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Original scientific paper

Anti-Shock Foundation of Naval Engines for Naval Vessels*

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Abstract: The foundation of engines for naval vessels ensures multi-axis stiffness values and optimal isolation of harmonic vibrations on all axes. Another factor is the shock impulses caused by the battle actions of the ships. Dynamic impulse loads result from underwater or surface detonation of sea mines, a hit by a missile or an artillery shell. Such an impact is a high-energy step impulse. The paper presents the problem of selecting the physical parameters of materials used for shock absorbers. The presented models of material properties enable the presentation of physical and mathematical shock absorbers models for both impulse and harmonic interactions.

Keywords: Shock absorber, Impact loads, Simulations.

1. Introduction

The use of shock absorbers in ship structures has quite a long history. Publications on the effects of underwater explosions and the methodology of testing resistance to impulse impacts appeared after World War II. An additional factor that significantly supported the research was the tests of nuclear weapons used in open sea waters. The effects of underwater shock waves (UNDEX effects) were the primary impulse for computational and simulation research. Unfortunately, the current defense standards, as well as STANAG do not provide information about calculation procedures but only about verifying the impact resistance (1). Published papers and industrial studies mainly focus on the issues of vibration damping and reduction of the hydroacoustic field (2). The small number of scientific publications on shock resistance does not fill the projected gap. N. Klatka conducted the last marine research in Poland in detonation wave identification in 1982, and it was continued by A. Grządziela and Szturomski from 2012 to 2020(3–7). The preliminary research results were the main impetus for detailed research on the UNDEX effect, including research on materials on shock absorbers for naval vessels.

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An underwater explosion is a process of sequential and parallel physical phenomena, leading to an imbalance of the system, which initially consists of the explosive and the liquid medium surrounding it. The process is accompanied by chemical and physical reactions, the release of a significant amount of heat, the formation of gaseous products and the emission of energy in a relatively short time. The first stage of the explosion process is the combustion reaction of the explosive, which results in a detonation wave constituting the surface of the discontinuity and combustion products in the form of gas. The detonation wave created as a result of chemical reactions spreads from the detonation centre to the surface of the gas bladder and transfers energy to the adjacent water molecules. The gas takes the form of a bubble and moves upward at a certain speed. Thus, it is called a gas bladder or gas ball. The front of this wave moves in the initial period of about 2.5 microseconds, at the detonation speed (6000 - 8000 m / s), and after a few milliseconds, it reaches the speed of sound in seawater (approx. 1500 m / s)(8).

The hull's response to the shock wave caused by an underwater explosion depends on whether it is a surface ship or submarine hull. Under the influence of the shock wave, the surface ships will move upwards in the direction perpendicular to the surface of the body of water (there will also be slight displacements in other directions). Submarines will respond to the pressure pulse by moving in the direction of the shockwave(9).

When the shock wave hits the hull, it takes over some of its energy, which is then transferred to the remaining elements of the hull (frames, decks, stringers, etc.). This energy propagates through the fuselage at a specific relative speed, releasing in the form of vibrating energy. For durability reasons, the element of the ship that is most susceptible to shock loading is the propulsion system, mainly the main engines due to their enormous mass.

The literature analysis shows extensive knowledge in the foundation field for engines and machines for marine vessels. Specialist companies offer chocking materials to create permanent cast-in-place machinery supports for all sizes and types of main engines and auxiliary machines(10). Publications indicate the need to ensure the suppression of harmonic vibrations, and selecting the suitable grade depends on the machinery's alignment requirements and the chock's average operating temperature. The presented solutions refer to the tolerated requirements in terms of static stress on a chock. The sum of the engine deadweight and the tension on all bolts, hardener ratio guide, and sometimes the calculations are presented(11).

The vibrations generated by the machine lead to various problems such as shortening the service life of engines through wear of parts and the transmission of these vibrations to other uninsulated adjacent structures, causing problems of noise and vibration transmission as well faster destruction of electronic components. The stiffness of a rubber anti-vibration mount is constant for harmonic excitation, but it changes when a dynamic force is applied to it. This parameter depends on the architecture, the rubber mixture used and even the frequency of excitation.

Proper mounting of the marine gear and propulsion engine in the vessel, once aligned, is critical to maintaining good alignment and consequent smooth, quiet operation and warrants close attention. The marine gear/engine foundation is that part of the boat's structure that supports the propulsion machinery and holds it properly. It generally consists of two longitudinal rails – with liberal transverse bracing – which carry the gear/engine's weight, thrust force, torque reaction, and inertial loads of the gear/engine. It is good design practice to make the foundation support structure as long as possible. This helps to limit hull deflection by distributing the loads over more of the hull length. The entire foundation must be strong enough to withstand operational forces due to torque, thrust, pitching, rolling, and occasional grounding. Since no structure is ideally rigid, the foundation must have greater rigidity than the shaft line so that none of the components of the driveline is stressed beyond their limits when flexing of the hull occurs. Depending on the vessel's hull composition, foundation structures may be of metal (steel or aluminium), wood, or fibreglass.

Generally speaking, dynamic stiffness is always greater than static stiffness, so calculations based on static stiffness may lead to wrong conclusions. However, in some cases, it is possible to reach limits of dynamic stiffness, which are two and even three times greater than the static stiffnesses.

The different stiffnesses of pads for each axis make it possible to offer significant flexibility in the direction perpendicular to the engine's crankshaft. This provides more effective isolation from vibrations of all types of engines.

The marine anti-vibration mounts work correctly when loaded at their 60% load capacity. This way, the vibration isolators offer correct stiffness properties and accept additional harmonic loads without premature deterioration(12).

The foundations of main engines for warships are considered a dynamic system consisting of a machine, parallel antishock pads, a foundation frame, and a floating ship's hull. The presented system indicates the need to analyse the materials' influence as elastic and damping elements. Several materials are used for antishock absorbers in marine applications. Below are the most commonly used materials and their brief characteristics:

1. Natural rubber provides high absorption of harmonic vibrations, resistance to fatigue and attractive elasticity. Therefore, it is often the most recommended material for insulating vibrations because it has the highest

modulus of elasticity (the ability to return to its original form). The disadvantage of natural rubber is the lack of resistance to temperatures above 50 ° C and the sensitivity of chemicals, including saturated hydrocarbons. Nevertheless, this material is one of the most cost-effective polymers to be used in a dry bilge and an effective ventilation system. Mainly applicable in shock absorbers for enclosure gas turbines. Another disadvantage is its high elasticity to impulse interactions, which results in flexible couplings between the motor and the gear.

2. Neoprene has high tensile strength and abrasion resistance. Thus, its main advantage is working in environments with constant exposure to lubricating oil and fuel. It should be noted, however, that continuous contact with saturated hydrocarbons over time degrades the elastomer and changes its physical properties.

3. EPDM (ethylene propylene diene monomer) polymer is sometimes used for foundations in small boats as EPDM resilient parts can have low-temperature requirements and tear resistance. In addition, EPDM is used in cases involving exposure to UV radiation, e.g. for elements directly attached to the open deck. An advantage of EPDM is that it can be manufactured over a wide tensile strength/stiffness range.

4. Silicone rubber is rarely used due to its low tear and abrasion resistance. However, research is currently underway on modifying silicone rubber to increase its damping coefficient, which will increase its attractiveness as a material for shock absorbers.

In conclusion, the most commonly used materials for shock absorbers are rubber materials, mixtures of rubber with polymeric materials, and more and more recently, specialized polymeric materials. The use of rubber as a material for shock absorbers brings many benefits, the most important of which are:

- Provides excellent damping and energy absorption of harmonic vibrations and pulse shock.
- Excellent noise and vibration damping and heat dissipation as dissipation of damping energy.
- Provides stability with appropriate hardness and initial deflection,
- A wide range of elastomers with different characteristics are available.
- Long service life even under constant pressure, vibration or vibration.
- Resistant to oil, water, ozone and other harmful factors.

This paper shows the importance of testing materials used for shock absorbers. At high deformation rates, catalogue physical properties may bring results far from those predicted in the calculations. Furthermore, due to the possible resonance effect during a UNDEX event, knowledge of the stiffness and damping of materials on shock absorbers may be crucial for maintaining the technical efficiency of engines.

2. Materials physical properties

Rubber or metal-rubber Vibro-isolators are relatively cheap, and it is this fact makes the most common. Rubber has both advantages and disadvantages. The most crucial benefit of rubber is, as already mentioned, the relatively high damping factor γ, much more significant than the factor characteristic for steel springs (γ ~ 0.005). In the case of rubber, the value of the damping coefficient γ depends primarily on the hardness of the rubber. Although it also depends on the frequency of forced vibrations, the average values of this coefficient can be made dependent on the Shore hardness, as shown in Fig. 1. The spring constant of a rubber Vibro-isolator also depends on the hardness of the rubber, but it changes with the static load in a non-linear manner. It defines the constant of elasticity as.

$$k = \frac{dQ}{dy} \ \frac{N}{m}; \tag{1}$$

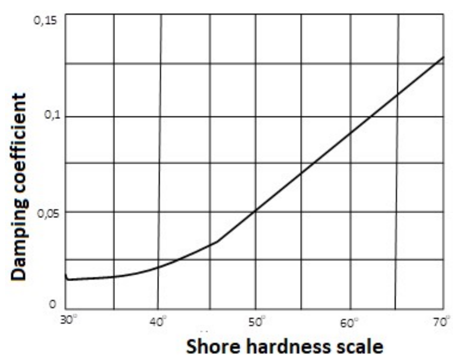

Fig. 1. *Diagram of dimensionless damping coefficient γ as a function of rubber hardness on the Shore scale*(13)

Figure 2 shows an example of the so-called complex characteristic, which consists of the fact that the value of k increases with increasing load. If the static load, expressed by the point Z0 or Z, does not exceed a specific value of Q, the characteristics of the Vibro-isolator can be treated as linear because k has a constant value in this load range.

But if the static load increases, e.g. to Q1, then the frequency of free vibrations of the system will be much higher because the value of k is then expressed as:

$$k' = \frac{Q_1}{\delta'}, \tag{2}$$

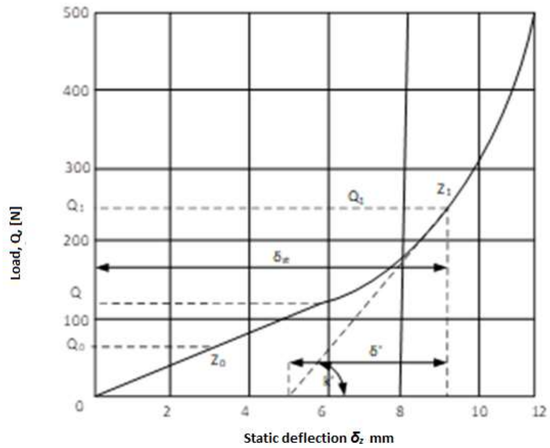

Fig. 2. *An example diagram of a nonlinear rubber characteristic*(13)

As the vibration amplitudes (dynamic displacements) are very small compared to the static deflection, changes in the dynamic load cause only a slight oscillation of the vibration isolator deflection in the vicinity of point Z1. The free vibration frequency of the system is then:

$$f' = \frac{0{,}5}{\sqrt{\delta'}} \text{ Hz} \tag{3}$$

and it is much greater than the ratio would suggest Q_1/δ_{st}.

Faultless calculation of the Vibro-isolator is possible only when the constructor has the appropriate experimental data. In general, the linear characteristic of a rubber Vibro-isolator can be assumed only when the static deflection δst is very small, i.e. it does not exceed 10% of the thickness (height) of the elastic element. Therefore, if not supported by relevant experimental data, the calculation of a rubber Vibro-isolator is only an approximate calculation. The biggest obstacle to obtaining accurate results is the discrepancy between the static and dynamic elastic modulus.

The dynamic modulus of rubber elasticity (and thus the active elasticity constant) is a function of the ingredients used in its production. For natural rubber, the dynamic to static modulus ratio varies within 1.2-1.4; for synthetic rubber, the ratio is 1.4-2.0. The free vibration frequency of the system, calculated taking into account the dynamic modulus, is, therefore, higher than the free vibration frequency calculated based on the static modulus. In the calculations of Vibro-isolators, the ratio of static modules is assumed(14):

$$\frac{E_{st}}{G_{st}} \approx 6{,}5 \tag{4}$$

where: E_{st} - Young's modulus, i.e. the modulus of longitudinal elasticity, MPa,
G_{st} - Kirchhoff modulus, i.e. shear modulus, MPa.

The dependence of these modules on the hardness of the rubber is shown in Figures 3 and 4. However, it should be remembered that the actual values of the static modules may differ from the values read from the charts (up to ± 15%) because the properties of rubber are not uniform, even with the same hardness. The ratio of dynamic modulus to static v is usually taken as a constant value in practice; for soft rubbers with a hardness lower than 550 Sh, v - 1.25, and hard rubbers, i.e. above 550 Sh - v = 1.75, where:

$$v = \frac{G_d}{G_{st}} = \frac{E_d}{E_{st}} \tag{5}$$

The modulus ratio v varies almost linearly, as shown in Figure 5. Therefore, compliance with the measurement results obtained after installing the machine is possible only if the vibration isolators are mass-produced as standard elements, provided with appropriate characteristics; otherwise, verification by measurement is always necessary.

The most commonly used material for the construction of ship shock absorbers is rubber of various hardness. Rubber is one of the elastomers. The spring element for rubber should be connected to other shock absorber elements only on the load-bearing surfaces. In contrast, the remaining parts should be free so that the material can deform in different directions. Parts of a shock absorber made of rubber are most often combined with metal elements that enable correct assembly.

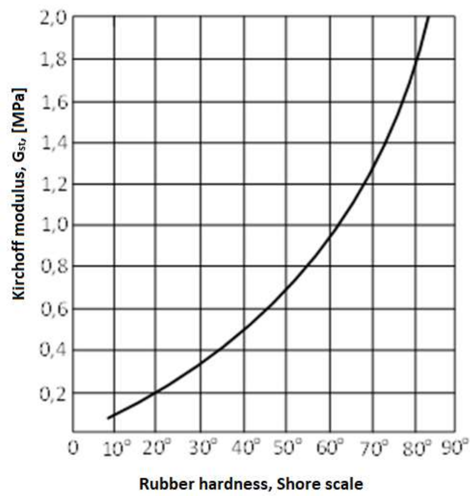

Fig. 3. *Diagram of the Kirchhoff static modulus as a function of rubber hardness*(13)

Such joints are made by the vulcanization method because the strength of the joint is practically equal to the strength of the rubber itself. The deflection of the shock absorber s under a statically loaded force F - see Figure 6, can be determined based on the formula:

$$s = \frac{4Fh}{E\pi d^2}\ , m \tag{6}$$

To maintain the safety conditions concerning damage to the shock absorber and the correct functionality, shock absorber manufacturers use the following relationship:

$$s < 0{,}1h. \tag{7}$$

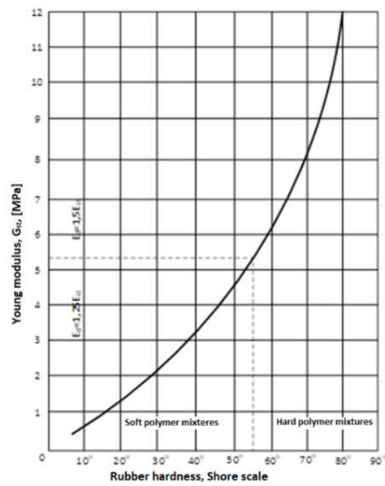

Fig. 4. *Diagram of the Young static modulus as a function of rubber hardness*(13)

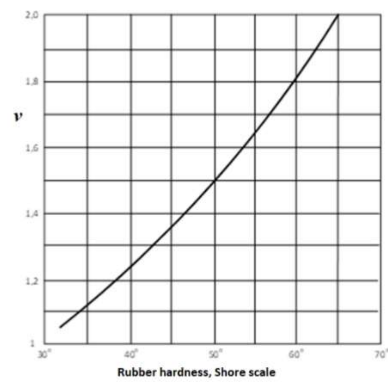

Fig. 5. *Diagram of the Ratio of dynamic to static modulus v as a function of rubber hardness* (13)

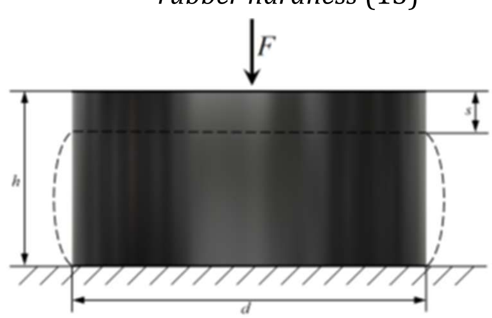

Fig. 6. *Shock absorber deflection under force F*(13).

For a given load is the maximum allowable force Fdop, the value of which should not be exceeded, also for safety reasons concerning damage. This force depends on the cross-sectional area of the shock absorber and the allowable stresses for the given rubber properties:

$$F_{dop} = \frac{\pi d^2}{4} \sigma_{dop}; \tag{8}$$

For a correct analytical solution of a rubber shock absorber, its shape coefficient k should be determined. This coefficient occurs where the dimension or shape of the loaded elements changes, where the stress distribution changes:

$$k = \frac{d}{4h_s}; \tag{9}$$

where: h_s - shock absorber height change in tension or compression.

As a result of the above analytical solutions for a rubber absorber, it is possible to determine the relationship between the calculated values - Young's Modulus, hardness according to the Shore scale and the shape factor.

3. Laboratory identification of the dynamic model forced with impulse load

Exemplary laboratory identification of a model of dynamic foundation of a ship engine loaded with an impulse from underwater detonation consists of three stages. The first part of the research on metal-rubber shock absorbers focused on four aspects, namely the measurement:

- deflection height,
- the height of the rebound.

The site's description is presented in Figure 7, and the research methodology is introduced in detail in the earlier publication of the Authors(13).

Two different shock absorbers were tested, all of them having the exact geometrical dimensions, i.e. cylinder diameter 20 mm and height 40 mm. In addition, five other materials were tested, each with three harnesses (55 Sha, 65 Sha, 75 Sha):

• NBR rubber, i.e. acrylonitrile-butadiene rubber.

• NR rubber (natural). Natural rubber is a flexible hydrocarbon polymer derived from latex. Latex is a milky colloid, the source of which is rubber trees.

The tests were carried out on the hammer drop machine shown in Figure 8. The mass of the falling element, including the mass of the shock absorber attached to it, was 2350 g. During the tests, vibration accelerations on the machine foundation were recorded using the BKSV 4514-B accelerometer. The analysis of vibration parameters was carried out in the Pulse Reflex environment. First, the height of deflections and rebound of shock absorbers were determined. Then, there were subjected to a free fall from a height of 300 and 450 mm. During the deflection distance and the height of reflection tests, a fast camera was used with automatic detection of the object in the frame and the image recording speed of 960 frames/second. Recorded movies were processed to obtain an image of the maximum deflection of the shock absorber or the full height of the reflection. A prepared measure was placed at the same distance from the camera lens as the falling shock absorber, which served as the distance standard for further processing. Thanks to the frame by frame processing of the images, it was also possible to determine the contact time of the shock absorber with the foundation of the drop hammer.

The next step in processing the images to determine the height of deflection and reflection was the cropping of the picture - Figure 9, to the width of the adopted distance standard for testing the deflection of shock absorbers, 100 mm was assumed, and for testing the height of rebound equal to 300 and 450 mm. In Autodesk Inventor, the prepared frame was inserted into the running sketch function as an image file.

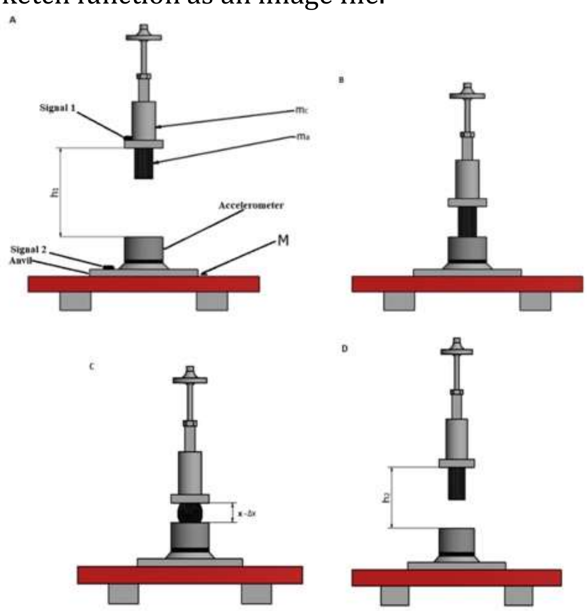

Fig. 7. *Diagrams of theoretical foundations for the conducted research*(13)

After giving it the assumed width of 100 mm, the scale 1: 1 was obtained, so the line drawn from the beginning to the end of the deflected shock absorber defined its deflection distance.

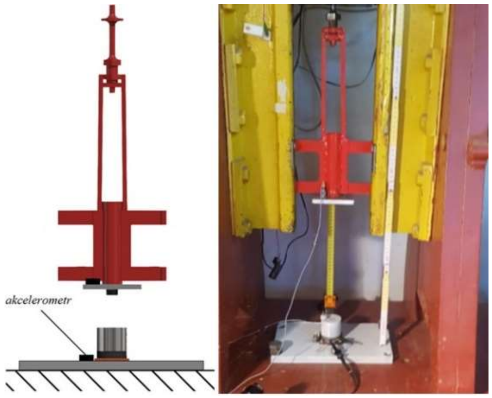

Fig. 8. *View of the drop hammer stand (right) and its virtual model (left).*

Fig. 9. *The maximum deflection of the shock absorber as read during post-processing*

The maximum deflection values of metal-rubber shock absorbers made of two different materials with three stiffnesses obtained during the tests are shown in Table 1.

Table 1. *The maximum values of deflection of shock absorbers were obtained during tests on the stand*

Material Sh A	Test 1 : 300 mm Deflection [mm]	Test 2: 300 mm Deflection [mm]	Test 3 : 450 mm Deflection [mm]	Test 4: 450 mm Deflection [mm]
NBR 55	25,4	25,7	22,8	23,1
NBR 65	27,6	26,7	24,3	24,2
NBR 75	31,9	32,0	30,9	27,1
NR 55	22,3	21,8	19,0	19,1
NR 65	22,6	22,8	20,5	20,4
NR 75	27,3	26,7	24,1	23,4

Tests were performed for both materials types. The dependence of more significant deflection in the case of lower stiffness expressed in the Shore A scale is visible here.

Table 2. *The maximum rebound values of shock absorbers were obtained during the tests.*

Material Sh A	Free fall height [mm]	Rebound height [mm]			Average reflection height [mm]
NBR 55	300	152	153	152	152
	450	214	213	217	215
NBR 65	300	125	124	126	125
	450	175	178	179	177
NBR 75	300	96	97	97	97
	450	130	139	141	137
NR 55	300	164	166	168	166
	450	213	247	251	237
NR 65	300	145	166	162	158
	450	230	234	245	236
NR 75	300	141	145	143	143
	450	175	202	209	195

This dependence occurs for all tested materials. The data collected in Table 1 also shows significant changes in the height of the shock absorber deflection depending on the material used.

The data presented in Table 2, shows a strong relationship between the stiffness of the material and the height of the rebound. The lower the stiffness of the material, the greater the rebound height, which clearly shows that the dissipation of the impact energy in the case of lower Shore stiffness of the material reaches smaller values.

The presented research results indicate the need for an individual adjustment of the shock absorber to dampen impulse loads. Catalogue selection of shock absorbers makes sense when the machine will be loaded only with harmonic interactions that require analysis in terms of fatigue and/or environmental wear.

4. Conclusion

The use of shock absorbers for the foundation of main engines on warships imposes additional technical requirements to suppress harmonics and shock loads. The conducted tests confirmed the necessity to carry out individual verification procedures of materials used in shock absorbers. The research results indicate that the essential factor for protecting against a shock impact is the damping factor, which determines the dissipated elastic energy. As a result, the main engine is not too displaced from the working position on the foundation. There is also a contradiction of the expected damping and deformation; hence the choice of shock absorbers is a computational process and the optimization of allowable deformations. However, selecting the shock absorber and its configuration in the foundation based on catalogues will result in a very high risk for the engines. It is caused in terms of resonance and the lack of resistance to the effects of UNDEX (Underwater Explosion).

The presented tests should be verified by SRS (Shock Response Spectrum) tests and the analysis of changes in physical parameters as a function of fatigue loads(15). The last two factors are currently being researched and analysed, which will be presented shortly in the following Authors publication.

Another factor confirming the need for a precise method of calculating shock absorbers is the need to reduce the physical fields by naval vessels. Warships in the design and operation phase are tested on test ranges to assess the hydroacoustic field emissions from the lower hemisphere to the marine environment. The primary source of emission is the ship's propeller, the emission of which has components from the propeller geometry and structure and vibration energy transmitted through the shaft line from the main engine. Correct installation of the engine also reduces the acoustic emission from the foundation through the hull to the water.

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