NUMERICAL ANALYSIS OF INFLUENCE OF PRECHAMBER GEOMETRY IN IC ENGINE WITH TWO-STAGE COMBUSTION SYSTEM ON ENGINE WORK CYCLE PARAMETERS

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Abstract

Numerical analysis results of influence of prechamber geometry in two stage combustion system SI engine on chosen engine work cycle parameters such as temperature and pressure are presented in the paper. Within the confines of calculations performed in KIVA-3V software, the modelling of processes occurring in combustion chambers of engine with prechamber powered by gasoline for four diameters of the duct connecting the prechamber with cylinder (3mm, 4 mm, 6 mm and 9 mm) were conducted. The research were performed for four values of excess air number equal: 1.4, 1.6, 1.8, 2.0. The modelling results showed that the most favourable case is the configuration with the connecting duct of diameter equal 6 mm. The maximal values of pressure and temperature were obtained during combustion in this engine. The maximal values of pressure and temperature influence the value of maximal indicated pressure and determine the engine performance. Performed simulations of two-stage combustion process delivered information concerning spatial and time-dependent pressure and temperature distribution in combustion chamber of modelled engine.

Keywords: *engine with two-stage combustion system, prechamber, excess air number*

1. Introduction

Introducing more rigorous toxic components emission standards contributes to improvement in piston engine design and combustion organizing. Currently manufactured gaseous, high-power SI engines use two-stage lean mixture combustion system with sectional combustion chamber in order to reduce toxic components content in exhaust gases and improve the engine efficiency. In such system the combustion chamber consists of two parts: the main chamber in cylinder of the engine and prechamber in engine head connected with the main chamber by connecting duct. Very lean mixture prepared in engine inlet system (λ=1.5÷3.0) is aspirated to the cylinder. However stechiometric mixture (λ=1.0) is delivered to the prechamber. The rich mixture ignition by spark discharge occurs in prechamber (large amounts of CO and HC and slight amounts of NO_x are produced). As a result of pressure increase the burning content of prechamber is forced by the connecting duct to the main combustion chamber in cylinder where many moving ignition kernels develop. In consequence lean flammable mixture, which could not be ignited by spark discharge, ignites in many regions. The ignition is fast enough to provide high engine cycle efficiency and avoid the disadvantages connected with combustion during expansion stroke. At the time of main combustion slight amounts of NO_x are produced and particles of CO and HC are burnt [6].

During recent years the research concerning numerical modelling using more and more advanced mathematical models has been intensively developing. The development of numerical modelling is heightened by increasing computational power that allows modelling not only flow processes but also combustion in 3D. One of more advanced numerical models used for combustion process in

piston engines modelling is KIVA-3V developed in Los Alamos National Laboratory in the United States of America.
The aim of the research was to investigate the influence of connecting duct geometry on chosen engine cycle parameters. Within the confines of calculations performed in KIVA-3V software, the modelling of processes occurring in combustion chambers of engine with prechamber powered by gasoline for four diameters of the duct connecting the prechamber with cylinder (3mm, 4 mm, 6 mm and 9 mm) were conducted. The courses of kinetic energy of turbulence, pressure and temperature variations in engine combustion chambers in function of crank angle were estimated. Moreover the distribution of above mentioned quantities in the combustion chambers was calculated for chosen crank angles. Characteristics of heat release rate and the total heat released during combustion in function of CA were presented.

2. Engine with prechamber modelling

Three dimensional model of engine with prechamber realizing two-stage combustion was prepared in accordance with S320SI test engine dimensions. The test engine was a rebuilt four-stroke high-pressure engine manufactured by Wytwórnia Silników Wysokoprężnych „ANDORIA" in Andrychów. This engine after modifications was adapted to fuel combustion as spark ignited engine due to equipping it with new ignition system [3].

Tab. 1. Main parameters of S320ZI research engine

Displacement volume	....	1800 cm^3
Number of cylinders	....	1
Cylinder arrangement	....	horizontal
Inside crank radius	....	80 mm
Cylinder bore	....	120 mm
Connecting-rod length	....	275 mm
Piston stroke	....	160 mm
Compression ratio	....	8.6
Engine speed	....	1000 rev/min

The geometric mesh of combustion chamber prepared in Cartesian co-ordinate system using KIVA pre-processor is depicted in figure 1. The mesh built out of 24500 cells and 27000 nodes consists of two combustion chambers: the prechamber in engine head and the main chamber in engine cylinder. The total volume of those chambers is 237 cm^3. The prechamber volume is approximately 4.5% of total volume above the piston at TDC and it is located asymmetrical regarding the cylinder axis. It was connected with the main chamber by cylindrical duct of diameter equal 3, 4, 6 i 9 mm [2].

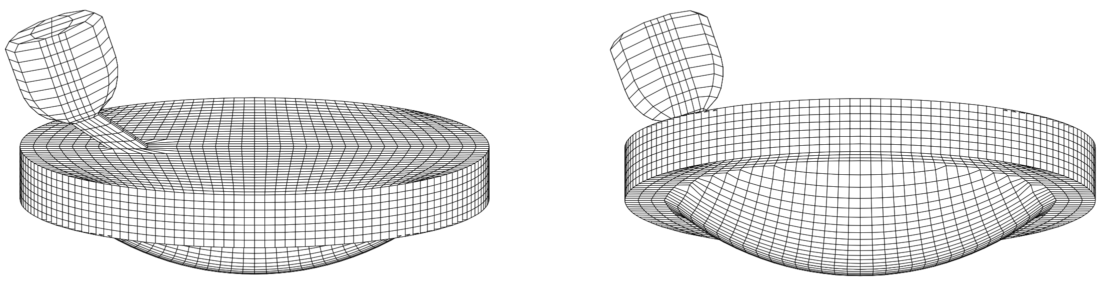


Fig. 1. Geometric mesh of combustion chambers at TDC

3. Course of calculations

The calculations started at BDC at the beginning of compression stroke and lasted for 360º CA since the end of engine power stroke. Lean combustible mixture prepared in inlet system was in engine combustion chambers at the time of starting the compression stroke. 45º CA before TDC the mixture in prechamber was enriched to stechiometric composition by injecting additional dose of fuel, which was approximately 2.5% of energetic share of all the fuel delivered to the engine.

The ignition by spark discharge 12º CA before TDC occurred in prechamber. As a result of pressure increase in prechamber, the burning mixture was forced by the connecting duct to the main chamber in cylinder and ignited the lean mixture. The research were conducted for four values of excess air number equal 1.4, 1.6, 1.8 and 2.0 averaged for main chamber and prechamber. Table 2 presents chosen input parameters of modelled process with their values corrected according to the literature [5],[9] and experimental research at the test stand [3].

Tab. 2. Chosen input parameters of modelled process

Quantity	KIVA name	Dimension	Value
Rotational speed	rpm	rpm	1,0e+3
Cylinder bore	bore	cm	12,0
Piston stroke	stroke	cm	16,0
Squish	squish	cm	1,38
Fuel	gasoline	-	gasoline
Initial pressure (180^o before TDC)	presi	MPa	0,1
Initial temperature (180^o before TDC)	tempi	K	365,0
Kinetic energy of turbulence (180^o before TDC)	tkei	J/m^3	0,17
Turbulence scale (180^o before TDC)	scli	cm	2,2
Ignition advance angle	calign	^{o}CA	-12,0

The prechamber and the main chamber connecting duct of diameter equal 3mm, 4mm, 6 mm and 9 mm was analysed during the simulations (fig. 2).

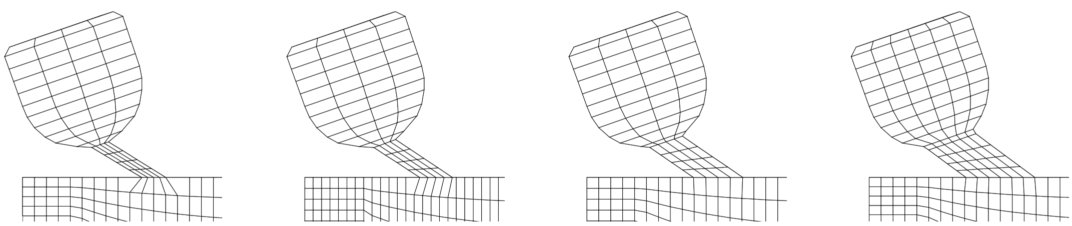


Fig. 2. Parts of geometric mesh with connecting duct of diameter equal: 3mm, 4 mm, 6 mm, 9 mm

4. Modelling results

One of the most important issues in combustion engine is fast air-fuel mixture combustion [7]. The combustion process in piston engine is elongated in the case of lean mixtures combustion. It is caused by greater value of heat transferred to the walls. The combustion speed is dependent on the quality of fuel and air mixing before ignition. The mixing takes place during compression stroke as a result of fresh charge swirl caused by the geometry of combustion chambers. The swirl in cylinder is accompanied with the increase of mixture turbulence level. One of the main parameters describing the turbulence in flow is the kinetic energy of turbulence k [9]. High but not excessive turbulence level before the ignition particularly in the area of spark plug is desired in the case of SI engine with charge prepared outside the working volume. Fig. 3 depicts the kinetic energy of turbulence in function of CA, averaged for both the cylinder and the prechamber during compression stroke for four connecting duct diameters equal: 3 mm, 4 mm, 6 mm and 9 mm.

Fig. 4 depicts the kinetic energy of turbulence in function of CA only in prechamber. The highest kinetic energy of turbulence value averaged for the cylinder and prechamber and for the prechamber only was obtained in the engine equipped with the connecting duct of 3 mm diameter. The lowest kinetic energy of turbulence value was obtained in the case of 9 mm connecting duct diameter. At the time of spark discharge 12ºCA before TDC the k value in the prechamber was: 1760 J/m^3 – 3 mm connecting duct diameter, 855 J/m^3 – 4 mm connecting duct diameter, 254 J/m^3 – 6 mm connecting duct diameter and 112 J/m^3 – 9 mm connecting duct diameter.

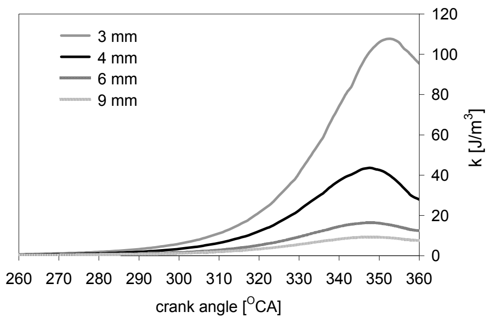


Fig. 3. The kinetic energy of turbulence averaged for both the cylinder and the prechamber of two-stage combustion system engine

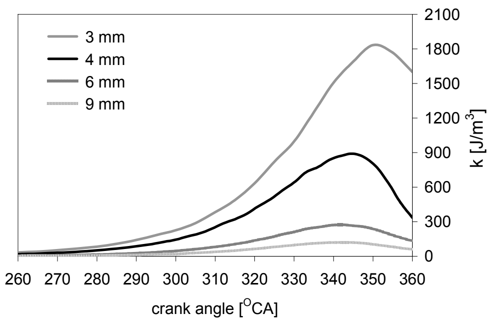


Fig. 4. The kinetic energy of turbulence in the prechamber of two-stage combustion system engine

Fig. 5 depicts the kinetic energy of turbulence distribution in combustion chambers of engine equipped with prechamber at the moment of spark discharge 12ºCA before TDC.

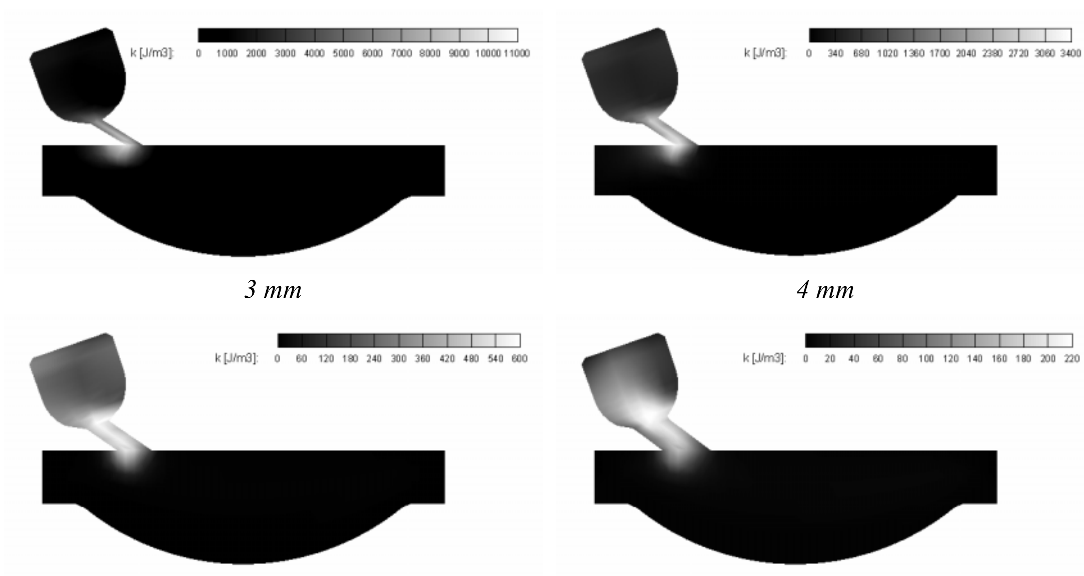


Fig. 5. The kinetic energy of turbulence distribution in combustion chambers of engine equipped with prechamber at the moment of spark discharge 12ºCA before TDC

Lean mixture combustion in traditional SI engine with single combustion chamber leads to deterioration of main engine cycle parameters such as pressure and temperature and as a result it causes the deterioration of engine efficiency revealed by decreased values of maximal torque and maximal indicated pressure. Engines with sectional combustion chamber provide increased combustion speed and in consequence such engines make possible significantly greater mixture impoverishment regarding conventional engines. Figures 6 to 9 depict courses of pressure and temperature averaged for cylinder and prechamber of modelled engine in function of CA at analysed range of mixture composition $\lambda = 1.4 \div 2.0$.

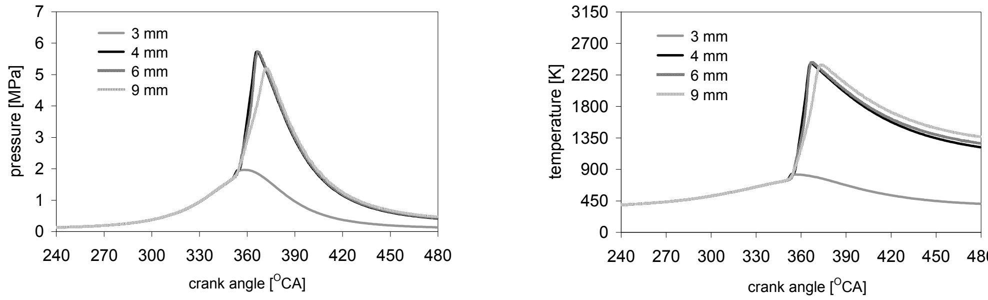


Fig. 6. Courses of pressure and temperature at λ = 1.4

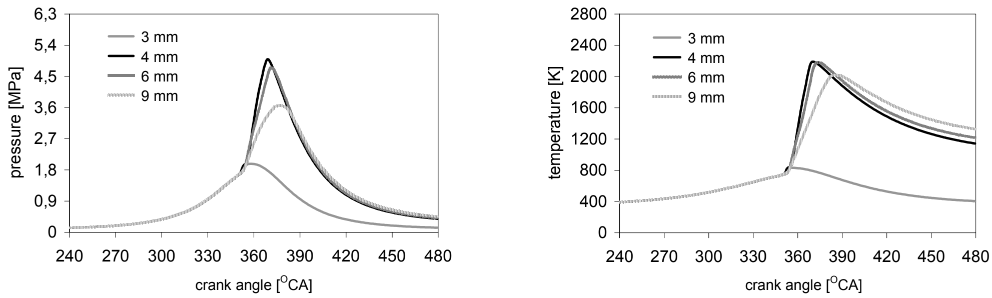


Fig. 7. Courses of pressure and temperature at λ = 1.6

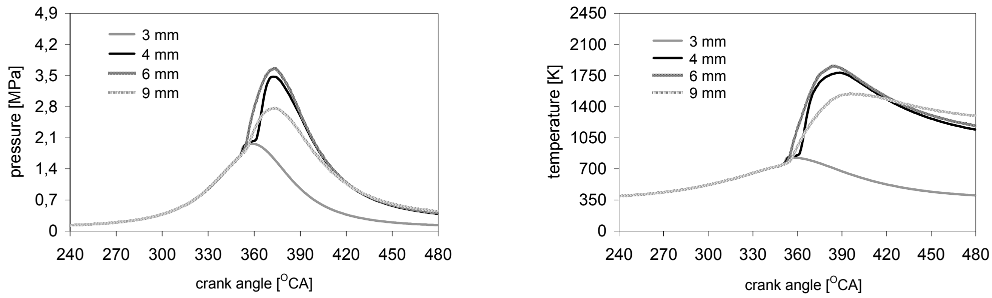


Fig. 8. Courses of pressure and temperature at λ = 1.8

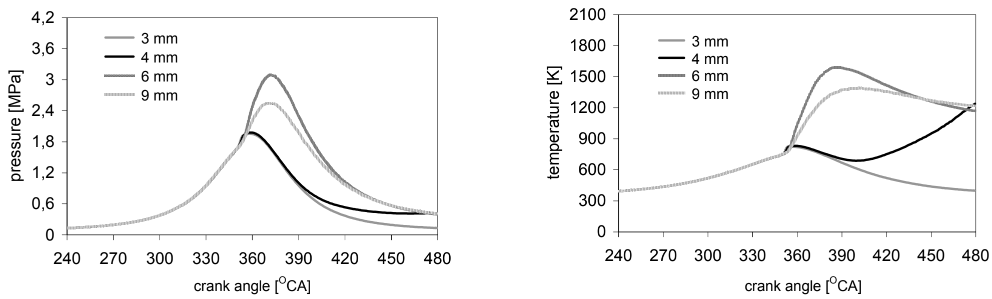


Fig. 9. Courses of pressure and temperature at λ = 2.0

The following charts show that in the case of engine with two-stage combustion system equipped with the connecting duct of 3 mm diameter the obtained values of pressure and temperature were comparable with the compression process. The mixture combustion took place only in the prechamber as the connecting duct of such small diameter did not allow the flame to propagate to the cylinder and ignite the charge in the cylinder.
The engine equipped with connecting duct of diameter equal 4 mm and 6 mm was characterized by similar values of maximal pressure and temperature while combustion mixture of λ=1.2÷1.8. Significant differences in obtained values of pressure and temperature occurred while the leanest mixture of λ=2.0 combustion. In the case of the leanest mixture and the connecting duct of diameter equal 4 mm, the combustion process in cylinder was delayed until the end of expansion stroke due to choking in connecting duct. Because of that fact, obtaining high values of pressure and temperature was impossible. The engine equipped with the connecting duct of the biggest diameter equal 9 mm working at λ=1.2÷1.8 obtained the lowest values of pressure and temperature. At λ=2.0 analysed parameters of engine cycle values were lower than those obtained in the engine equipped with duct of 6 mm diameter and higher than values obtained in engine with duct of 4 mm diameter.
Temperature distribution in working areas of modelled engine with prechamber while combusting mixture of λ=2.0 at selected CA and for four analysed connecting duct diameters are depicted in figures 10 to 12.

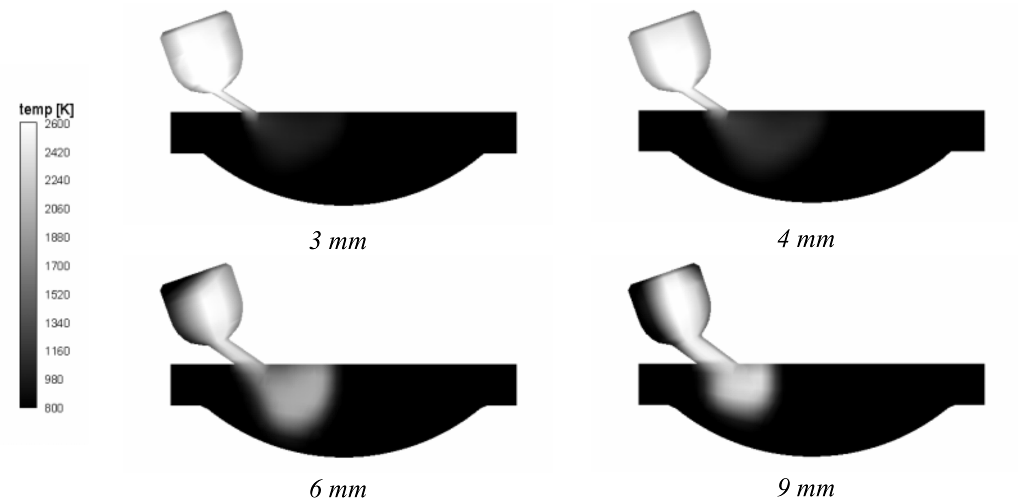
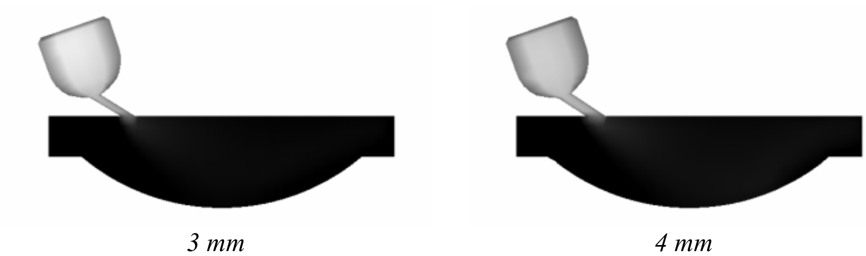


Fig. 10. Temperature distribution in combustion chambers of modeled engine for λ = 2.0 at 5°CA before TDC



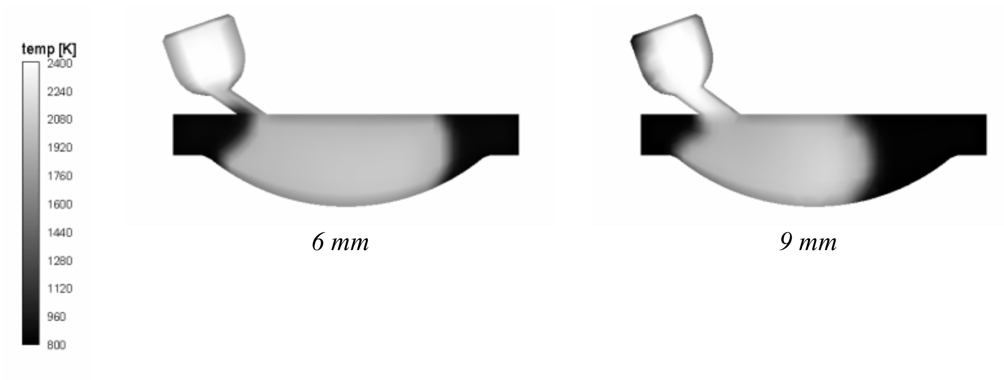


Fig. 10. Temperature distribution in combustion chambers of modeled engine for $\lambda = 2.0$ at 5°CA after TDC

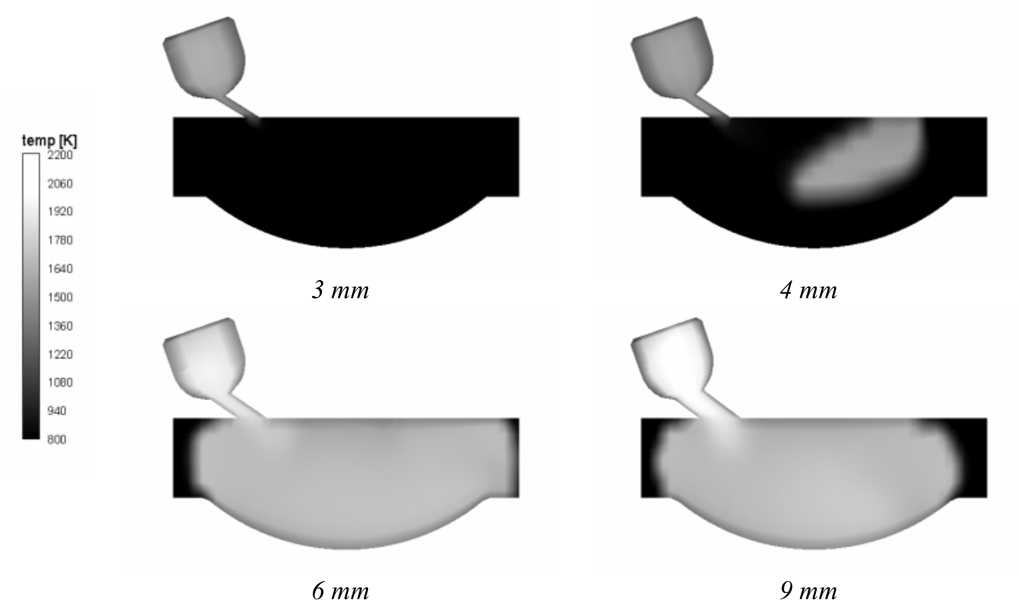


Fig. 10. Temperature distribution in combustion chambers of modeled engine for $\lambda = 2.0$ at 30°CA after TDC

Differences in combustion speed in reference to the diameter of the duct connecting the main chamber and the prechamber can be revealed while analysing the temperature distribution in combustion chambers of modelled engine. The most favourable configuration was the one with connecting duct of diameter equal 6 mm. In the case of this engine the combustion process was the most dynamic one, and the time in which flame propagates in the volume of combustion chamber was the shortest. Too small diameter of the duct didn't allow the flame to propagate fast from prechamber to the main chamber in cylinder and prevented the mixture in the main chamber to ignite fast enough in the configuration of engine equipped with 4 mm duct. In the case of engine with 9 mm duct, the deterioration of flame propagation speed and the decrease of engine cycle parameters maximal values is revealed in reference to the engine with 6 mm duct even though the combustion process was proper.

The combustion process in the case of engine with the connecting duct of 3 mm diameter occurred only in prechamber as the connecting duct of such small diameter did not allow the flame to propagate to the cylinder. The deterioration of maximal engine work cycle parameters is connected with low velocity of flame propagation and insufficient cheat produced during the first phase of lean, homogeneous mixture combustion. Figures 11 to 14 depict heat release rate during combustion and amount of heat released during combustion in function of CA of modelled engine equipped with connecting duct of 3 mm, 4 mm, 6 mm and 9 mm diameter for lambda equal

λ=1.4÷2.0. These values were calculated using the first thermodynamics law for ideal gas on the basis of pressures and temperatures averaged for the whole volume of combustion obtained during numerical modelling.
Heat release rate:

$$\frac{dQ}{d\varphi} = \frac{dU}{d\varphi} + p\frac{dV}{d\varphi},$$

$$\frac{dQ}{d\varphi} = \frac{1}{\chi - 1}\left(\chi p\frac{dV}{d\varphi} + V\frac{dp}{d\varphi}\right), \tag{1}$$

$$\chi = 1{,}392 - 8{,}13\ 10^{-5}\,T \ \ [4], \tag{2}$$

where:

- U - internal energy [kJ],
- p - pressure [kPa],
- T - temperature [K],
- V - volume [m^3],
- χ - isentrope ratio.

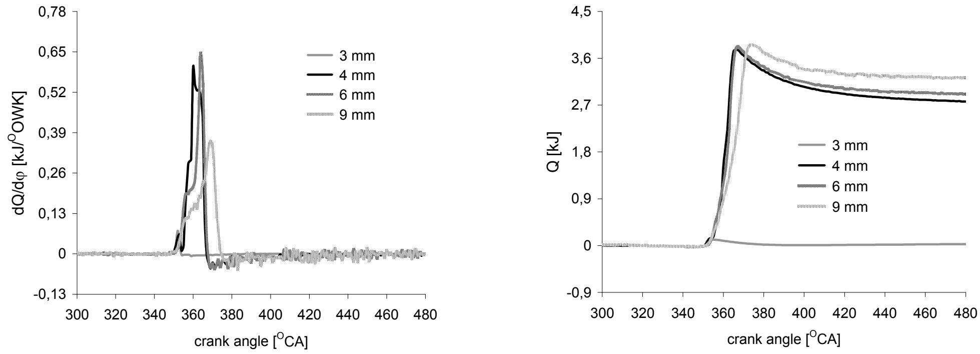


Fig. 11. Heat release rate and amount of heat released during combustion in function of CA for λ=1.4

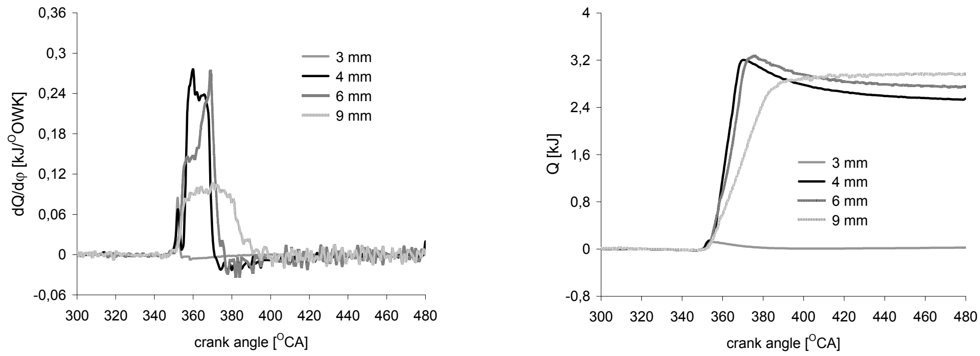


Fig. 12. Heat release rate and amount of heat released during combustion in function of CA for λ=1.6

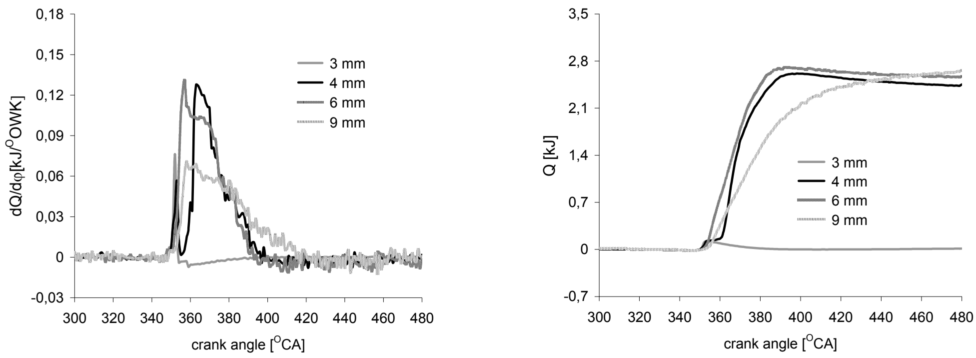


Fig. 13. Heat release rate and amount of heat released during combustion in function of CA for λ=1.8

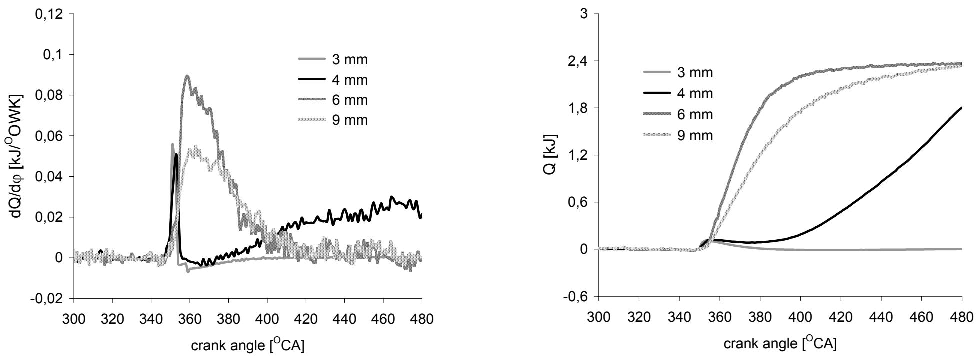


Fig. 14. Heat release rate and amount of heat released during combustion in function of CA for λ=2.0

The analysis of depicted graphs confirms the fact that the most favourable case was the one of engine equipped with connecting duct of 6 mm diameter. In this engine heat release rate curve and the heat released during combustion curve for the whole analysed mixture composition range of λ=1.4÷2.0 are the most favourably and prove the proper curse of combustion.

4. Conclusions

The KIVA-3V numerical code used during research made possible to generate 3D geometric mesh of combustion chambers of S320ZI test engine with prechamber and allowed to perform numerical calculations of processes occurring in this engine. Performed simulations of two-stage combustion process delivered information concerning spatial and time-dependent pressure and temperature distribution in combustion chamber of modelled engine. This information would be extremely difficult to obtain by experimental methods. Maximal values of chosen engine parameters such as pressure and temperature for four different diameters (3 mm, 4 mm, 6 mm, 9 mm) of duct connecting the prechamber and the main chamber were compared in the paper. The analysis of obtained results revealed that the most favourable configuration was the one with connecting duct of 6 mm diameter. Maximal values of pressure and temperature were obtained during the mixture of λ=1.4÷2.0 combustion in engine of 6 mm duct. Even though the mixture was leant to the level of λ= 2.0, the combustion process in this engine was fast enough to provide obtaining high values of indicated pressure and proper engine efficiency in consequence.

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