

Vibration and noise minimisation in living and working ship compartments

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ABSTRACT

The article discusses the problem of minimisation of the noise emitted by mechanical devices installed on a ship to its living and working compartments. Basic features of the abovementioned mechanical systems are defined taking into account frequencies generated by them. Passive and active methods of noise minimisation via interference into the propagation path are presented. In the conclusion, expected effects are formulated.

Key words: vibroacoustic signals generated by mechanical devices on a ship;
minimisation of vibroacoustic signals

INTRODUCTION

A ship is a complex technical object in which the principles of propagation of acoustic signals are complicated. Therefore minimising the propagation of internal noise is a difficult task, due to specific conditions taking place on floating objects of this type. The hull is a separate structure inside which numerous sources of strong noise are located. This is especially true for the engine room, limited in space, in which the access is to be secured to the main engine, gear, propeller shaft and auxiliary units, such as electric current generator, pumps, fans, etc. These limitations make potential use of some methods to minimise vibroacoustic effects accompanying the operation of the above devices extremely difficult. Of high importance in ship noise fighting is the identification of both the sources and propagation paths of the vibroacoustic energy acting in a direct way, or having the form of reflection and/or diffraction waves, which generate sound in the air and materials.

The main source of vibroacoustic signals on a ship are internal combustion engines, i.e. the main engine and auxiliary engines driving, for instance, electric current generators. Other sources of remarkable vibrations of this type include auxiliary devices such as pumps, compressors, hydraulic systems, main engine gears, propellers, and air conditioning and ventilation systems. The contribution of particular sources in the total level of noise in the engine room depends on their location inside the hull and the type and dimensions of the ship. These sources are most frequently located in the after body, from which they act

on the living and working ship compartments. The dominating sounds in the engine room compartment are the air sounds – direct and reflected. In a larger distance from the source, the dominating effects in the noise are material sounds.

Due to mechanical power transmission, the vibrating main engine, excited by the action of varying forces, is the main source of the acoustic energy of both the air and material type. The exciting forces come from:

- ✦ pressure pulsations in the inlet and outlet ducts
- ✦ pressure changes in cylinders during the combustion process
- ✦ operation of the timing gear
- ✦ pressure changes in fuel and lubrication systems
- ✦ inertia of moving engine elements
- ✦ toothed gears
- ✦ auxiliary devices.

Frequencies of these forces are connected with the rotational speed of the engine crankshaft, and are defined as [2, 3]:

$$f_N = \frac{nZ_C}{60s} k \quad (1)$$

where:

- n – rotational speed of the crankshaft [rev/min]
- Z_C – number of cylinders
- s = 2 – for four-stroke engines
- k = 0,5; 1; 2; 3... – harmonics of the exciting forces.

After defining the rotational frequency caused by the unbalance of the rotating masses as:

$$f_N = \frac{n}{60} \quad (2)$$

we get the frequencies connected with the operation of the engine sub-assemblies:

❖ camshaft frequency

$$f_r = f_n i = 0.5 f_n \text{ [Hz]} \quad (3)$$

where:

i – camshaft drive transmission ratio ($i = 0,5$).

❖ valve closing frequency

$$f_z = f_n \frac{Z_z}{s} \text{ [Hz]} \quad (4)$$

where:

Z_z – number of separately working valves.

❖ toothed gear frequency

$$f_p = f_n z \text{ [Hz]} \quad (5)$$

where:

z – number of pinion teeth.

❖ fan blade frequency

$$f_w = f_n l \text{ [Hz]} \quad (6)$$

where:

l – number of blades.

Like for fans, the frequencies of the impeller pumps are connected with the number of pump blades l and the frequency f'_n of the driving motor revolutions.

$$f_p = f'_n l \text{ [Hz]} \quad (7)$$

The frequency of the electric current generators is most frequently equal to 50 [Hz].

It results from the above relations that different engine sub-assemblies generate the same frequencies, in particular when the energy spectrum reveals harmonic frequencies of the exciting forces caused by nonlinearities of the mechanical systems.

The above list of exciting sources includes low-frequency sources, below 100 [Hz] (main engine with the exhaust system and generators) and high-frequency forces, above 1 kHz (fans, pumps, toothed gears, etc.) This makes using any general method to minimise the vibrations and noise generated by those devices very difficult.

The level of the acoustic pressure generated by the engines used on ships can be approximately calculated from the relation [2]:

$$L_p = 12 \lg N + 30 \lg n - 10.7 \text{ [dB]} \quad (8)$$

where:

N – engine power [kW]

n – rotational speed of the crankshaft [rev/min].

This level depends on a number of design factors, such as, for instance, geometrical dimensions of the pistons, type of hull material, etc.

The level of the acoustic power of the low- and medium-power Diesel engines is approximately equal to [2]

$$L_N = 59 + 10 \lg N_z + 10 \lg n_z - 30 \lg \frac{m}{n} \pm 4 \text{ [dB]} \quad (9)$$

where: N_z corresponds to the nominal engine power [kW] recorded at its nominal rotational speed n_z [rev/min].

Major part of the acoustic energy is transmitted from the engine via a material path. The vibrations are transmitted to the elements of the steel hull structure not only through the foundations, but also through the propeller shaft, engine fittings, fuel, lubrication and cooling systems, and other auxiliary devices. Low-speed engines working with rotational speed ranging between 100 and 250 rev/min emit vibroacoustic signals ranging from a few to about 200 Hz, while the medium-speed engines ($n = 350 - 1000$ rev/min) cooperating with the toothed gear – up to 2000 Hz. Moreover, these engines, when cooperating with a gear, reach the sound level higher by up to 15 dB than that recorded for the low-speed engines [6].

The origin of the acoustic effects generated during the operation of the ship propeller is turbulence and cavitation, which both appear on the edges of the propeller when the circumferential velocities are sufficiently high. The cavitation noise spectrum is within the frequency limits of 20 – 500 Hz. The turbulence and the cavitation act on the ship hull and make it vibrate. The excited large-area plate panels become secondary sources of the broadband noise of large acoustic power.

MINIMISATION OF THE LEVEL OF NOISE AND VIBRATIONS GENERATED IN THE ENGINE ROOM

The vibroacoustic signals emitted by the driving system can be minimised by:

- ☆ changes of aerodynamic coefficients at device's inlet and outlet and inside its working space,
- ☆ minimisation of the exciting forces and their spectra.

Changes of the aerodynamic conditions can be obtained by the use of covers, to damp the vibrations, and silencers at inlets and outlets of the main and auxiliary engines, while the exciting forces can be minimised using vibration absorbers.

Minimisation of the vibration level

Damping is connected with the dissipation of the mechanical energy converted, for instance, to the thermal energy, i.e. with decreasing the general efficiency of device's operation. That is why the damping is only introduced when the minimisation cannot be obtained via structural and parametrical modifications. Such an approach accompanies active methods, unlike those based on changes of the transmission path – by introducing vibroinsulation. Three groups of methods of structure modification can be named. The first method consists in introducing additional internal connections to the system using elastic absorbing elements (disc connections, for instance). In the second method, additional masses (Frame type, for instance) are introduced to the object, while in the third method the required effect is obtained by rearranging the structure continuity using intermediate flexible elements (vibroinsulators).

The main disadvantage of the dynamic vibration eliminators is that they can only be used for harmonic excitations (they are tuned to a precisely defined frequency). They do not bring profits in the mechanical systems used in unsteady conditions. Similar disadvantages are revealed by the parametric modification of a mechanical system in which the parameters are the load vector variables: inertia „ M ”, stiffness „ K ” and damping „ C ” of the system. During its operation a real nonlinear mechanical system is subject to the action of an exciting force having a broad spectrum and large number of harmonics. In order to reduce the amplitude of the vibrations, broad-spectrum exciting forces are to be decreased.

In mechanical conditions the damping forces are, as a rule, smaller than the elasticity and inertia forces. These forces are directed opposite to the velocity vectors and tend to decrease them, thus decreasing the kinetic energy of the system. This energy depends on the type of friction (viscous, coulomb or material friction).

Methods which are frequently used for minimising vibrations of a mechanical system include passive (displacement) vibroinsulation and active (forced) vibroinsulation. However, it is passive methods which are most frequently used in practice. A machine fixed on vibroinsulators has 6 degrees of freedom, which means 6 resonance frequencies in case of a linear system. The vibroinsulators are selected in such a way as to eliminate the machine operation in the resonance band. There exist a number of methods in which mechanical devices can be connected with the foundations using vibroinsulators. These methods include lifting, vertical, skewed, and mixed systems.

Various materials are used for vibroinsulation. The elements used for heavy mechanical devices are steel, rubber, or pneumatic springs. In case of large loads, steel springs are most often used, as their parameters can be easily calculated. The springs used in vibroinsulation have linear characteristics, in which the force is proportional to the deflection. The type of the applied springs (coil springs, disc springs, etc.) depends on the assumed vibroinsulation system.

Rubber springs are used in cases where high frequencies of the exciting forces and low loads are observed, and their use is limited to springing. Rubber elements under constant load should not exceed 15% for shearing, 25÷40% for hard mixtures, and 40÷70% for soft mixtures. Their hardness is to be within the limits of 50÷60 Sh. Pneumatic springs are rarely used in industrial practice because of their large geometric dimensions.

Effectiveness of vibroinsulation

Irrelevant of the excitation characteristics, the effectiveness of the vibroinsulation can be defined using the model of:

- ♦ displacement vibroinsulation,
- ♦ forced vibroinsulation.

The vibroinsulation model for the kinetic excitation with the displacement $z(t)$ can be illustrated using a simple single-mass system, with mass „ m ”, damping „ k ” and elasticity „ c ”. This system is shown in Fig. 1.

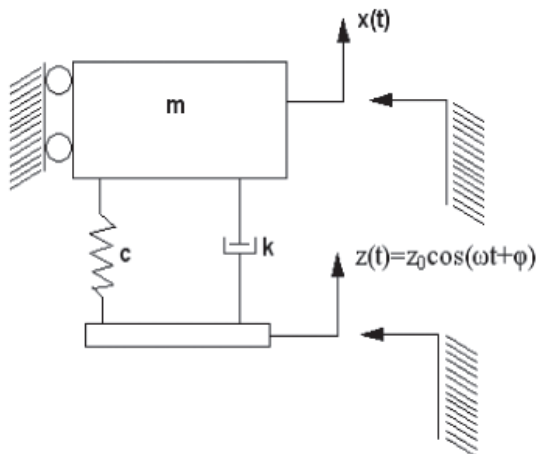


Fig. 1. Displacement vibroinsulation model

In case of the forced vibroinsulation model, the kinetic excitations are converted to forces, as illustrated in Fig. 2.

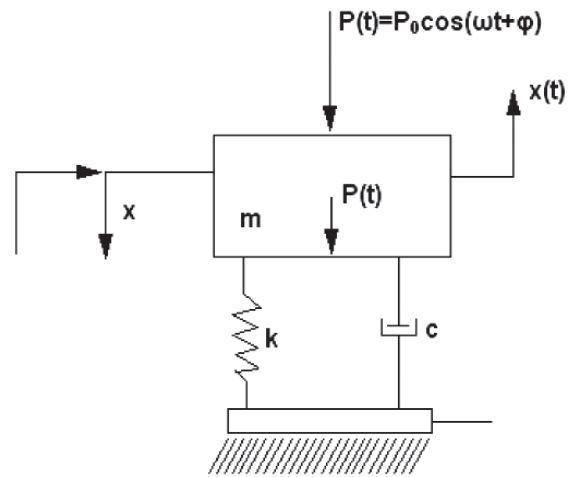


Fig. 2. Forced vibroinsulation model

The vibroinsulation criterion is defined by the relation:

$$T_x = \left| \frac{x(t)}{Z(t)} \right| \quad (10)$$

The adopted criterion is to fulfil the conditions:

$$T_p \leq 1 \text{ and } T_x \leq 1$$

When $T_p = T_x = T$ i.e. the vibration displacement coefficients are the same, $T < 1$ is practically observed when the ratio between the vibration frequency ω and the free vibration frequency ω_0 is larger than 3 [2]. Introducing the relative damping factor ξ to the evaluation we can conclude that when $\xi \leq 0.1$ the value of the transmission coefficient T practically does not depend on damping. These cases happen most frequently in technical practice. When $\xi = 1$ the vibroinsulation effect disappears. The transmission coefficient depends almost solely on damping.

The vibroinsulation systems can be divided into active and passive.

The principle of action of the active vibroinsulation is to generate control forces which act on the vibroinsulation object. The passive system can only dissipate the vibration energy or store it temporarily. The active systems have external energy sources, which, controlled automatically, can deliver or absorb the energy. In general, the active vibroinsulation methods can be divided into excitation controlled methods and vibration field parameter controlled methods. The vibroinsulation system of this type includes an instrument that measures vibration parameters and controls the external power source and the final control element. This vibroinsulation system is a complex automatic control system – a complex and expensive technical system, which limits its application. Cheaper are semi-active systems, which do not generate forces, but modify the controlled vibroinsulator's damping and elasticity parameters. The passive systems, on the other hand, can only separate the energy or temporarily store it [4].

Minimisation of the noise level

The incident acoustic wave approaching a partition can be treated as a beam of acoustic energy acting across the border between two media of different impedance. The total energy E_c of the beam can be divided into the absorbed energy E_p , the reflected energy E_o and the energy E_{pr} penetrating through the partition.

$$E_c = E_p + E_o + E_{pr} \quad (11)$$

The measure of the acoustic energy is the intensity of sound, therefore the above relation can be written as the sum

of intensities of sound components, using identical subscripts for sound absorption, reflection, and penetration through the partition.

$$I_c = I_p + I_o + I_{pr} \text{ [W/m}^2\text{]} \quad (12)$$

In acoustics, the above phenomenon is evaluated using dimensionless estimators bearing the name of coefficients defined in the following way [3, 5]:

✦ sound absorption coefficient

$$\alpha = I_p / I_c$$

✦ acoustic wave reflection coefficient

$$\beta = I_o / I_c$$

✦ sound penetration coefficient

$$\gamma = I_{pr} / I_c$$

Each of those coefficients takes values from between 0 and 1, and their total energy balance is to be equal to 1. The spectra of the above coefficients are determined in acoustic tests performed for materials used for both one-layer and multi-layer partitions.

Like for vibrations, the noise can also be minimised using active and passive methods. In the passive methods, the minimisation of the effect of the noise on human beings can be achieved via:

- ⇒ limiting the noise emitted by the source of sound
- ⇒ reducing the acoustic energy on its propagation path.

The vibroacoustic energy of the noise sources can be reduced without interference into the technological process by:

- ➔ changing aero- and hydrodynamic conditions of machine operation,
- ➔ reducing propagation efficiency coefficients.

The term “change of medium flow conditions in the source” is to be understood as the change of the velocity of the flowing medium and the resultant noise (acoustic power of the acoustic noise is proportional to 6 ÷ 8th power of the gas flow velocity). On the other hand, the reduction of the propagation efficiency coefficient is connected with changes of materials, protecting coats, intermediate emission factors, etc. In case of internal combustion engines, the vibroacoustic analysis makes the basis for formulating modifications to be introduced to inlet and outlet ducts by changing their geometries to decrease the energy of the flowing media. This refers to the use of noise eliminators, such as silencers, of instance.

The silencers can be divided, in general, into absorptive silencers and reflective silencers. The absorptive silencers act against the propagation of the acoustic wave by absorbing remarkable part of its acoustic energy. In most cases this effect is obtained by lining relevant surfaces with sound absorbing material. Silencers of this type can be used as suction duct silencers.

The principle of operation of the reflective silencers consists in installing an acoustic discontinuity in the channel. The acoustic resistance of this discontinuity is either much smaller or much larger than the characteristic resistance of the channel. Most often, the discontinuity of this type takes a form of a single or double stepwise change of channel diameter (chamber silencers, or resonator silencers). Silencers of this type can be used as exhaust duct silencers.

It is noteworthy that the silencer reveals high efficiency when its mobility (the inverse of the impedance) is much higher than the sum of inlet and outlet mobilities. Other recommended ways of noise minimisation include:

- machine body vibration damping coatings
- sound absorbing housings
- acoustic screens
- changes of acoustic absorption capacity of the compartment
- changes of insulating ability of the partition.

The coatings which damp the vibration of the engine, propulsion system, and current generator units are not used, in practice, on ships, as they would make continuous supervision of operation of those machines very difficult. The sound absorbing housings cover the entire machine. From inside, the material should reveal high sound absorption coefficient, which is obtained by covering these surfaces with a layer of damping material. The walls of the housing should reveal high reflection coefficient β , which, along with the inner damping material, produces the effect equal to [2]

$$\Delta L_u = L_1 - L_2 = \beta_u = 20 \log f \rho + 10 \log \alpha + \rho \quad (13)$$

where:

- ρ – surface density of the partition
- f – frequency of the emitted sound [Hz]
- α – sound absorption coefficient
- L_1, L_2 – sound level in front of and behind the partition [dB].

The effect of the use of housings depends on the tightness of housing elements. Unlike the sound insulation housings, the acoustic screens are not practically used inside the ship engine rooms.

Good results in noise minimisation in a closed compartment can be obtained by changing the sound absorption coefficient α . The effect of such changes is shown in Fig. 3.

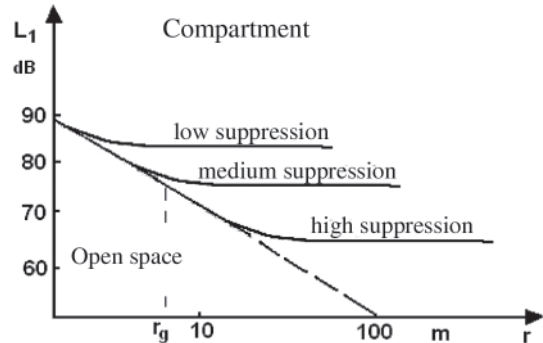


Fig. 3. Noise level reduction in the open space and in the compartments revealing different suppression levels

In the stray field, at a distance larger that the limiting value r_g from the source the level of noise depends on the acoustic absorption capacity A [m²] of the compartment. After denoting by α the average coefficient of sound absorption in the compartment with constraints of total surface S , we can write the definition [5]

$$A = S\alpha \text{ [m}^2\text{]} \quad (14)$$

After increasing the sound absorption coefficient from α_1 to α_2 we obtain the effect of noise reduction by

$$\Delta L = 10 \log \frac{R_2}{R_1} \text{ [dB]} \quad (15)$$

where:

$$R_{1,2} = \frac{A_{1,2}}{1 - \bar{\alpha}_{1,2}} \text{ [m}^2\text{]} \quad (16)$$

The penetration of the acoustic energy through partitions is a complicated phenomenon. It is assumed that the sound

penetration is affected by dynamic phenomena, as well as by constructional and material characteristics of the partition. The applied coefficient – acoustic insulation ability of the partition is of approximate nature. The effect of material characteristics on the insulating ability of the partition is shown in Fig.4.

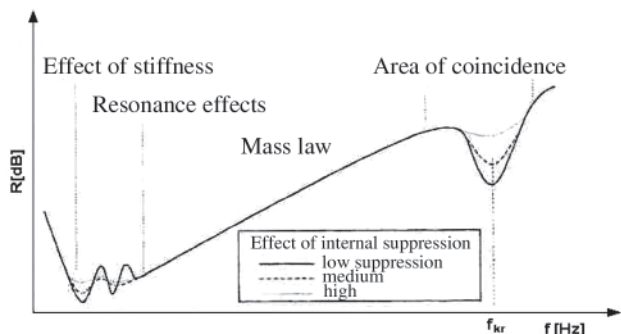


Fig. 4. The effect of material characteristics of the partition on its acoustic insulation ability with respect to air sounds.

Physical properties of the materials which affect the insulation characteristic include: elasticity, specific gravity, internal suppression of the materials, etc. Within the low frequency range the insulating ability mainly depends on the stiffness of the partition. Then the effect of the free vibration frequencies f_0 takes place. At the frequency $2f_0$ – the dominating effect is the effect of mass, and then, for higher frequencies, the effect of coincidence is observed. The acoustic insulation ability of the uniform (homogeneous and isotropic) partition is equal to [6].

$$R = 20 \lg m + 20 \lg f - C \text{ [dB]} \quad (17)$$

where:

- f – frequency [Hz]
- m – superficial mass of the partition [kg/m^2]
- C – constant coefficient (equal to 48 for normal atmospheric conditions).

It results from the equation bearing the name of the *mass law*, that the acoustic insulation ability of the uniform (homogeneous and isotropic) partition is proportional to the superficial mass of the partition expressed by the weight of one m^2 of its surface and increases with the frequency, approximately by 6dB/octave.

Relating the acoustic insulation ability of the partition only to its superficial mass is large simplification of the problem. Errors in the obtained results are caused by the phenomenon consisting in the presence, in some conditions of the sound wave incidence on the plate, of the equality between the velocity c_g of the wave diffracted in the plate and the velocity c_0 of the incident acoustic wave approaching the plate ($c_g = c_0$).

The condensation phenomenon acts towards the reduction of the input impedance of the plate, i.e. the value of the acoustic insulation ability. Theoretically, this value is expected to decrease to zero for the coincidence frequency, but because of internal material losses of the partition, only stepwise decrease of the insulating ability is observed. In hard partitions it reaches a few or, sometimes, up to twenty decibels. For the area of coincidence ($f > f_{kr}$) the scale of insulation decrease depends on internal material losses, defined by the coefficient η .

The *coincidence frequency* or the *critical frequency* is determined from the formula [5].

$$f_{kr} = \frac{c_0^2}{2\pi} \sqrt{\frac{m}{B}} = \frac{c_0^2}{2\pi h} \sqrt{\frac{12\rho(1-\nu^2)}{E}} \text{ [Hz]} \quad (18)$$

where:

- c_0 – the sound speed in the air [m/s]
- m – superficial mass of the plate [kg/m^2]
- B – plate stiffness to bending to a cylindrical surface $B = Eh^3/12(1-\nu^2)$ [Nm]
- ρ – plate material density [kg/m^3]
- h – plate thickness [m]
- ν – Poisson ratio.

Decreasing the mass of the partition, without simultaneous reduction of its acoustic insulation ability, is possible by building a multi-layer partition. To obtain the maximum possible insulating ability of the multi-layer partition, certain conditions are to be fulfilled with respect to the number of layers, thickness, resultant stiffness E, and superficial mass of the partition. Due to the coincidence phenomenon, the partition should reveal low stiffness to bending B.

When designing multi-layer partitions, we should aim at the highest possible increase of energy suppression in the soft layer (large ηh) by using materials that reveal high internal loss coefficient η . Increasing the layer thickness h is not recommended as it increases the stiffness of the partition and shifts the coincidence phenomenon down into the frequency band below 5 kHz.

Unlike passive noise minimisation methods, another group of methods comprises active methods. These methods, in the application to noise reduction, are complementary to the classical (passive) methods. In general, they make use of additional (secondary) sources of sound, which work simultaneously with the basic (primary) sources. As a result, mutual compensation or destructive interference of the primary and secondary wave takes place. To obtain the maximum possible (complete, theoretically) suppression of the primary wave, the generated secondary wave should have the same frequency and amplitude as the primary wave, but be opposite in phase. For instance, for harmonic waves the suppression of an order of 20 dB is obtained when the difference between the acoustic pressure levels is smaller than 1 dB, and the phase shift does not differ from 180° by more than 5° . This example shows that the secondary source controller is to meet high requirements.

The source of a control signal is a primary signal detector, a microphone for instance. It should be located at another point than the observation point, otherwise the system will be unstable and open to self-excitations. From the primary detector the signal reaches an electronic controller which activates the secondary source to change the amplitude and phase of the signal. It has a form of a filter with a relevant amplitude–phase characteristic.

CONCLUSION

- The main engines and auxiliary units used on ships are most often connected with the hull via auxiliary frames. The operation of the above machines is unsteady and instationary by nature. Vibrations generated by those machines are minimised using passive and active methods. The group of active methods includes: vibration dampers, vibroinsulators, and multi-layer rigid skin plates. These constructions dissipate the energy of vibrations or partially store it. The active vibration reduction systems have external sources of energy. These methods are expensive, as they require complicated automatic control systems. In unsteady conditions their efficiency can be low.
- Similarly, the engine room noise in motor yachts is minimised using passive methods, such as sound absorbing

housings installed for the main engine and auxiliary units, and multi-layer wall partitions.

- For sound absorbing housings, it is worth remembering about necessary disposal of the heat and gases to protect the object thermally. The air is to be changed 60 times per hour in case of gases lighter than the air, and 120 times per hour for heavier gases. Correctly manufactured housings make it possible to reduce the sound level by 15 ÷ 20 dB(A). A sample of the correctly designed housing is shown in Fig. 5.

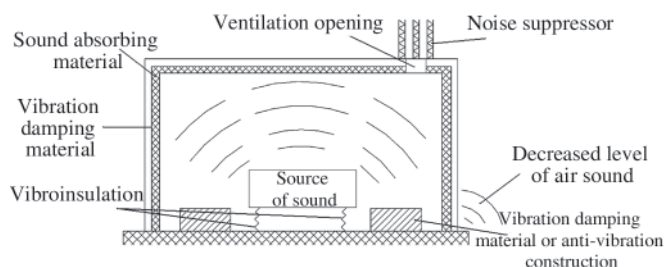


Fig. 5. A sample correctly designed sound absorbing housing

- Multilayer wall partitions are materials which reveal high insulating ability. For the finishing materials used in cabins with increased sound absorption coefficients, the average sound absorption coefficient from the source side should be equal to $\alpha_{sr} \approx 0.6$. Proper selection of the insulating ability of the walls and their acoustic absorptivity make it possible to reduce the noise level by over 20 dB(A).

- In a comprehensive approach to the problem of noise and vibration minimisation in motor sea-going yachts, passive methods can bring noise reduction up to 30 dB(A). Each type of yacht needs an individual approach when selecting methods to maximise insulating effects. Changes to the acoustic environment can be introduced both in the engine room and in crew compartments. In order to achieve maximum possible effects, passive and active methods are to be used together, as the active part acts within the low-frequency range, while the passive part – in higher frequencies. Such an approach provides opportunities for gaining the best possible acoustic effects.

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