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METHOD FOR ASSESSING THE RELATIONSHIP BETWEEN THE CHARACTERISTICS OF VIBROACTIVITY AND THE DESIGN PARAMETERS OF A MARINE DIESEL

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A B S T R A C T

Protection of the crew and passengers from vibration and noise of machines, auxiliary mechanisms and propeller, as well as reduction of sound emission into the surrounding space, are paramount tasks in solving the problem of vibration and noise in shipbuilding. The magnitude of the level of vibration and noise affects the performance, well-being and health of people, the operation of equipment and mechanisms, etc. It should be noted that vibration not only has a negative physiological effect on the human body, but also serves as a source of airborne noise. Noise values are determined mainly by the acoustic and vibrational power generated by the source. The determination of defects by vibration parameters is one of the most intellectually intensive sections of diagnosing reciprocating machines, which include marine engines. The studies presented in this paper are aimed at eliminating these gaps, primarily on the basis of a more complete description of vibrational processes. The article presents the results of a study related to the control, diagnostics and evaluation of the technical condition of an internal combustion engine running on diesel fuel. The possibility of establishing a connection between the vibrational parameters of the engine and the change in its design characteristics during operation is shown. The results of the work of the analytical and experimental information system for monitoring the vibration activity of diesel engines are presented.

1. INTRODUCTION

Currently, one of the priority national goals for the next decade is digital transformation (Cherepovitsyn, & Evseeva, 2021; Yurak, Dushin & Mochalova, 2020), therefore, a trend has been established to improve existing and introduce new modern technologies in the field of information support in all industries of Russia (Afanasyev, 2020; Tsvetkova & Katysheva, 2019; Kozmenko, Masloboev & Matviishin, 2018), including those that ensure the efficient functioning of mineral resource enterprises. (Afanasev, 2020; Filatova, Nikolaichuk, Zakaev & Ilin, 2021).

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Currently, information, computing, telecommunication systems and tools permeate literally all spheres of human activity and are experiencing a stage of rapid development (Vasilev, Cherepovitsyn, Tsvetkova, & Komendantova, 2021; Tsvetkov & Fedoseev, 2020). The emergence in recent years of the "Internet of things", "smart" objects for various purposes, computing facilities for processing and exchanging large amounts of information and data used to solve various problems of human activity, is due to the results achieved in the field of digital technologies (Resniova & Ponomarenko, 2021; Plotkin & Khaikin, 2017).

2. MATERIALS AND METHODS

Increasing the power plant's power while decreasing their weight causes an increase in vibration and noise levels on ships and complicates the task of meeting the requirements for noise and vibration (Zhukov et al, 2020; Tusov, Bezyukov & Afanaseva, 2012). At present, diesel and gasoline internal combustion engines (ICE) are most often used as sources of mechanical energy in ship power plants (SPP). The noise of diesel engines is more intense than the noise of other types of internal combustion engines at equal powers.

The degree of vibration influence depends on its spectral composition, direction, place of application and exposure duration (Zhukov et al, 2020; Godin, Zabotin & Zabotina, 2020).

Each ICE consists of numerous parts (pistons, valves, etc.), which perform rapid reciprocating movements or, in other words, receive strong accelerations, which leads to the formation of significant variable forces. The most unpleasant are the impacts of some details on others (Tusov, Bezyukov & Afanaseva, 2012). The most characteristic parameters of these processes can be: shifting of pistons, blows of valves and a fuel pump in a diesel engine. Experiments with an internal combustion engine operating in a forced mode have shown that these processes generate noise of approximately the same order of magnitude as the combustion noise.

It is known that vibrations in the air environment are wave-like propagating movements of air particles and changes in its pressure (Tusov, Bezyukov & Afanaseva, 2012; Bochkarev, Kamenskikh & Lekomtsev, 2020). In the frequency range $1000 - 5000$ Hz, a person hears sounds if the sound pressure is not less $p_0 = 2 \cdot 10^{-5}$ N/m^2 (hearing threshold), and the pain threshold is estimated by pressure $p^* \approx 20$ N /m^2 (Sun, Lu, Chen & Jiao, 2022; Hou & Xia, 2021). Sound level (L_v) can be estimated by the ratio of the strength of a given sound to the strength of sound corresponding to the hearing threshold:

$$L_v = 10lg\frac{I}{I_0} \text{ (dB)}. \quad (1)$$

Where $I_0 = 10^{-12}$ W/m^2 - sound intensity corresponding to the audible threshold (sound power threshold for an airborne noise source). The sound pressure level can be estimated using the formula:

$$L_p = 20lg\frac{p}{p_0} \text{ (dB)}. \quad (2)$$

Where p_0- audible pressure, p- effective value of the real sound pressure.
Changing the sound level by 1 dB corresponds to a 26% change in sound intensity, and a 12% change in pressure (Esu, Wang & Chryssanthopoulos, 2021; Godin, Zabotin & Zabotina, 2020).

The vibrational velocity of a sound wave, along with sound pressure, is one of the most important measured quantities. The level of vibrational speed can be determined by the formula:

$$L_{\dot{q}} = 20lg\frac{\dot{q}}{\dot{q_0}} \text{ (dB)}. \quad (3)$$

Where $\dot{q}$ - the effective value of the speed in the corresponding frequency band, and $\dot{q_0} = 5 \cdot 10^{-8}$ m/s - speed threshold.

When studying vibrational processes, vibrational displacements are also used as vibrational parameters q and vibrational accelerations $\ddot{q}$, related (taking into account the phase) by the relations $\ddot{q} = j\omega\dot{q} = -\omega^2 q$ (Chen, H., Hu, N., Cheng, Zhang & Zhang, 2019; Godin, Zabotin & Zabotina, 2020). The threshold levels for them are chosen in such a way that at a frequency of *1000* Hz the levels of vibration displacement, vibration velocity and vibration acceleration in decibels have the same values, that is:

$$L_q = 20lg\frac{q}{q_0} \text{ (dB)}, L_{\ddot{q}} = 20lg\frac{\ddot{q}}{\ddot{q_0}} \text{ (dB)}. \quad (4)$$

Where $q_0 = 8 \cdot 10^{-12}$(m), $\ddot{q}_0 = 3 \cdot 10^{-4}$(m/s^2).
Known empirical relationships between the main characteristics of internal combustion engines and their noise:

$$L_p \approx (57 + 10\lg(N_N \cdot P_N) + 30\lg(N/N_N)) \text{ (dB)}. \quad (5)$$

Where L_p - sound power level; N_N – rated engine speed, rpm; P_N – rated power, kW;N – engine operating speed.
The third octave sound power level is calculated using the formula:

$$L_p \approx \left(52 + 10\lg\left[\left(N_N \cdot P_N\left(1 + \frac{P_N}{m}\right)\right) / \left(\frac{f}{1000} + \frac{1000}{f}\right)\right] + 20\lg(N/N_N)\right) \text{ (dB)}. \quad (6)$$

The third octave level of sound vibration at the engine foot above the shock absorber can be determined from the expression:

$$L_v \approx \left(44 + 10lg\left(N_N P_N^{0,55}\left(1 + {}^{P_N}/_{m}\right) / \left(1 + (f/1500)^3 {}^{m}/_{P_N}\right)\right) + 30lg\left({}^{N}/_{N_N}\right)\right) \text{(dB)}. \quad (7)$$

Where L_v - speed relative level $v_0 = 5 \cdot 10^{-8}$ m/s, m – engine weight, kg; f – geometric mean frequency of the third octave band.

Below are the results of experimental studies of the vibration level caused by the piston shift carried out for the 5D4 (4Ch8.5/11) diesel engine, which is a single-row vertical four-stroke non-reversible four-cylinder internal combustion engine with the following technical characteristics power (N) *17.8* kW; rated speed (n) *1500* rpm; mean effective pressure (P_e) *0,575* MPa; clearance between piston and bushing (δ) *0,0003* m; piston stroke (S_{xn}) *0,11* m; cylinder diameter (D_c) *0,085* m; maximum cycle pressure (P_z) *8* MPa; sleeve thickness (h_{vt}) *0,005* m; block thickness (h) *0,007* m; block stiffness (D_{czb}) $6542 \left(\frac{kg \cdot m^2}{s^2}\right)$; lateral force ($N_{max}$) $2{,}268 \cdot 10^3$ (N); bushing stiffness (D_{czvt}) $1{,}278 \cdot 10^4 \left(\frac{kg \cdot m^2}{s^2}\right)$ (Tusov, Bezyukov & Afanaseva, 2012).

The presented results were obtained using an experimental setup, which includes an object of research, an information-measuring complex, a control-computing complex and an output recording printing device, as shown in Fig. 1. Feedback was carried out using the "operator" block. This block allows control over the measurement process.

In the experimental setup, the measuring device recorded vibration accelerations at a frequency of *125* Hz in the range from *0* to *165* dB. As a result of the experiment, the numerical values of vibration were obtained, given in tables 1 and 2.

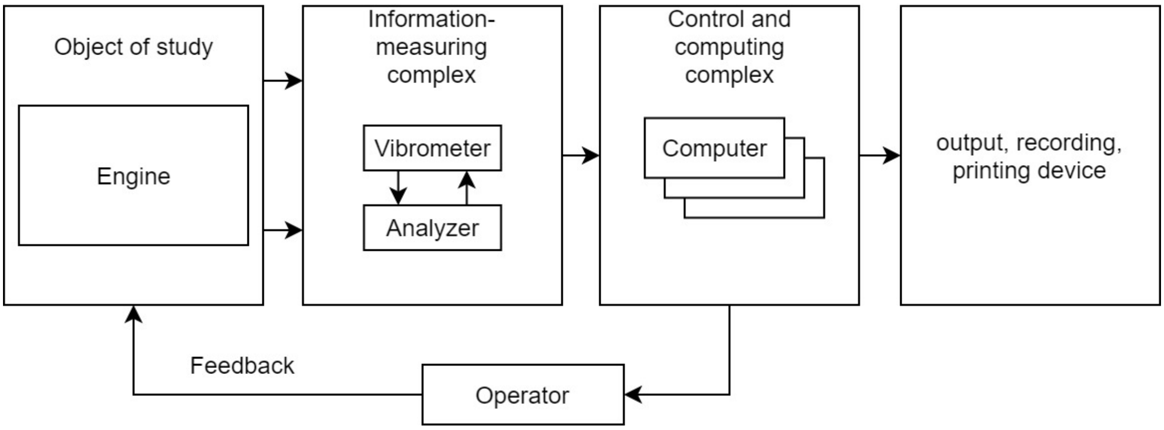

Figure 1. Experimental setup for measuring, registration and processing of diesel vibration. *Source:* "Compiled by the authors".

Table 1. Vibration velocity values obtained during the experiment on a 5D4 diesel engine (4Ch8.5/11) in the mode: *1500* rpm under different loads

load, kW	Root mean square value of vibration velocity (RMS), mm / s	
	Filter Vel3	Filter Vel10
0.3	7.67	6.17
1.2	15.5	12.9
1.96	18.8	16.2
3.96	19	17
5.28	20.7	18.2
7.04	27.5	21.4
9.24	30.9	22.1
9.6	16.6	12.4
10.12	19.5	14.6

Source: "Compiled by the authors".

Table 2. Vibration velocity values obtained during the experiment on a 5D4 diesel engine (4Ch8.5/11) in idle mode

mode	Root mean square value of vibration velocity (RMS), mm / s	
	Filter Vel3	Filter Vel10
1500 rpm	13.5	11.1
1300 rpm	5.95	3.59
1000 rpm	11.4	8.81

Source: "Compiled by the authors".

3. RESULTS

As an example, fig. 2, 3 and 4 show graphs of vibration velocity and three-dimensional diagrams of frequency characteristics obtained during the experiment, so fig. 2 shows a graphical representation of vibration velocity measured in idle mode, a) crankshaft rotation speed 1500 rpm, b) crankshaft speed 1300 rpm, c) crankshaft speed 800 rpm.

Fig. 3 shows a graphical representation of the vibration velocity measured at a crankshaft speed of 1500 rpm, power 10.12 kW.

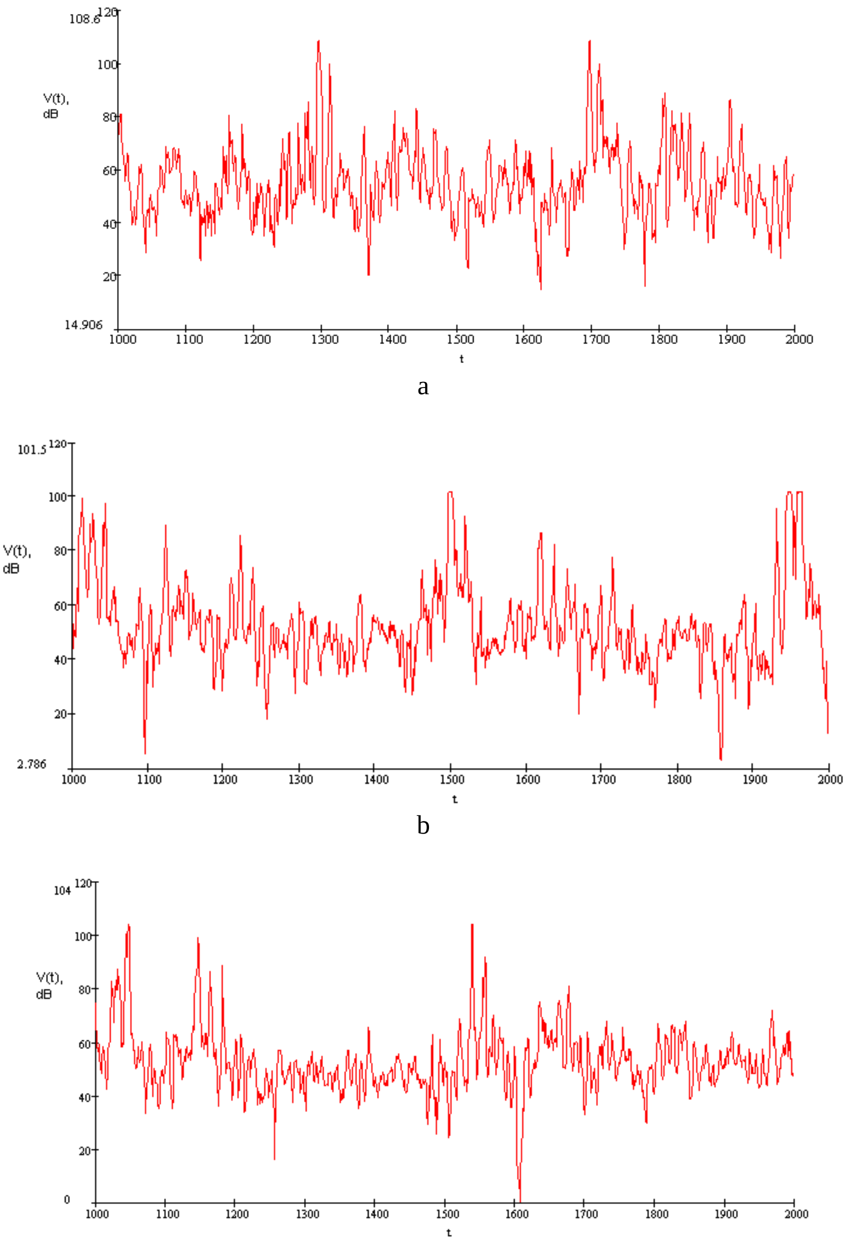

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Figure 2. Graphical representation of the vibration velocity measured on a 5D4 diesel (4Ch8.5/11).

Source: "Compiled by the authors".

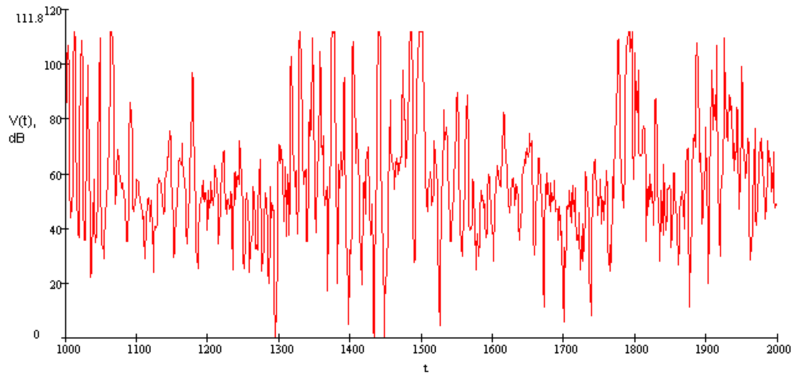

Figure 3. Graphic representation of vibration velocity measured on a 5D4 (4Ch8.5/11) diesel engine by load characteristic: crankshaft rotation speed 1500 rpm; power 0.3 kW.
Source: "Compiled by the authors".

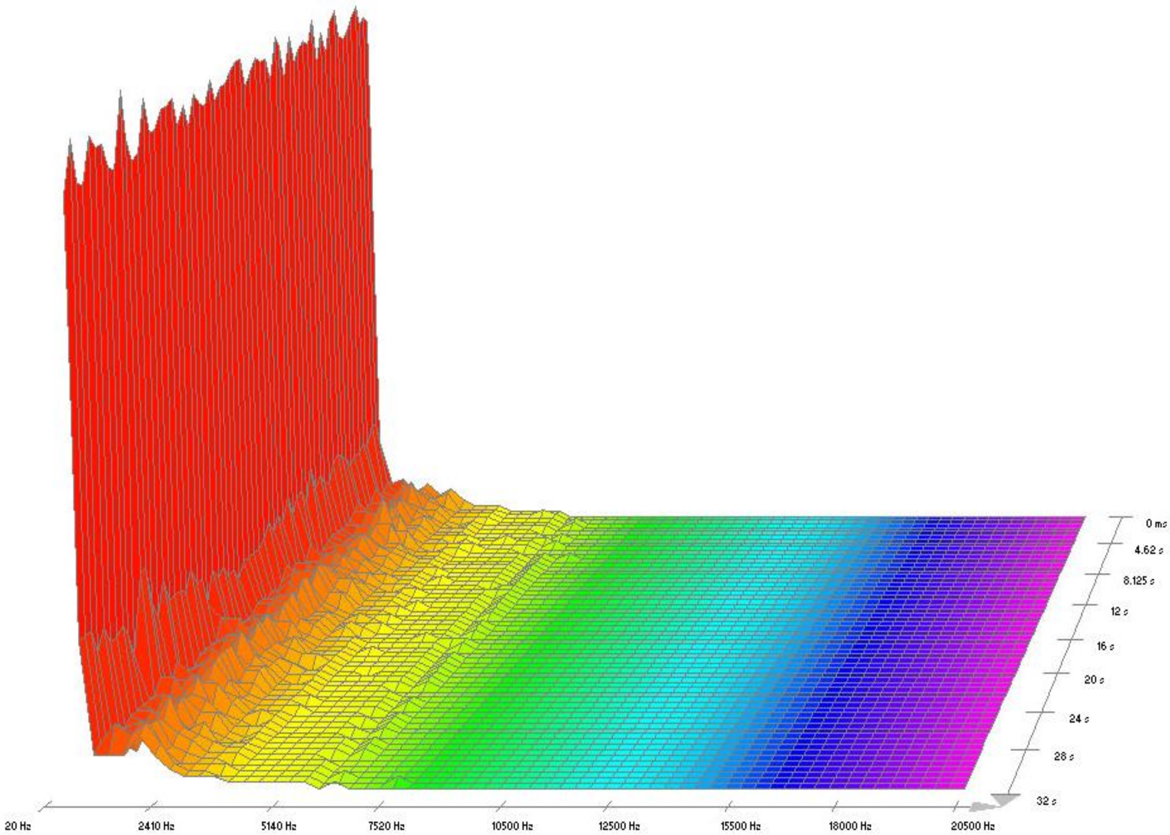

Figure 4. Three-dimensional diagram of the frequency response of the signal over time (variable idle mode).
Source: "Compiled by the authors".

Of greatest interest is the root-mean-square value of vibration velocity (RMS). To determine the vibration parameters, the threshold level is $V_{v_0} = 5 \cdot 10^{-8-8}$ m/s. Let's note that the relative units of vibration are 20-fold decimal logarithms of the vibration velocity root-mean-square value ratios (V_v) to some initial level V_{v_0} (threshold level) and are measured in decibels:

$$L = 20lg\frac{V_v}{V_{v_0}}\text{ (dB)}. \tag{8}$$

As a result of the experiment, it was found that the average value of the vibration level, in idle mode at a frequency of 1500 rpm, is 111,2 dB.

Analysis of the processes occurred during the operation of a diesel engine made it possible to identify the main design and operating parameters that affect the level of vibration activity of a diesel engine. Note that reducing the gap between the piston and the cylinder wall (Tusov, Bezyukov & Afanaseva, 2012) can be considered as one of the ways to reduce vibration and noise emitted directly by the engine, that is, a decrease in the vibration activity of the internal combustion engine can be achieved both by reducing the disturbing forces acting in the engine, and by changing and improving its design.

Investigation and modeling of vibrations caused by the operation of the cylinder-piston group can be carried out using a criterion equation built on the basis of criteria obtained as a result of applying the methods of the similarity theory and analysis of dimensions. The equation of vibration velocity, depending on the intensity of both mechanical and gas-dynamic effects on the details of the skeleton, has the form (Tusov, Bezyukov & Afanaseva, 2012):

$$V = C \cdot \omega \cdot S_{xn} \cdot \left[\frac{P_z}{\rho \cdot h \cdot n^2}\right]^r \cdot \left[\frac{N_{max}\delta}{D_{czvt}+c \cdot D_{czb}}\right]^t \cdot \left[\frac{S_{xn} \cdot D_c^2 \cdot P_z}{D_{czvt}+k \cdot D_{czb}}\right]^m \text{(m/s).} \quad (9)$$

This equation takes into account such important design and operating parameters as piston stroke (S_{xn}), cylinder diameter (D_c), block stiffness (D_{czb}) and bushings (D_{czvt}) of cylinders, the size of the gap between the piston trunk and the mirror of the cylinder bushing (δ), maximum cycle pressure (P_z), maximum lateral force value (N_{max}) and corner frequency ($\omega = 2\pi f$, f – cyclic frequency), as well as unknown coefficients C, r, t, m, c and k, depending on the design features of the internal combustion engine and the damping properties of its materials.

On the basis of the obtained criterion equation, a method for analyzing the vibration activity of the engine, a method for assessing the technical condition of a cylinder-piston group and a method for studying the gap between the piston trunk and the mirror of the cylinder bushing have been developed.

Table 3 shows the values of the vibration level for the diesel engine 4Ch8.5 / 11, calculated using the criterion equation.

Table 3. Values of the vibration level for a diesel engine 4ch8.5 / 11, calculated using the criterion equation.

Frequency, Hz	**Vibration level values calculated on the basis of the criterion equation, dB**		
	Minimum clearance 0.0002 m	**Average clearance value 0.0003 m**	**Limit clearance 0.0005 m**
63	112.902	115.25	118.208
125	110.892	113.24	116.198
250	108.728	111.076	114.034
500	111	113.348	116.306
1000	109.131	111.479	114.437
2000	104.791	107.139	110.097
4000	92.793	95.141	98.099
8000	80.343	82.691	85.649

Source: "Compiled by the authors".

4. DISCUSSION

The relevance of the study is determined by the complexity of establishing a relationship between vibration characteristics and changes in the technical condition of engine parts. Increasing the competitiveness of Russian water transport, shipbuilding and ship engineering is inextricably linked with the improvement of methods for modeling and analyzing complex processes, tools for monitoring, diagnosing and assessing the technical condition of units and mechanisms.

Obviously, the maximum effect in increasing competitiveness can only be achieved with an integrated approach aimed at improving all, or at least most, scientific and technical indicators of a modern vessel. This approach does not exclude the need to select priorities, which, first of all, include the task of improving the scientific and technical level and quality of ship power.

One of the most important indicators of the technical condition of diesel engines is the level and nature of the change in vibration parameters, as the most sensitive to various deviations of the technical condition from the norm. it gives base claim that _ themes work is up-to-date.

The results of the experiment carried out on the diesel engine 4Ch8.5/11 show that the proposed method for analyzing the vibration activity of the internal combustion engine makes it possible to estimate the vibration level and predict the parameters of changes in the cylinder-piston group operation, for example, to calculate the change in the size of the gap between the piston trunk and the bushing mirror. The mounting gap between the piston trunk and the cylinder bushing mirror for this engine is 0.0002 m. The calculated value of the gap at a frequency of 125 Hz is 0.0002109 m, which confirms the reliability of the results obtained and indicates that the calculation error is 5%.

5. CONCLUSION

The need to implement real-time technical systems for monitoring, diagnosing and managing complex objects, information about the state of which is processed, transmitted and used to improve the efficiency of the functioning of these objects in solving the tasks assigned to them, is beyond doubt. One from such applied tasks is the task of ensuring the minimum possible vibration activity of ship power plants (SPP) in all modes of its operation. At present, this problem can be solved only with the use of modern digital systems and tools for continuous monitoring and diagnosing of power plants operating in real time.

The article is devoted to the problem associated with the negative impact on the environment of vibration and noise caused by the operation of ship mechanisms.

The existing methods for calculating the permissible levels of vibration and noise, together with the methods of modeling, the theory of similarity and dimensional analysis, have made it possible to create a method that allows monitoring and assessing the vibration level of a marine internal combustion engine. The method is based on a criterion equation that allows, on the basis of design and operating parameters, to calculate the level of vibration caused by the piston shifting, as well as to predict the change in the gap between the piston bore and the cylinder liner mirror. The experiment was carried out on a diesel engine 5D4 (4Ch8.5/11), it showed the effectiveness of the proposed method (the error did not exceed 5%).

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