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RESEARCH ON ENERGETIC PROCESSES IN A MARINE DIESEL ENGINE DRIVING A SYNCHRONOUS GENERATOR FOR DIAGNOSTIC PURPOSES PART 2 – MATHEMATICAL MODEL OF THE PROCESSES

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Abstract

The paper presents the formation steps to the mathematical model of the gas-dynamic processes occurring in the cylinders of the marine diesel engine driving a synchronous generator. All steps have been illustrated by charts of the modeled parameters as a function of the crankshaft angle. In addition, the paper presents the problem that occurs during modeling quasi-stationary dynamic processes.

Keywords: energetic processes, marine diesel engine, mathematical model

List of the most important markings

a	–	linear acceleration	R	–	individual gas constant
c	–	carbon mass	s	–	mass fraction of sulfur
cp	–	specific heat	S	–	cross sectional area
d	–	diameter of the valve	t	–	time step
D	–	diameter of the cylinder	T	–	temperature
g	–	gravitational acceleration	w	–	linear velocity
h	–	mass fraction of hydrogen / valve lift	x	–	route
K	–	bulk modulus	W	–	calorific value
l	–	rod length	V	–	volume / capacity
L	–	vaporization heat of fuel	α	–	crankshaft angle
L_0	–	theoretical air requirement	κ	–	heat capacity ratio
L_r	–	actual air requirement	λ	–	excess air ratio
m	–	mass	μ	–	outflow/inflow ratio
$\dot{m}$	–	mass flow	ρ	–	density
n	–	mass fraction of nitrogen	τ	–	the amount of fuel supplied to the air cylinder unit
p	–	pressure	ω	–	angular velocity
Q	–	heat			
r	–	crank radius of the crankshaft			

Applied indexes

cyl	–	applies to the cylinder
dol	–	applies to the intake
ks	–	applies to the combustion chamber
pal	–	applies to the fuel
wtr	–	applies to the injection
wyl	–	applies to the outlet
w	–	applies heat used for heating fuel
zaw	–	applies to the valves

1. Introduction

Gas-dynamic processes occurring in the 4-stroke cylinder of the marine diesel engines of compression ignition is an extremely complicated issue. Therefore, most researchers use ready-made, commercial computer tools such as KIVA or FLUENT which allows to conduct the analysis of processes in engines [1, 2, 3]. This software was developed for the design of internal combustion engines, therefore using it in order to assess the technical condition of existing engines is problematic. This is due to the fact that each modelling change in the structure design of the engine (corresponding to its degradation) involves the need of modifying the complex elements of the grid. Expenditure of time necessary to carry out the calculations and the amount of work with model modifications seriously hampers the use of such applications for diagnostic purposes.

It is reasonable to develop a specialized diagnostic model allowing for relatively simple modifying parameters of the engine structure for simulation purposes for technical fault conditions. It should also allows the analysis of the impact of a given set of unsuitability on the key parameters of the engine such as pressure and temperature inside the cylinder, the rotational speed of the crankshaft, the speed of air flow into the cylinder, the flow rate of the exhaust gas from the cylinder, etc.

The authors had focused on the development of such model and computer program based on equations used during the model description .

2. Modeling of piston movement in the cylinder

The first stage of model development of the marine diesel engine is to determine the course of change in the volume of the engine cylinder as a function of crankshaft angle. The mathematical description of movement of the piston in the engine has been in details described in the literature [6, 7].

Firstly, it is necessary to enter the parameters describing the geometry of the crank-piston mechanism i.e.:

- rod length *l*,
- crank radius of the crankshaft *r*.

On this basis, it is possible to calculate the length of the path of the piston (displacement) as a function of the crankshaft revolution angle:

$$x_{cyl} = \mathrm{r} \cdot \left(1 - cos\alpha_{cyl} + \frac{\mathrm{r}}{2 \cdot \mathrm{l}} sin^2\alpha_{cyl}\right), \tag{1}$$

In equation (1) is the value of the angle of rotation of the crankshaft α_{cyl}, variable with each iteration of every $0{,}1^0$ OWK. In the case of multi-cylinder engines, angles which shift between the cylinders should be included. The length of piston as a function of crankshaft revolution angle is based on the calculated value of the equation 1.

The value of the instantaneous velocity of the piston as a function of crankshaft revolution angle is described in the formula:

$$w_{cyl} = \frac{\mathrm{d}x}{\mathrm{d}t} = r \cdot \omega \cdot \left(sin\alpha_{cyl} + \frac{\mathrm{r}}{2 \cdot \mathrm{l}} \cdot \sin 2 \cdot \alpha_{cyl} \right), \tag{2}$$

where:

$$\omega = \frac{1}{10 \cdot t}$$

However, the acceleration of the piston is described by the equation:

$$a_{cyl} = \frac{\mathrm{d}w}{\mathrm{d}t} = \mathrm{r} \cdot \omega^2 \cdot \left(cos\alpha_{cyl} + \frac{\mathrm{r}}{\mathrm{l}} \cdot \cos 2 \cdot \alpha_{cyl} \right), \tag{3}$$

3. Modeling of gas-dynamic processes in the engine cylinder without taking into account the exchange process of the medium

Knowing the value of piston displacement as a function of crankshaft angle calculated according to the formula (1) the capacitance change of the cylinder as a function of crankshaft revolution angle can be calculated (equation 4).

$$V_{cyl} = \left(\frac{\pi \cdot \mathrm{D}^2}{4} \right) \cdot \left(x_{cyl} + \mathrm{x_{ks}} \right), \tag{4}$$

Having changes in the course of the combustion chamber volume, it is possible to calculate the weight of the medium contained in the cylinder for the first iteration. The medium mass was calculated according to the formula:

$$m = \rho \cdot V_{cyl}, \tag{5}$$

assuming that V_{cyl} is equal to the maximum capacity of the cylinder defined by the equation (4), where x_{cyl} is equal to twice the length of the crank, it has been assumed that the air density is ρ=1,2 $\frac{kg}{m^3}$.

In the initial phase of model development it was assumed that the thermodynamic medium is only compressed and expanded (the inflow and outflow of the cylinders of the engine is not taken into account and does not include fuel injection). Having the mass medium, it was possible to calculate the temperature and pressure inside the cylinder according to the formula:

$$p_{cyl} \cdot V_{cyl} = m_{cyl} \cdot \mathrm{R_{cyl}} \cdot T_{cyl}, \tag{6}$$

and:

$$\frac{T_{cyl}}{T_{cyl-1}} = \left(\frac{p_{cyl}}{p_{cyl-1}} \right)^{\frac{\kappa_{cyl-1}-1}{\kappa_{cyl-1}}}, \tag{7}$$

In order to use equations (6 and 7) it was necessary to determine the value of the individual gas constant R_{cyl} for the medium contained within a cylinder and the adiabatic exponent κ, which is calculated from the dependence (8).

$$\kappa_{cyl} = \frac{1}{0{,}7267+6{,}1\cdot10^{5}\cdot(T_{cyl-1}-327)-2\cdot10^{-8}\cdot(T_{cyl-1}-327)^{2}} - 0{,}6\cdot\tau_{pal}, \tag{8}$$

where:
τ_{pal} – the amount of fuel supplied to the cylinder per unit of air (in the first iteration of was zero),
$T_{cyl\text{-}1}$ – the temperature of the medium inside the cylinder of the previous iteration (the first iteration assumes that it is 293 K).

Whereas the value of the individual gas constant is calculated from (9).

$$R_{cyl} = 287 + 18{,}8\cdot\frac{1}{\lambda\cdot L_0}, \tag{9}$$

The value of the individual gas constant for the medium if the fuel injection amount is not taken into account is: $287\,\frac{J}{kg\cdot K}$.

The value of the coefficient of excess air λ for the intake and compression stroke reaches an infinitely high value; however, during the injection of infinite value, it changes to the value adopted for the type of engine and torque load. At the early stage of modeling, if fuel injection is not included into the cylinders of the engine, the air excess ratio is treated as infinitely large. Theoretical air consumption L_0 occurring in the equation (9), is described by the dependence:

$$L_0 = 11{,}84\cdot c + 34{,}214\cdot h, \tag{10}$$

where: the mass share value of carbon c and hydrogen h depend on the type of fuel used.

Knowing the mass of air in the cylinder, it is possible to calculate the parameters inside the cylinder such as temperature and pressure. The temperature was calculated based on equation (7) which, after transformation, took the form [4]:

$$T_{cyl} = \left(\frac{V_{cyl-1}}{V_{cyl}}\right)^{\kappa_{cyl}-1}\cdot T_{cyl-1}, \tag{11}$$

after substituting the equation (6) the form of:

$$T_{cyl} = \left(\frac{V_{cyl-1}}{V_{cyl}}\right)^{\kappa_{cyl}-1}\cdot\frac{p_{cyl-1}\cdot V_{cyl-1}}{m_{cyl}\cdot R_{cyl}}, \tag{12}$$

The fluid pressure within the cylinder is calculated from the dependence:

$$p_{cyl} = \frac{m_{cyl}\cdot R_{cyl}\cdot T_{cyl}}{V_{cyl}}, \tag{13}$$

Its course is carried out as a function of crankshaft angle illustrated in Figure 1 (the inflow and outflow medium from the cylinder and fuel injection was not taken into account).

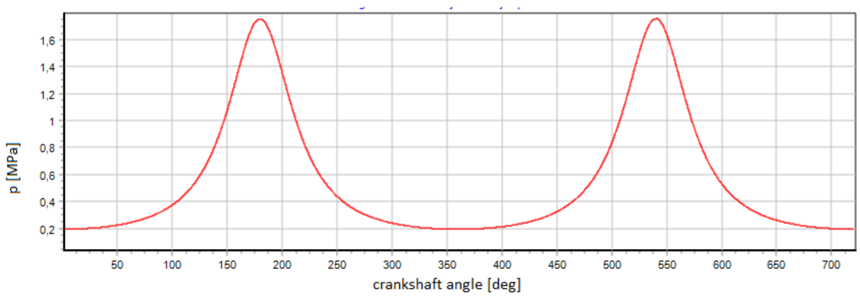

Fig. 1. Course of pressure change within a cylinder as a function of crankshaft angle

4. Modeling the process of air intake and outlet medium from the cylinder

The next step in the construction of the model was to consider the possibility of air intake and exhaust of the medium. It was necessary to know the opening and closing angles of the intake and exhaust valves. In model, data of the engine Sulzer type 6 AL 20/24 had been used. Engine is installed in the Laboratory of Marine Power Plant Operation of the Polish Naval Academy. Apart from the opening and closing angles of the valves, information regarding the change of the active cross sectional area of the valve as a function of crankshaft angle was also necessary. When the valves are closed, the active cross-sectional area of the valves is zero, whereas when the valves are open, the active cross-sectional area is calculated according to the following equation [8]:

$$S_{zaw} = \pi \cdot \left(d + \frac{d}{2} \cdot sin2 \cdot \gamma\right) \cdot h \cdot cos\gamma, \tag{14}$$

where markings were adopted as shown on Figure 2.

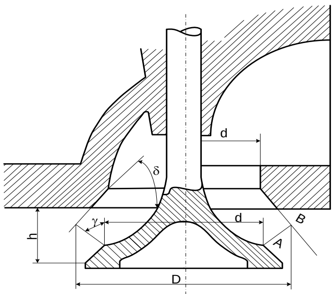

Fig. 2. Scheme of valve used in the calculation of the active cross-sectional area of refrigerant flow [8]

To use the equation (14) it was necessary to be familiar with the course of changes in the value of raising the valve as a function of crankshaft angle. In most of the models proposed in the literature, this value is determined as a two-state, i.e. when the valve is closed, the value of valve lift is equal h = 0 and when the valve is open h is equal to the average amount of valve lift [7]. This approach of modeling the changes in the field of the cross section area of the exhaust valve is illustrated in Figure 3. This simplification assumes that the area denoted by A must be equal to the area B.

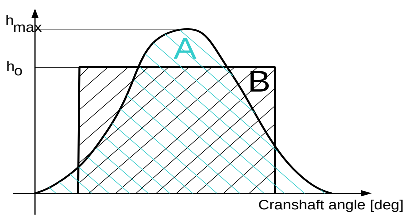

Fig. 3. Diagram of the actual (A) and theoretical (B) lift of the exhaust valve

Due to the need to develop a model, it was found that binary model is insufficient and it was decided to model the operation of the valve according to the According to the lack of knowledge of the geometry of the cam in the timing system, it was decided to simplify the assumption that the valve lift is changed in accordance with a second power function, whose characteristic points are: the h=0 value for the angle of opening and closing the valve, and the value of h=max for the angle corresponding to half the angle of opening the valve [10]. The method of determining the valve lift is presented on Figure 4.

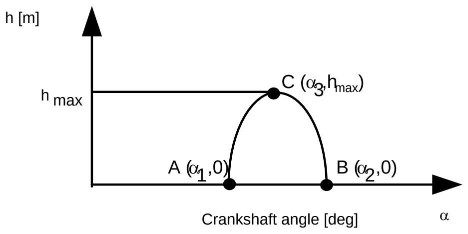

Fig. 4. The method of determining the course of valve lift as a function of crankshaft revolution angle

Changes in the lift value of the intake valves for the engine with firing order 1-5-3-6-2-4 was shown in Figure 5 a, while the waveforms of the active change of the sectional areas of the intake valves as a function of crankshaft angle was shown in Figure 5 b.

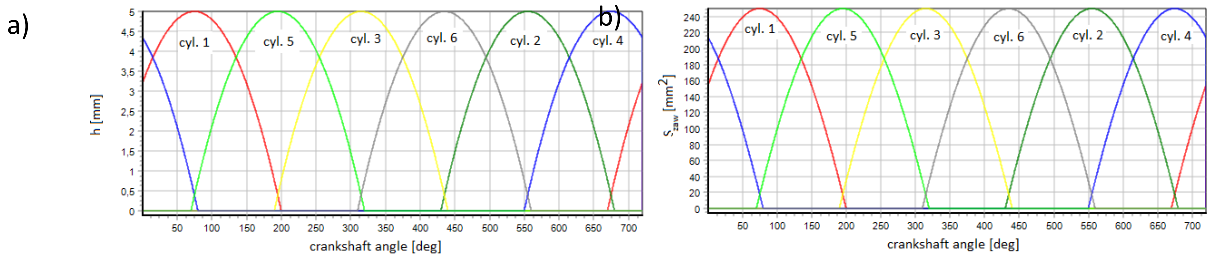

Fig. 5. Change diagram a) the intake valve lift, the change of the active sectional area of the intake valve as a function of engine crankshaft revolution angle

If we dispose of the course of the active sectional area of the intake valve, the value of the internal pressure of the cylinder, the air supply pressure of the cylinder and counter pressure of exhaust gas, it is possible to calculate the flux of the medium flowing through the intake and exhaust valves. The flux value and inlet medium flowing out of the cylinder describes the equation:

$$\dot{m} = w \cdot S_{zaw} \cdot \rho \cdot \mu \tag{15}$$

The density of the medium can be calculated using the equation:

$$\rho = \frac{p}{R \cdot T} \tag{16}$$

While the velocity of flow of the medium is determined from the de Saint-Venant equation [4]:

$$w = \sqrt{\frac{2 \cdot \kappa_1}{\kappa_1 - 1} \cdot R_1 \cdot T_1 \cdot \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\kappa_1 - 1}{\kappa_1}}\right]}, \tag{17}$$

In the case of air flow into the cylinder, the parameters of index 1 relate to inflow air, whereas index 2 relates to the parameters inside the cylinder. Index 1 shows the performance index within the cylinder for gas outflow via the exhaust valves whereas index 2 relates to a exhaust gas in the outlet channel.

Masses flow suppling cylinders could be calculated on the basis of equation 15 as a $\dot{m}_{dol}$ and the medium leaving the cylinders as a $\dot{m}_{wyl}$. In order to calculate the increase in mass in the cylinder, it is necessary to incorporate the inflowing mass with air into the cylinders, which is described by the equation:

$$\Delta m_{dol} = \dot{m}_{dol} \cdot t, \tag{18}$$

while the medium flow mass is described by the equation:

$$\Delta m_{wyl} = \dot{m}_{wyl} \cdot t, \tag{19}$$

The mass in the cylinder is calculated based on the equation:

$$m_{cyl} = m_{cyl-1} + \Delta m_{dol} - \Delta m_{wyl}, \tag{20}$$

The course of changes in the mass in the engine cylinder after taking into account the inflow and outflow medium for the intake and outlet valves is shown in Figure 6.

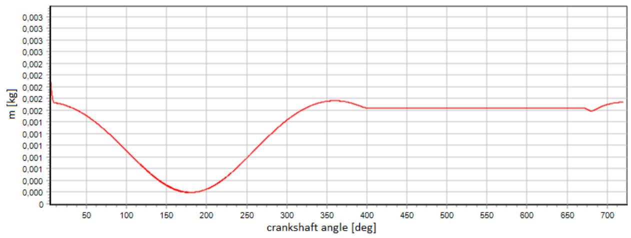

Fig. 6. Course of mass changes in the engine cylinder as a function of crankshaft revolution angle

5. Modeling the process of fuel supply into the engine cylinder

The inclusion of fuel injection into the cylinder was a further step in creating the model. The basic problem was to determine the amount of fuel injected into the cylinder during implementation of engine operating cycle. It was determined on the basis excess air ratio λ. As far as the value of coefficient λ is known it is possible to calculate the mass of air requirement L_0 for consuming 1 kg of fuel, which has been described in dependence 10. At the same time the actual air mass L_r needed to consume 1 kg of fuel, can be calculated from the dependence [5]:

$$L_r = \lambda \cdot L_0, \tag{21}$$

Hence the mass of the fuel injected into the cylinder during one cycle of the engine is calculated from the equation:

$$m_{pal} = \frac{m}{l_r}, \tag{22}$$

where:
m – air mass in the cylinder (calculated in the first iteration).

The opening angle of the injector is known, the angle of the closure remains to be calculated. In order to determine it, it is necessary to calculate the mass flow of fuel flowing from the injector. If the fuel dose attributable to the operating cycle is known, it is possible to calculate the duration of fuel injection. The duration of the fuel injection can be determined degrees of crankshaft angle for a given engine speed. Initially, it was attempted to determine the flow rate of fuel on the basis of the Bernoulli equation:

$$p_{cyl} + g \cdot \rho_{cyl} \cdot h_{cyl} + \frac{\rho_{cyl} \cdot w_{cyl}^2}{2} = p_{wtr} + g \cdot \rho_{wtr} \cdot h_{wtr} + \frac{\rho_{wtr} \cdot w_{wtr}^2}{2}, \tag{23}$$

which took the form:

$$w_{cyl} = \sqrt{\frac{2 \cdot (p_{wtr} - p_{cyl}) + \rho_{wtr}}{\rho_{cyl}}}, \tag{24}$$

The calculations are compared with the speed of sound in the fuel, calculated according to the formula:

$$w_{cylkr} = \sqrt{\frac{K}{\rho}}, \tag{25}$$

where:
K – bulk modulus for the fuel amounting to $2{,}2 \cdot 10^9$ Pa [5, 6]

Calculations have shown that the duration of the fuel injection flow rate is significantly higher than the speed of sound, and therefore assumed that the fuel outflow injector takes place at a speed equal to the speed of sound w_{cylkr}. Additionally, a series of simulations in which the influence of injection pressure on the speed of sound in the fuel was determined. Studies have shown that this effect is small - the value of the speed of sound is in the range of 1290 m/s with a pressure of 20 MPa to 1350 m/s to 90 MPa. Therefore, it was reasonable to adopt an average fuel flow rate, equal to the average sound velocity of 1320 m/s. The flow of fuel was calculated on the basis of the speed of fuel injection into the cylinder:

$$\dot{m}_{pal} = w_{cylkr} \cdot S_{wtr} \cdot \rho_{pal} \cdot \mu, \tag{26}$$

Mass gain in the cylinder caused by the fuel injection is calculated according to the formula:

$$\Delta m_{pal} = \dot{m}_{pal} \cdot t, \tag{27}$$

In contrast, mass in the cylinder:

$$m_{cyl} = m_{cyl-1} + \Delta m_{pal}, \tag{28}$$

Aside from the mass of fuel supplied to the cylinder, it is necessary to determine the amount of heat supplied with the fuel. This value is increased by the heat medium contained in the cylinder which can be calculated from the formula:

$$Q_{wtr} = m_{pal} \cdot \left(W_{pal} + c_{pal} \cdot T_{pal}\right) + m_{cyl} \cdot c_{pcyl} \cdot T_{cyl}, \tag{29}$$

where the calorific value is calculated from the dependence [9]:

$$W_{dol} = 340 \cdot c + 1017 \cdot h + 63 \cdot n + 191 \cdot s - 106 - 25 \cdot w, \tag{30}$$

where *c*, *h*, *n*, *s* – are the mass shares of the individual elements contained in the composition of the fuel, and determines the moisture content of the fuel.

At the same time, it is necessary to consider the required amount of heat supplied to the fuel. This value is described in the dependence:

$$Q_{par} = Q_w + m_{pal} \cdot \left[L + c_p \cdot \left(T_{cyl} - T_{pal}\right)\right], \tag{31}$$

where:

Q_w – heat used to warm the fuel to the evaporation temperature,

m – mass of injected fuel,

L –vaporization heat of fuel,

c_p – specific heat of fuel vapors,

T_{cyl} – medium temperature in the engine cylinder,

T_{pal} – temperature of fuel.

Besides the heat supplied to the medium with fuel and consumed for its evaporation, it is necessary to take into account the heat transferred by the medium inside the cylinder to the engine cooling system $Q_{chł}$. The engine cylinder is treated as the three surfaces penetrated by heat: the surface of the cylinder liner, piston and cylinder head. (Fig. 7a).
The heat penetrating through the surface of the cylinder liner was calculated according to the formula:

$$Q_T = 2 \cdot \pi \cdot \lambda \cdot (x + x_{ks}) \cdot \frac{T_{cyl} - T_{chł}}{ln\frac{0,5 \cdot D + x_{cyl}}{0,5 \cdot D}}, \tag{32}$$

Equation 32 is the sum of the height of the combustion chamber and piston travel. In each iteration, piston travel in the cylinder is changed according to dependence 1. Indications used in dependence 32 are shown in Figure 7 b.

The heat penetrating through the piston head and the cylinder head is calculated according to the formula:

$$Q = \frac{\lambda}{\delta} \cdot \frac{\pi \cdot D^2}{4} \cdot \left(T_{cyl} - T_{chł1}\right), \tag{33}$$

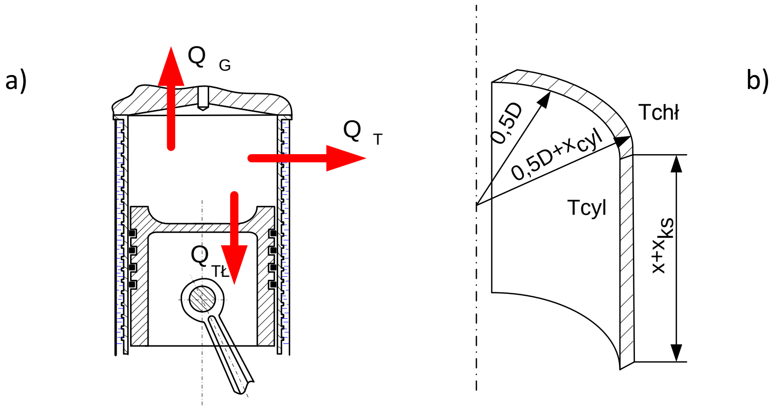

Fig. 7. a) Heat transfer through the walls of the cylinder marine engine, b) signs used in dependence 32

Where markings were adopted as shown in Figure 8.

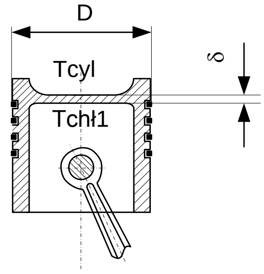

Fig. 8. Figure for calculating the amount of heat penetrating the piston head

In order to calculate the heat flow given to the cylinder walls, it is necessary to sum up the heat flows transmitted to the walls of the cylinder liner, piston and cylinder head. This-could be written using the equation:

$$Q_{chł} = Q_T + Q_G + Q_{TŁ}, \tag{34}$$

Having the heat input value of the fuel, the heat collected at its heating and evaporation and heat transferred to the walls of the cylinder, it is possible to calculate the temperature of the medium in the cylinder from the dependence:

$$T_{cyl} = \frac{Q_{wtr} - Q_{par} - Q_{chł}}{m_{cyl} \cdot c_p}, \tag{35}$$

The specific heat of the thermodynamic medium in the cylinder calculated from the dependence:

$$c_p = \frac{\kappa_{cyl}}{\kappa_{cyl} - 1} \cdot R_{cyl}, \tag{36}$$

The pressure of the working medium is calculated from the equation 13. The course of cylinder pressure variation as a function of crankshaft angle is shown in Figure 9.

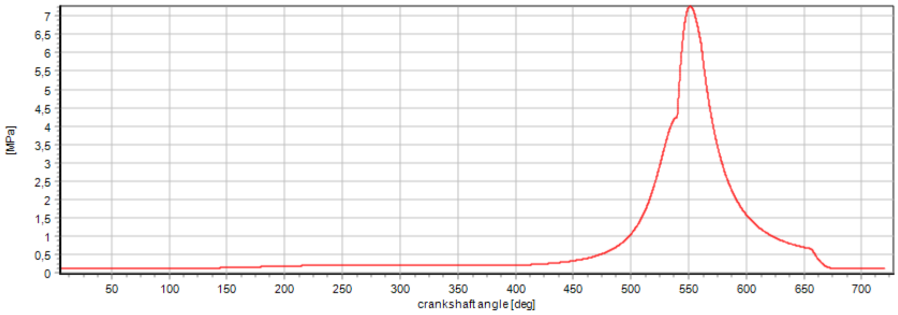

Fig. 9. The pressure profile within the cylinder as a function of crankshaft revolution angle

6. Summary

The mathematical model of physical processes in the cylinders of marine diesel engine described in the article is a fragment of the constructed model of the marine engine. The authors plan to develop this model by enriching it with a gas-dynamic phenomena occurring in the air intake and exhaust duct. In addition, it is planned to take into account the inertia forces of mases carrying out rotational, reciprocating and complex movement. In principle, it will allow to modeling fluctuation of the angular velocity during engine operation cycle. This will determine the impact of a possible change to simulate the technical condition of the engine on measurable waveforms of diagnostic parameters.

The results obtained from the model at the stage of its creation are verified with tests conducted on real objects. The great versatility of the model can allow it to be used for the study of the phenomena did not into consideration at the stage of construction, such as analysis of the impact of changes in engine load on the emission of toxic compounds, etc.

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