Detection of Changes in the Opening Pressure of Marine Engine Injectors Using Vibration Methods

Otkrivanje promjena u početnom tlaku ubrizgača goriva brodskog motora s pomoću vibracijskih metoda

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Summary

Injectors of marine engines are, apart from injection pumps, a key element of the fuel injection system. Their proper operation affects the quality and time of producing the fuel-air mixture in engine cylinders. The technical condition of the injectors also has an impact on engine performance and the emission of harmful compounds into the atmosphere. One of the basic injector damages occurring during their normal operation is the change of the injector opening pressure. The article presents mathematical relations indicating the occurrence of vibration diagnostic parameters that could potentially determine changes in the opening pressure of individual injectors. A two-stage passive-active experiment was carried out during which an injector working with different spring preload force was tested. The presented results of experimental tests confirm the possibility of determining whether the injector opening pressure is within the permissible range. The presented method does not require interference in the engine operation. measurements can be made during regular operation.

KEY WORDS

marine diesel engine injectors
diagnostic
vibrations

Sažetak

Ubrizgači goriva brodskih motora, uz pumpe ubrizgača, predstavljaju ključan element sustava za ubrizgavanje goriva. Njihov pravilan rad utječe na kvalitetu i vrijeme potrebno za stvaranje mješavine goriva i zraka u cilindrima motora. Tehničko stanje ubrizgača također utječe na rad motora i ispuštanje štetnih smjesa u atmosferu. Jedno je od osnovnih oštećenja ubrizgača tijekom normalnoga rada promjena u početnome tlaku ubrizgača. U ovome radu prikazani su matematički odnosi koji ukazuju na pojavu vibracijskih dijagnostičkih parametara, koji potencijalno mogu odrediti promjene u početnome tlaku pojedinih ubrizgača. Proveden je pasivno-aktivni eksperiment u dvjema fazama, tijekom kojega je testiran jedan ubrizgač koji je radio s različitim preopterećenjem opruge. Dobiveni rezultati eksperimentalnih testova potvrđuju mogućnost da se odredi je li početni tlak ubrizgača u dopuštenim okvirima. Predstavljena metoda ne traži prekid rada motora; mjerenja se mogu obaviti za vrijeme normalnog rada.

KLJUČNE RIJEČI

ubrizgači brodskog dizelskog motora
dijagnostika
vibracije

1. INTRODUCTION / *Uvod*

Considering the machinery service strategy in a function of its technical condition it is necessary to have a set of reliable diagnostic methods that determine the technical condition of all vital components important for reliability [13]. One of such methods is skilfully used vibration diagnostics. The use of vibration measurements in diagnostic systems allows determining the optimal moment of stopping the machinery to perform necessary repairs. The moment of the appearance of the defect and its propagation to the catastrophic damage can be illustrated using the so-called P-F curve - figure 1.

Analysing Figure 1, it should be assumed that the moment of defect does not eliminate the engine from further operation. However, responsible personnel have to be aware of the defect and have tools to monitor its development and assess the impact on the technical condition of the engine. The main objective of the injection installation is to prepare appropriate doses of fuel with their proper spraying in the cylinder at a certain crankshaft angle before TDC. Each of the mentioned parameters describing the operation of the injection installation is characterized by specific limit values given each time by the engine producers. As long as these values do not exceed the permissible ones, the engine is in a state of technical efficiency. After exceeding the limit values, the object changes its state to the state of technical disability, however, it is still able to be operated. It is crucial that the technical service personnel are aware of the need to quickly repair the resulting and developing defect. If an existing defect

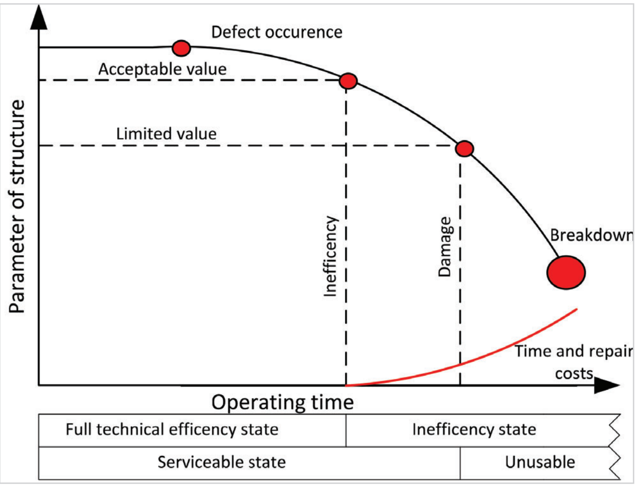

Figure 1 Machine life P-F chart
Slika 1. Trajanje stroja P-F karta

is not detected at this stage, it usually develops very quickly leading to severe engine damage.

Based on many years of diagnostic experience gathered at the Institute of Ship Design and Exploitation of Polish Naval Academy, analysis of the occurrence of operational incapacity of marine engines operated in the Navy was carried out – Figure 2 [10]. The conducted analysis showed that the most common defects occur in the following functional systems:

- engine fuel supply system - 72%,
- timing system - 19%,
- engine power supply system - 9%.

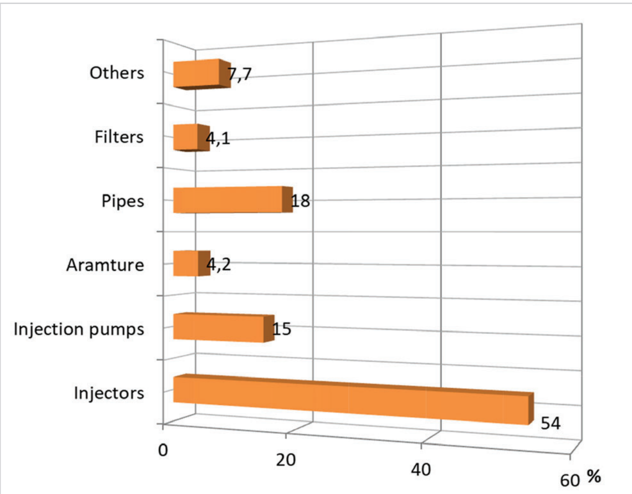

Figure 2 The percentage structure of faults of components the engine fuel installation
Slika 2. Postoci pojava grešaka na sastavnicama sustava goriva motora

Damage to injection systems is 49.68% of all damages that occurred in the operation of marine engines in Polish Navy in recent years, of which 51% was damage to the injectors [11].

Exceeding the permissible values characterizing the operation of injectors of marine diesel engines has from the diagnostic point of view two consequences, i.e. leads to an increase or decrease in combustion pressure and changes its distribution as a function of crankshaft angle (°CA). Of course, there are also many other phenomena that are unfavourable to the engine, but they are not the subject of this paper. Changes in an injectors technical condition cause not only a decrease in the quality of cylinder processes but also in the long-term. They can cause severe secondary damage to other components of the naval propulsion system (e.g. as a result of drastic lowering of the injector opening pressure poorly sprayed fuel can mix with the oil and lead to seizing of the bearings).

The exploitation of the marine engine fuel supply system takes place in marine conditions, which are considered the most difficult, and it is reflected in the high intensity of damage [1, 8]. This is due to the destructive impact and use of the engine at low and variable loads, the possibility of water in the fuel, as well as salt and seawater in the intake air, cyclical overloads, etc. The operation of the engine with injectors in a not perfect condition always leads to a decrease in the performance and efficiency of the marine propulsion system. The reasons for this condition can be varied [12]:

- bunkering of low-quality fuel (low quality of bunkered fuel, ineffective operation of centrifuges, contaminated filters, contaminated tanks, leaks of other substances to the fuel supply system, etc.),
- lack of immediate reaction of technical personnel to minor primary damage of structural elements, affecting the quality of the fuel system operation,
- repeated engine starts, frequent changes of its load and long-lasting work at low loads,
- chemical corrosion caused by the aggressive influence of the factors contained in fuel (sulphur and vanadium), especially during long engine stoppages,
- erosive effect of solid particles contained in the fuel, as well as cavitation erosion of flow channels,
- thermal effect of the heated engine on precise dosing elements.

Many scientists devote their attention to issues related to the diagnostics of diesel engine injectors [1,2,3,4,5,6,7,8,9,12,14,15,16,17]. Currently, the most effective way to assess the technical condition of injectors is their disassembly and testing on the dedicated test benches [14]. However, this means the engine has to be stopped for the duration of the test. In addition, engine manufacturers recommend testing during reviews carried out at specific intervals of time, unfortunately, the damage is very common in the inter-review period.

2. INJECTOR OPERATION MATHEMATICAL MODEL / *Matematički model rada ubrizgača*

Before the research, a thorough mathematical analysis of dynamic interactions occurring during the operation of the injector was carried out. Fuel for the injector (Figure 3 shows the injector of the Sulzer 6 AL. 20/24 engine, which was the object of tests), is supplied by high-pressure pipes in which high pressure is generated by the injection pump. Due to the small volume of the injector chamber and the use of a non-return valve to prevent a pressure drop in the line between successive piston stroke of the pump, high pressure can be obtained very quickly. The injector spring pushes on the needle. Injection starts when the force acting on the needle caused by the fuel pressure exceeds the force generated by the initial spring tension. Injection stops when the spring tension force is sufficient for the injector to close rapidly. The force of the needle's impact on the seat is directly proportional to the elastic force of the spring. Also, there is a damping force in the system which acts in opposition to the return of the needle speed vector during the injection process.

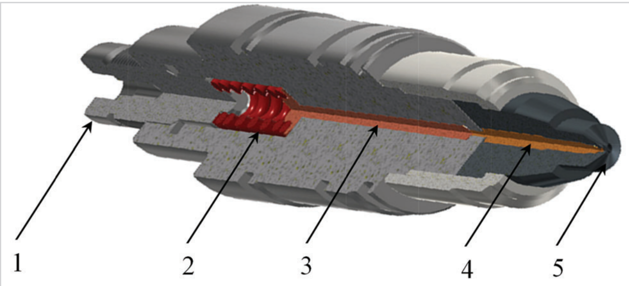

Figure 3 Sulzer 6AL 20/24 engine injector, 1 – adjusting screw, 2 – spring, 3 – rod, 4 – needle, 5 – nozzle

Slika 3. Sulzer 6AL 20/24 ubrizgač goriva, 1 – vijak za reguliranje, 2 – opruga, 3 – motka, 4 – igla, 5 – sapnica

In the physical model, it was assumed that the system had one degree of freedom, connected with the direction of the injector needle movement. The flow at the injector's nozzle is laminar. The spire and the spindle were included as one mass - m.

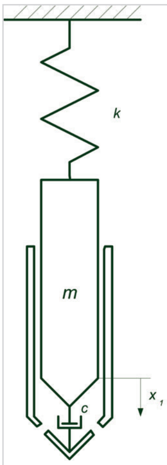

Figure 4 Kinematic scheme of a precise injector pair

Slika 4. Kinematska shema preciznog para ubrizgača

The injector (Fig. 4) consists of an outer cylinder, a piston of mass *m*, and a nozzle with holes of diameter *d* (in the case of the 6 AL 20/24 engine the nozzle has 7 holes). The injector chamber is filled with a viscous fuel with dynamic viscosity μ [Ns/m^2]. The first task is to formulate the polar equation of the injector. It is the relationship between the force acting on the piston and its speed in relation to the housing of the injector. Due to the occurrence of damping force in the model, the law of Hagen-Poiseuille was used. It assumes that at a stationary, laminar flow of incompressible, viscous fluid in a cylindrical pipe (i.e. in a pipe with a constant, circular cross-section), flow volume is proportional to the pressure gradient along the duct, and therefore to the pressure difference at the ends of the duct (1) [10]:

$$\Delta p = \frac{128 \cdot \mu l}{\pi d^4} Q \tag{1}$$

where: Δp - pressure drop over the length of one-meter pipe of diameter *d*, μ - dynamic viscosity of fuel, *Q* - fuel flow rate in the pipe, which might be described as:

$$Q = 0{,}25\pi D^4 u \tag{2}$$

where: *u* - fuel speed in the pipe

Speed is determined by the law of flow continuity, with:

$$0{,}25\pi D^4 u = 0{,}25\pi n d^4 v \tag{3}$$

where: – number of nozzle holes, – diameter of nozzle hole.

It has resulted:

$$u = \frac{nd^4 v}{D^4} \tag{4}$$

Assuming that the diameters of the injector holes are much smaller than the diameter of piston *D*, force *F* acting on the piston reaches the value of:

$$F = 0{,}25\pi D^2 \Delta p \tag{5}$$

By substituting the formula for velocity *u*, the polar equation of fuel's damping was obtained in the formula:

$$F = c \cdot v \tag{6}$$

where the coefficient of viscous damping is defined as:

$$c = 8nd^4 \mu l \pi \tag{7}$$

The model of needle impact into the cylindrical body of the injector was simplified to the form of a vibration forced by instantaneous force. The measure of the short-term impact of a needle is the impulse *S*, i.e.:

$$S = \int_{t_1}^{t_2} F(t)dt, \tag{8}$$

Because the pulse duration is insignificantly short, i.e.

$$\Delta t = t_2 - t_1 \tag{9}$$

A simplifying assumption about the occurrence of instantaneous force was made. If the value of force F_0 is constant then the condition is:

$$S = F_0 \Delta t = 1\, N \cdot s \tag{10}$$

It is a unit impulse.

Determining the impulse response of the system with the damping of fuel inside the injector chamber in oscillating systems with subcritical damping allows recording the motion of the piston in the form of:

$$x(t) = Ae^{-bt}\sin(\omega_D t + \alpha) \tag{11}$$

where: – constant, – initial phase.

The equation of motion takes the form:

$$m\ddot{x} + c\dot{x} + kx = F(t) \tag{12}$$

Applying the theorem on momentum change:

$$m\ddot{x} - m\ddot{x}_0 = S - \int_{t_1}^{t_2} c\dot{x}dt - \int_{t_1}^{t_2} kxdt \tag{13}$$

Any finite impulse acting on a body with mass *m* will result in a finite increase in the speed of its body. If the body had an initial speed of *0* at the end of the pulse will be:

$$\dot{x} = \frac{\check{S}}{m} \tag{14}$$

When estimating the impulses of and forces that act on the injector in time Δt, it should be noted that the mass of the plunger moving during the infinitely small Δt changes its position during an end impact of Δx in the time of Δt, hence:

$$\int_{t_1}^{t_2} c\dot{x}dt < c\dot{x}(t_2)\Delta t = 0(\varepsilon) \qquad \int_{t_1}^{t_2} kxdt < kx(t_2)\Delta t = k\Delta x\Delta t = 0(\varepsilon^2) \quad 15)$$

where: *0(ε)* is the small size of the first order.

This means that the pulses from the injector spring and the fuel damping are the first and the second-order impulses and do not affect the mass speed after the pulse. Thus, the initial conditions are:

$$x(0) = 0, \quad \dot{x}(0) = \frac{S}{m} \tag{16}$$

Substituting it result:

$$x(t) = Ae^{-bt}\sin(\omega_D t + \alpha) = \frac{S}{m\omega_D}e^{-bt}\sin\omega_D t \tag{17}$$

Which could be defined as:

$$x(t) = Sh(t) \tag{18}$$

where:

$$h(t) = \frac{1}{m\omega_D}e^{-bt}\sin\omega_D t \tag{19}$$

Finally:

$$\frac{S}{\omega}(-b^2e^{-bt}sin\omega t - 2be^{-bt}\omega cos\omega t - e^{-bt}\omega^2 sin\omega t)c \tag{20}$$

$$\left[\frac{S}{m\omega}(-be^{-bt}sin\omega t + e^{-bt}\omega\ cos\omega t)\right] + k\left(\frac{S}{m\omega_D}e^{-bt}\sin\omega_D t\right) = F(t)$$

The obtained equation (20) indicates that the change of the value of the force acting on a needle connected with the injector pre-tension spring changes the form of the oscillation equation for the needle. This means that there is a vibration parameter that identifies the injector adjustment status.

3. COURSE OF RESEARCH / *Tijek istraživanja*

As a form of experiment realization, a two-stage passive-active experiment was adopted, with the simulated variable state which was the injector opening pressure. Along with the change of the fuel injector opening pressure, the fuel injection angle was also changed indirectly, and thus the moment and value of the maximum combustion pressure. During the first stage of the tests, characteristics of the injector opening pressure as a function of the position of the adjusting screw were made. It was established that change of the position of the adjusting screw by 90° causing the reduction of the spring preload and results in a change of injector opening pressure by 3 MPa. During the adjustment of injector on the test stand, the time waveforms of vibration accelerations were recorded at two points. The first one was located on adjusting screw (signal2). Adjusting screw in this injector type is equipped with an internally screwed hole, thanks to which it was possible to screw in a threaded rod, the length of which was chosen so that after mounting the injector it was possible to discharge its upper part through the hole in the engine's head cover. As a result, it was possible to measure the vibrations of the injector which is not available during normal engine operation. The second measuring point was located on the injector high-pressure pipe (signal3). The tested injector during the regulation at the testbed is shown in Figure 4. Measurements of vibration parameters were carried out for the nominal injector opening pressure of 24,5 MPa and lower pressures (21,5; 18,5; 15,5; 12,5; 9,5; 6,5; 3,5 MPa). The vibration parameters for pressures above the nominal pressure have not been verified because situations of increase the injector opening pressure during its normal operation do not occur (they can only result from incorrect adjustment).

Figure 4 View of the injector disassembled from the Sulzer 6 AL 20/24 engine at the injectors testing and adjustment station

Slika 4. Prikaz ubrizgača skinutog sa Sulzer 6 AL 20/24 motora na stanici za testiranje i prilagodbu ubrizgača

During the second stage of tests, the injector with known characteristics of pressure changes as a function of the initial tension of the spring was mounted on the engine. Measurements of vibration accelerations were carried out in three points - two identical as during the first stage and third located on the engine head tighten bolt (signal4). The engine torque receiver was a water brake. Tests were carried out for three engine crankshaft's speeds and loads with a torque due to the marine engine propeller characteristics:

1. n = 500 rpm, M = 2050 Nm;
2. n = 600 rpm, M = 3420 Nm;
3. n = 700 rpm, M = 4390 Nm.

For all engine speeds, the spring preload was changed according to the characteristics determined during the first test stage. The technical solution used allowed to change the opening pressure of the injector without stopping the engine. This procedure made it possible to carry out measurements for different injector opening pressures under the same engine operating conditions. The authors did not meet in the literature description of an implementation of injector tests in a similar way. Usually the researchers, in all the changes in the spring characteristics of the injector, disassemble it from the engine. Such operation makes the engine conditions impossible to be reproduced during the next steps of measurements (the exception is the use of an electro-eddy rotor brake). To determine the effect of changes in the injector opening pressure on the indicated pressure parameters, continuous measurement of this pressure was carried out. The obtained results of changes in the indicated pressure at the rotational speed of 700 rpm are shown in Figure 5.

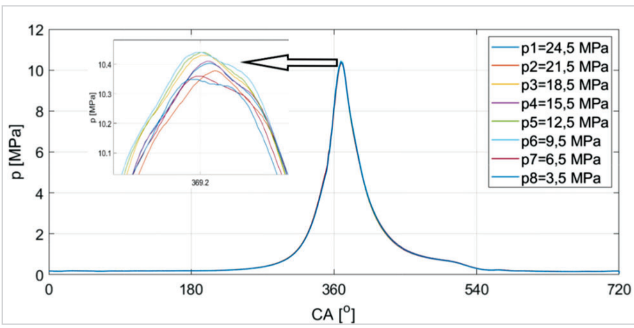

Figure 5 Indicator diagrams corresponding to the variable opening pressure of the injector for a constant engine load with the torque, at the rotational speed of 700 rpm

Slika 5. Indikatorski dijagrami prema varijabilnom početnom tlaku ubrizgača na stalnom opterećenju motora sa zakretnim momentom, na rotacijskoj brzini od 700 o/m

Analysing Figure 5, it can be observed that the changes in indicated pressure are mainly connected with maximum values. The pressure distribution in the remaining parts of the indicator diagrams remains virtually unchanged. Contrary to expectations, changes in maximum pressure are not significant. The range of changes for maximum pressures in the range of injector opening pressure changes from the nominal value (24,5 MPa) to the minimum (3,5 MPa) does not vary by more than 15% for 500 rpm.

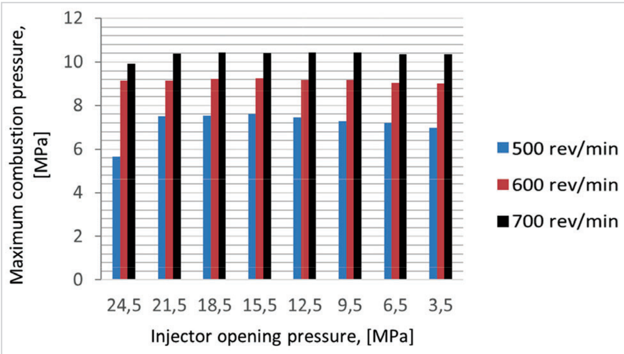

Figure 6 Graph of maximum indicated pressure as the function of the injector opening pressure for rotational speeds used during tests

Slika 6. Grafički prikaz maksimalno dobivenog tlaka kao početnog tlaka ubrizgača na rotacijskim brzinama koje su se koristile tijekom testiranja

In the case of higher rotational speeds (600 and 700 rpm) the maximum pressure spread is even smaller and does not exceed 5%. These are the quantities that generally cannot raise the crew's concern. What is more, on the indicator diagrams there was also no change in the crank angle occurrence of the maximum combustion pressure depending on injector opening varying pressure. According to the above, detection of injectors damage connected with changes of spring tension using the indication of the engine may be impossible regardless of the value changes value of this tension.

4. CHOSEN TEST RESULTS / *Rezultati odabranoga testa*

Vibration accelerations time courses recorded during both stages of the tests were subjected to a two-stage analysis. First, the overall levels of vibration accelerations recorded on injector test and regulation station were compared. Signals were averaged in two-second periods. The waveforms recorded on engine test bed during its operation were averaged over the time corresponding to the full cycle of its operation (two revolutions, this time was different for each rotational speeds). An example of the obtained results for the measurement carried out on the high-pressure injector pipe, carried out at the injector test and regulation station is shown in Figure 7.

Analysis of Figure 7 allows indicating the relationship between the tested vibration parameters and the injector's opening pressure. The higher the opening pressure, the higher the vibration acceleration values. This is unusual situation for vibration diagnostics, where diagnostics have become accustomed to a situation in which increasing vibration parameters indicate a deterioration of the technical condition of the machine or its element. This phenomenon can be explained easily, namely that the higher the injector opening pressure, the greater the dynamics of the injection process, causing the creation of force with higher values. The next stage of the analysis was an analysis of vibration accelerations amplitude spectra in the full range of measured frequencies - up to 25,6 kHz. An example of spectra obtained from the time courses recorded at the injector test stand is shown in Figure 8. These spectra were then examined for the determination of the frequency band sensitive to changes in the injector opening pressure values. A range of different frequency bands was used, from which the average amplitude of vibration accelerations was calculated. As a result of the analysis, the frequency band in the range of 2000-2800 Hz was selected as the most sensitive.

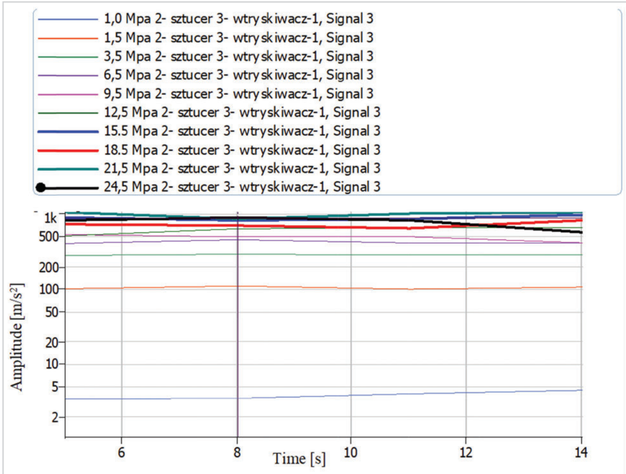

Figure 7 Overall values of vibration acceleration a_{rms} for two-second time periods counted for the 1Hz-10kHz frequency band and all tested injector opening pressures for measurements carried out at the high-pressure injector pipe

Slika 7. Ukupne vrijednosti vibracijskog ubrzanja a_{rms} *za periode od dvije sekunde izračunate za frekvencije 1Hz – 10kHz i sve testirane početne tlakove ubrizgača za mjerenja provedena na visokotlačnoj cijevi ubrizgača*

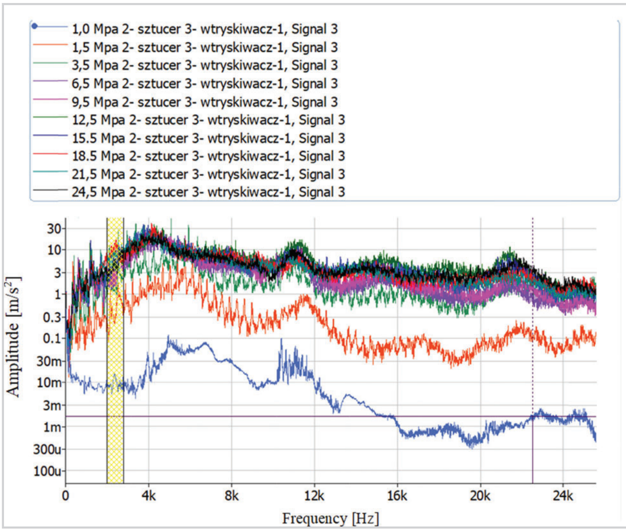

Figure 8 Vibrations of accelerations amplitude spectra recorded on the high-pressure connector pipe and all injector opening pressures tested on the engine test stand

Slika 8. Amplitude vibracijskih ubrzanja snimljene na visokotlačnoj spojnoj cijevi i svim početnim tlakovima ubrizgača testiranog na stanici za testiranje ubrizgača

Table 1 Results of average values of vibration accelerations in the range of 2 – 2,8 kHz

Tablica 1. Rezultati prosječnih vibracijskih ubrzanja u opsegu 2-2,8 kHz

p_{inj} [MPa]		1,0	1,5	3,5	6,5	9,5	12,5	15,5	18,5	21,5	24,5
a_{rms}	sig. 2	0,13	3,43	17,30	13,73	13,95	18,65	17,07	19,35	16,87	21,82
[m/s^2]	sig. 3	0,11	7,67	53,23	56,74	52,05	56,45	76,58	105,45	89,08	62,09

Table 1 summarizes the results of the average values of vibration accelerations (a_{rms}) obtained from spectra calculated of signals recorded on the injector control bolt (signal 2 – sig. 2) and the high-pressure connector pipe (signal 3 – sig. 3).

An analysis of values presented in table 1 allows to state that the chosen frequency range is a band sensitive to changes in vibrations accelerations amplitude associated with decreasing the opening pressure of the investigated injector. A significant reduction in the average amplitude values calculated in the indicated frequency range occurs when the injector opening pressure decreases by at least 12 MPa. In the same way, the waveforms recorded during engine operation on test stand were analysed. An example of the results obtained with reference to the average values of vibration accelerations calculated from time courses was shown in Figure 9.

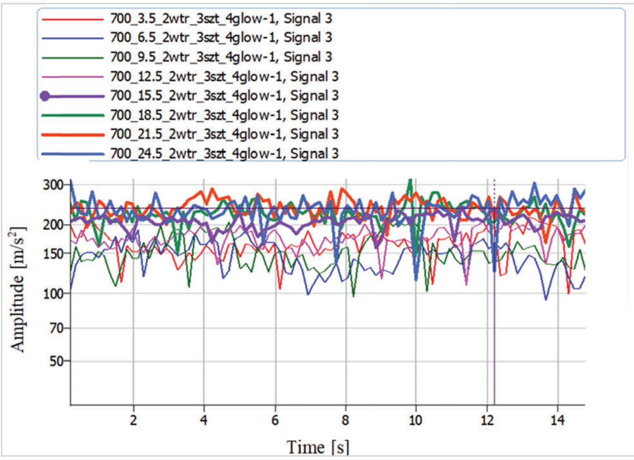

Figure 9 Average values of vibration acceleration a_{rms} for 160 ms time fragments counted for the 1Hz-10kHz band and all tested injector opening pressures.

Slika 9. Prosječne vrijednosti vibracijskih ubrzanja a_{rms} za 160 ms vremenske fragmente izračunate za 1 Hz – 10kHz i svi početni tlakovi testiranoga ubrizgača

The best results in terms of average values of vibration accelerations calculated from time course fragments were obtained for the rotational speed of 700 rpm and the measuring point located on the high-pressure connector pipe. In the case of other rotational speeds and other measuring points, the results are ambiguous. Therefore, the spectral analysis of all vibration acceleration recorded during the operation was carried out. The example was shown in Figure 10.

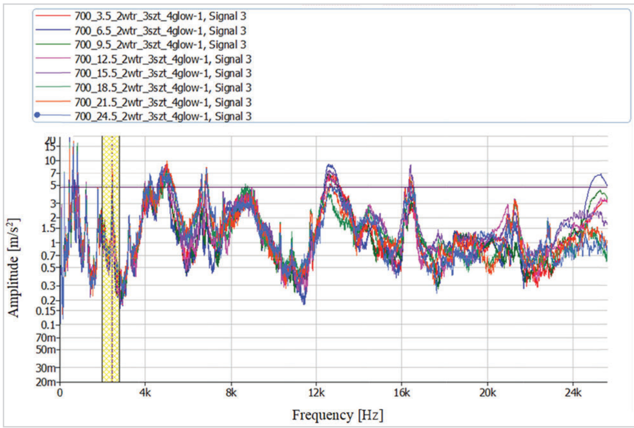

Figure 10 Amplitude spectra of vibrations accelerations recorded on the high-pressure connector pipe and all injector opening pressures obtained during engine operation

Slika 10. Amplituda vibracijskih ubrzanja snimljena na visokotlačnoj spojnoj cijevi i svi početni tlakovi ubrizgača dobiveni tijekom rada motora

The analysis of spectra presented in Figure 10 draws attention to their utterly different course than in the case of spectra obtained during measurements on injectors testing and adjustment station. There are two main reasons for this. The first of them is an entirely different mounting of the injector on the engine and thus a change in the stiffness of the system. The second is the interaction of other working engine components, including the remaining injectors

Table 2 Average values of vibration acceleration a_{rms} calculated for 2-2,8 kHz band for all measurements recorded during engine operation no test stand

Tablica 2. Prosječne vrijednosti vibracijskih ubrzanja a_{rms} izračunate za 2 – 2,8 kHz za sva mjerenja zabilježena tijekom rada motora na stanici za testiranje

p_{inj} [MPa]		3,5	6,5	9,5	12,5	15,5	18,5	21,5	24,5
$a_{rms;}$	signal 2	56,83	48,21	57,15	106,03	123,64	55,23	56,54	43,13
500 [rev/min]	signal 3	14,11	14.03	11,89	17,10	16,90	17,41	15,84	16,81
	signal 4	17,85	21,94	18,90	20,13	18,05	19,43	21,03	23,61
$a_{rms;}$	signal 2	57,89	65,55	78,42	93,64	108,84	120,25	142,59	119,54
600 [rev/min]	signal 3	11,90	13,45	12,46	16,17	18,32	14,51	18,00	15,25
	signal 4	16,82	16,59	17,10	18,78	19,46	18,19	19,44	19,40
$a_{rms;}$	signal 2	63,88	61,25	67,15	94,07	145,22	156,55	94,51	128,06
700 [rev/min]	signal 3	15,46	13,59	15,29	15,26	15,87	19,52	21,67	18,56
	signal 4	12,55	10,800	13,25	14,95	17,22	18,35	18,17	19,91

working with the same frequency. Despite the occurrence of these phenomena significantly changing the obtained vibration acceleration spectra, the frequency range of 2-2,8 kHz remains sensitive to changes in the injector opening pressure. This is evidenced by the results of all measurements, which were collected in Table 2. To facilitate the analysis of the obtained results, they were also presented graphically on figures 11 - 13.

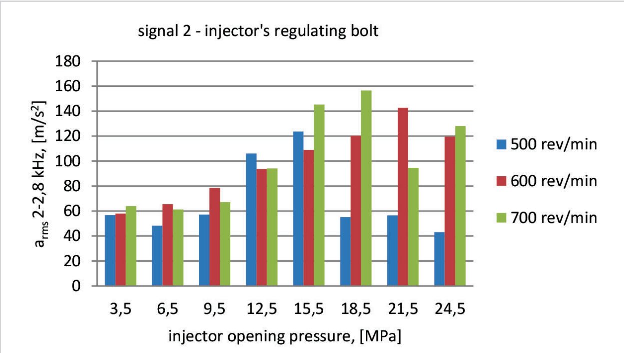

Figure 11 Values of average vibration accelerations in the 2-2.8 kHz band obtained during measurements on the injector regulating bolt

Slika 11. Vrijednosti prosječnih vibracijskih ubrzanja za 2 – 2,8 kHz dobivene tijekom mjerenja na regulacijskom vijku

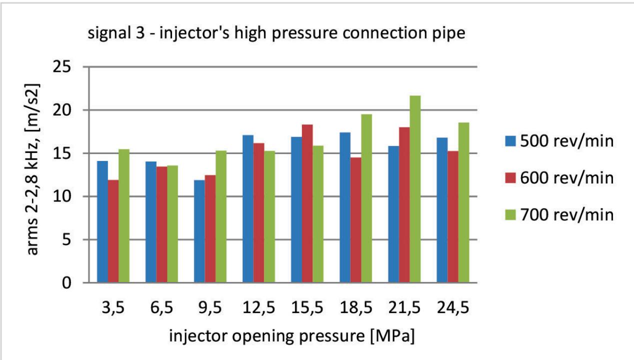

Figure 12 Values of average vibration accelerations in the 2-2.8 kHz band obtained during measurements on the injector's high-pressure connection pipe

Slika 12. Vrijednosti prosječnih vibracijskih ubrzanja za 2 – 2,8 kHz dobivene tijekom mjerenja na visokotlačnoj spojnoj cijevi ubrizgača

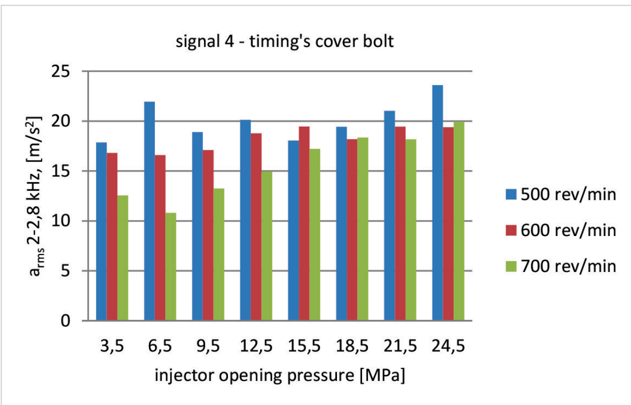

Figure 13 Values of average vibration accelerations in the 2-2.8 kHz band obtained during measurements on the timing's cover bolt

Slika 13. Vrijednosti prosječnih vibracijskih ubrzanja za 2 – 2,8 kHz dobivene tijekom mjerenja na vijku za vremensko podešavanje

Analysis of the presented results allows to state that the most unambiguous dependencies occur for measurements conducted at engine rotational speed of 700 rpm. This situation is repeated for all measuring points, however, the most visible dependencies are observable for the measuring point located on the timing cover bolt. The signals obtained for the rotational speed of 500 rpm are ambiguous and cannot be used as a diagnostic material due to the low sensitivity to the propagation of simulated damage. Slight changes in the opening pressure (which may be, for example, due to the natural "subsidence" of the spring) are impossible to detect. They even cause an increase in the value of the vibration parameter being studied, where its further propagation causes its reduction.

5. CONCLUSIONS / *Zaključci*

The conducted tests and its results confirm the effectiveness of vibration diagnostics and the ability to implement the analysis of average values of vibration accelerations from selected frequency ranges to detect inefficiencies associated with the change (reduction) of the injector opening pressure in marine diesel engines.

The possibility of detection of changes in the opening pressure occurs when the value of the opening pressure changes by at least 50% to the nominal pressure. This value may seem to be large, however, vibration acceleration as a diagnostic parameter is characterized by a higher sensitivity than indicated pressure measurement, especially at higher rotational speeds. The equation 20 presented in the paper is correct, which is confirmed by the results collected in Table 1, but it does not take into account the influence of other engine components, but only the changes within the hull of the tested injector. It should be noted that marine diesel engine is a challenging object from a diagnostic point of view, mainly due to the high complexity of the structure. The extortion in multi-cylinder engines occurring at the same frequency is also a critical difficulty. A solution might be the analysis of vibration signals in the function of crankshaft rotation, which the authors plan to do in the near future.

REFERENCES / *Literatura*

[1] Arroyo, J., Muñoz, M., Moreno, F., Bernal, N., Monné, C. (2013). "Diagnostic method based on the analysis of the vibration and acoustic emission energy for emergency diesel generators in nuclear plants". *Applied Acoustics*, Vol. 74, No. 4, pp. 502-508. https://doi.org/10.1016/j.apacoust.2012.09.010

[2] Carlucci, A. P., Chiara, F. F., Laforgia, D. (2006). "Analysis of the relation between injection parameter variation and block vibration of an internal combustion diesel engine". *Journal of Sound and Vibration*, Vol. 295, pp. 141-164. https://doi.org/10.1016/j.jsv.2005.12.054

[3] Delvecchio, S., Bonfiglio, P., Pompoli, F. (2018). "Vibro-acoustic condition monitoring of Internal Combustion Engines: A critical review of existing techniques". *Mechanical Systems and Signal Processing*, Vol. 99, pp. 661-683. https://doi.org/10.1016/j.ymssp.2017.06.033

[4] Dykas, B., Harris, J. (2017). "Acoustic emission characteristics of a single cylinder diesel generator at various loads and with a failing injector". *Mechanical Systems and Signal Processing*, Vol. 1, pp. 397-414. https://doi.org/10.1016/j.ymssp.2017.01.049

[5] Grządziela, A., Kluczyk, M. (2017). "High and medium speed marine diesel engines injection installation vibration diagnostics". *Diagnostyka*, Vol. 18, No. 2, pp. 37-42. https://doi.org/10.29354/diag/109793

[6] Jafarian, K., Mobin, M., Jafari-Marandi, R., Rabiei, E. (2018). "Misfire and valve clearance faults detection in the combustion engines based on a multi-sensor vibration signal monitoring". *Measurement*, Vol. 128, pp. 527-536. https://doi.org/10.1016/j.measurement.2018.04.062

[7] Jianmin, L., Yupeng, S., Xiaoming, Z., Shiyon, X., Lijun, D. (2011). "Fuel Injection System Fault Diagnosis Based on Cylinder Head Vibration Signal".

Procedia Engineering, Vol. 16, pp. 218-223. https://doi.org/10.1016/j.proeng.2011.08.1075

[8] Kluczyk, M., Grządziela, A. (2017). "Vibration diagnostics of common rail injectors". *Journal of Marine Engineering & Technology*, Vol. 16, No. 4, pp. 177-184. https://doi.org/10.1080/20464177.2017.1387088

[9] Kluczyk, M., Grządziela, A. (2019). "Marine diesel engine Common Rail injectors monitoring with vibration parameters". *Diagnostyka*, Vol. 20, No. 3. https://doi.org/10.29354/diag/109793

[10] Landau, L. D., Lifszyc, J. M. (2009). *Hydrodynamika*. Warszawa: PWN.

[11] Łutowicz, M. (2006). *Identyfikacja procesu sprężania okrętowego tłokowego silnika spalinowego dla potrzeb diagnostyki jego przestrzeni roboczych*. Rozprawa doktorska. Gdynia.

[12] Mancaruso, E., Sequino, L., Vaglieco, B. M. (2016). "Analysis of the pilot injection running Common Rail strategies in a research diesel engine by means of infrared diagnostics and 1d model". *Fuel*, Vol. 178, pp. 188-201. https://doi.org/10.1016/j.fuel.2016.03.066.

[13] Mihanović, L., Komar, I., Gržan, M. (2016). „Methodology Analysis Using Exploitation Reliability with the use of the RTOP Main Diesel Engine". *Naše more*, Vol. 63, No. 2, pp. 48-55. https://doi.org/10.17818/NM/2016/2.2

[14] Qiang, W., Peilin, Z., Chen, M., Huaiguang, W., Cheng, W. (2019). "Multi-task Bayesian compressive sensing for vibration signals in diesel engine health monitoring". *Measurement*, Vol. 136, pp. 625-635. https://doi.org/10.1016/j.measurement.2018.07.074

[15] Szymański, G. M., Tomaszewski, F. (20016). "Diagnostics of automatic compensators of valve clearance in combustion engine with the use of vibration signal". *Mechanical Systems and Signal Processing*, Vol. 68-69, pp. 479-490. https://doi.org/10.1016/j.ymssp.2015.07.015

[16] Taghizadeh-Alisaraei, A., Mahdavian, A. (2019). "Fault detection of injectors in diesel engines using vibration time-frequency analysis". *Applied Acoustics*, Vol. 143, pp. 48-58. https://doi.org/10.1016/j.apacoust.--2018.09.002

[17] Więcławski, K., Mączak, J., Szczurowski, K. (2018). "Fuel injector diagnostics based on observations of magnetic flux changes". *Diagnostyka*, Vol. 19, No. 3, pp. 89-93. https://doi.org/10.29354/diag/94040