-5,3
1600	112,2	118,0	-4,8
1800	120,6	125,3	-3,7
2000	130,6	127,5	2,5

5. Conclusions

Based on the model calculations and the tests in a engine test bench done by the author it can be stated, that while constructing a new engine or modernise already existing one a number of analysis regarding constructional parameters of the approaching system and their influence on the cylinder's filling and on operational parameters of the engine such as power, rotational moment or fuel consumption have to be done.

The constructional parameters of the approaching system can be established based on the experimental tests, however, numerousness of parameters will require multiple tests. Probability of obtaining the satisfactory results already during the first series of tests is practically equal to zero, realisation of the whole predicted tests for the series of altered parameters will be unusually time-consuming and expensive. So, good practice is to initially estimate the required parameters and then to make tests confirming the pertinence of choice.

1. For the quantity of filling fairly significant influence have the parameters of the camshaft system, particularly the moment of closing of the approaching valve. According to the author, speeding up a little bit the moment of

closing the valve, the outflow of air's mass from the cylinder to the approaching system can be limited.

2. The numerical simulations allow to shorten time and cost of introducing the constructional changes of the approaching system, but they really cannot replace the experimental tests in full.
3. It's worth of mentioning to obtain considerable compatibility of the engine's parameters calculated based on the results of the simulation tests with those determined during the experimental tests.

The calculation program, using the Matlab environment, is equipped with friendly graphic interface helpful in its operation.

The possibility of changing basic constructional parameters of the engine and thermodynamical parameters of medium and surroundings, makes possible extension of possibility of model applications for tests of other engines and tests of influence of remaining elements of the approaching system.

Nomenclature/*Skróty i oznaczenia*

S - tensor of tension /strain/
T - temperature
Γ - thermal conductivity
U - speed /velocity/
F - mass force
C_v - specific heat with constant capacity
q - elementary productivity /efficiency/ of the internal source of heat
m - mass of charge in the cylinder
m_d - mass delivered /fed/ to the cylinder
m_w - mass transferred from the cylinder
E_d - energy fed /delivered/ to the charge in the cylinder
E_w - energy transferred from the system
ΔU - change of internal energy of the system
D - a differentiate matrix, which after left-sided multiplication by the ρ vector gives the vector of partial differences, where the arrow in right direction means differentiate in front , and the arrow in left direction means differentiate in back
W_d - a calorific value of the fuel,
ρ_{pow} - a density of the air,
V_s - cubic capacity of a cylinder
i - number of cylinders
n - rotational speed of the engine's crankshaft,
L_t - theoretical demand for air,
τ - number of terns of the engine's crankshaft for one stroke of operation /work/,
η_i - indicated efficiency, internal efficiency,
η_m - mechanical efficiency,
η_v - filling coefficient /factor/,
λ - coefficient of air excess,
m_{sym} - air's mass in the cylinder received from simulation,
q_{pal} - mass of the injected fuel to the cylinder during one cycle /circulation/.
m_{pal} - mass of the fuel established during the engine test bench,
t - time needed by the engine to consume the mass of m_{pal},
n - rotational speed.

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