Analysis of mass transfer in marine engine with prechamber combustion spark ignition system

Ireneusz PIELECHA*

Poznan University of Technology, Faculty of Civil and Transport Engineering, Piotrowo 3, 60-965 Poznan, Poland

Abstract. The development of combustion systems construction is associated with the possibility of increasing the thermal or overall efficiency of an internal combustion engine. The combustion systems currently in use (mainly related to direct fuel injection) are increasingly being replaced by hybrid systems, including direct and indirect injection. Another alternative is the use of prechambers in new combustion systems. This article concerns the thermodynamic aspect of this issue – namely, the assessment of the inter-chamber flow of a marine engine equipped with a prechamber combustion spark ignition system. The research was carried out using mainly one-dimensional simulation apparatus, and detailed analyses were presented using three-dimensional modeling. The tests included the engine model at medium load. Differences in mass flows were shown at different diameters and different numbers of holes from the preliminary chamber (while maintaining the same cross-sectional area). Similar values of excess air coefficient during ignition of the fuel dose in the prechamber were observed, which resulted in changes in the flow between the prechamber and the main chamber. The differences in mass flow affected the temperatures achieved in the individual combustion chambers. Based on three-dimensional analyses, the mass transfer rate between the chambers and the temperature distribution were assessed during fuel ignition initiated in the prechamber.

Key words: marine engine; prechamber combustion spark ignition; combustion process modeling; combustion thermodynamics.

1. Introduction

Combustion systems using prechambers in internal combustion engines are mainly used due to their potential to allow the combustion of lean mixtures. The increase in the system efficiency when burning lean mixtures results from the physics and chemistry of engine processes (described in simplified form by Otto cycle equation; efficiency for the Otto cycle reached 51% [1]). As the compression ratio increases, the cycle efficiency increases; however, friction losses and heat losses play a large role. On the other hand, lean mixture combustion can increase the adiabatic exponent (specific heat ratio c_p/c_v), and thus increase engine efficiency. Combustion with a high excess air ratio leads to a limitation of the maximum cylinder temperature and reduces the occurrence of knock. At the same time, it also enables an increase in the compression ratio (which again increases the efficiency).

The classification of prechambers depends on the way they are supplied: chambers with fuel delivered (active) or more common – passive (unpowered) [2, 3]. Passive chambers are used mainly for the combustion of lean mixtures at low and medium engine loads in stationary engines with power between 0.8 and 2.5 MW [3].

Due to the wide flammability range of methane, prechamber systems are much more often used for methane combustion [4]

than when burning gasoline. When burning methane, it is possible to obtain an air excess ratio of 2.5 [4, 5], ensuring that the engine work irregularity remains below 5%. The theoretical analysis of the formation of the air-fuel mixture in a gas engine with a two-stage combustion system was carried out by Tutak and Jamrozik [6]. Analyses regarding passive prechambers confirmed that similar values of excess air ratio were obtained in both prechambers [7]. This type of fuel supply forces the use of an excess air ratio of around 1.5.

The prechamber geometry analysis was conducted by Shah et al. [8]. It was found that an increase in the prechamber volume from 1.4 to 2.4% causes substantial reduction in the flame development angle (which is defined as time duration between the beginning of the spark discharge and 10% accumulated heat release in the main chamber) and main chamber combustion duration, and an increase in initial heat release in the main chamber.

In terms of emission analyses, it was found that the indicated specific NO_x emissions increase with an increase in prechamber volume and reduction in nozzle diameter.

Attard et al. conducted extensive analyses with directly supplied chambers [9]. With up to 2% of the fuel dose injected into the prechamber (turbulent jet ignition system), it was found that such a combustion system was very useful in automotive engines. Using the TJI system compared to the standard system, the mixture leaning range was increased to 2.1 (1.4 standard) coefficient of variation, with engine irregularity of up to 10% CoV(IMEP).

Tests of a turbulent jet ignition system with an active prechamber in relation to the amount of fuel burned in the

*e-mail: ireneusz.pielecha@put.poznan.pl

prechamber was also investigated by Pielecha et al. [10]. It was then demonstrated that the overall efficiency of the engine can be increased by limiting the dose burned in the prechamber. The tests were carried out on a single-cylinder engine with a displacement of 0.5 dm^3, for which an overall efficiency of $\eta_o = 0.35$ was obtained at a dose of 0.3 mg/injection (with an excess air ratio $\lambda = 1.65$).

The analysis of inter-chamber flows in a gasoline (0.5 dm^3) engine using a prechamber was conducted by Gombosuren et al. [2]. He showed the pressure differences in both chambers and on this basis, he drew conclusions on the inter-chamber flows. Similar work was also carried out in [3].

2. Purpose of research

The aim of the performed research was to assess the inter-chamber flows using the prechamber combustion spark ignition system (PCSI), considering the different geometry connecting the prechamber with the main chamber. For this purpose, a 1D (one-dimensional) simulation done in AVL Boost was used. The results were then verified using 3D tests (AVL Fire) and the outflow velocity of burning exhaust streams through flow channels was determined. The flow analysis was based on the simulation results of mass flow between the chambers. Additionally, heat dissipation in both chambers and the associated average temperature were analyzed. The obtained results regarding inter-chamber flows were analyzed in terms of changes in the outflow geometry of the holes from the prechamber. This case was subjected to a detailed 3D analysis.

3. Methods

3.1. Research equipment.
Simulation tests were carried out using one- and three-dimensional simulations. One-dimensional studies of the combustion process were carried out in AVL BOOST. Using the PCSI module for engines with a large displacement, the pressure changes in the main chamber and in the prechamber were analyzed. The prechamber volume was less than 1% of the main chamber. A visual representation of the combustion chamber was provided in Fig. 1a. The system implemented in AVL BOOST 2019 R2 (Fig. 1b) contained a model of a single-cylinder engine with a PCSI combustion chamber. Valve movement was modeled in the system, a constant value of excess air ratio was adopted for the dose reaching the cylinder (simulation of full mixing of the dose of fuel fed to the intake manifold). The prechamber was an active chamber into which natural gas (methane) was injected. One-dimensional tests were supplemented with tests in AVL FIRE 2019 R2 (Fig. 1c), after the creation of a mobile combustion chamber mesh (polymesh mesh), with a minimum number of 74 647 cells present in TDC, and a maximum number of 398 384. Additionally, the mesh was compacted for the preliminary chamber and discharge channels (the maximum mesh size was 0.5 mm there).
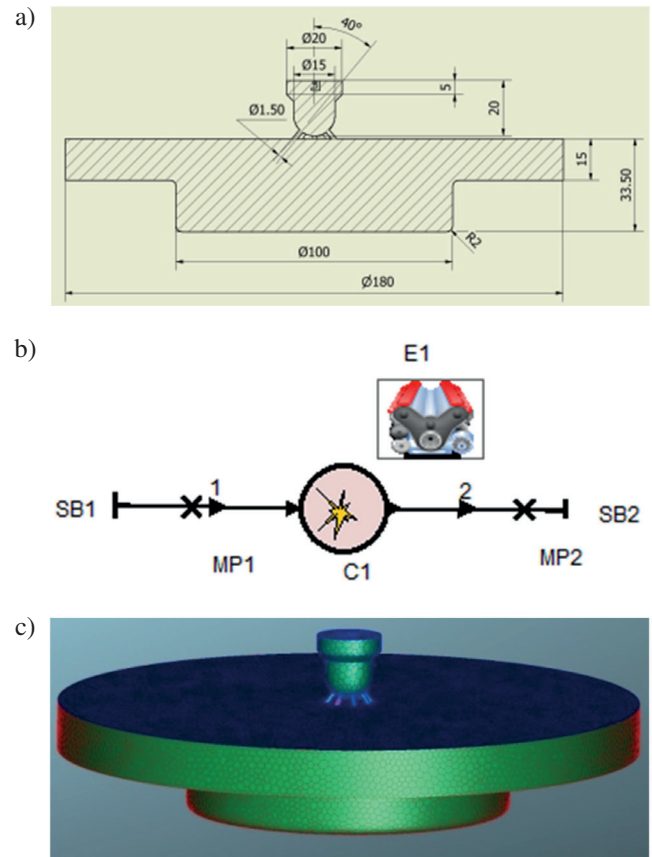

Fig. 1. Combustion system models: a) geometric; b) computational one-dimensional made in AVL Boost; c) three-dimensional implemented in AVL Fire

3.2. Test object.
Simulation tests were carried out for a single-cylinder, supercharged four-stroke engine, whose parameters were given in Table 1. The fuel dose fed to the prechamber accounted for 0.1% of the entire fuel dose. The tests were performed at an average engine load, indicating mean effective pressure IMEP = 1.2 MPa at a rotational speed of 1500 rpm.

Table 1
Datasheet of the tested engine

Parameter	Value
D	180 mm
S	230 mm
V_{PC}	3.18 cm^3
V_{PC+MC} (GMP)	530.458 cm^3
V_{cyl}	6383 cm^3
ε	12
n	1500 rpm
IMEP	1.02 MPa
Equivalence ratio (λ-value)	0.692 (1.445)
Injection to prechamber	90 deg bTDC
Injection dose (% of full)	0.5 mg (0.1%)

The analysis of inter-chamber flows was carried out with different outflow geometry from the prechamber. The variable parameters were the number of holes and their outflow diameters from the prechamber (9 variants). The basic variants adopted are 8 holes with three diameters (1.0, 1.5 and 2 mm). Changes were made in such a way so as to maintain a constant cross-section for the basic hole surface area; by changing the number of holes, their diameter was changed to obtain A = const. The values of the variability of the outflow geometry from the preliminary chamber are presented in Table 2.

Table 2
Prechamber parameters variation

Parameter	Value
No. of holes/diameter	6 × 1.154 mm (P_6_1.154); 6 × 1.732 mm (P_6_1.732); 6 × 2.309 mm (P_6_2.309); (base) 8 × 1.000 mm (P_8_1.000); 8 × 1.500 mm (P_8_1.500); 8 × 2.000 mm (P_8_2.000); 10 × 0.894 mm (P_10_0.894); 10 × 1.341 mm (P_10_1.341); 10 × 1.788 mm (P_10_1.788);
Area (const)	6.283 mm^2 (P_6_1.154; P_8_1.000; P_10_0.894) 14.137 mm^2 (P_6_1.732; P_8_1.500; P_10_0.341) 25.132 mm^2 (P_6_1.2.309; P_8_2.000; P_10_1.788)

4. Results

4.1. Research method analysis. Choosing the PCSI model makes it necessary to specify combustion parameters for the cylinder and the prechamber. The PCSI model is a predictive combustion model for spark-ignition gas engines equipped with prechambers [11].

The heat release rate in the prechamber was based on the Magnussen formula [12], where the hemispherical flame front volume is taken for calculations [13]

$$\frac{dQ_{\text{pre}}}{dt} = C_{\text{c,pre}} \cdot H_{\text{u}} \cdot V_{\text{F}} \cdot \frac{m_{\text{f,pre}}}{V_{\text{pre}}} \cdot C_{\text{p}} \cdot \frac{\sqrt{k_{\text{pre}}}}{\sqrt[3]{V_{\text{pre}}}}, \qquad (1)$$

where C_{p} represents thermal correction with the combustion products concentration (fuel mass fractions of carbon, hydrogen, and oxygen); V_{F} – flame front volume; k – turbulent kinetic energy; m_f – fuel mass in the prechamber.

Its thickness is constant in the AVL BOOST program and is 3 mm. The flame radius is calculated by integrating the turbulent flame speed [13].

The heat release rate in the main chamber was divided into two zones

$$\frac{dQ_{\text{cyl}}}{dt} = \frac{dQ_{\text{cyl,I}}}{dt} + \frac{dQ_{\text{cyl,II}}}{dt}, \qquad (2)$$

Therefore, the fuel supply is also divided into two areas. In each of them, the mass balance is determined separately considering the fuel flow between the chambers. Therefore, the total value of the released heat is the sum of both parts. The first phase of heat release results from the outflow of flame streams from the prechamber

$$\frac{dQ_{\text{cyl,I}}}{dt} = C_{\text{c,I}} \cdot H_{\text{u}} \cdot m_{\text{f,I}} \cdot \frac{\sqrt{k_{\text{spray}}}}{\sqrt[3]{V_{\text{cyl}}}}. \qquad (3)$$

The second combustion phase is also described by the Magnussen equation considering the mass of fuel in the flame front volume as a function of flame surface and fuel density $(m_{\text{f,II}}/V_{\text{cyl}})$ [13]

$$\frac{dQ_{\text{cyl,II}}}{dt} = C_{\text{c,II}} \cdot H_{\text{u}} \cdot A_{\text{f}} \cdot s_{\text{f}} \cdot \frac{m_{\text{f,II}}}{V_{\text{cyl}}} \cdot C_{\text{p}} \cdot \frac{\sqrt{k_{\text{cyl}}}}{\sqrt[3]{V_{\text{cyl}}}}. \qquad (4)$$

In the initial phase, the flame surface adopts a semi-spherical shape. When it reaches the piston, the flame surface is calculated as an equivalent spherical layer of the same volume as the hemispherical flame.

4.2. Evaluation of prechamber geometry change effect on engine operating indicators. The research began by comparing the combustion pressure changes in the main chamber obtained by independent calculation systems: AVL BOOST and AVL FIRE. The fuel injection to the prechamber was determined in the 90 to 80 deg angular range before TDC (triangular fuel flow) and ignition at 16 deg bTDC.

A comparative analysis of the pressure changes in the cylinder indicates that similar values of pressure changes in the entire analyzed crankshaft angle range were obtained (Fig. 2). The maximum pressure difference was delta $(P_{\text{cyl max}} = 0.34$ MPa (around TDC), while the maximum pressure difference obtained from both methods was delta $(P_{\text{cyl max}}) = 0.18$ MPa. The compression process and initial combustion period were comparable (pressure changes did not exceed 0.15 MPa, which can be considered as similar values and correct verification of test methods can be accepted).

Positive verification of the pressure changes in the cylinder facilitated assessing the average useful pressure in the cylinder when changing the diameter of the holes, considering that their
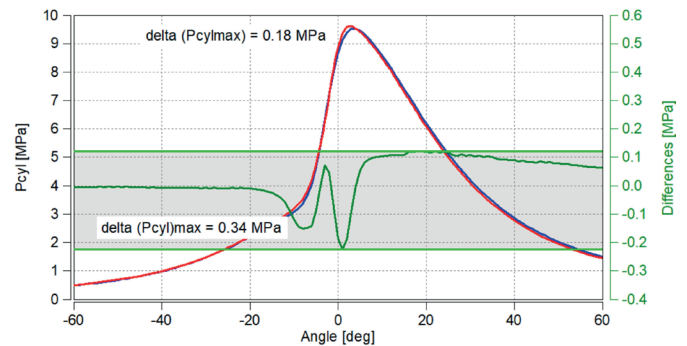
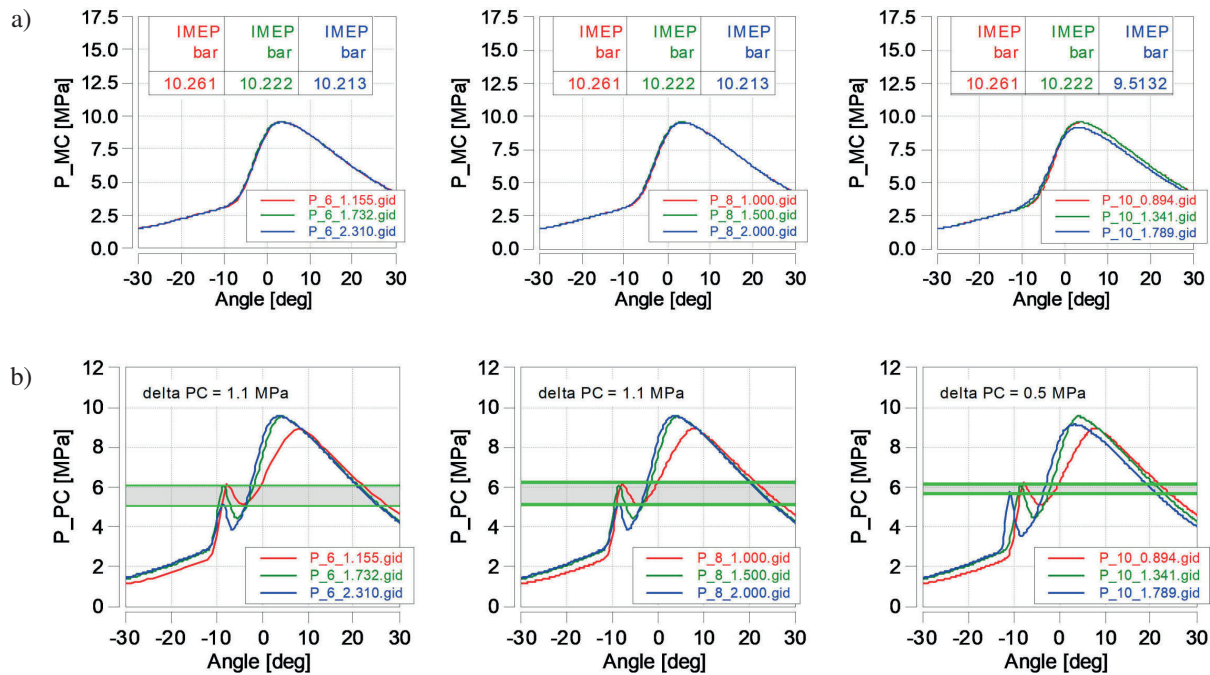Fig. 2. Model pressure verification for 1D (AVL Boost) and 3D (AVL Fire)

Bull. Pol. Acad. Sci. Tech. Sci. 69(2) 2021, e136744

3

Fig. 3. The effect of changing the number of holes and their diameter: a) on the pressure and IMEP value in the cylinder main chamber (P_MC); b) on the pressure changes in the prechamber (P_PC), while maintaining a constant flow surface area (area = const)

number is constant (Fig. 3). Very similar IMEP values were obtained with differences below 0.2%. This means that with the same number of holes, the pressure changes in the main chamber did not fluctuate greatly. The analysis showed a similar (or very steady) pressure change characteristic in the main chamber (Fig. 3a), while the pressure variations in the prechamber are significant (Fig. 3b).

Small prechamber holes (irrespective of the number) limit the mixture flow during the compression (the lowest compression curve values in the range of up to 10 deg bTDC). Also, they result in a prolonged combustion initiation process and a slow pressure build-up in this chamber. This may be due to the fact that the excess air ratio is limited. A larger flow cross section through the chamber openings results in a more rapid combustion process (its initiation), a large drop in the pressure in the first combustion phase, with the pressure increase not showing a large delay. This means that the smallest flow diameter selected for testing was too small.

The comparison of the pressure changes in the chambers of the two-stage PCSI system allows further thermodynamic assessment of the process presented in the following parts of the article.

5. Thermodynamic evaluation of the geometric changes in the combustion chamber

Absolute combustion pressure values do not directly determine thermodynamic process indicators. The analysis of the pressure increase rate in both combustion chambers showed much higher rates in the prechamber than in the main chamber for each di-

ameter of the holes (Fig. 4a). The pressure increase rates were twice as high in the prechamber (solid lines – Fig. 4a) than in the main chamber (dashed lines – Fig. 4a). This was mainly due to the small volume of this chamber.

The heat release rate (HRR) analysis (Fig. 4b) points to the two-zone model described above. In the prechamber these rates were low as a result of a limited dose of fuel in this area. Much higher HRR occurred in the main chamber (due to a much higher fuel dose), with the difference in these rates was by a factor of 15.

Based on the pressure changes in both chambers, it was possible to determine the changes in the mixture flow rate between them (Fig. 5). The analysis of the mixture flow rate after ignition into the main chamber (a few degrees before TDC) was almost symmetrical in relation to the backflow (from the main chamber to the prechamber when the pressure in the cylinder reached its maximum). For large hole diameters, these differences were reduced.

This also results from the value of the excess air ratio created in the prechamber (Fig. 6). Small holes lead to a limited amount of combustible mixture coming from the main chamber, and thus the lambda is also large. However, the maximum differences in lambda values (around 80 deg before TDC) are not large.

The value of the excess air ratio should affect the average temperature in the combustion chamber. Figure 7 is an assessment of the average temperature values in the PC and MC (prechamber and main chamber). Additionally, the values of appropriate temperature in the burn zones of both chambers were presented. The highest temperature values occur in the prechamber with small diameter holes (Fig. 7 – PC).
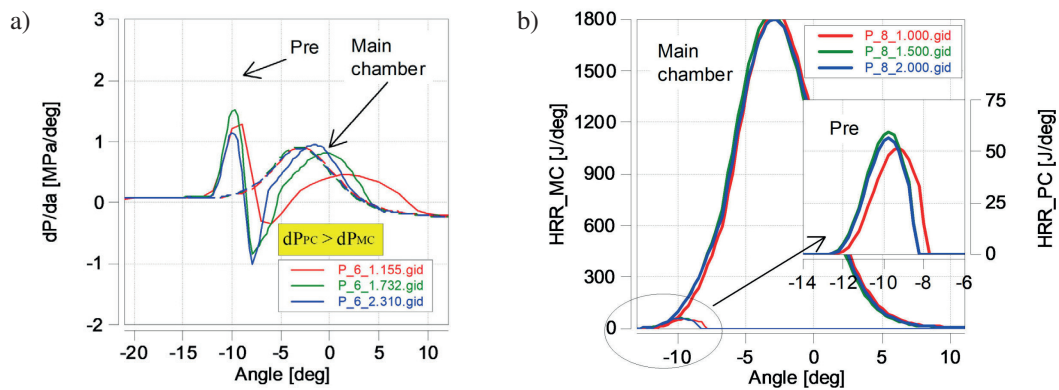
Fig. 4. Change of thermodynamic indicators of the combustion process with the selected variants of the number of holes and their diameters: a) the pressure rising (dP/da) change; b) the heat release rate (HRR) in both combustion chambers: main (HRR_MC) and pre (HRR_PC)
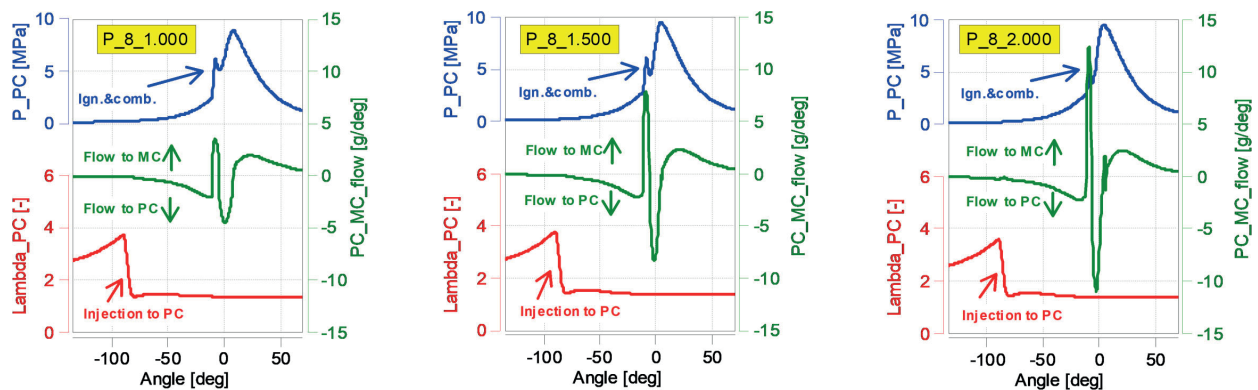

Fig. 5. Mass flow (PC_MC_flow) between chambers: main and prechamber and the resulting change in the excess air ratio (Lambda_PC) in the prechamber against the background of the combustion pressure (P_PC) process at different outlets diameter: 8×1 mm, 8×1.5 mm, 8×2 mm
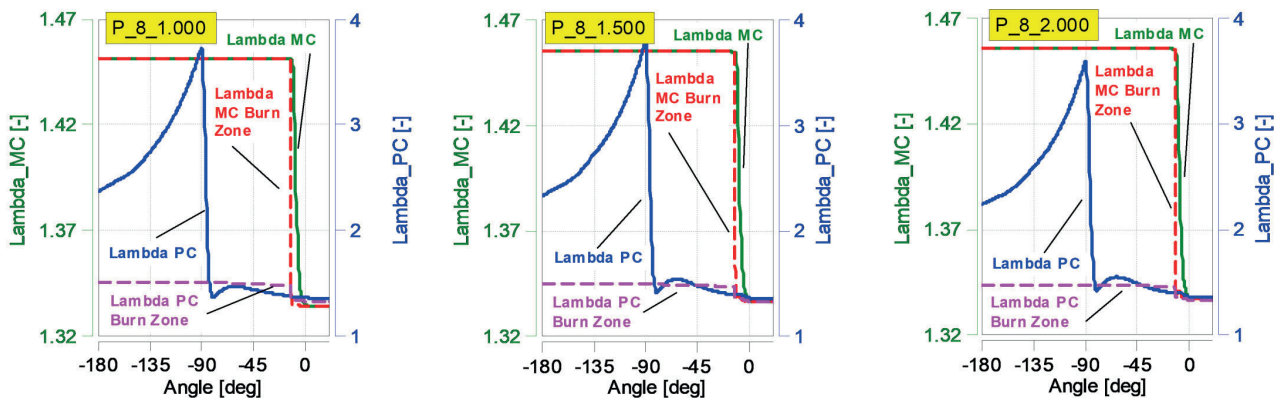

Fig. 6. Change of the total excess air ratio (Lambda_MC and Lambda_PC) and in the combustion zone in the chambers: prechamber (PC) and main chamber (MC) (test results for $n = 8$ holes, hole diameters: 1 mm, 1.5 mm, 2 mm)

As the diameters in the holes increase, the maximum temperature peak found at around 8 deg bTDC decreases. Then this value decreases further, reaching about 2200 K (at $\alpha = 5$ deg bTDC) with the smallest diameter holes. The minimum temperature value in this angular range appears earlier when increasing the diameter of the holes. This indicates an increase in the mass flow, which explains the decrease in the temperature. The analysis of the charts in Fig. 7 indicates a decrease

in the average temperature value in the prechamber to 2000 K with a large diameter of holes (PC size in Fig. 7). Changes in the average temperature values in the main chamber (MC) were small (the line marked MC – Fig. 7). It is possible to state that they are independent of the hole diameter size. This indicates that the greatest differences in the combustion process occur in the prechamber, but they do not always result in the changes in the values of thermodynamic indicators for the main chamber.
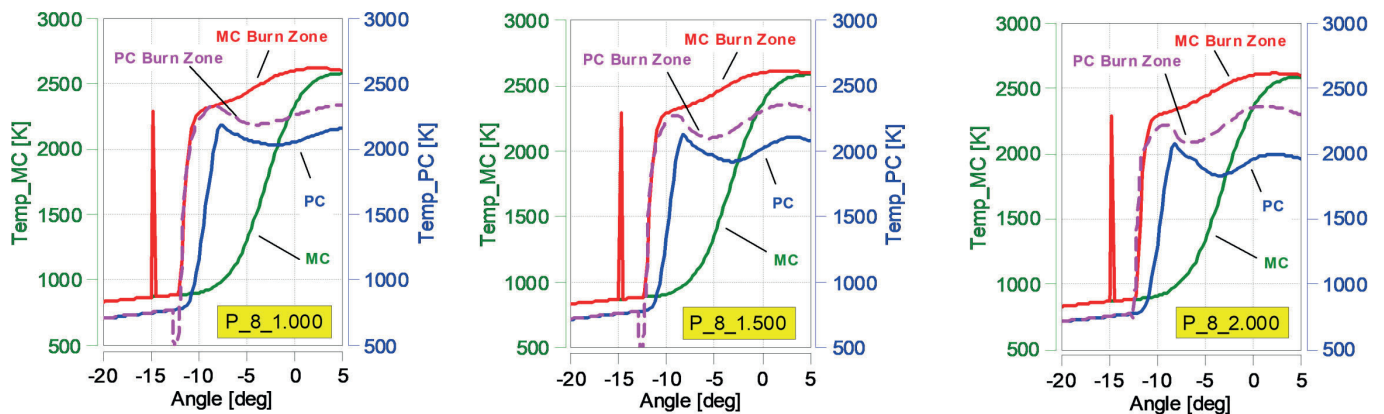
Fig. 7. Temperature change (Temp_MC and Temp_PC) in individual chambers: PC and MC and in combustion zones PC and MC (results for $n = 8$ holes, hole diameters: 1 mm, 1.5 mm, 2 mm)
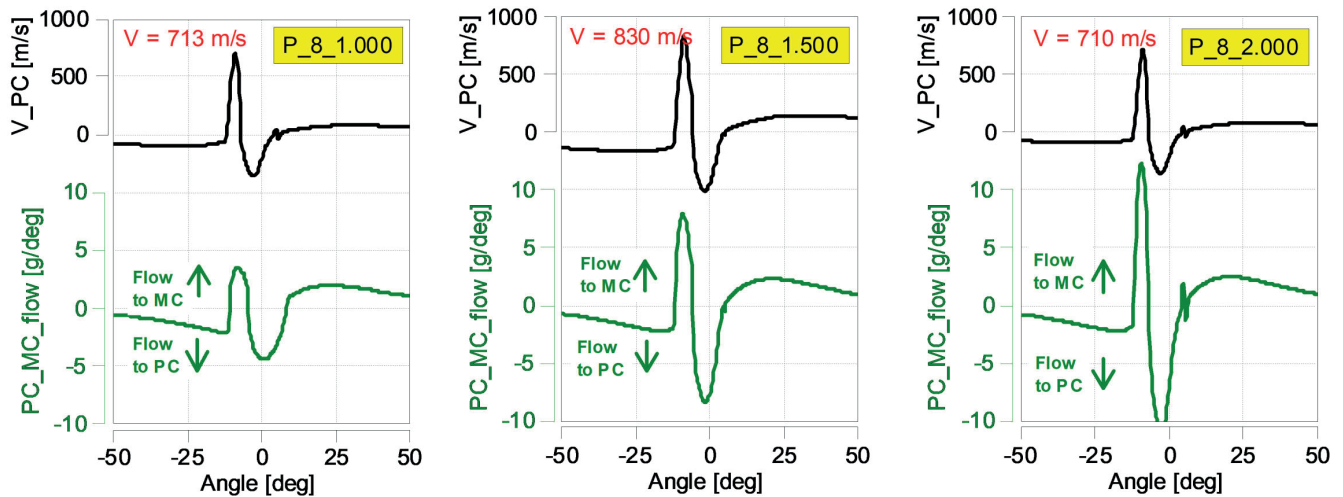

Fig. 8. Conditions of mass flow velocity (burning streams) from the PC to the MC (V_PC) and mass flow rate values from the prechamber to the main chamber (PC_MC_flow) (tests for $n = 8$ holes, hole diameters: 1 mm, 1.5 mm, 2 mm)

A particularly interesting thermodynamic quantity is the mass flow rate from the prechamber holes. One-dimensional analysis indicated the occurrence of high speeds of about 700–800 m/s (Fig. 8) that was irrespective of the size of the holes. There is a certain diameter of the holes at which the flow speed is the highest. However, it is difficult to draw conclusions from this relationship with the results for only three diameter sizes available. Since many publications provide the value of the outflow speed of a burning flame [14, 15] (typically in the range of 300–500 m/s), this quantity was used for comparison with one- and three-dimensional tests. AVL Fire software was used to perform a set of analyses and the obtained results of the mass flow rate through the prechamber holes for the tests with AVL BOOST and AVL FIRE were very consistent (Fig. 9). It should be noted that the obtained results were of the mass flow rate and not the direct outflow speed of the burning flame flowing through the prechamber holes. However, based on the proportions, it can be assumed that the flame outflow values were smaller than the mass outflow values, which may indicate that their values are correct.

The mass flow direction (see Fig. 9) is consistent with the one-dimensional flow direction (Fig. 8). Additionally, with a similar crank angle in both chambers, there are maximum flow values.

A comparison of the average temperature values with their instantaneous values in the prechamber and main chamber shows that the values obtained are consistent between these tests (Figs. 7 and 10). Figure 10 does not indicate the temperatures in the combustion zone; however, the average temperature values are similar. Visible temperature changes around the holes connecting the two chambers indicate the outflow of burning mass, which confirms the general conditions of the combustion process using prechambers.

The thermodynamic conditions analyzed above relate to the combustion of gaseous fuels in marine engines, in which the excess air ratio is much greater than one. Knowing the specificity of inter-chamber flows, it will be possible to determine the critical volumes that contribute most to the creation of toxic components of exhaust gases, in particular nitrogen oxides and particulate matter.
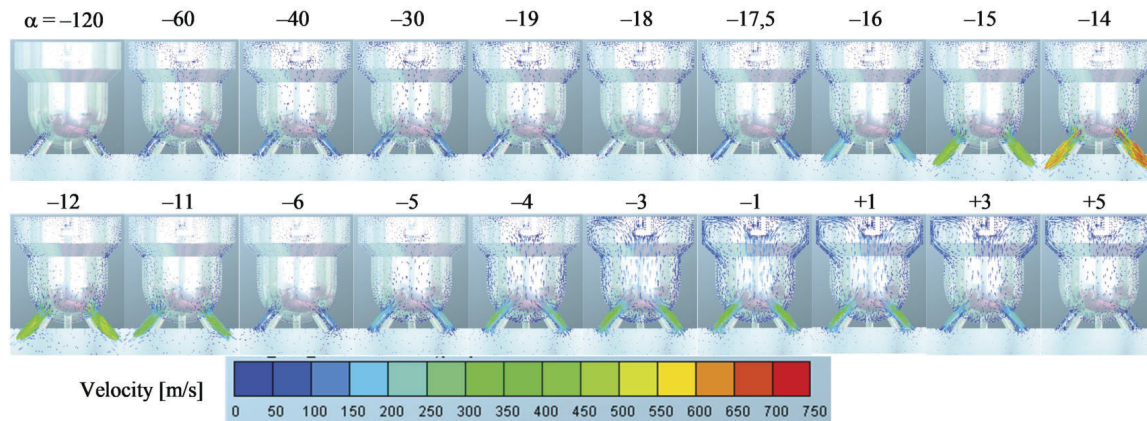
Fig. 9. Analysis of the velocity from the prechamber holes to the main chamber (the vectors indicate the mass flow direction) (tests for $n = 8$ holes, hole diameters: 1 mm, 1.5 mm, 2 mm)
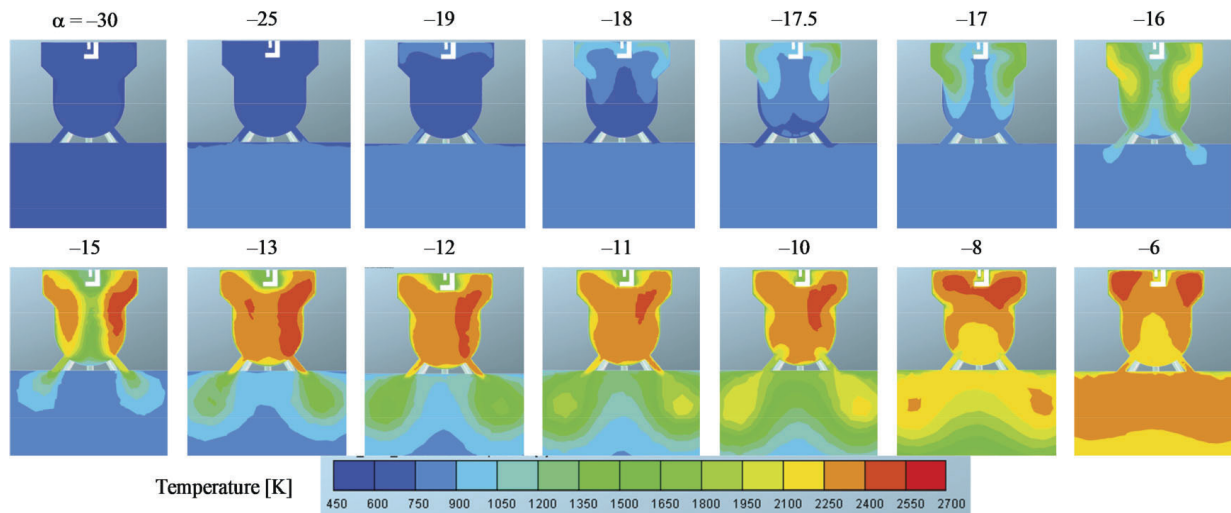

Fig. 10. Temperature analysis during initialization of the combustion process in the prechamber and its spread into the main chamber ($n = 8$ holes, $d = 1.5$ mm)

6. Conclusions

Simulation tests for the two-stage PCSI combustion system allowed a full assessment of the flow and thermodynamic phenomena.

The analysis of the results of the tests of the PCSI combustion system in a single-cylinder engine led to several main conclusions:

- The effect of changing the number and size of the prechamber holes is significant in relation to the flow processes between the chambers. It is also particularly important in relation to the thermodynamic processes occurring in the prechamber.
- The small number of holes in the prechamber means that the maximum rate of the pressure increase in it is much higher than in the main chamber; this value decreases with an increasing hole diameter and earlier start of mass flow to the main chamber.
- Increasing the diameter of the prechamber holes results in an increase in mass transfer between the two chambers with a double increase in the diameter of the openings, more than a double increase in flow rate has been obtained. The most intense flows were observed during the ignition phase in the prechamber (flow to the main chamber) and the combustion in the main chamber (inflow to the prechamber).
- The initialization of ignition in the prechamber, supplied with a small dose of fuel, results in a low heat release rate (Fig. 4); however, the analysis of the temperature distribution indicated similar values to those obtained in the main chamber.
- One- and multi-dimensional analysis of the mass flow between the chambers indicated that similar flow velocities have been obtained (Figs. 8 and 9): about 700 m/s at an angle of about 14 deg bTDC (from PC to MC) and slightly smaller at the flow return – 400–450 m/s at an angle of 3–1 deg bTDC (for flow from MC to PC).

Bull. Pol. Acad. Sci. Tech. Sci. 69(2) 2021, e136744

7

REFERENCES

[1] P. Tarnawski and W. Ostapski, "Pulse powered turbine engine concept – numerical analysis of influence of different valve timing concepts on thermodynamic performance", *Bull. Pol. Acad. Sci. Tech. Sci.* 66(3), 373–382 (2018), doi: 10.24425/123444.

[2] N. Gombosuren, O. Yoshifumi, and A. Hiroyuki, "A charge possibility of an unfueled prechamber and its fluctuating phenomenon for the spark ignited engine", *Energies* 13(2), 303 (2020), doi: 10.3390/en13020303.

[3] M. Günther (ed.), *Ignition Systems for Gasoline Engines*, 4th International Conference. Verlag expert, Berlin, 2018. doi: 10.5445/IR/1000088324.

[4] A. Shah, "Improving the efficiency of gas engines using pre-chamber ignition", PhD Thesis, Lund University, 2015.

[5] P. Hlaing, M.E. Marquez, V.S.B. Shankar, E. Cenkar, M.B. Houidi, and B. Johansson, "A study of lean burn pre-chamber concept in a heavy duty engine", *SAE Tech. Paper* 2019–24–0107 (2019), doi: 10.4271/2019-24-0107.

[6] A. Jamrozik and W. Tutak, "Theoretical analysis of air-fuel mixture formation in the combustion chambers of the gas engine with two-stage combustion system", *Bull. Pol. Acad. Sci. Tech. Sci.,* 62(4), 779–790 (2014), doi: 10.2478/bpasts-2014-0085.

[7] J. Benajes, R. Novella, J. Gomez-Soriano, P.J. Martinez-Hernandiz, C. Libert, and M. Dabiri, "Evaluation of the passive pre-chamber ignition concept for future high compression ratio turbocharged spark-ignition engines", *Appl. Energ.* 248, 576–588 (2019), doi: 10.1016/j.apenergy.2019.04.131.

[8] A. Shah, P. Tunestal, and B. Johansson, "Effect of pre-chamber volume and nozzle diameter on pre-chamber ignition in heavy duty natural gas engines", *SAE Tech. Paper* 2015–01–0867 (2015), doi: 10.4271/2015-01-0867.

[9] W. Attard, N. Fraser, P. Parsons, and E. Toulson, "A turbulent jet ignition pre-chamber combustion system for large fuel economy improvements in a modern vehicle powertrain", *SAE Int. J. Engines* 3(2), 20–37 (2010), doi: 10.4271/2010-01-1457.

[10] I. Pielecha, K. Wisłocki, W. Cieślik, and Ł. Fiedkiewicz, "Prechamber selection for a two stage turbulent jet ignition of lean air-gas mixtures for better economy and emission", *17th International Conference Heat Transfer and Renewable Sources of Energy (HTRSE-2018), E3S Web of Conferences,* 70, 03010 (2018), doi: 10.1051/e3sconf/20187003010.

[11] L.O. Guelder, "Turbulent premixed flame propagation models for different combustion regimes", *23rd Symposium (International) on Combustion*, The Combustion Institute, Pittsburgh, USA, 1990, doi: 10.1016/S0082-0784(06)80325-5.

[12] B.F. Magnussen and B.H. Hjertager, "On mathematical modeling of turbulent combustion with special emphasis on soot formation and combustion", *16th Symposium (International) on Combustion*, The Combustion Institute, Pittsburgh, USA, 1976, doi: 10.1016/S0082-0784(77)80366-4.

[13] AVL BOOST. AVL AST Documentation 2019.

[14] M. Gholamisheeri, I.S. Wichman, and E.Toulson, "A study of the turbulent jet flow field in a methane fuelled turbulent jet ignition (TJI) system", *Combust. Flame* 183, 194–206 (2017), doi: 10.1016/j.combustflame.2017.05.008.

[15] G. Gentz, B. Thelen, M. Gholamisheeri, P. Litke, A. Brown, J. Hoke, and E. Toulson, "A study of the influence of orifice diameter on a turbulent jet ignition system through combustion visualization and performance characterization in a rapid compression machine", *Appl. Therm. Eng.* 81, 399–411 (2015), doi: 10.1016/j.applthermaleng.2015.02.026.

8

Bull. Pol. Acad. Sci. Tech. Sci. 69(2) 2021, e136744