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ASSESSMENT OF THE 3-D COMBUSTION MODEL IN THE MARINE 4-STROKE ENGINE

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Abstract

The aim of this research is to formation of the model of combustion process in the marine, 4-stroke, diesel engine. The chosen object of research is a laboratory AL25/30 engine. For achievement of the aim, laboratory measurements are made and the results are used to determine the boundary and initial conditions. In addition, measurements of the fuel injection shape are made on the test bench and are made a faithful geometric model of structural components of the engine cylinder. The obtained data has been implemented in the three-dimensional model comprising the fuel injection phenomena, the brake-up and the evaporation of fuel, auto-ignition, flame propagation and heat exchange with the structural elements of the engine cylinder. As a result of activities succeeded in creating a model of the combustion process in the cylinder, which has been positively validated due to the maximum combustion pressure and the temperature of the exhaust gases. Obtained results of calculations also allowed verifying the assumption of negligible auto-ignition delay. The adopted ECFM-3Z model of the ignition and combustion, used for modeling of combustion in diesel engines, showed the calculated size of the auto-ignition delay of 7-8 ° CA.

Key words: *marine engine, multidimensional model, Dukowicz evaporation model, TAB brake-up model, ECFM-3Z combustion model*

1. Introduction

The development of the construction of the marine piston engines in the XXI century goes towards the growth of the combustion process efficiency while gaseous emission reduction. Both of mentioned objectives can be achieved through the development of engine design for optimal combustion process throughout all engine load conditions. Optimization of the combustion process should proceed towards the obtaining a certain fuel ignition in possible large ranges of thermodynamic parameters in the engine cylinder. The fuel ignition should be provided during the engine operation at malfunction conditions such as leakage of the engine cylinder or malfunction of the air exchange system. Moreover, combustion process in the engine cylinder should be complete. Such state of affairs provides, inter alia, the high temperature of the combustion process [1]. Unfortunately, an excessive of the combustion temperature increase results in an increase of the nitric oxides emission. For this reason, the combustion process parameters should be governed for the engine load, the engine rotational speed and the technical condition of the engine. Such correction provides electronic control of the valve timing and the fuel injection process [2].

To the optimal control of the combustion process requires knowledge about the phenomena occurring in the engine cylinder. Suitable for this purpose are the methods based on the direct measurement of engine operating parameters. It should be noted, that the measurement parameters of fluids from the engine operation may be insufficient. The temperature of the cooling water, oil, and exhaust gas does not reflect the temperature of the engine cylinder and the measurement of cylinder pressure may be burdened with a significant error [3]. Due to the rapidity of phenomena occurring in the engine cylinder direct measurement of the combustion temperature is not possible. Applying optical methods [4] is possible only after modernization of the engine construction in the laboratory conditions and the cost of this type measurements may be relatively high [5].

In recent years, numerical methods are gaining in popularity. The reason for this is the increase in computing power and significant decrease in the modeling cost. Numerical methods can be applied to both qualitative and quantitative evaluation of the phenomena occurring in the engine cylinder [6]. It should be noted, that such models must be used with great care and only after a positive validation of the combustion process parameters concerned with the data obtained through the direct measurements.

The aim of the work is to create a multi-dimensional model of the combustion process in the cylinder of the marine 4-stroke engine. This model is intended to allow the appointment of thermodynamic parameters of the engine exhaust gases and analysis of phenomena occurring during the combustion process.

2. The model description

Model phenomena occurring in the engine cylinder was prepared as Euler description [8]. The base model of the combustion process in the engine cylinder is a geometric grid, including the shape of the cylinder with the air intake duct, the outlet duct and exhaust and inlet valves. The geometric grid was built based on the technical documentation of the research object. As a research object was selected laboratory 4-stroke engine, type Sulzer Al25/30. Analysis and selection of spatial grid parameters are presented in [9]. Prepared grid is presented in Fig.1 and parameters of the Al25/30 engine in Tab.1.

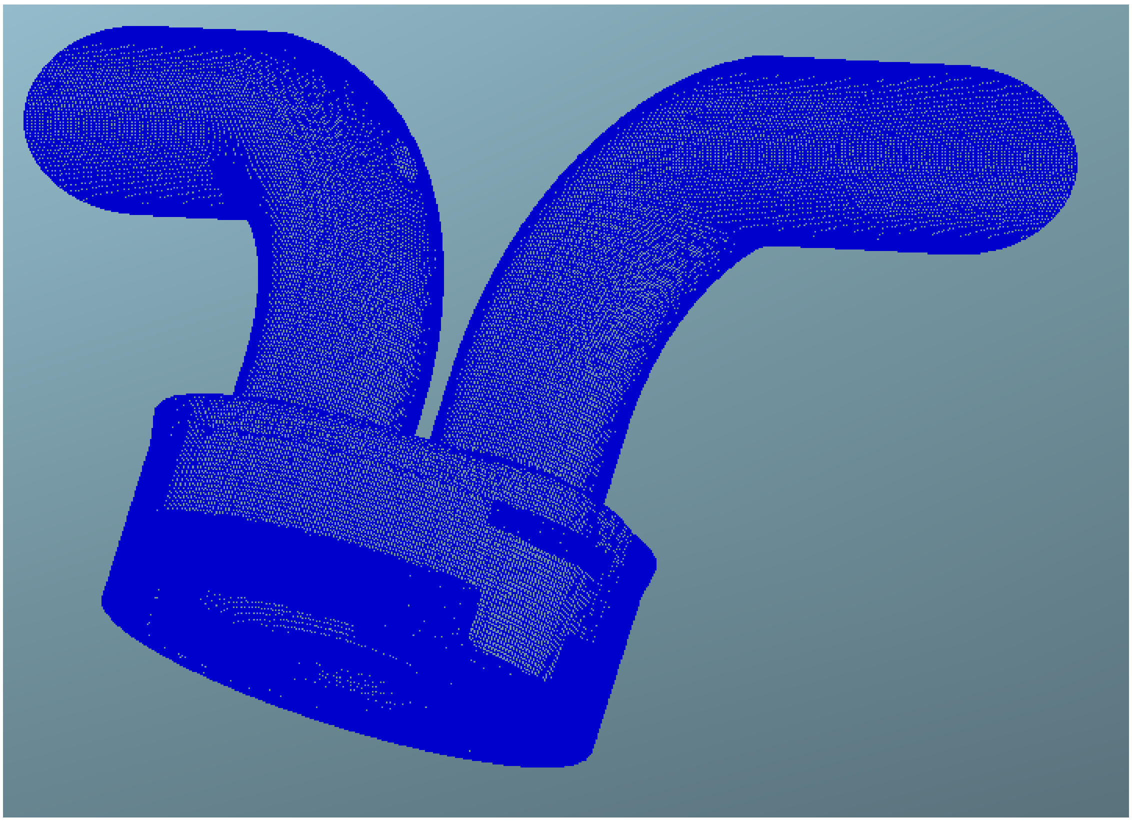

Fig.1. The grid of the engine cylinder

Tab. 1. The laboratory engine parameters

Parameter	Value	Unit
Max. electric power	250	kW
Rotational speed	750	rpm
Cylinder number	3	–
Cylinder bore	250	mm
Stroke	300	mm
Compression ratio	12,7	–
Injector nozzle		
Holes number	9	–
Holes diameter	0.325	mm
Holes position diameter	7	mm
Holes position angle	150	deg.
Spray cone angle	6	deg.
Opening pressure	25	MPa

2.1 The fuel injection model

The fuel injection model is based on the geometrical dimensions of the injector nozzle, which are presented in Tab.1. According to [10], page 526, the injected fuel cone angle depends on the differential pressure in the nozzle and in the cylinder chamber and the ratio of the diameter to the length of the injector holes. Due to the low credibility of the measurement of the length of the injector holes, the injected fuel cone angle was determined empirically on a test stand. The tests were conducted at atmospheric pressure, assuming that the injection pressure is many times greater than the pressure prevailing in the cylinder chamber. Fuel injection was photographed at a frequency of 60 frames per second, and measurement was made on the collected photographic material.

Mass of fuel injected was calculated on the basis of fuel consumption. Fuel injection characteristic was determined based on the measured characteristics of injection pressure, according to the following relation:

$$\dot{m} = f\left(\sqrt{(p_w - p_c)}\right), \tag{1}$$

where:
m – mass of injected fuel in [g/s],
p_w – injection pressure in [Pa],
p_c – pressure in the cylinder chamber in [Pa].

Assumed injection timing equals to 25 degrees before top dead center of the piston position. Start of injection equals to 695 degrees angular position of the crankshaft. The basis for this decision was to analyze the characteristics of the pressure on the indication valve. I assumed that the pressure signal in the indicator valve is not delayed relatively to the pressure in the engine cylinder. In addition, it is assumed that self-ignition delay is negligibly small.

2.2 The brake-up and the evaporation model

The initial value of the droplet diameter of the fuel injection is taken as 0.325 mm, which corresponds to the diameter of the nozzle holes. A further break-up of fuel droplets has been described by the Lagrange description [8] with TAB model application [11]. This model specifies the conditions for breaking-up of fuel droplets as a dimensionless factor that depends on the density of the fuel and the surrounded air, the viscosity of the fuel droplet, the relative velocity and diameter of droplets. If the value of the mentioned ratio is greater than 1, the drop breaks up. Distribution of mean droplet diameter, determined by the Sauter method [10], is assumed as Chi^2.

Simultaneously with the fuel atomization process begins the process of evaporation. This process results from the heating of fuel droplets. For modeling of heat flow from the air to fuel droplets and mass flow of fuel vapors from the droplets to air the Dukowicz's model is adopted [12]. The spherical shape of fuel droplets (microgravity conditions) and a constant temperature and heat transfer conditions on the surface of the droplet is assumed.

2.3 The combustion process

Evaporated fuel is mixed with air in the engine cylinder. To modeling these phenomena the k-ε model [6] were using. The combustion process was described by the ECFM-3Z model. Mentioned model is prepared by Groupement Scientifque Moteurs consortium [13]. It is a model developed for modeling the combustion in diesel engines and it belongs to the CFM (Coherent Flame Model) class of models. The model can be used for both direct injected [14], and spark ignited engines [15]. This model assumes that the chemical reactions take place in the relatively narrow layer of the flame. The flame progresses to the direction of fresh mixture of air and fuel. Mentioned flame layer is defined a homogeneous mixture of fuel and air and its shape and size is diffusion phenomena. In the present model, the self-ignition delay is determined by air temperature, the density of the mixture and the molar concentration of oxygen and fuel. Chemical kinetic calculations are prepared for assumed substitute fuel composition in the form of hydrocarbon $C_{13}H_{23}$.

3. The laboratory stand

Boundary conditions and initial conditions, as well as data necessary for validation of this model were collected during laboratory tests. The laboratory researches are carried out on a laboratory engine Al25/30, which basic parameters are presented on Tab.1. During the laboratory research the engine operate at a constant speed and load equal 250kW. The engine was fueled by diesel oil with known specifications. The research covered the registration of parameters of turbocharger, fuel system, lubrication, cooling and air exchange systems. All parameters are measure with a sampling time equal to 1 second and used for modeling the data was the arithmetic average of 3 observations. Closer description of laboratory research can be found in [16].

4. The validation model and obtained results

Initial conditions of the model are measured during laboratory research pressure and temperature of the charge air in the air intake duct. These conditions were adopted for the entire volume of the intake manifold and the engine cylinder at the crank shaft angular position corresponding to beginning of the closing of the intake valve. Moreover, boundary conditions are determined, as the temperature of individual structural elements of the engine cylinder. The temperature of the intake valve with the inlet channel is assumed as measured temperature of the charge air. The boundary temperatures of the cylinder liner, piston and cylinder head are assumed as the lubricating oil temperature, measured behind the engine. For the temperature of the surface of exhaust valve with the exhaust duct is assumed temperature of the exhaust gas, measured behind the modeled cylinder of the engine.

The calculations are carried out from the beginning of the intake valve closing (575 degrees of the contractual angle of crankshaft position - CA), by compression and combustion processes until the end of the opening of the exhaust valve (contractual 850° CA). The top dead center of the piston position during the power stroke engine has the contractual CA equal 720°. Each step of the calculation corresponds to the CA required from 17 to 100 iterative calculations of the balance equations of momentum, energy and continuity for each grid cells. The convergence criterion for momentum, pressure and energy was assumed as the change in values of mentioned parameters is not larger than 0.01 after each iteration. The variable step of calculation was assumed from 0.01 to 1 degree of CA also. The results of calculations of each step are the input to the calculations for the next step. Calculations were performed using Fire software in 2013.1 version from AVL manufacturer.

On the 2 drawing the results of model validation are presented. Criteria of the model validation are assumed temperature of the exhaust gas after cylinder at the opening of the outlet valve (contractual 840° CA) and maximum combustion pressure in the engine cylinder.

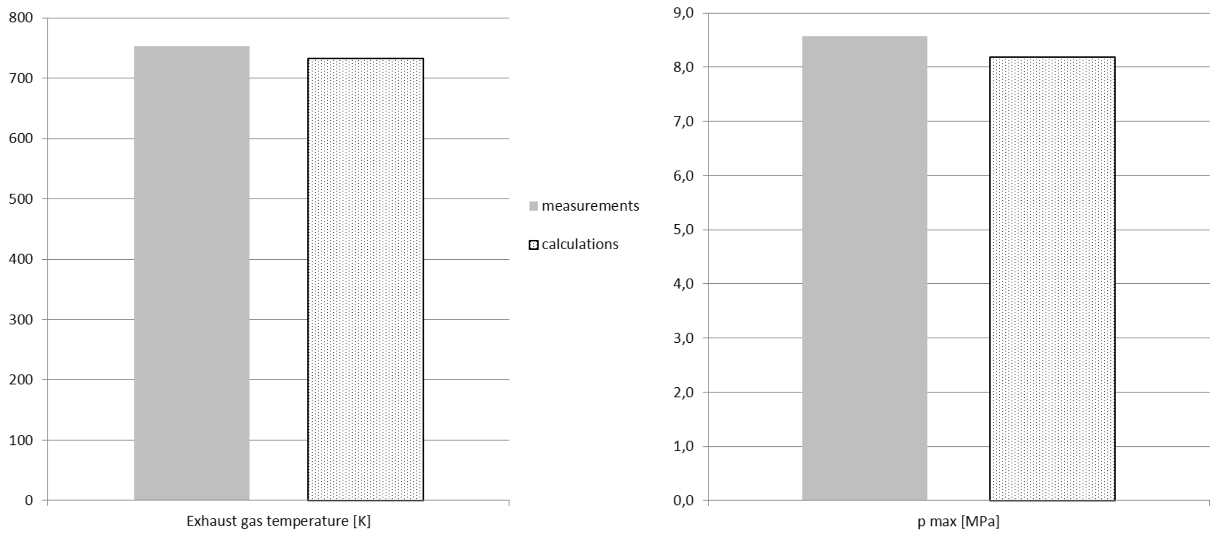

Fig.2. The calculated exhaust gas temperature and maximum pressure (p max)

According to the presented results, calculated values are smaller from measured by 2.8% in the case of the exhaust gas temperature and by 4.5% in the case of the maximum in cylinder pressure.

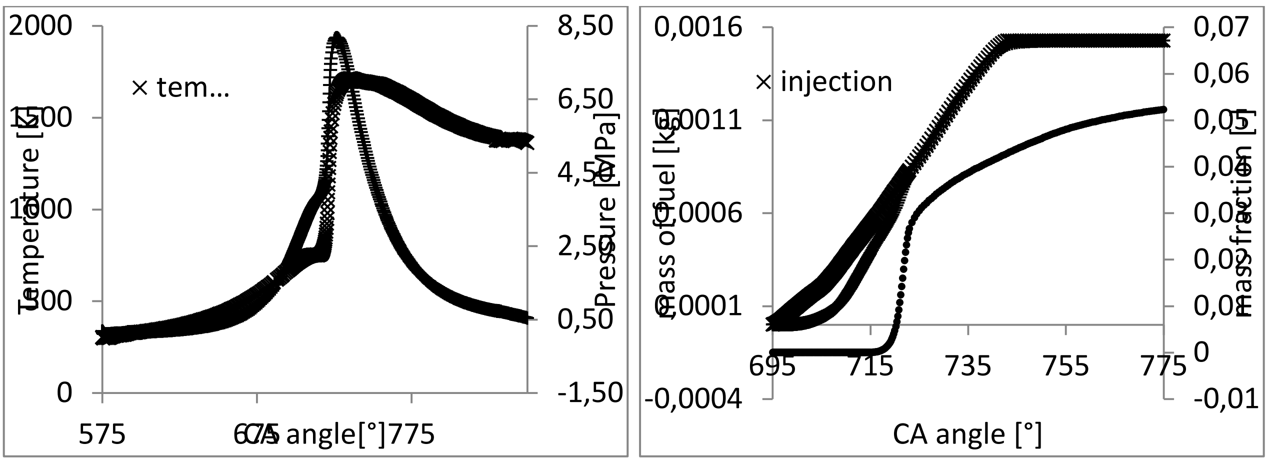

Fig.3. The mean parameters in all volume of the engine cylinder

On the left side of the Fig.3 the average values of temperature and pressure calculated for the whole volume of the engine cylinder are shown. It should be noted that obtained results are qualitatively consistent with results available in the literature [10]. On the right side of the Fig.3 the characteristics of the injection, vaporization and combustion of the fuel in the engine cylinder are shown. As previously mentioned, the fuel injection characteristic is a function of the (1) equation.

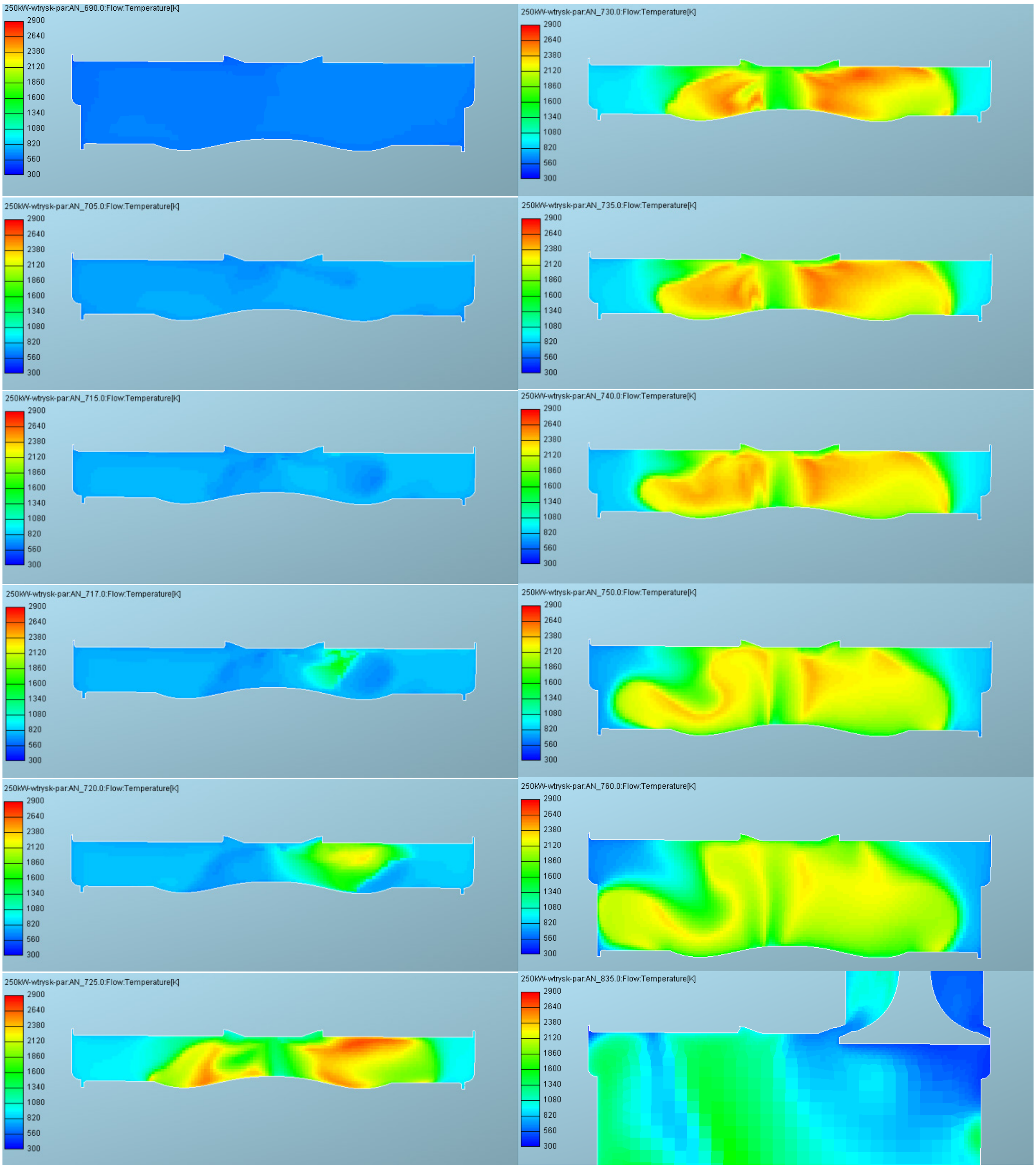

Fig.4. The temperature of the combustion process

In the initial period of fuel injection at a relatively low pressure and temperature in the engine cylinder, the amount of the evaporating fuel is less than the amount of injected fuel. Ignition of the fuel, referred to the sudden increase in the fraction of fuel burned, causes an increase in temperature and pressure in the cylinder. The result of this is an increase of the amount of evaporating fuel to a value equal to the amount of injected fuel. The residual value of the fuel burned fraction in the initial stage of the combustion process is the result of chemical dissociation of fuel before auto-ignition. It should be noted that illustrated in Fig.3 temperature values are the average values of temperatures for the entire volume of the cylinder. Local temperatures differ significantly from the average values.

Figure 4 shows temperature changes in the engine cylinder with the rotation of the crankshaft on the example of the axial cross-section of the cylinder. Cross-section of the cylinder passes through the axis of the right side of the nozzle hole. The lower profile of the cross section corresponds to the shape of the piston bottom. According to the presented the results, auto-ignition of the fuel occurs 3-4° before TDC of the piston position and causes a sudden increase of temperature. Since then, the development of combustion occurs towards a cooler piston bottom and the cylinder walls. The initial combustion period is very fast, because the combustion rate is determined by kinetic phenomena that are taking place in a mixture of fuel and air. At the first, the turbulent combustion is slowed down at the moment of lack of combustible mixture in the cylinder. According to the result from Fig.3, starting from the TDC of the piston position (contractual 720° CA), the combustion rate is determined by the fuel injection characteristics and diffusion phenomena, occurring on the surface of the liquid fuel droplets. A symptom of this state of affairs is to cover the characteristics of fuel injection and fuel evaporation.

It should be noted that calculations show a significant delay of auto-ignition. Assumed models and boundary and initial conditions have resulted in auto-ignition delay equal to 7-8° CA. For this reason, it must be assumed that the auto-ignition delay is not negligible, as assumed. Furthermore, according to the cited measurement results carried out on the same research facility [3] there is a significant delay between the measured pressure signal on the indicator valve and the signal measured in the engine cylinder. Therefore, the conclusion must accept that the determination of injection based on the characteristics of the pressure signal on the indicator valve gives a result considerably delayed due to two discussed phenomena.

According to results, presented in Fig.4, the maximum combustion temperature comes to 2500K. It should also be noted that at large part of the cylinder volume the temperature of the combustion process is high enough to allow oxidation of the nitrogen in accordance with the thermal Zeldovich mechanism [10].

5. Conclusions

The aim of this research was to formation of the model of combustion process in the marine, 4-stroke, diesel engine. The chosen object of research was a laboratory AL25/30 engine. For achievement of the aim, laboratory measurements were made and the results were used to determine the boundary and initial conditions. In addition, measurements of the fuel injection shape were made on the test bench and made a faithful geometric model of structural components of the engine cylinder. The obtained data has been implemented in the three-dimensional model comprising the fuel injection phenomena, the brake-up and the evaporation of fuel, auto-ignition, flame propagation and heat exchange with the structural elements of the engine cylinder. As a result of activities succeeded in creating a model of the combustion process in the cylinder, which has been positively validated due to the maximum combustion pressure and the temperature of the exhaust gases. This model allowed for a qualitative assessment of the processes occurring in the engine cylinder. The analysis of the temperature distribution in the cylinder showed significant differences of temperature in the volume of the cylinder. As a result of these differences in the maximum temperature reached in the cylinder combustion to 2500K, and the average for the entire volume of the cylinder is lower than 600K.

Obtained results of calculations also allowed verifying the assumption of negligible auto-ignition delay. The adopted ECFM-3Z model of the ignition and combustion, used for modeling of combustion in diesel engines, showed the calculated size of the auto-ignition delay of 7-8 ° CA. Therefore, further work is needed on the quantitative validation of the obtained model.

Acknowledgments

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