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

I. 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 possibilities 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 pre-chambers in new combustion systems. This article is concerned with the thermodynamic aspect of this issue – namely the assessment of 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 analyzes were presented with the use of three dimensional modeling. The tests concerned the engine model at medium load. Differences in mass flows were shown at different diameters and different number 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 pre-chamber 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 analyzes, 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 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 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]. Analyzes 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 beginning of 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 analyzes, it was found that indicated specific NOx emissions increase with an increase in prechamber volume and reduction in nozzle diameter.

Attard et al. conducted extensive analyzes 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) – with engine irregularity of up to 10% CoV(IMEP) – coefficient of variation.

Tests of a Turbulent Jet Ignition system with an active prechamber in relation to the amount of fuel burned in the 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

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


This article has been accepted for publication in a future issue of this journal, but has not been fully edited. Content may change prior to final publication.

2. Prechamber. The tests were carried out on a single-cylinder engine with a displacement of 0.5 dm³, 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 dm³) 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 inter-chamber flows using the PCSI (Prechamber Combustion Spark Ignition) system, taking into account the different geometry connecting the prechamber with the main chamber. For this purpose, 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 simulation results of mass flow between chambers. In addition, heat dissipation in both chambers and the associated average temperature were analyzed. Obtained results regarding inter-chamber flows were analyzed in the aspect of changes in the outflow geometry of the holes from the prechamber. This case was subjected to 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 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 fuel dose 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) (minimum number of cells – 74647 present in TDC, maximum – 398384). 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>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>D</td>
<td>180 mm</td>
</tr>
<tr>
<td>S</td>
<td>230 mm</td>
</tr>
<tr>
<td>( V_{PC} )</td>
<td>3.18 cm³</td>
</tr>
<tr>
<td>( V_{PC+MC} ) (GMP)</td>
<td>530.458 cm³</td>
</tr>
<tr>
<td>( V_{cyl} )</td>
<td>6383 cm³</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>12</td>
</tr>
<tr>
<td>( n )</td>
<td>1500 rpm</td>
</tr>
<tr>
<td>IMEP</td>
<td>1.02 MPa</td>
</tr>
<tr>
<td>Equivalence ratio (( \lambda )-value)</td>
<td>0.692 (1.445)</td>
</tr>
<tr>
<td>Injection to prechamber</td>
<td>90 deg bTDC</td>
</tr>
<tr>
<td>Injection dose (% of full)</td>
<td>0.5 mg (0.1%)</td>
</tr>
</tbody>
</table>

Table 1

Data sheet of the tested engine

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 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 $A_{\text{pre}} = \text{const}$. The values of the variability of the outflow geometry from the preliminary chamber are presented in Table 2.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of holes/diameter</td>
<td>$6 \times 1.154 \text{ mm (P}_{6,1.154}\text{);}$</td>
</tr>
<tr>
<td></td>
<td>$6 \times 1.732 \text{ mm (P}_{6,1.732}\text{);}$</td>
</tr>
<tr>
<td></td>
<td>$6 \times 2.309 \text{ mm (P}_{6,2.309}\text{);}$</td>
</tr>
<tr>
<td>(base)</td>
<td>$8 \times 1.000 \text{ mm (P}_{8,1.000}\text{);}$</td>
</tr>
<tr>
<td></td>
<td>$8 \times 1.500 \text{ mm (P}_{8,1.500}\text{);}$</td>
</tr>
<tr>
<td></td>
<td>$8 \times 2.000 \text{ mm (P}_{8,2.000}\text{);}$</td>
</tr>
<tr>
<td></td>
<td>$10 \times 0.894 \text{ mm (P}_{10,0.894}\text{);}$</td>
</tr>
<tr>
<td></td>
<td>$10 \times 1.341 \text{ mm (P}_{10,1.341}\text{);}$</td>
</tr>
<tr>
<td></td>
<td>$10 \times 1.788 \text{ mm (P}_{10,1.788}\text{);}$</td>
</tr>
<tr>
<td>Area (const)</td>
<td>$6.283 \text{ mm}^2 (P_{6,1.154}; P_{8,1.000}; P_{10,0.894})$</td>
</tr>
<tr>
<td></td>
<td>$14.137 \text{ mm}^2 (P_{6,1.732}; P_{8,1.500}; P_{10,0.341})$</td>
</tr>
<tr>
<td></td>
<td>$25.132 \text{ mm}^2 (P_{6,2.309}; P_{8,2.000}; P_{10,1.788})$</td>
</tr>
</tbody>
</table>

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]:

$$\frac{dQ_{\text{pre}}}{dt} = C_{P,\text{pre}} \cdot H_u \cdot V_F \cdot \frac{m_{f,\text{pre}}}{V_{\text{pre}}} \cdot C_p \cdot \frac{k_{\text{pre}}}{\sqrt{V_{\text{pre}}}}, \quad (1)$$

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 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

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

therefore also the fuel supply is divided into two areas. In each of them, the mass balance is determined separately taking into account the fuel flow between the chambers.

The total value of heat released is, therefore, the sum of both parts. The first phase of heat release results from the outflow of flame streams from the prechamber

$$\frac{dQ_{\text{cyl},I}}{dt} = C_{C,\text{I}} \cdot H_u \cdot m_{f,\text{I}} \cdot \frac{\sqrt{k_{\text{spray}}}}{3 \sqrt{V_{\text{cyl}}}}, \quad (3)$$

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

$$\frac{dQ_{\text{cyl},II}}{dt} = C_{C,\text{II}} \cdot H_u \cdot A_f \cdot S_f \cdot \frac{m_{f,\text{II}}}{V_{\text{cyl}}} \cdot C_p \cdot \frac{k_{\text{cyl}}}{3 \sqrt{V_{\text{cyl}}}} \quad (4)$$

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. 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 have been obtained (Fig. 2). The maximum pressure differences was $\Delta (P_{\text{cyl}})_{max} = 0.34 \text{ MPa (around TDC)}$, while the maximum pressure difference obtained from both methods was $\Delta (P_{\text{cyl}})_{max} = 0.18 \text{ MPa}$. 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).

![Fig. 2. Model pressure verification for 1D (AVL Boost) and 3D (AVL Fire)](image-url)
Positive verification of the pressure changes in the cylinder allowed assessing the average useful pressure in the compression process and a slow pressure build-up in this chamber. This may be due to the fact that the excess air ratio is limited. The larger flow cross section through the prechamber (solid lines – Fig. 4a) than in the main chamber for each diameter of the holes showed much higher rates in the prechamber than in the main chamber (dashed lines – 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.

Small prechamber holes (irrespective of the number) limit the mixture flow during compression (the lowest compression curve values in the range of up to 10 deg bTDC). In addition, they result in a prolonged combustion phase (its initiation) being more rapid, the pressure drop in the first combustion phase was large, then the pressure increase did not show a large delay. This means that the smallest flow diameter selected for testing was too small.

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 diameter 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 much higher fuel dose) – the difference in these rates was by a factor of 15.

Based on the pressure changes in both chambers, it was possible to determine 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 back flow (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. Small holes result in the amount of combustible mixture coming from the main chamber being limited, 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. Fig. 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](image1.png) 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](image2.png) 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](image3.png) 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 holes diameters increase, the maximum temperature peak found at around 8 deg bTDC decreases. Then this value further decreases, reaching about 2200 K (at α = 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 mass flow, which explains the decrease in 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 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 holes diameter size. This indicates that the greatest differences in the combustion process occur in the prechamber, but they do not always result in changes in the values of thermodynamic indicators for the main chamber.

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 on this relationship with the results for only three diameter sizes available. Due to the fact that 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 analyzes 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 (in 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’s values.

![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)](image)

![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)](image)
Analysis of mass transfer in marine engine with prechamber combustion spark ignition system

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)

Comparison of 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 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 most contribute to the creation of toxic components of exhaust gases, in particular nitrogen oxides and particulate matter.

6. Conclusions

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

Analysis of the tests results of the PCSI combustion system in a single-cylinder engine resulted in a number of main conclusions:

- The effect of changing the number and size of prechamber holes is significant in relation to 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 pressure increase in it is much higher than in the main chamber; this value decreases with 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 during combustion in the main chamber (inflow to the prechamber).
• 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 that were 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 (Fig. 8 and Fig. 9): about 700 m/s at an angle of about 14 deg bTDC (from PC to MC) and slightly smaller at flow return – 400–450 m/s at an angle of 3–1 deg bTDC (for flow from MC to PC).

Acknowledgements. This work has been done under AVL University Partnership Program.

REFERENCES


