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Engine Working Cycle Analysis for Diagnostic and Optimisation Purposes

Original scientific paper

In this paper the most relevant engine characteristics have been determined and the analysis of diesel engine working process through the elaboration of the indicated pressure diagrams has been done[1]. Relevant characteristics have been calculated and an analysis of the obtained indicated pressure diagrams during engine exploitation has been performed. The obtained results provide the possibility for evaluation of the engine working cycle efficiency and further engine performance optimisation. Today's development of electronically controlled engines and diagnostic equipment enables the control and precise setting of relevant parameters that are important in engine process optimisation such as fuel injection, air inlet and exhaust, lubrication, cooling, etc.

Keywords: *diagnostic, engine working cycle analysis, expert system, marine diesel engine, optimisation.*

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Analiza radnog ciklusa dizelskog motora u svrhu dijagnosticiranja i optimiziranja rada motora

Izvorni znanstveni rad

U radu se određuju relevantne značajke motora, te uz obradu indiciranih dijagrama analizira radni proces dizelskog motora. Proračunavaju se bitne značajke radnog procesa, analiziraju dobiveni indicirani dijagrami na motorima tijekom eksploatacije. Iz dobivenih rezultata izvode se zaključci i ocjenjuje se pravilnost odvijanja radnog procesa, te se ukazuje na mogućnost optimiziranja rada motora. Današnji stupanj razvoja suvremenih elektronički upravljanih motora i dijagnostičkih sustava omogućava on-line kontrolu bitnih procesa motora poput ubrizgavanja goriva, ispuha, usisa, podmazivanja, hlađenja i sl.

Ključne riječi: *analiza radnog ciklusa motora, brodski dizelski motor, dijagnostika, ekspertni sustav, optimizacija.*

1 Introduction

The most effective way of engine diagnosing and optimisation is an engine working cycle analysis [1], [2], [3]. In an internal combustion engine, cylinder pressure is the parameter having a crucial impact on engine performance. Cylinder pressure dynamics has been explored and correlated with other important parameters. On the basis of the measured cylinder pressure and recorded data, an engine working condition analysis has been made. A method based on experimental results has been used.

2 Analysis of the diesel engine cylinder pressure dynamics

A cylinder pressure diagram analysis (p-α diagram) is of crucial importance in investigating the diesel engine working cycle and diagnosing the behaviour of engine performance [2], [4], [5].

2.1 Technical data on the marine diesel engine *MAN B&W* 6S70MC

Two-stroke, in-line, low speed, main propulsion diesel engine with direct injection, *MAN B&W*, type 6S70MC belongs to the high efficiency engines with thermal efficiency higher than 50% and specific fuel oil consumption below 160 g/kWh.

Technical data [4]:

- Effective power : 13364 kW
- Engine speed: 85 min^{-1}
- Mean effective pressure: 15.27 bar
- Number of cylinders: 6
- Stroke: 2.674 m
- Cylinder diameter: 0.700 m
- Connecting rod length: 3.066 m
- Compression ratio: 17.8
- Scavenging ports opening angle: 39.3° before BDC)
- Scavenging ports closing angle: 39.3° after BDC

1 The results presented in the paper have been derived from the scientific research project ''New Technologies in Diagnosis and Control of Marine Propulsion Systems'' supported by the Ministry of Science, Education and Sports of the Republic of Croatia.

- Exhaust valve opening angle: 119° after TDC
- Exhaust valve closing angle: 249° after TDC.

2.2 Calculation of combustion related parameters

Crank angle calculation of fuel injection start was calculated for the engine *MAN B&W* type 6S70MC at 100%, 85%, 75%, and 50% load.

Input parameters are:

ε [-] - compression ratio;
χ [-] - polytrophic exponent;
p_2 [bar] - compression pressure;
p_{max} [bar] - maximum cylinder pressure;
$dp/d\alpha$ [bar/°] - cylinder pressure rate;
αp_{max} [°] - crank angle at max. cylinder pressure p_{max};
t_k [s] - combustion time delay (defined value for *MAN B&W - MC* series engines, according to experiments done with fuel H_d = 42 446 [kJ/kg])
n [1/min] - engine speed;
H_d [KJ/kg] - fuel heat value.

Calculated values:

1) Crank angle at the start of combustion,
2) Crank angle at the start of fuel injection.

2.2.1 Calculation of the crank angle at the start of combustion (α_{comb})

The combustion starts when cylinder pressure rises intensively.

The duration of combustion (α_1 in Figure 2) is calculated using the following equation [6]:

$$\alpha_1 = \frac{p_{max} - p_2}{dp / d\alpha}. \tag{1}$$

The calculated crank angle at the start of combustion (α_2 in Figure 2) is obtained analytically:

$$\alpha_2 = \alpha p_{max} - \alpha_1. \tag{2}$$

Actual values of: crank angle at the start of combustion (α_{comb}), compression pressure (p_2), maximum cylinder pressure (p_{max}), cylinder pressure rate ($dp/d\alpha$), are taken from experimental values obtained in [1] and [7]. The start of combustion is shown as example in Figure 2 at the intersection point of two straight lines drawn with two points (A_1 (αp_{max}, p_{max}) and A_2(α_2, p_2)) and compression pressure line.

The compression pressure line has been defined using the following method:

Pressure at compression end (p_2) can be expressed as:

$$p_2 = p_1 \cdot \varepsilon^{\chi}, \tag{3}$$

Scavenge air pressure (p_1) can be calculated as:

$$p_1 = \frac{p_2}{\varepsilon^{\chi}}, \tag{4}$$

where the compression pressure (ε) is

$$\varepsilon = \frac{V_k + V_s}{V_k}, \tag{5}$$

and

$$V_k = \frac{V_s}{\varepsilon - 1}, \tag{6}$$

where:
V_s [m^3] - cylinder stroke volume,
V_k [m^3] - cylinder compression volume.

Compression ratio at x point (ε_x), has been calculated as follows:

$$\varepsilon_x = \frac{V_k + V_s}{V_k + V_x}, \tag{7}$$

where
V_x [m^3] - cylinder volume with piston in x position – instant volume (Figure 1) is expressed as follows:

$$V_x = \frac{\pi D^2}{4}(l + r - (r\cos\alpha + \sqrt{l^2 - r^2 \cdot \sin^2\alpha}), \tag{8}$$

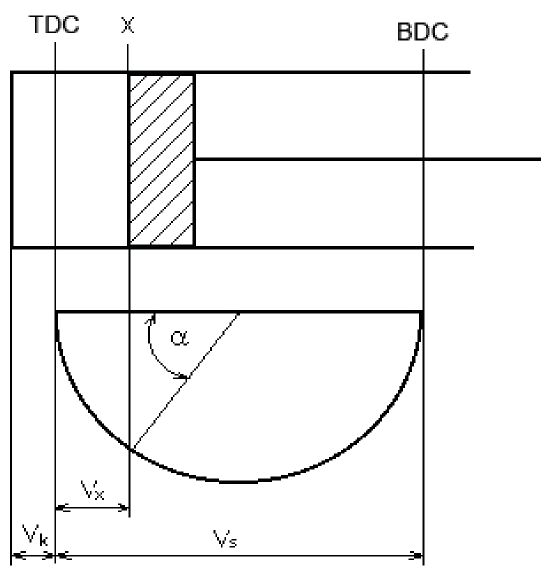
where
l [m] - connecting rod length
r [m] - crankshaft radius.

Now we can express the instant compression ratio relative to the crank angle (ε_x)

$$\varepsilon_x = \frac{\varepsilon}{1 + \frac{1}{2}(\varepsilon - 1)\cdot(r/l + 1 - \cos\alpha - \sqrt{(r/l)^2 \cdot \sin^2\alpha})}, \tag{9}$$

where
α[°] – crank angle.

Figure 1 **Crank angle in correlation with the piston position** [6]
Slika 1 **Kut zakreta koljenastog vratila u korelaciji s položajem klipa** [6]



The equation (9) has been incorporated in the following expression:

$$p_{2x} = p_1 \cdot \varepsilon_x^{\chi} \tag{10}$$

whereas the compression pressure relative to the crank angle (p_{2x}) has been expressed as:

$$p_{2x} = p_1 \cdot \frac{\varepsilon}{1+\frac{1}{2}(\varepsilon-1)\cdot(r+1-\cos\alpha-\sqrt{r^2\cdot\sin^2\alpha})}, \tag{11}$$

according to which the compression pressure can be calculated and p-α graph can be drawn as shown in Figure 2.

A straight line should be drawn through the two points according to the expression:

$$y - y_1 = \frac{y_2 - y_1}{x_2 - x_1}(x - x_1) \tag{12}$$

where the actual points are A_1 (αp_{max}, p_{max}) and $A_2(\alpha_2, p_2)$, so the expression should be:

$$y = \frac{p_2 - p_{max}}{\alpha_2 - \alpha p_{max}}(\alpha - \alpha p_{max}) + p_{max}. \tag{13}$$

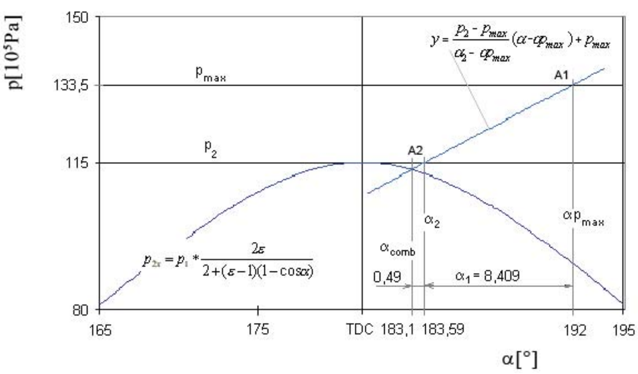


Figure 2 **Compression pressure curve in correlation with the crank angle and straight line construction at 100% load**

Slika 2 **Krivulja tlaka kompresije u korelaciji s kutom zakreta koljenastog vratila i pravca koji prolazi kroz točke konačnog tlaka kompresije i maksimalnog tlaka izgaranja pri 100% opterećenju**

2.2.2 *Determination of the crank angle at the beginning of fuel injection*

The coefficient ε has been determined to be 17.8 and the coefficient polytrophic exponent χ has been determined to be 1.38.

The crank angle at the pre-combustion phase (α_3) is calculated as follows:

$$\alpha_3 = 6 \cdot t_k \cdot n \tag{14}$$

where combustion time delay (t_k) is taken from experimental values obtained in [1] (for MAN B&W - MC series engines, according to experiments done with fuel *Hd* =42 446 [kJ/kg]).

The crank angle at the beginning of fuel injection in the cylinder (α_{inj}) is

$$\alpha_{inj} = \alpha_{comb} - \alpha_3 \tag{15}$$

Table 1 **Marine diesel engine *MAN B&W* type 6S70MC - Input and output values for calculation**

Tablica 1 **Brodski dizelski motor *MAN B&W* tip 6S70MC – Ulazne i izlazne vrijednosti proračuna**

INPUT VALUES: Engine load					
Variable	**Units**	**100%**	**85%**	**75%**	**50%**
p_2	[bar]	115	102	95	70
p_{max}	[bar]	133.5	132.5	126	95
dp/dα	[bar/°]	2.2	3.55	3.55	4.1
αp_{max}	[°]	192	192	192	192
t_k	[s]	0.01	0.01	0.01	0.01
n	[rpm]	85	80.5	77.2	67.1
CALCULATED VALUES:					
Values	**Units**	**100% load**	**85% load**	**75% load**	**50% load**
α_1	[°]	8.4090909	8.5915493	8.732394	6.097561
α_2	[°]	183.59091	183.408451	183.2676	185.90244
p_1	[bar]	2.1633074	1.91875961	1.78708	1.3167958
α_{comb}	[°]	182.1	182.0	182.2	183.9
α_3	[°]	5.1	4.83	4.632	4.026
α_{inj}	[°]	176.8	177.7	177.6	179.9

The analysis gives crank angle values at the start of fuel injection (α_{inj}) of about 177° after BDC, with differences of 0.9° for 100%, 85% and 75% loads. Crank angle values at the start of combustion (α_{comb}) are about 182° after BDC with difference of 0.2° for 100%, 85% and 75% loads. For 50% load there is a delay of about 2° for the beginning of injection and combustion.

3 Experimental research of working cycle of the marine diesel engine *MAN B&W* 6S70MC

Experimental research of the working cycle was performed on a two stroke, low speed, marine diesel engine with direct injection, featuring characteristics described in Part 1 of this paper. Measuring instruments for indicating cylinder pressure were added to ordinary measurement equipment on the main marine engine. Pressure in the cylinder was indicated on all cylinders in correlation with the engine crank angle for different revolutions and loads. Pressure transducers *Kistler 7061* were installed on cylinders and a crank angle marker *Leine Linde* was installed at the crankshaft to measure the crank angle in correlation with cylinder pressure. During the thermodynamic analysis, 50 cycles were measured using an analogue-digital measuring system and the mean value was taken into consideration. Self developed software for analysing the diesel engine thermodynamic cycle has been developed and used [1]. During analyse it was confirmed that the angle difference between cylinders is equal to the theoretical crank shaft angle with tolerance of +/- 0.06 ºCA. Because of various crank shaft twisting stresses on the journals, real angle tolerance is greater. Regarding revolution angle, the referential flywheel angle is obtained by means of integration of the revolution speed, after running the harmonic analyses and syntheses. It was assumed that flywheel rotates at constant speed. Indicated cylinder pressures are shown on p - α diagrams in Figures 3 to 7.

The maximum combustion pressure depends on compression ratio, air-fuel ratio, ambient conditions, and heat value of the fuel. But, in this case, an adjustment injection timing system (Variable Injection Timing VIT) was fitted to the engine, enabling us to influence the maximum combustion pressure [8], [9]. A higher maximum combustion pressure results in a better fuel consumption. Because of fuel preparation after the beginning of injection, the cylinder pressure slightly drops (lost heat for fuel evaporation), but the pressure increases rapidly when a large amount of fuel is burnt. In order to decrease the maximum combustion pressure in this engine, the injection timing had to be delayed. Maximum combustion pressures were limited by the mechanical properties of cylinder materials [10].

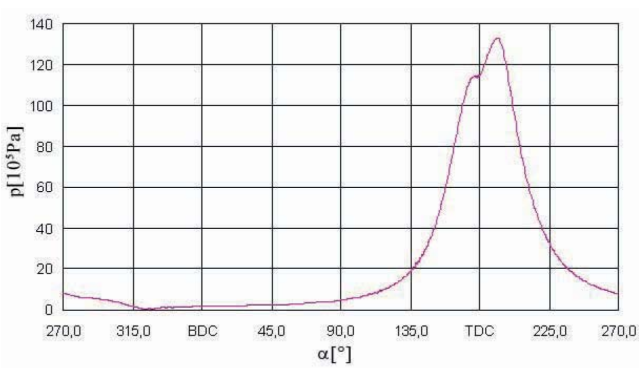
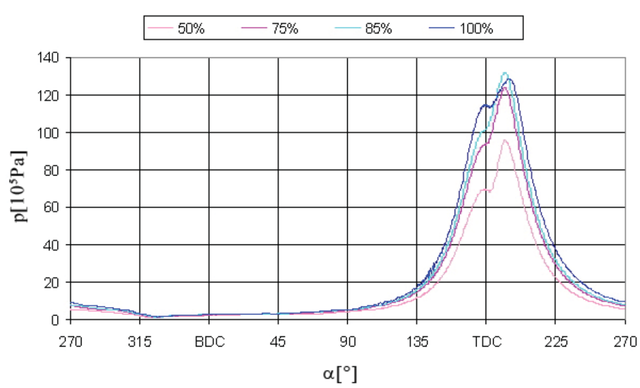


Figure 3 **Cylinder pressure in the sixth cylinder at 100% load**
Slika 3 **Tlak u šestom cilindru pri 100% opterećenju**

Figures 4, 5, and 6 show cylinder pressure curves in the cylinders under different loads.

Figure 4 **Cylinder pressure in the first cylinder at different loads**
Slika 4 **Tlak u prvom cilindru pri različitim opterećenjima**



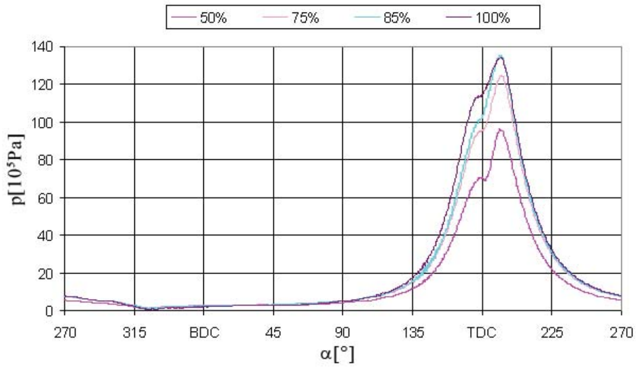


Figure 5 **Cylinder pressure in the third cylinder at different loads**
Slika 5 **Tlak u trećem cilindru pri različitim opterećenjima**

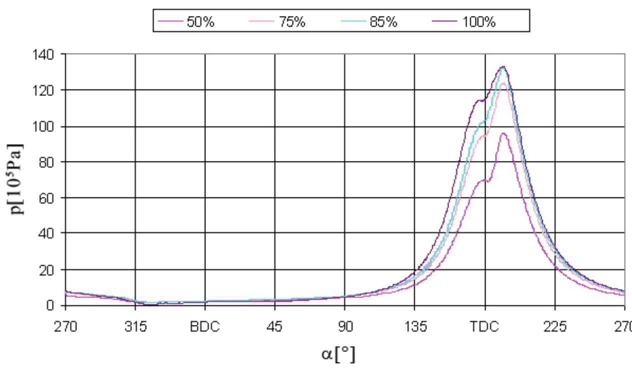


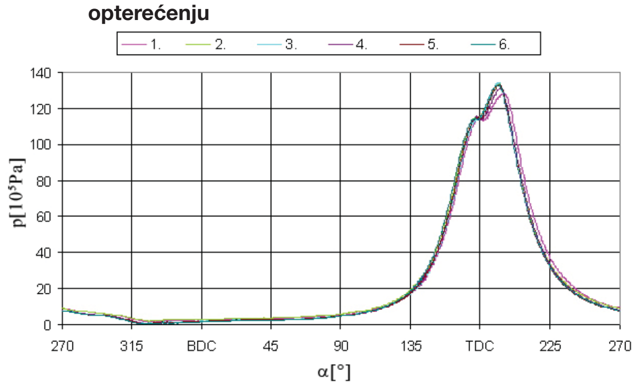
Figure 6 **Cylinder pressure in sixth cylinder at different loads**
Slika 6 **Tlak u šestom cilindru pri različitim opterećenjima**

In the first cylinder the maximum combustion pressure at 85% of the engine load was greater than at 100% load as shown in Figure 6. Likewise, the crank angle, when the maximum pressure had been reached, occurred earlier at 85% load than in the case of 100% load. Maximum pressure and the angle, at which the maximal pressure was achieved, were almost the same in the third and sixth cylinders at 85% and 100% loads.

3.1 Analysis of the results obtained at 100% load

The indicating cylinder pressure curves have been analysed and results have been compared with the conclusion achieved in

Figure 7 **Cylinder pressure curves in all cylinders at 100% load**
Slika 7 **Krivulje tlaka u cilindru za sve cilindre pri 100% opterećenju**



the previous research to see if the assumptions about the deviation of parameters are correct.

Figure 7 clearly shows that the cylinder pressure curve in cylinder 1 is not overlapping with the other cylinder pressure curves, starting from the beginning of injection until exhaust valve opening. In the following figures we can see the deviation value at characteristic points.

3.1.1 Injection and combustion segment

Cylinder pressures range between 133 and 134 bar as shown in Figure 8. The pressure in cylinder 4 amounts to about 131 bar, whereas an even greater deviation occurs in the first cylinder where the maximum cylinder pressure reaches 128.5 bar. The maximum pressure in most of the cylinders is obtained at 191° after BDC except in the fourth cylinder (193°) and the first cylinder (195°).

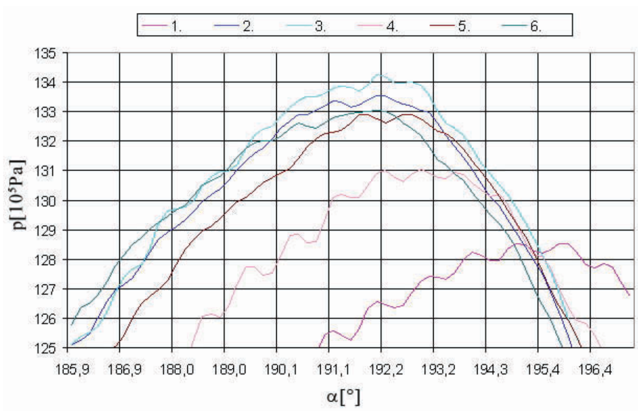


Figure 8 **Cylinder pressure curves, combustion segments, in all cylinders at 100% load**
Slika 8 **Krivulje tlaka u cilindru, za vrijeme izgaranja, za sve cilindre pri 100% opterećenju**

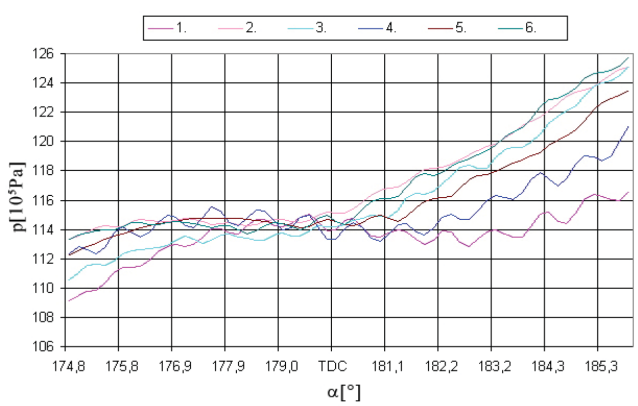


Figure 9 **Cylinder pressure curves, injection segments, in all cylinders at 100% load**
Slika 9 **Krivulje tlaka u cilindru, za vrijeme ubrizgavanja, za sve cilindre pri 100% opterećenju**

In Figure 9 a start of combustion angle deviates in ° from one cylinder to the other but the pressure amounts to 113-114 bar for all the cylinders. In order to determine exactly the beginning of combustion, and the influence on the maximum cylinder pressure, the curves in Figures 10 to 15 have been analysed and conclusions on the engine performance have been drawn.

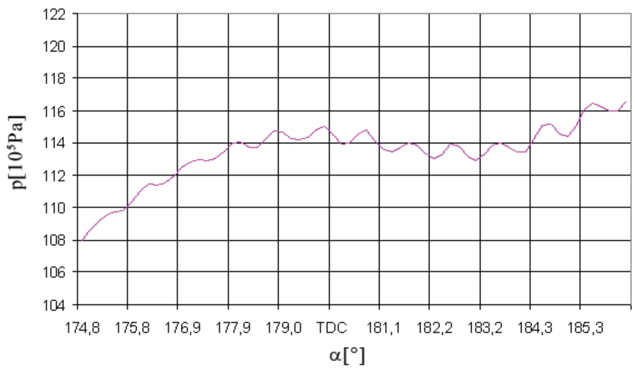


Figure 10 **Cylinder pressure curve segments in the first cylinder at 100% load**
Slika 10 **Segmenti krivulje tlaka u prvom cilindru pri 100% opterećenju**

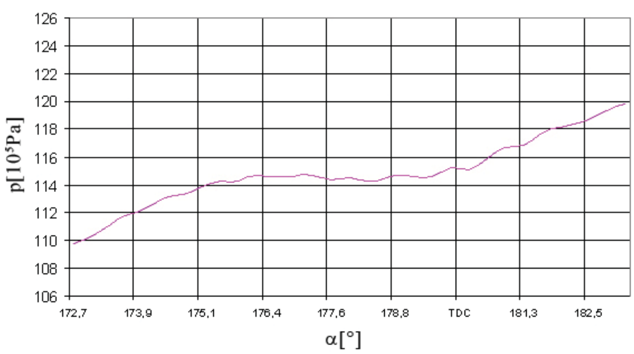
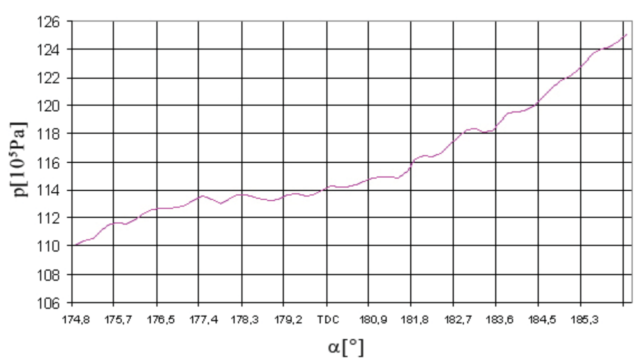


Figure 11 **Cylinder pressure curve segments in the second cylinder at 100% load**
Slika 11 **Segmenti krivulje tlaka u drugom cilindru pri 100% opterećenju**

Figure 12 **Cylinder pressure curve segments in the third cylinder at 100% load**
Slika 12 **Segmenti krivulje tlaka u trećem cilindru pri 100% opterećenju**



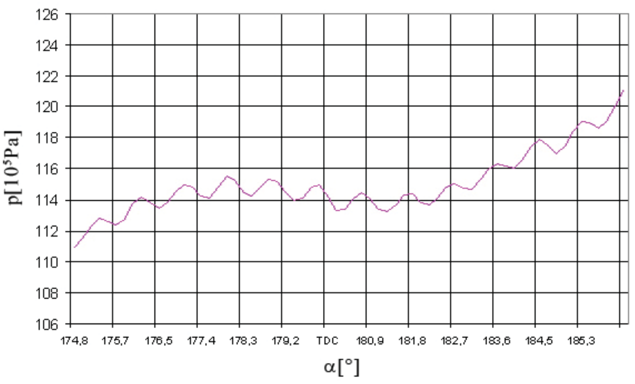


Figure 13 **Cylinder pressure curve segments in the fourth cylinder at 100% load**
Slika 13 **Segmenti krivulje tlaka u četvrtom cilindru pri 100% opterećenju**

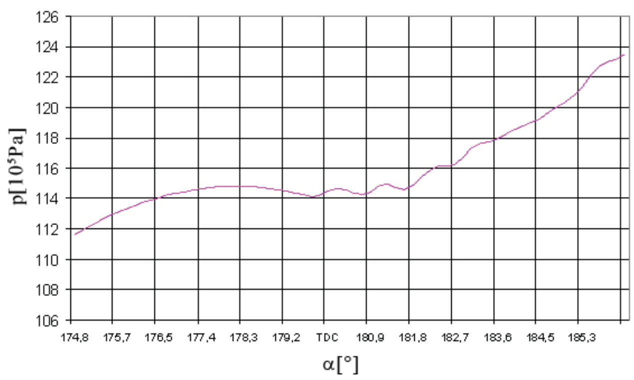


Figure 14 **Cylinder pressure curve segments in the fifth cylinder at 100% load**
Slika 14 **Segmenti krivulje tlaka u petom cilindru pri 100% opterećenju**

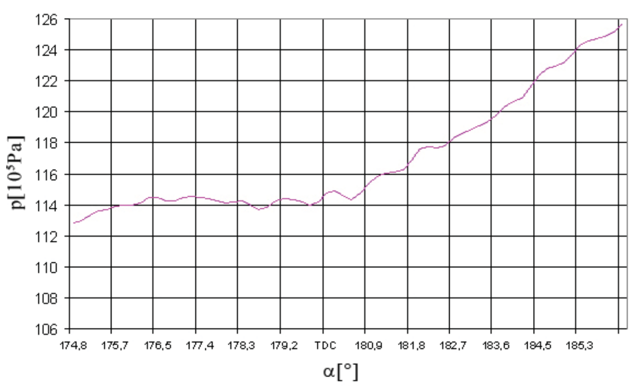


Figure 15 **Cylinder pressure curve segments in the sixth cylinder at 100% load**
Slika 15 **Segmenti krivulje tlaka u šestom cilindru pri 100% opterećenju**

Crank angles at the combustion start and end as well as relevant cylinder pressures are important in analysing the combustion process. The values of the measured combustion beginning angles are the following:

1st cylinder - 183° ; 2nd cylinder - 178.3° ; 3rd cylinder - 179°; 4th cylinder - 181°; 5th cylinder - 180°; 6th cylinder - 178.7°.

When comparing the above results with the cylinder pressure curves shown in Figure 8 , it appears that the maximum combustion pressure in the third cylinder is at 192.2°, whereas the combustion starts at 179°. The maximum pressure in the fifth cylinder is achieved later and it has a lower value. Also, the start of combustion period is achieved later. In the fourth cylinder, the combustion process begins last; hence the maximum combustion pressure is the lowest. The combustion in the first cylinder starts 4° earlier than in the third cylinder, so that the maximum combustion pressure is 6 bar lower in the first cylinder and 3° later, compared with the third cylinder. The indicating cylinder pressure curve is lower in the first cylinder than in the other, starting from the beginning of injection, as shown in Figure 7.

The analysis shows the regularity regarding the positions and values of the maximum combustion pressures being closely related to the combustion beginnings. The injection pressure diagrams do not follow the developed combustion pressure diagrams for all the cylinders. When comparing the maximum combustion pressures in the same cylinder under different loads, we cannot find any regularity; the values differ from load to load and, in one cylinder, the maximum combustion pressure is higher under a specific load, compared to the results obtained in the other cylinder. Likewise, when the maximum pressure has been reached in a cylinder, the crank angle differs as loads change. Wave dynamics in the intake and exhaust manifold influences in-cylinder thermodynamic parameters and thus consequently influences combustion parameters. An earlier combustion crank angle gives an earlier crank angle at which the maximum pressure is reached. The crank angle beginning is very important as this angle makes it possible to regulate the maximum pressure as well as the indicating pressures, thus influencing the thermal and mechanical loads and the overall efficiency of the engine working process.

The combustion characteristics and combustion period depend on the physical and chemical fuel properties, compression ratio, scavenging pressure, cooling process, resistance in inlet and outlet air-exhaust flow, piston speed, cylinder space volume and shape, turbulence in cylinder space, exhaust space, fuel atomisation, fuel injection crank angle beginning and duration, and other parameters.

Intelligent engines include systems to control fuel injection timing, exhaust valve opening, cooling and lubrication capabilities. An expert system installed on intelligent engines will enable us to change the actions of fuel injection, fuel injection duration, valve timings, real-time cooling and lubrication intensity under different loads for each cylinder, in order to reach the optimum performance.

In this particular case, the maximum pressure on the fourth cylinder should be raised by 6 bar to be on a level with other cylinders. An intelligent system should be able to control the injection beginning and give an earlier crank angle by about 2.5°. The intelligent system should also control other relevant parameters and optimise all values that deviate from the others.

3.1.2 Opening of the exhaust valve and scavenging channel segment

The period between opening the exhaust valve (which should be 299° after BDC, according to the manufacturer) and opening scavenging ports (320.7° after BDC) has been analysed to see the behaviour of cylinder pressure (see Figure 16).

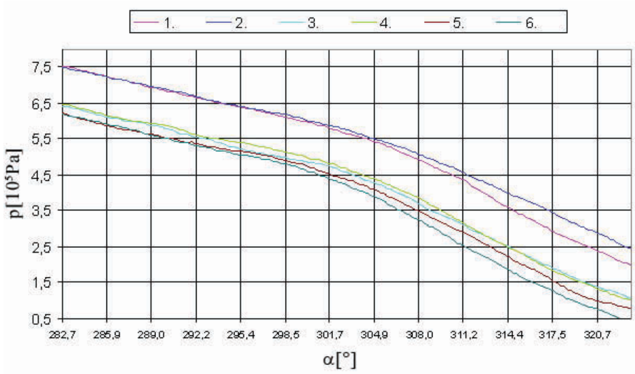


Figure 16 **Cylinder pressure curve at exhaust valve opening segments on all cylinders at 100% load**
Slika 16 **Krivulja tlaka u cilindru za vrijeme otvaranja ispušnih ventila, za sve cilindre pri 100% opterećenju**

Cylinder pressure curves for all cylinders drop at 299° after BDC. In the first and second cylinders, cylinder pressures are 1 bar higher than in other cylinders. In the first, fifth and sixth cylinders, pressure curves drop faster than in others.

Theoretically, it is important that exhaust valve opens on time and cylinder pressure should fall below scavenging pressure in order not to allow exhaust gas enter in the cylinder through the scavenging port. The quality of scavenging, fuel-air ratio and cylinder temperature depend on exhaust timing. Too early exhaust opening causes reduction of efficiency during expansion and loss of power necessary for exhaust valve open due to higher pressure in the cylinder.

For the observed engine, scavenging pressure is 2.1 bar (gauge pressure) at 100% load. It is important to reach cylinder pressure of 2.1 bar (gauge pressure) at 320.7^o after BDC. According to diagrams in Figure 16, in the first and second cylinder we should correct the exhaust opening timing to an earlier time, and in other cylinders, the exhaust opening time should be later to achieve 3.1 bar (absolute pressure). Although the engine features constant pressure turbocharging, it is necessary to consider that mean intake or exhaust manifold pressure does not represent adequately instant manifold pressure at valve opening or closing. Fluctuations around the mean value might thus influence mass, momentum and enthalpy transfer

Cylinder pressures (gauge pressure) at 320.7° after BDC are measured as follows:

1st cylinder – 2.3 bar; 2nd cylinder - 2.8 bar; 3rd cylinder – 1.2 bar; 4th cylinder – 1.2 bar; 5th cylinder – 1.0 bar; 6th cylinder – 0.7 bar

3.1.3 Closing of the exhaust valve and scavenging channel segment

Figure 17 shows measured cylinder pressure in all cylinders at 100% load in the period between scavenging port closing at 39.3° after BDC and exhaust valve closing at 69° after BDC.

The cylinder pressure values at 39.3° are 2.0 bar (+/- 0.2 bar) in all cylinders, while at 53° the cylinder pressure rises faster as a result of the beginning of exhaust valve closing.

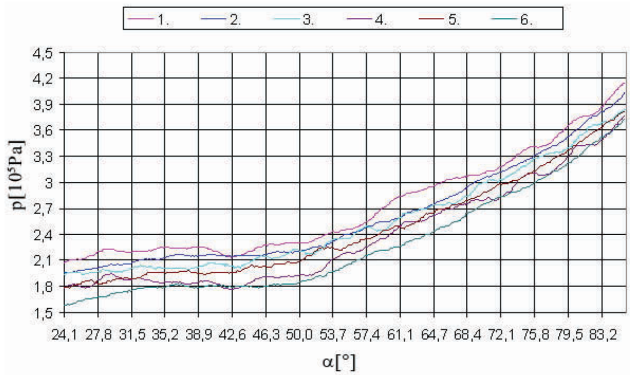


Figure 17 **Cylinder pressure curve at scavenging port closing segments on all cylinders at 100% load**
Slika 17 **Krivulje tlaka u cilindru za vrijeme zatvaranja usisnih kanala, za sve cilindre, pri 100% opterećenju**

The above analysis does not show any deviation of cylinder pressure due to irregularity of the exhaust valve closing, so it is not necessary to implement any corrective procedure or action.

3.2 Analysis of the results obtained at 85% load

The observed engine has been designed to provide optimal performance at 85% load. To reach maximum efficiency, the engine is adjusted to produce maximum cylinder pressure (130 bar). Figure 18 shows good adjustment of cylinder pressures in all cylinders, and there are no deviations to be analysed. The variable injection time – system that has been fitted to this engine shows good characteristics.

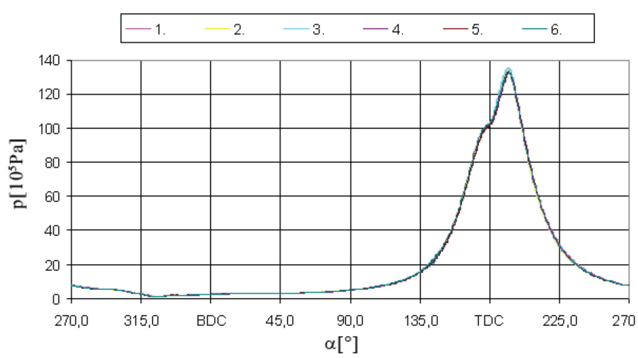


Figure 18 **Cylinder pressure in all cylinders at 85% load**
Slika 18 **Tlak u cilindru za sve cilindre pri opterećenju od 85%**

3.3 Analysis of the results obtained at 75% load

Figure 19 shows the deviation in cylinder pressure curves at 75% load.

Here we have a deviation in the fourth cylinder, starting at the beginning of injection until exhaust closing. It is important to analyse this difference and propose a corrective action.

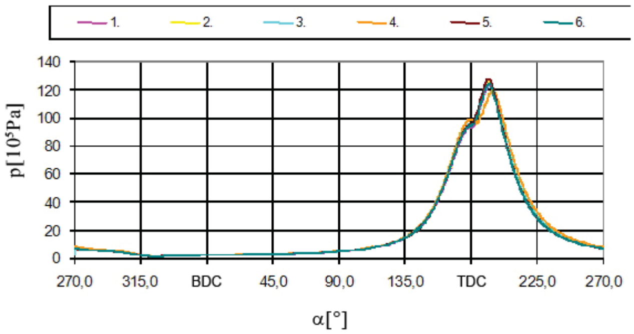


Figure 19 **Cylinder pressure in all cylinders at 75% load**
Slika 19 **Tlak u cilindru za sve cilindre pri opterećenju od 75%**

3.3.1 Injection and combustion segment at 75%

In the following figures the cylinder pressure curves have been analysed in more detail measurements at injection and combustion phases.

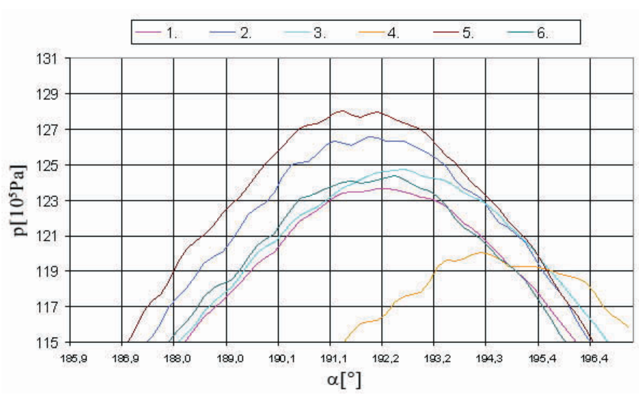


Figure 20 **Cylinder pressure curves, combustion segments, in all cylinders at 75% load**
Slika 20 **Krivulje tlaka u cilindru za vrijeme izgaranja, za sve cilindre pri opterećenju od 75%**

Figure 20 indicates maximum cylinder pressures ranging from 124 to 128 bar at 192° after BDC, except in the fourth cylinder where the maximum combustion pressure is 120 bar at 194.5° after BDC. In the fifth cylinder, the maximum pressure is greater than in others and has been reached 0.5° before than in the case of other cylinders. According to the engine manufacturer's recommendations, the maximum pressure difference should not exceed +/- 3 bar from the average. In this case the average pressure is 126 bar, therefore the cylinder pressure in the fourth cylinder is beyond the recommended limits as the difference is -6 bar.

If we compare the results at 100% and 75% load, there is not any regularity that would give us the possibility to decide what kind of a corrective action to take. For example, at a 100% load, the maximum pressure is in the third cylinder, and at a 75% load it is in the fifth cylinder. Also, at a 100% load, the greatest deviation is in the case of the first cylinder, whereas at a 75% load, it is in the case of the fourth cylinder. Moreover, cylinders at different loads have different combustion timing: at a 100% load it is in the second and sixth cylinder, while at a 75% load, it is in the fifth cylinder.

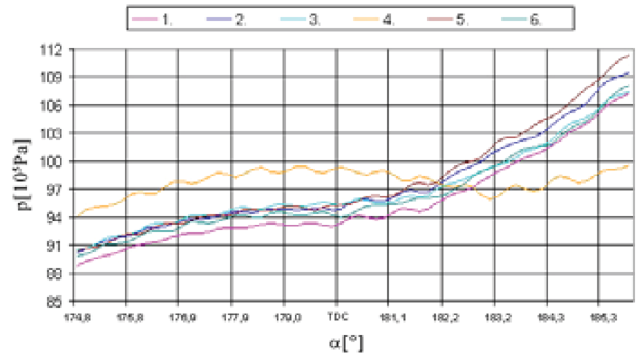


Figure 21 **Cylinder pressure curves, injection segments, in all cylinders at 75% load**
Slika 21 **Krivulje tlaka u cilindru za vrijeme ubrizgavanja, za sve cilindre pri opterećenju od 75%**

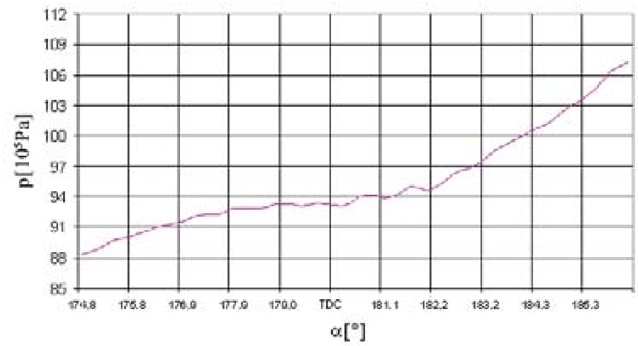


Figure 22 **Cylinder pressure curve segments in the first cylinder at 75% load**
Slika 22 **Segmenti krivulje tlaka u prvom cilindru pri opterećenju od 75%**

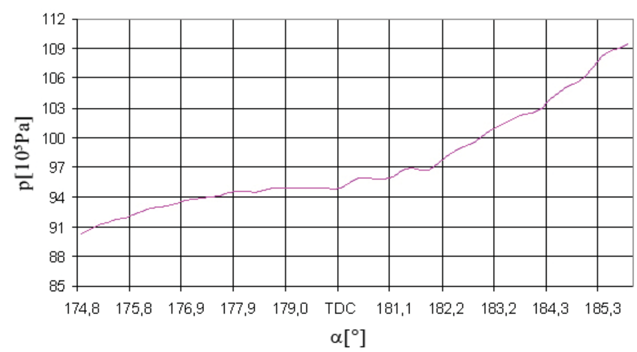


Figure 23 **Cylinder pressure curve segments in the second cylinder at 75% load**
Slika 23 **Segmenti krivulje tlaka u drugom cilindru pri opterećenju od 75%**

The values of measured combustion beginning angles are the following:

1st cylinder – 180.2° ; 2nd cylinder - 180° ; 3rd cylinder - 181° ; 4th cylinder - 183°; 5th cylinder – 179.5°; 6th cylinder – 180.5°.

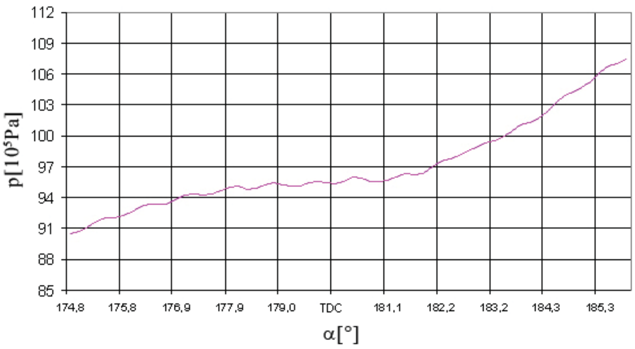


Figure 24 **Cylinder pressure curve segments in the third cylinder at 75% load**

Slika 24 **Segmenti krivulje tlaka u trećem cilindru pri opterećenju od 75%**

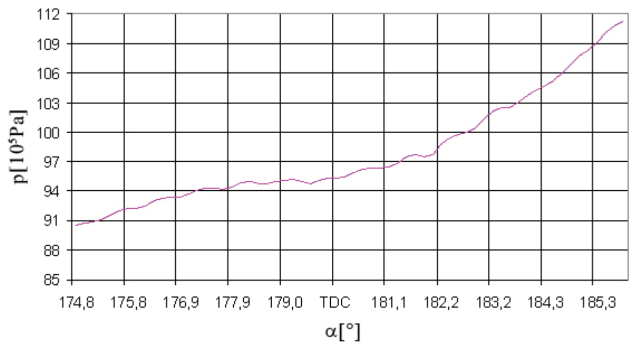


Figure 25 **Cylinder pressure curve segments in the fourth cylinder at 75% load**

Slika 25 **Segmenti krivulje tlaka u četvrtom cilindru pri opterećenju od 75%**

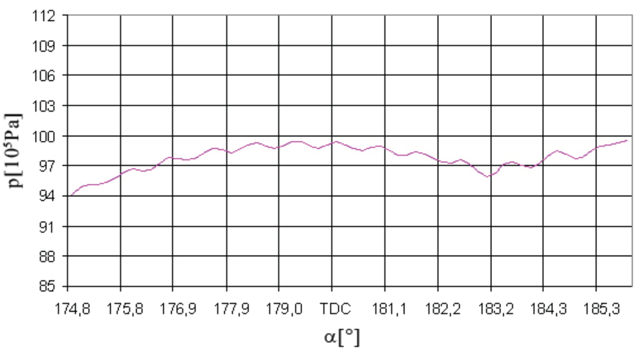


Figure 26 **Cylinder pressure curve segments in the fifth cylinder at 75% load**

Slika 26 **Segmenti krivulje tlaka u petom cilindru pri opterećenju od 75%**

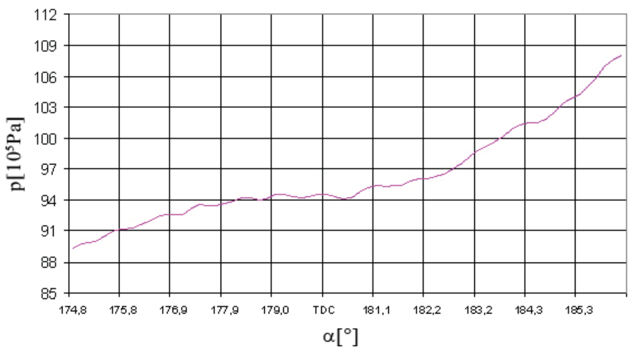


Figure 27 **Cylinder pressure curve segments in the sixth cylinder at 75% load**

Slika 27 **Segmenti krivulje tlaka u šestom cilindru pri opterećenju od 75%**

At these angles we can notice the lowest combustion pressure at the end of latent combustion and the cylinder pressure starts to rise at the start of combustion.

When analysing Figures 21 to 27, several conclusions can be made:

- the fifth cylinder has the combustion beginning at 179.5° and the maximal combustion pressure at 191.5°;
- the first, second and sixth cylinders have the combustion beginning at 180° and the maximal combustion pressure at 192°;
- the third cylinder has the combustion beginning at 181° and the maximal combustion pressure at 192.5°;
- the fourth cylinder has the combustion beginning at 183° and the maximal combustion pressure at 194.3°.

At a 75% load an irregularity appears, showing that the beginning of an earlier combustion results in a higher and earlier maximum combustion pressure. The same conclusion can be made from the previous analysis of occurrences at a 100% load. This is important as the combustion pressure value and timing have a great influence on the efficiency of the engine and thermal load. Lower engine efficiency causes lower lubrication quality (due to higher temperatures during die down period, hence lower viscosity). The combustion beginning influences the mechanical load and pressure rise.

3.3.2 Opening of the exhaust valve and scavenging ports segment

Figure 28 shows the cylinder pressure diagrams during opening of the exhaust valve (299° after BDC -factory settings) and scavenging ports (320.7° after BDC - factory settings).

The diagrams show that all cylinders have the same opening exhaust valve time as the shapes are identical. In the fourth cylinder the pressure is 0.5 bar higher than in other cylinders.

At 320.7° crank angle after BDC, the pressure developed in the cylinder should be 1.5 bar, as scavenging pressure has that value at 75%. The cylinder pressures at 320.7° after BDC are measured as follows:

1st cylinder – 1.9 bar
2nd cylinder – 1.7 bar
3rd cylinder – 1.9 bar
4th cylinder – 2.3 bar
5th cylinder – 1.5 bar
6th cylinder – 1.5 bar

To obtain pressure at 1.5 bar, the exhaust valve in the first, third and fourth cylinders should open earlier.

For the optimisation of the engine working cycle, an intelligent system capable to control electronically the exhaust valve should be implemented. This system could use analyses

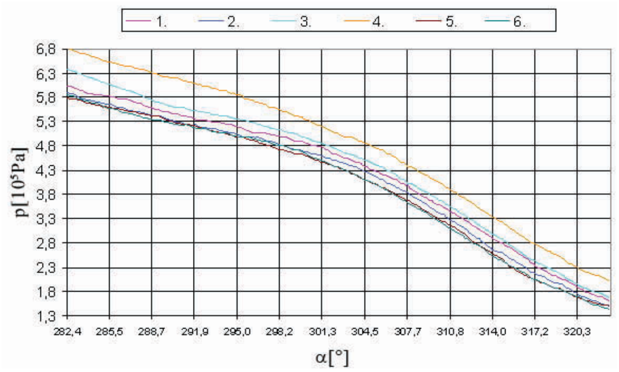


Figure 28 **Cylinder pressure curves at exhaust valve opening segments in all six cylinders at 75% load**
Slika 28 **Krivulje tlaka za vrijeme otvaranja ispušnih ventila, kod svih šest cilindara pri opterećenju od 75%**

of engine parameters (cylinder pressure) and change the timing for exhaust opening for each cylinder separately.

3.3.3 Closing of the exhaust valve and scavenging ports segment

Figure 29 presents cylinder pressure in all cylinders at 75% load during the period between scavenging port closing at 39.3° and exhaust valve closing at 69° after BDC.

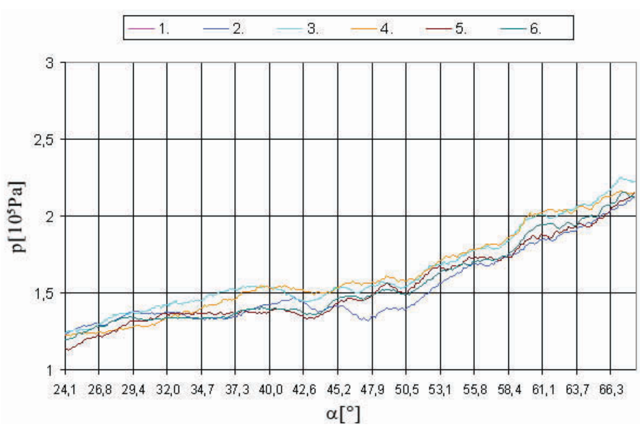


Figure 29 **Cylinder pressure curves at scavenging port closing segments in all six cylinders at 75% load**
Slika 29 **Krivulje tlaka za vrijeme zatvaranja usisnih kanala, kod svih šest cilindara pri opterećenju od 75%**

Figure 29 shows that cylinder pressures in all cylinders are equally developed and that at the scavenging ports closing time (39.3° after BDC - factory value) the value of cylinder pressures is 1.5 bar (+/- 0.15 bar). Pressure curves rise faster at 51°, due to the beginning of exhaust valve closing.

In these measurements there are no irregularities that would be important to do corrective actions and optimisation. However, the exhaust valve closing time is an important factor due to its influence on the compression ratio (ε). An intelligent system applied to the electronic engine should be able to control the injection timing, the exhaust valve opening and closing, and the cooling water capability. The variation of the following parameters in different cylinders should be possible: fuel injection law, injection beginning, compression ratio, cooling intensity.

4 Conclusion

A cylinder pressure diagram analysis with different engine loads on several cylinders has been carried out for the purpose of diesel engine working cycle optimisation. A two-stroke main propulsion engine has been analysed. During the analysis, several deviations of relevant parameters have been determined. An expert system applied to intelligent engines could make use of the deviation, solve problems and give advice for parameter correction in order to achieve optimum combustion.

The analysis has indicated certain rules in the working cycle process, e.g. an earlier beginning of injection giving a higher maximum cylinder pressure and vice versa.

To achieve the optimal combustion, rapid pressure and temperature rise should be reduced. For obtaining a "soft" and more economical engine operation, combustion must be observed and ignition timing as well as the amount of injected fuel must be controlled. An intelligent engine featuring an electronically controlled fuel injection system should be able to reduce the amount of fuel at the start of injection. An efficient operation with better combustion is achieved by proper adjusting the fuel injection system and a higher pressure in the combustion chamber.

An equable working cycle between cylinders, obtained by comparing and equalising relevant parameters such as fuel injection and exhaust valve opening and closing timing, could result in the optimal engine performance. The results of the engine working cycle analysis will be included in the future development of expert systems for engine diagnostics and optimisation.

An intelligent system [11] applied to the electronically controlled engine should be able to control injection timing, exhaust valve opening and closing, load dependent lubrication and cooling water capability. Variation of the following parameters on different engine cylinders should be possible: fuel injection law, injection beginning and timing, compression ratio, lubrication oil flow rate, cooling intensity, etc.

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