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A METHOD OF VIBRATION DAMPING FOR DIESEL ENGINE CYLINDER LINERS TO PREVENT THE CONSEQUENCES OF EROSION

METODA TŁUMIENIA DRGAŃ TULEI CYLINDROWYCH SILNIKÓW O ZAPŁONIE SAMOCZYNNYM ZAPOBIEGAJĄCA KONSEKWENCJOM EROZJI*

This article presents the utilitarian need to determine the free vibrations of marine Diesel engine cylinder liners and the authors' own studies in this area. Theoretical investigations on free vibrations and experimental ones on forced vibrations have been described. Theoretical studies have been conducted with the application of characteristic dimensionless numbers in the Elektroniks Workbench and Wis Sim digital environment enabling virtual modeling of cylinder liner vibrations and determination of their characteristics: amplitudes, frequencies and accelerations. In the theoretical examination mechanical and electrical system analogues have been applied. A calculation method for the cylinder liner vibration damper, developed as a result of the study, has been discussed. Electrical oscillation damping filter design methods basing on the Bessel, Batterword and Chebyshev polynomials have been used. The course of the experimental examinations has been described and their results have been presented. Validation of the developed method has been executed applying measurement results concerning the parameters of Diesel engine cylinder liner vibration with various elastic elements. The results of the authors' own, theoretical and experimental, examinations have been confronted with those obtained by other scholars.

Keywords: *cylinder liner, vibration damping, damper, cooled surface erosion, virtual modelling.*

W artykule uzasadniono potrzebę określenia drgań własnych tulei cylindrowych silników okrętowych o zapłonie samoczynnym. Opisano badania własne w tym zakresie: teoretyczne dotyczące drgań własnych i eksperymentalne, dotyczące drgań wymuszonych.W badaniach teoretycznych do modelowania zastosowano liczby kryterialne podobieństwa z wykorzystaniem środowiska cyfrowego Elektroniks Workbench i Wis Sim. Umożliwiły one modelowanie wirtualne drgań tulei cylindrowych i określenie ich charakterystyk: amplitudy, częstości i przyśpieszenia. W badanach tych zastosowano analogi układów mechanicznych i elektrycznych. Opracowano metodę tłumienia drgań tulei cylindrowych z zastosowaniem filtrów tłumienia drgań elektrycznych w oparciu o wielomiany Bessela, Batterworda i Czebyszewa. Opisano przebieg badań eksperymentalnych i pokazano otrzymane wyniki. Walidację opracowanej metody przeprowadzono wykorzystując wyniki pomiarów parametrów drgań tulei cylindrowej silnika spalinowego z różnymi elementami sprężystymi. Skonfrontowano wyniki badań własnych z wynikami badań innych badaczy.

Słowa kluczowe: *tuleja cylindrowa, tłumienie drgań, tłumik, erozja powierzchni chłodzonej, modelowanie wirtualne.*

1. Introduction

Cylinder line vibrations bring about the occurrence of cavitation in cooling systems of ship Diesel engines [1, 10] especially in the medium- and high-speed ones. Therefore, damping cylinder liner vibrations is an important issue already at the designing and engine construction stage and at the same time it is an important way to maintain durability of such engines especially the trunk piston ones. In those engines, the passage of the connecting-rod through the upper and lower dead centres in the cylinder generates vibrations of high frequency. They cause erosive damage – surface degradation of the cooled part of the liner due to cavitation in the cooling liquid [3, 4, 5, 7, 16] . Erosive degradation of the surface significantly lowers durability of the cylinder [9] liner and at the same time the hourly service life of the whole engine. An effective method of preventing erosion of the surface of chilled internal combustion engines is presented in [1, 4], where for protection of metal cylinder liners and block mantle it is recommended to add chilled water containing nickel salts. The nickel layer well protects the metal from the impact of implosive gas-vapor bubbles [4]. However, it is not always possible to use this additive in cooled water due to the difficulty of its production. In literature the solution to the problem of cylinder liner vibrations, and at the same time protection of their cooled surfaces, is realized throughout mathematical models of the studied process reflecting with the highest probability real operational conditions of cylinder liners [5, 12, 16], including the application of simulation [13] with ANSYS software[11], vibration analysis based on clusters [17] and the method of finite elements [6].

The aim of the study is to find another, less labour-consuming utilitarian method of cylinder liner vibration damping which at the same time prevents the erosion wear of the Diesel engine cylinder jacket.

In this case, the purpose of damping the surfaces of the chilled internal combustion engines is to reduce the vibration of the cylinder liners. To prevent such phenomena it is necessary to develop a model of the studied process which will with high probability correspond with real operational conditions of the liner during engine operation.

Applying theories of probability to construct engineering models enables the transfer of results obtained in a theoretical way and in laboratories to technical scale. The use of similarity criterion numbers

(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl*** simplifies the studies and lowers their costs. To do this, the following similarities are used [15]:

- geometric one which is fulfilled when shapes and respective geometric dimensions are proportional. As a criterion of similarity one takes the ratio of two characteristic linear dimensions (e.g. diameters),
- kinematic one (of physical fields) having geometric similarity (e.g. distribution of pressure or velocity lines), with the maintained similarity of physical fields (distribution of lines of those fields). As a coefficient of similarity one takes the ratio of two characteristic values,
- dynamic one which is fulfilled when coefficients of similarity of different characteristic values significant for a given phenomenon e.g forces of inertia or viscosity remain in strictly determined relationships (criterion numbers e.g. Froud number, Reynolds number for flows).

2. Determination of cylinder liner natural vibrations throughout the virtual modelling method

Determination of geometric or kinematic similarity coefficient is not usually problematic. The situation is more complex in the case of dynamic similarity. While solving the problem the possibility to represent a cylinder liner as a string of linked cylinder rings of a simple shape was used. An example of cylinder liner decomposition for a Diesel engine of the 4Cz8,5/11 type is shown in Fig 1 [15].

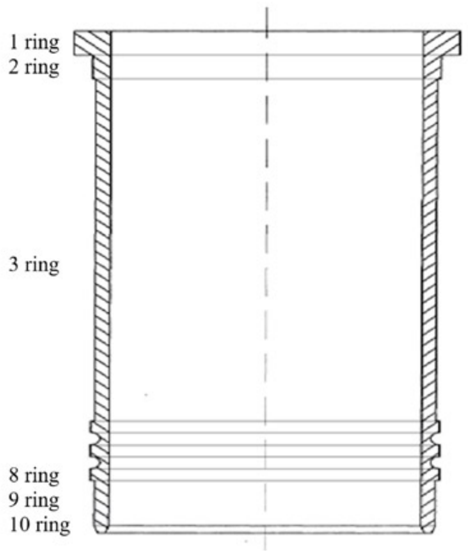

Fig. 1. Cylinder liner decomposition for the 4Cz 8,5/11 type engine

Such treatment of a cylinder liner enabled the application of the electromechanical analogue method to construct its virtual model [2]. The electromechanical analogue method was created on the basis of similarity of differential equations of electrical and mechanical systems. That method was used by Faraday and Maxwell, who using mechanical models tried to present in a more understandable way the hardly identified at that time electromagnetic processes.

Analogy of similarity between mechanical and electric phenomena was illustrated using mechanical and electrical systems corresponding to the following equations:

- for the linear electrical circuit:

$$L\frac{\mathrm{d}^2 q}{\mathrm{d}t^2}+R\frac{\mathrm{d}q}{\mathrm{d}t}+\frac{q}{c}=e \tag{1}$$

- for the oscillatory electrical circuit:

$$C\frac{\mathrm{d}e}{\mathrm{d}t}+\frac{e}{R}+\frac{1}{L}\int e\,\mathrm{d}t=i \tag{2}$$

- for a mechanical oscillating system of a mass suspended on a string:

$$m\frac{\mathrm{d}v}{\mathrm{d}t}+b\frac{\mathrm{d}x}{\mathrm{d}t}+kx=f \tag{3}$$

or

$$m\frac{\mathrm{d}v}{\mathrm{d}t}+bv+\frac{1}{k}\int v\,\mathrm{d}t=f \tag{4}$$

From the mathematical point of view equations (1)-(4) are similar. From the identity of equations of the mechanical system and the similarity of the electrical one results that the inductance L corresponds to mass m; electrical charge q to displacement x; electrical resistance R to dissipation coefficient b; electrical capacity C to rigidity k; voltage e to force f. It is the first form of the analogy. The second one suggested later by Chienla and Fajierstonoma, is based on the identity of equations for a mechanical system and a similar electrical circuit [13, 14]. It results from them that electrical capacity C is an analogue of mass m; voltage e of velocity v; conductivity $1/R$ of dissipation coefficient b; inductance L of elasticity k; electrical current I of force f.

Table 1 shows basic values and their corresponding analogues[14].

Table 1. Values and their corresponding analogues for the force-voltage and force- electrical current similarities

String	***Analogues***	**Values–analogues**				
Mechanical		Force (F)	Velocity (V)	Mass (m)	Dissipation coefficient (b)	Rigidity (k)
Electrical	***Force–voltage***	Voltage (U)	Current (I)	Inductance (L)	Resistance (R)	$1/C$
	Force–current	Current (I)	Voltage (U)	Electrical capacity (C)	Conductivity $1/R$	$1/L$

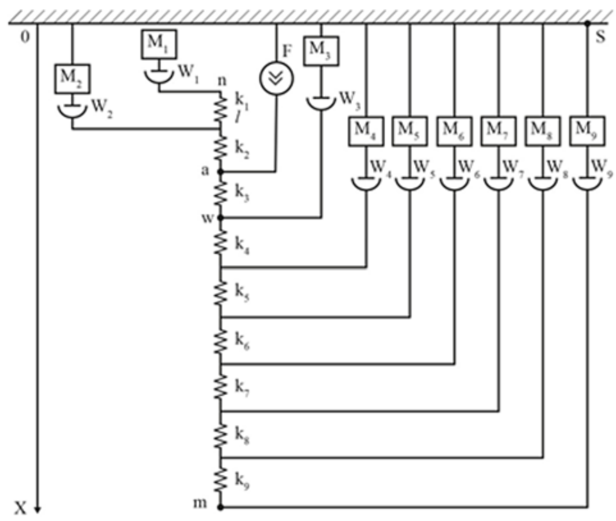

Fig. 2. A diagram of a mechanical representation of a cylinder liner of a 4Cz8,5/11 type engine in the form of a two-terminal network

Replacing real elements with their respective ones in the form of two-terminal networks characterising mass, energy dissipation and rigidity, a string of decomposed elements of the studied cylinder liner was constructed. An example of a mechanical network of a cylinder liner of the 4Cz8,5/11 type engine is shown in Fig 2.

Symbols $M_1, M_2, \ldots, M_9$ denote two-terminal networks characterizing ring mass; symbols $k_1, k_2, \ldots, k_9$ denote two-terminal networks characterizing ring rigidity; symbols $W_1, W_2, \ldots, W_9$, denote two-terminal networks characterizing dissipative properties of the rings, F – active two-terminal network of force, determined by the magnitude of piston force interaction with the liner, evoking its vibrations when the piston passes through the upper and lower dead centres. The values of rigidity, mass and dissipation of cylinder liner rings were determined on the basis of an equation developed for a certain natural ring frequency according to Timoszenko's equation [12].

$$\omega^2 = \frac{D_{sz}}{R^4 \rho \delta} \frac{n^2 \left(n^2 - 1\right)^2}{\left(n^2 + 1\right)} \tag{5}$$

where: D_{sz} – cylindrical rigidity of the liner;
R – external liner radius;
ρ – ring material density;
δ – ring crosswise area;
n – the number of radial half-waves appearing with ring deformation resulting from vibrations.

Rigidity of the ring liners was determined depending on the value of the r/l ratio where:
l – the length of a ring liner. For the third ring, for which $r/l \le 0{,}1$ equation (5) after transformations becomes:

$$k = \frac{EJ}{r^4} \frac{n^2 \left(n^2 - 1\right)^2}{\left(n^2 + 1\right)} \tag{6}$$

where: E – modulus of elasticity of the ring material;
J – moment of inertia of the ring cross-section surface.

For all the remaining rings with the $r/l \ge 0{,}1$ ratio elasticity was determined from equation [14]:

$$k = \frac{E\delta \left(\frac{\pi r}{l}\right)^4 + \frac{EJ}{r^2}\left(n^2 - 1\right)^2 n^4}{r^2 \left(\left(\frac{\pi r}{l}\right)^2 + \left(n^2 + 1\right) n^2\right)} \tag{7}$$

whereas ring mass was found from:

$$m_i = \rho_i \, \delta_i L_i \tag{8}$$

Dissipation of particular rings b_i was determined using the logarithmic damping decrement:

$$b_i = \frac{1}{\pi} \theta \sqrt{m_i k_i} \tag{9}$$

Table 2. Values of mass, elasticity and dissipation of particular cylinder liner rings of a Diesel engine of the 4Cz 8,5/11 type

Ring number	δ_i(m)	$m_i = \rho\delta_i$ (kg)	k_i (N/m)	$b_i = \frac{1}{\pi}\theta\sqrt{m_i k_i}$ (N·s/m)
1	$2{,}985\ 10^{-03}$	23,399	$6{,}741\ 10^{11}$	$3{,}641\ 10^{04}$
2	$1{,}414\ 10^{-03}$	11,084	$3{,}529\ 10^{11}$	$1{,}813\ 10^{04}$
3	$4{,}000\ 10^{-03}$	31,36	$1{,}462\ 10^{09}$	$6{,}013\ 10^{03}$
4	$1{,}414\ 10^{-03}$	11,084	$3{,}529\ 10^{11}$	$1{,}813\ 10^{04}$
5	$6{,}872\ 10^{-04}$	5,388	$1{,}811\ 10^{11}$	$9{,}055\ 10^{03}$
6	$1{,}414\ 10^{-03}$	11,084	$3{,}529\ 10^{11}$	$1{,}813\ 10^{04}$
7	$6{,}872\ 10^{-04}$	5,388	$1{,}811\ 10^{11}$	$9{,}055\ 10^{03}$
8	$1{,}414\ 10^{-03}$	11,084	$3{,}529\ 10^{11}$	$1{,}813\ 10^{04}$
9	$1{,}118\ 10^{-03}$	8,768	$2{,}852\ 10^{11}$	$1{,}450\ 10^{04}$
10	$8{,}388\ 10^{-04}$	6,576	$2{,}137\ 10^{11}$	$1{,}087\ 10^{04}$

Assuming that the liner is made of steel of 7480 kg/m^3 density, mass, rigidity and dissipation of particular rings were calculated for the studied Diesel engine of the 4Cz8,5/11 type from equations (6)-(9) and listed in Table 2.

The second stage of cylinder liner model development was to change the mechanical two-terminal network into electrical network in Elektronics Workbench (EWB) virtual environment . That approach was based on physical analogies between mechanical and electrical vibrations described by differential equations (1)-(4).

Resistors, condensers without charging losses and inductance without resistance are the basic electrical two-terminal networks. Mathematical model of a resistor is an algebraic equation (Ohm's Law) $\Delta U = IR$, where: ΔU - voltage drop at the resistor in V ; I – current intensity in A; R – resistance of a resistor in Ω. Models of a condenser and inductance are ordinary differential equations of the first order : $I = C(\mathrm{d}\Delta U/\mathrm{d}t)$; $\Delta U = L(\mathrm{d}I/\mathrm{d}t)$, where: C – condenser capacity in Farads; L – conductivity in Henries. Besides passive elements of the electrical circuit there are also active elements to which belong the sources of voltage and current. For the mechanical system, an analogue to Ohm's Law is:: $F = R_m \gamma$, where; γ – internal friction coefficient, R_m – analogue of electrical resistance. A change of force in time leads to a change of displacement. After differentiating the equation x = F/c versus time the following was obtained:

$$\frac{\mathrm{d}x}{\mathrm{d}t} = V = k_m \frac{\mathrm{d}F}{\mathrm{d}t} \tag{10}$$

where: V – velocity of displacement of the point of application of force;
k_m – rigidity.

Spring susceptibility was considered as an analogue of electrical condenser capacity because in elastic systems the relationship between the velocity of displacement and the velocity of force change is analogical to $I = C\frac{d\Delta U}{dt}$ for an electrical condenser. In accordance with the second Newton's Law:

$$F = m\frac{\mathrm{d}V}{\mathrm{d}t} = L\frac{\mathrm{d}V}{\mathrm{d}t} \tag{11}$$

body mass is an equivalent of electrical inductivity in modeling the anologue *Force –Voltage*. In accordance with the induced analogues,

Table 3. Calculated values of properties of elements of an electrical network

Ring number	Inductivity L (mHn)	Electrical capacity C (nF)	Resistance R (Ω)
1	1,484	23,399	2,308
2	2,833	11,084	4,635
3	72,87	31,36	13,97
4	2,833	11,084	4,635
5	5,522	5,388	9,281
6	2,833	11,084	4,635
7	5,522	5,388	9,281
8	2,833	11,084	4,635
9	3,506	8,768	5,797
10	4,679	6,576	7,733

between the mechanical and electrical values for the sake of determining parameters of an electrical circuit the following correlations were established: mass element m_i, in kg corresponds to electrical capacity C_i in nano-Farads; elasticity elements k_i, in N/m correspond to inductivity in Henries, as $1/L = k_i$, and dissipation of elements b_i, in Ns/m to electrical resistance R_i, defined as $R=1/b_i$ in Ohms. The calculated parameters of equivalent elements of an electrical circuit for a cylinder liner of a 4Cz8,5/11 type engine are shown in Table 3.

On the basis of the values of elements of an electrical network listed in Table 3 values of elements (resistors, condensers and induction coils) necessary to construct a network were determined. Representation of a cylinder liner of the 4Cz8,5/11 type engine in an electrical form is shown in Figure 3 [13].

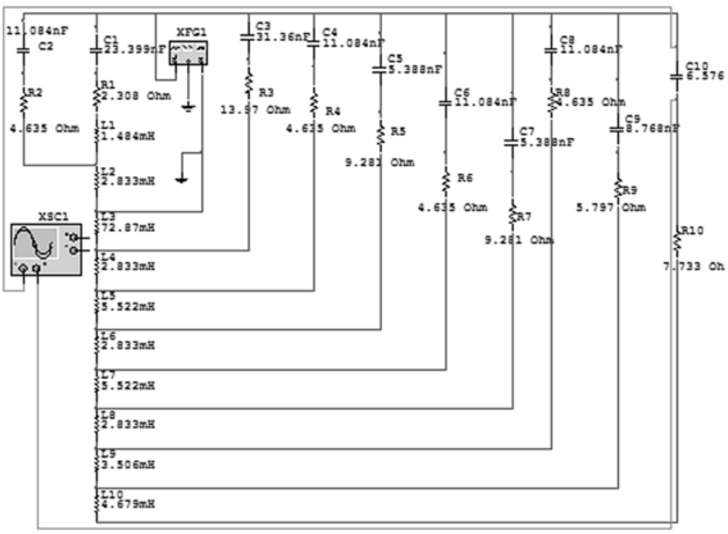

Fig. 3. A diagram representing a cylinder liner of a 4Cz 8,5/11 type engine in the form of an electric network

Using the representation of the cylinder liner in the form of an electrical network to determine the frequency of the first harmonic vibration of the cylinder liner, a virtual experiment has been carried out which used a generator inducing sinusoid, triangular and rectangular signals of a known frequency, amplitude and displacement. To monitor the experimental results, a two-channel oscillograph was used in the network. A piston stroke in the cylinder liner in the experiment was simulated by a short rectangular impulse whose duration was equal to the real piston stroke. For this reason in the physical model the duration of the stroke signal was estimated using a "Bordeaux " type oscillograph. Rectangular impulses in compliance with the frequency of piston strokes against the liner and the rotational speed of the engine shaft were set on the signal generator. The duration of stroke time was imitated by the value of the impulse-width signal modulation coefficient. In further studies, short strokes were imitated by Dirac's impulses. Experimental results of vibration frequency of the first harmonics of a cylinder liner for the 4Cz8,5/11 type Diesel engine turned out to be close to those obtained in studies carried out at an experimental stand (further described in part 3) and with the results from another study [7], which is shown in Table 4.

Table 4. Vibration frequencies of a cylinder liner of the 4Cz8,5/11 type Diesel engine

Frequency of cylinder liner natural vibrations in the virtual experiment	Frequency of cylinder liner natural vibrations on a technical scale [15]	Vibration frequency of a cylinder liner of the 4Cz8,5/11 engine in an experiment on a technical scale [7]
1720,5 Hz	1727±198 Hz	1874 Hz

3. Modelling cylinder liner vibration damping

While considering a cylinder liner of a diesel engine as a series of subsequently connected rings (see Fig1) they were treated as a string of vibrating elements with their own transmittance. (Transmittance is the ratio of the output value versus the input values of variables at zero initial conditions.) If the vibrating element is described by a differential equation of the second order in the following form:

$$T_2^2 y'' + 2b_i T_1 y' + y = kx \tag{12}$$

then the transmittance of vibrating elements is described by the equation below:

$$W_p = \frac{k}{T_2^2 p^2 + 2b_i T_1 p + 1} \tag{13}$$

where: p – Laplace's operator,
k – amplification coefficient,
b_i – dissipation of the rings.

$T_1 \mathrm{i} T_2$ transmittance coefficients were calculated solving the iterative equations (14)-(15):

$$b_i \frac{T_1}{2T_2} \quad (14) \qquad \omega = \frac{\sqrt{4T_2^2}}{2T_2^{\,2}}, \tag{15}$$

where: $\omega = 2 \cdot \pi \cdot f$ – angular velocity;
f – known natural vibration frequency of rings (Hz).

Time constants T_e can be determined solving the equation:

$$T_2 = \frac{T_e}{\sqrt{\omega^2 \cdot T_e^2 + 1}} \tag{16}$$

In the studies a simpler method was applied using the Vis Sim library. Determining experimentally the frequency of natural vibrations and the time constant (14), transmittance was calculated (13). The calculated values of time constants and natural vibration frequencies are listed in Table 5.

On the basis of parameters listed in Table 5, in the Vis Sim environment a functional

Table 5. Values of time constants and natural vibration frequencies of particular rings of a cylinder liner of the 4Cz8,5/11 type engine

Ring liner number	T_e (s)	f (Hz)	T_1 (s)	T_2 (s)
1	$14{,}00 \cdot 10^{-6}$	24820	$4{,}86 \cdot 10^{-6}$	$5{,}83 \cdot 10^{-6}$
2	$17{,}00 \cdot 10^{-6}$	23421	$4{,}69 \cdot 10^{-6}$	$6{,}31 \cdot 10^{-6}$
3	$3100{,}00 \cdot 10^{-6}$	1619	$6{,}23 \cdot 10^{-6}$	$98{,}31 \cdot 10^{-6}$
4	$6{,}24 \cdot 10^{-6}$	46567	$2{,}88 \cdot 10^{-6}$	$2{,}99 \cdot 10^{-6}$
5	$6{,}26 \cdot 10^{-6}$	45487	$2{,}98 \cdot 10^{-6}$	$3{,}06 \cdot 10^{-6}$
6	$6{,}24 \cdot 10^{-6}$	46567	$2{,}88 \cdot 10^{-6}$	$2{,}99 \cdot 10^{-6}$
7	$6{,}26 \cdot 10^{-6}$	45487	$2{,}98 \cdot 10^{-6}$	$3{,}06 \cdot 10^{-6}$
8	$6{,}24 \cdot 10^{-6}$	46567	$2{,}88 \cdot 10^{-6}$	$2{,}99 \cdot 10^{-6}$
9	$58{,}84 \cdot 10^{-6}$	12183	$5{,}57 \cdot 10^{-6}$	$12{,}76 \cdot 10^{-6}$
10	$2{,}30 \cdot 10^{-6}$	77121	$2{,}05 \cdot 10^{-6}$	$1{,}54 \cdot 10^{-6}$

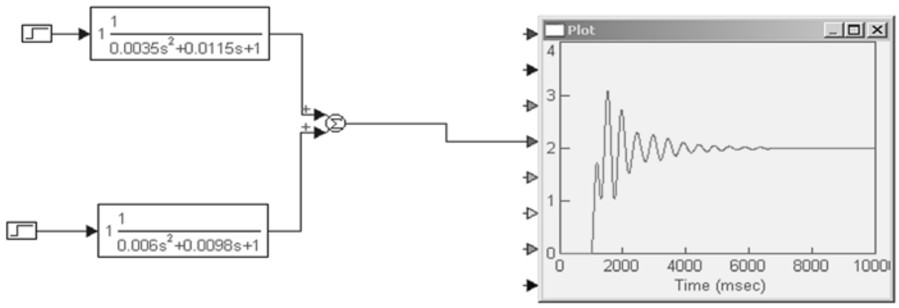

Fig. 4. A diagram of a functional model of a cylinder liner of the 4Cz8,5/11 type engine in the Vis Sim environment

model of an engine cylinder liner was constructed and shown in Figure 4. The model comprises blocks simulating piston strokes in the upper and lower dead point, a block summing up the results of the two strokes and a block reflecting the results.

In order to damp electrical signals electronic systems contain the most frequently used filters from the Vis Sim electronic environment library: Bessel's, Batterword's and Chebyshev's filters differing because of the used polynomials. The results of cylinder liner of the 4Cz8,5/11 type engine vibration damping obtained from the studies carried out on the liner model pointed to Chebyshev's filter as the most effective one. It is confirmed by the runs of results of cylinder liner vibration damping efficiency obtained using the above mentioned filters which are shown Figure 5.

4. Experimental validation of study results

In engineering practice the most effective are the filters which are the result of synthesis of ready-made step structures with inductivities as lengthwise elements and capacities as crosswise structures. Figure 6 shows examples of Chebyshev's filters diagrams of the third order consisting of three types of elements: resistance, electrical capacity and inductivity [15].

Performing a reverse transition to the one described at the beginning i.e. going from electrical to mechanical analogies, it was stated that the actual damper of cylinder liner mechanical vibrations constructed using Chebyshev's polynomial must possess as subsequent linked elements: rigidity and dissipation separated by mass elements -metal foil as separators [8]. Appropriateness of the studied damper construction was checked at the experimental stand whose diagram is shown in Fig. 7 together with a description [13].

Mounting the liner in the coat was done throughout pushing the cylinder liner into a ring groove in the coat. The vibrator generated dynamic impulses in the upper and lower part of the cylinder liner inducing vibrations of 50 Hz frequency. During the study washers were used as elastic elements. They are shown together with study results in Table 6. In order to check out the effectiveness

Fig. 6. Examples of Chebyshev's filters diagrams of the third order

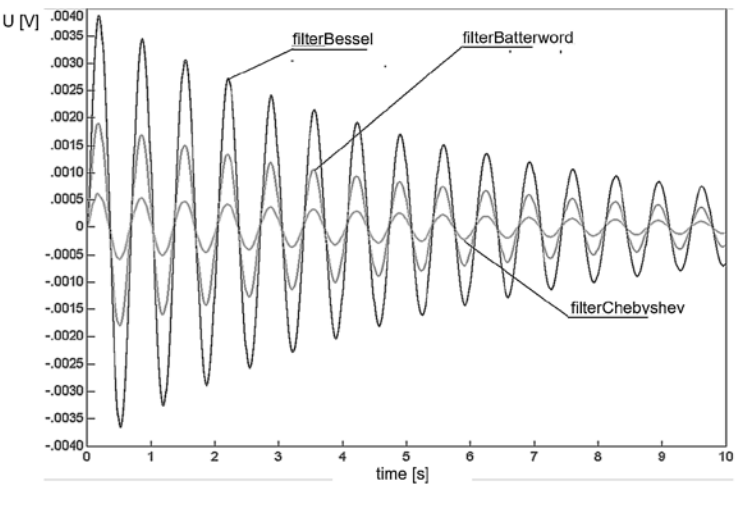

Fig. 5. Results of vibration damping effectiveness of a cylinder liner of the 4Cz8,5/11 type engine obtained using different filters

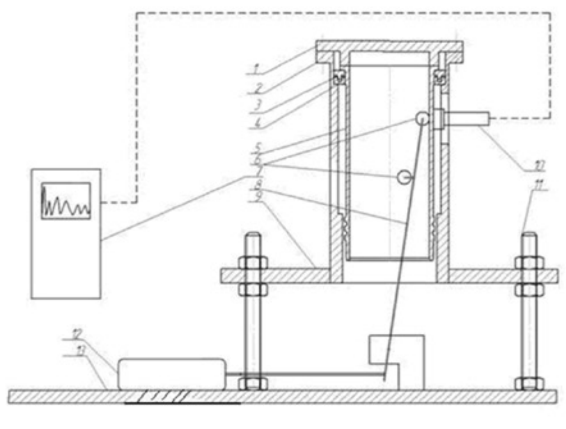

Fig. 7. Experimental stand for simulation studies of cylinder liner of the 4Cz 8,5/11 type engine vibration parameters 1 – cylinder head; 2 – cylinder coat; 3 – upper clamping ring; 4 – lower clamping ring; 5 – liner; 6 – upper and lower firing pin; 7 – vibrometer TV-300; 8 – bolt; 9 – support plate; 10 – TSV-01 vibration detector; 11 – screws; 12 – vibrator; 13 – base plate

Table 6. Results of measurements of vibration parameters of a cylinder liner of the 4Cz8,5/11 type Diesel engine with different elastic elements at the upper dead point of the liner

No	Elastic washer type	Top frequency (Hz)	Stable frequency (Hz)	Vibration amplitude (mm)	Displacement speed (m/s)	Vibrational acceleration (m/s²)
1	Without elastic washers	20000	58	0,593	10,027	8,26
2	Uniform rubber (δ = 4 mm)	67,5	0,323	0,04	0,64	0,57
3	Rubber set (4δ = 1 mm)	3,9	0,338	0,028	0,625	0.52
4	Polytetrafluoroethylene (δ = 4 mm)	5,5	0,325	0,027	0,424	0,36
5	Sillicon set(2δ = 2 mm)	7,4	0,324	0,055	0,879	0,73
6	Uniform paronit (δ = 3 mm)	3,8	0,338	0,07	1,069	0,9
7	Paronit set (3δ = 1 mm)	3,9	0,337	0,028	0,41	0,35

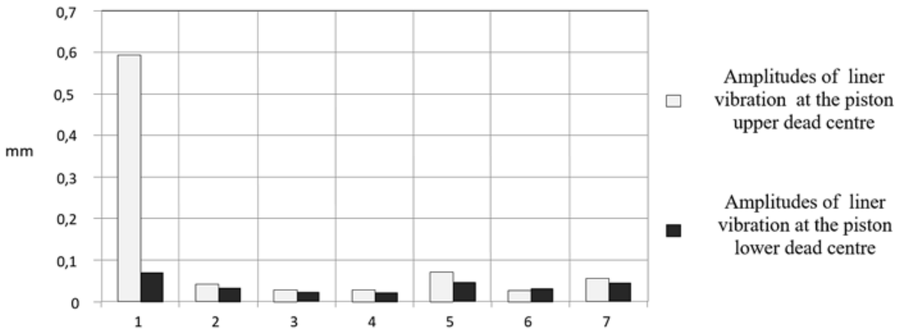

Fig. 8. Amplitudes of cylinder liner vibrations measured during the experiment

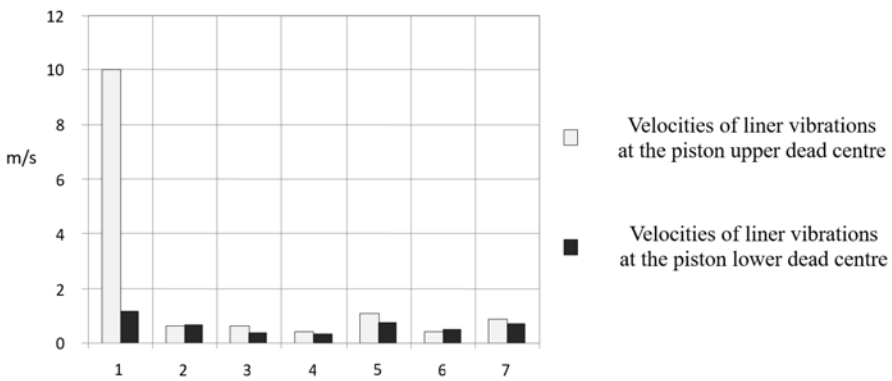

Fig. 9. Velocities of cylinder liner vibrations measured during the experiment

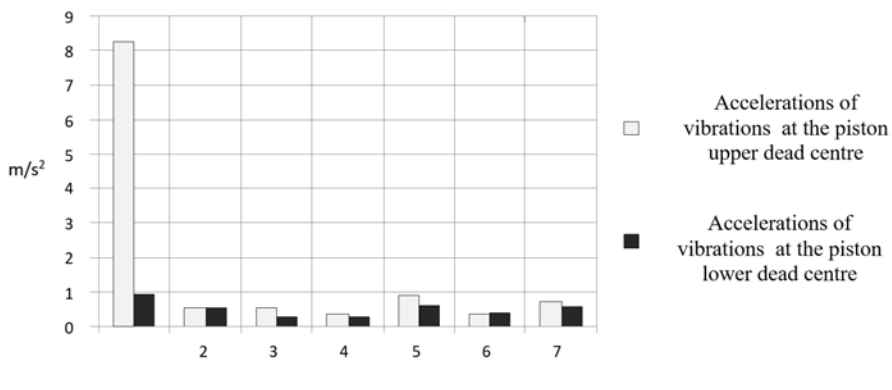

Fig. 10. Accelerations of cylinder liner vibrations measured during the experiment

of damping using a set of elastic elements separated with middle masses, an experiment was carried out comparing damper operation with uniform elastic elements made of rubber and paronit with the same as far as the thickness goes sets of rubber and paronit washers (δ = 1 mm), separated with metal foil. The total washer thickness varied from 3 to 4 mm.

Measurements of vibration frequencies of the cylinder liner were performed using the TSV-01 detector and oscillograph. Measurements of vibration parameters of the cylinder liner like amplitude, velocity and acceleration of vibrations were performed in the planes of strokes of the firing pins in the upper and lower dead centres in perpendicular directions using the TV-300 vibrometer with the TSV-01 detector. Results of measurements of vibration parameters of a cylinder liner are presented graphically as bar diagrams in Fig. 8, 9, 10.

5. Conclusion

The obtained theoretical and experimental results of the study showed real possibilities of increasing resistance to cavitation erosion, preventing erosive damage to cylinder liners throughout limiting their vibrations.

The results of studies on natural vibration parameters of cylinder liners of the 4Cz8,5/11 type engine in Elektronics Workbench virtual environment turned out to be similar to experimental results obtained during the study at an experimental stand on a technical scale, with simple measurements of actual liner vibrations and results from other studies close to those obtained in [7] and about ten times below those given, among others, in Table 5 in [17].

Thus, they confirmed the appropriateness of the applied method of virtual studies for determining the characteristics of cylinder liner vibrations and further studies based on analogues derived from probability criterion numbers more reliable.

During the experiment, the effectiveness of damper construction, consisting of a set of elastic elements was verified. Amplitudes of cylinder liner vibrations, speed of its displacement and acceleration when using a damper with uniform elastic elements (paronit and rubber) are 1.5 to 2.5 times bigger than those for analogical values for a damper with separated elastic elements.

Ananalysis of results of studies on different kinds of elastic washers as elements of damper construction shown in Table 6 proves that the use of any elastic material lowers vibration parameters of cylinder liners minimizing them practically to zero values. Results obtained during the experiment confirmed the appropriateness of adjusting the elastic damper set to the reverse mode to the cylinder liner resonance. Such behaviour of a cylinder liner is accompanied by minimum cavitation wear in the cooling fluid and as a result the liner will not undergo erosive degradation.

Metal surface scan obtain increased resistance to cavitation erosion throughout making the working medium flow laminar, covering metal surfaces with protective coats and relatively in the most simple way, in the case of vibration cavitation, through outdamping the vibrations of the washed cylinder liner.

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