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PROBLEMS CONNECTED WITH MINIMIZATION OF INTERNAL VIBRATION AND NOISE GENERATED BY POWER UNITS IN SHIPS

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Abstract

The paper discusses some problems connected with minimization of vibration and noise emitted by power units or ancillary aggregates in ships. Basic properties of the material systems are defined giving due attention to fundamental frequencies generated by these sources. Passive and active methods of noise minimization through interference with propagation path are presented. Anticipated effects are specified in the summing-up.

Key words: *power units in ships, minimization of vibro-acoustic signals*

1. Introduction

Minimization of internal noise and free vibration in sea motor yachts is a difficult task. The reasons are strongly connected with specific conditions existing in such objects, namely very small machine-room with necessary availability of transmission gear, main engine shafting and ancillary aggregates such as current generator, pumps, fans etc. These restrictions are a serious impediment to using some of the methods to minimize the vibro-acoustic effects. Generally, a multi-cylinder engine with spontaneous ignition is used where vibration is caused by variable exciting forces. The forces are induced by:
- pressure fluctuation within inlet and outlet channels,
- pressure variations in cylinders concomitant with combustion process,
- timing gear system at work,
- pressure variations in fuel system and lubrication system,
- inertia in mobile components of the engine,
- toothed gears,
- ancillary units.

Frequencies of these forces are related to rotational speed of the engine crankshaft as follows:

$$f_N = \frac{nZ_C}{60s}k \tag{1}$$

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... exciting force harmonics.

If the frequency of rotations, induced by non-balancing of the rotating masses, is shown as:

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

the following frequencies connected with operation of the engine sub-assemblies will be as below:
- frequency of the camshaft

$$f_r = f_n i = 0{,}5 f_n [Hz] \quad (3)$$

where:
i - relocation of the camshaft drive;
- frequency of closing the valves.

$$f_Z = f_n \frac{Z_Z}{s} [Hz] \quad (4)$$

where:
Z_Z - number of valves working separately,
- frequency of the toothed gear.

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

where:
z - number of the pinion teeth,
- blade frequency of the fans.

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

where:
l - number of blades.
Frequency of impeller pumps, similarly to that of fans, is connected with the number of pump blades „l" and frequency of main engine revolutions „f'_n":

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

Frequency of current generators is usually 50 [Hz].

As shown by the above relationships, various sub-assemblies of the engine generate the same frequencies, in particular when harmonic frequencies of the exciting forces are seen in the spectrum in result of nonlinear material systems.

Among the sources mentioned above there are low-frequency exciting sources, below 100 Hz (engine together with the exhaust, generators) and high-frequency exciting sources, above 1 kHz (fans, pumps, toothed gears etc.). Using uniform methods to minimize vibration and noise emitted by the units is very difficult in such situation.

Level of acoustic pressure in engines used for ships can be assessed approximately basing on:

$$L_p = 12 \lg N + 30 \lg n - 10{,}7 [dB] \quad (8)$$

where:
N - power of the engine [kW],
n - rotational speed of the crankshaft [rev./min.].
The level depends on many structural factors e.g. geometrical dimensions of the pistons, type of the body material etc.
Acoustic power of compression-ignition engine is about:

$$L_N = 59 + 10 \log N_Z + 10 \log N_Z - 30 \log \frac{m}{n} \pm 4 [dB] \quad (9)$$

where N_Z corresponds to rated power of the engine [kW] with its rated speed n_Z [rev./min.].

2. Methods to minimize the level of noise and vibration in machine-rooms

Minimization of vibro-acoustic signals emitted by power transmission systems can be obtained through:

1. changing aerodynamic indicators on the inlet, outlet and working space of the units,
2. minimization of the exciting forces and their spectra.

Aerodynamic conditions can easily be changed through using covers to dampen vibration and dampers on the engine inlet and outlet and on the aggregates. Minimization of the forces: through vibration absorbers.

2.1. Minimization of vibration levels

Damping is connected with dissipation of mechanical energy converted e.g. into thermal energy, thus also with reduction of general efficiency of the unit. Therefore damping is introduced when minimization cannot be obtained through modification in structure or parameters. Such approach is typical for active methods. On the other hand, passive methods are based on changes in transmission path, namely vibroinsulation. Among the methods of structural modification there are three groups. The first of them consists in introducing additional internal connections e.g. absorbing springs (e.g. disk connections). The second of them consists in introducing additional masses (e.g. Frame type). The third is based on redevelopment of the structure continuity e.g. through intermediate flexible elements (vibroinsulators).

A major disadvantage of dynamic vibration eliminators is that they can only be used for harmonic excitations (being adjusted to strictly specific frequency). They are not useful for material systems in nonstationary conditions. Similar disadvantages are seen in parameter modification of material systems where the parameters are variables of loading vectors such as: inertia of the system „M”, rigidity „K”, damping „C”. While at work, the true nonlinear material system is affected by exciting forces having a wide spectrum with a large number of harmonics. Wide spectrum exciting forces must necessarily be reduced in order to reduce vibration amplitude.

In mechanical conditions, the values of damping forces are usually lower than those of elasticity or inertial forces. Such forces are directed opposite to direction of speed vectors in an attempt to reduce or limit kinetic energy of the system. Type of friction (viscous, columbian or material) plays an important role here.

Minimization of vibrations in material systems is usually based on passive (relocation) or active (by force) vibroinsulation. In practice passive methods are prevalent. When fixed on vibroinsulators, the machine has 6 degrees of freedom or, in case of linear systems, 6 resonance frequencies. Vibroinsulators are so selected that the machine is prevented from working within resonance band. There are many methods to join the machine and its foundation together by way of vibroinsulators. Lifting systems or vertical, oblique, mixed types are fairly common.

A wide variety of materials can be used for vibroinsulation. With large-mass machines, usually pneumatic springs or those made of steel or rubber are used. With high loadings, steel springs are most common as they can easily be determined through computation. Most preferable types for vibroinsulation are linear springs where the force is proportional to deflection. Type of spring (helical, disk etc.) depends on the accepted vibroinsulation system.

Rubber springs are used for high-frequency exciting forces and low loadings, thus usually for springing. Rubber elements under stable loading should not exceed 15%, with coagulation 25-40% (hard blends) or 40-70% (soft blends), and their rigidity should be 50-60 Sh. Pneumatic springs are rarely used in industrial practice on account of their large geometric dimensions.

Effectiveness of vibroinsulation

Effectiveness of vibroinsulation can be defined, apart from excitation characteristics, through:

a) relocation vibroinsulation,
b) force vibroinsulation.

A model of vibroinsulation with kinematic excitation and relocation z(t) can be illustrated by a simple single-mass system with mass „m", damping „k" and elasticity „c". Such system is presented in fig.1.

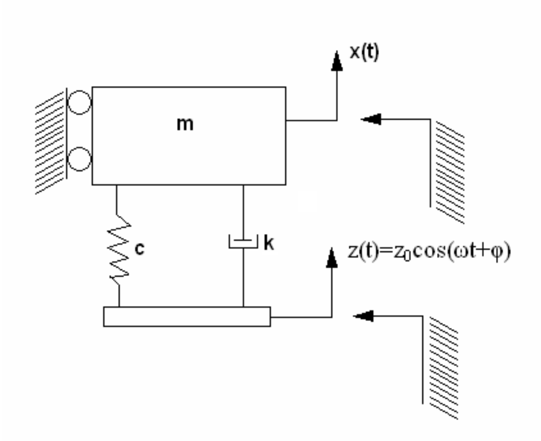

Fig.1. Model of relocation vibroinsulation

With force vibroinsulation, kinematic excitations are transformed into force type as shown in fig.2.

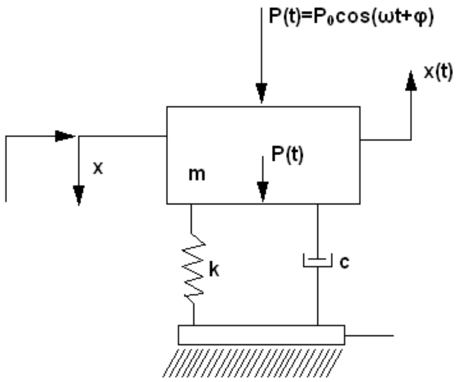

Fig.2. Model of force vibroinsulation

Vibroinsulation criterion is described by the following relationship:

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

This criterion should meet the following conditions:

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

If $T_p = T_x = T$, i.e. if vibration relocation factors are the same, it practically occurs for T<1 where free vibration frequency ω - to - free vibration frequency ω_o ratio is greater than 3. Upon adding a relative damping coefficient ξ to the assessment, it can be seen that if ξ $\leq$ 0.1, the value of relocation factor „T" is nearly independent on damping. Such situations occur in most cases. If
ξ = 1, the effect of vibroinsulation will disappear. Relocation factor depends almost entirely on damping.

Vibroinsulation systems can be divided into:
- passive,
- active.

Passive systems can only separate or temporarily store the energy.

Active vibroinsulation is based on generating control forces which tend to affect the unit in question. Passive systems can only dissipate or temporarily store the energy. Active systems contain external source of energy which can provide or absorb the energy through automatic control. In general, active methods of vibroinsulation can be divided into methods controlled by excitation and methods controlled by parameters of vibratory field. This type of vibroinsulation system contains a converter to measure vibration parameters and to control both the external power source and the force element. Such vibroinsulation system is a complex automatic control system. Due to its complexity and high costs, it has fairly limited applications. Semiactive systems are far simpler. Instead of generating forces, they modify the controlled damping parameters and vibroinsulator elasticity.

2.2. Minimization of noise level

Noise level can be minimized through the following methods:
- passive,
- active.

Passive minimization methods are based on:
- reduction of noise emission generated by a sound source,
- reduction of acoustic energies on their transmission paths.

Reduction of noise source vibroacoustic energy can be obtained without interference in manufacturing process through:
- changes in aerodynamic and hydrodynamic conditions of the machines,
- reduction in relocation efficiency factor.

Changes in conditions of media flow in the source should be understood as transformation of media speed into noise (acoustic power of aerodynamic noises is proportional to 6-8th power of the gas stream speed). On the other hand, reduction in relocation efficiency is connected with a change of materials, protective covers, emission media etc. If combustion engines are considered, a modification of inlet and outlet systems through changes in their geometry is recommended upon vibroacoustic analysis in order to reduce the media energy. In this aspect it is useful to consider installation of noise eliminators, including suppressors.

Suppressors can be roughly divided into absorption and reflection types. Absorption suppressors prevent transmission of acoustic waves through absorbing a large portion of their acoustic energy. In a majority of cases such effects are obtained through using internal sound deadener lining. Such suppressors can be sucking suppressors.

Reflection suppressors are included in acoustic non-continuity channel where their acoustic resistance is either much lower or much higher than characteristic resistance of the channel. Most often it is a single or dual change in section stroke (cavity or resonator suppressors). Such suppressors can be used as exhaust silencers.

It is important to note that noise suppressors will only be highly effective if their mobility (opposite to impedance) is much higher than the sum of inlet and outlet mobilities. Apart from suppressors the following methods are recommended to minimize the level of noise:
- coverings to damp vibration in bodies,
- sound absorbing casings,
- acoustic screens,
- changes in acoustic absorptivity of the rooms,
- changes in insulating power of the barriers.

Coverings to damp vibration in engine bodies, power transmission systems and aggregates have practically no chance to be used in ships due to the requirement that they should continually undergo operating supervision. Sound absorbing casings protect the entire machine. The material should have a high sound absorption coefficient on its internal side. This can be obtained through spreading sound deadener on respective surfaces. Walls of the casing should have high insulating power of the barrier („β”). This, together with the internal sound deadener, will provide the following effect:

$$\Delta L_u = L_1 - L_2 = \beta_u = 20\log f\rho + 10\log\alpha + \rho \quad (11)$$

where:p - surface density of the barrier,
f - frequency of the emitted sound [Hz],
ά - sound absorption coefficient,
L_1, L_2 - level of sound before and after the barrier [dB].
Effects of the casings depend on leak tightness of their components. Unlike sound insulating casings, acoustic screens are practically not used inside machine-rooms in ships.
Good effects of noise minimization in closed spaces can be obtained through changing noise absorption coefficient „ά". Effects of such changes are shown in fig.3.

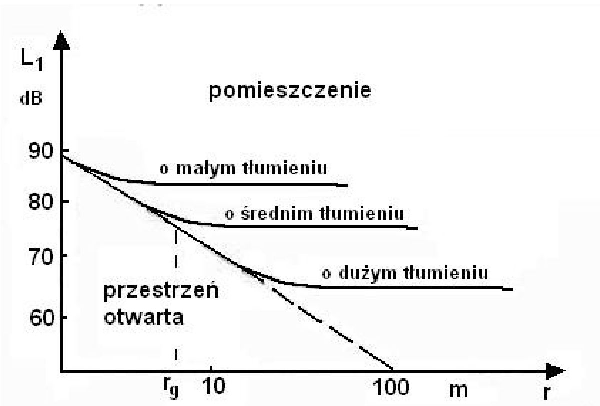

Fig.3. Reduction of noise level in open space and in rooms provided with various degrees of suppresion

If the distance from noise source is longer than „rg" (limit), the noise level will depend on acoustic absorptivity of room „A" [m²]. If „α" = mean sound absorption coefficient for a room floor „S", the definition can be as follows:

$$A = S\alpha\ [m^2] \quad (12)$$

Upon increasing the absorption coefficient from α1 to α2, the noise level will be reduced by:

$$\Delta L = 10\log\frac{R_2}{R_1}[dB] \quad (13)$$

where:

$$R_{1,2} = \frac{A_{1,2}}{1-\overline{\alpha}_{1,2}}[m^2] \quad (14)$$

Penetration of acoustic energy through barriers is a complicated phenomenon. It is generally accepted that such penetration may be affected by dynamic factors together with structure and material of the barrier. The coefficient (acoustic insulating power of the barrier) is approximate in character. Influence of material on insulating power of the barrier is presented in fig.4.

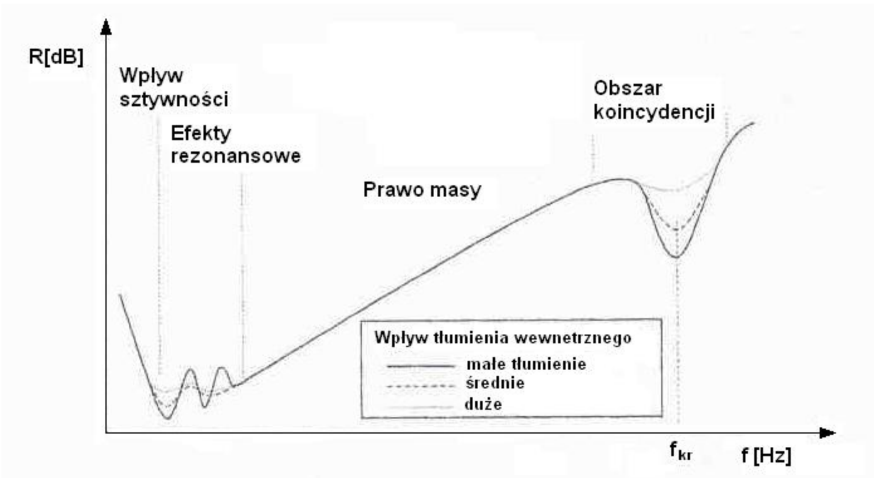

Fig.4. Influence of barrier material properties on its acoustic insulating power against air sounds

Physical properties of materials that affect insulating power include: elasticity, specific gravity, internal suppresion etc. As far as low frequencies are concerned, the insulating power depends chiefly on rigidity of the barrier. Next in importance is the influence of free vibration frequency „f_o". The influence of mass begins to work when said frequency is $2f_o$. Later stage is connected with coincidence effects. Acoustic insulating power of the barrier (homogeneous and isotropic) is:

$$R = 20\lg m + 20\lg f - C[dB] \qquad (15)$$

where: f - frequency [Hz], m - surface mass of the barrier [kg/m²], C - constant coefficient (equal to 48 for normal weather conditions).

Basing on *mass law* equation, acoustic insulating power of the barrier (homogeneous and isotropic) appears to be directly proportional to surface mass of the barrier (expressed by the mass of 1m² of its surface), growing with frequency approximately by 6dB/octave.

Approaching acoustic insulating power of the barrier in terms of its surface mass only seems to be a simplification of the problem. The reason for discrepancy between the results may be that sound wave will fall, in some conditions, onto the plate with bent wave speed in the plate (c_g) being equal to sound wave speed (c_o) falling onto the plate ($c_g = c_o$).

Imput impedance of the plate becomes reduced, together with reduction of acoustic insulating power, in result of condensation. Theoretically, insulating power level for coincidental frequency should drop to zero. However, the drop is of stepwise character due to internal losses of the barrier material. In case of hard barriers, such drop can even go up to more than ten dB. Within coincidence area ($f>f_{kr}$) the insulating power will deteriorate in accordance with internal losses in the material (coefficient ŋ).

Coincidental frequency or *resonance frequency* can be determined basing on the following formula:

$$f_{kr} = \frac{c_0^2}{2\pi}\sqrt{\frac{m}{B}} = \frac{c_0^2}{2\pi h}\sqrt{\frac{12\rho(1-v^{2)}}{E}}[Hz] \qquad (16)$$

where: c_o - sound speed in the air [m/s], m - surface mass of the plate [kg/m²], B - rigidity of the plate on bending cylindrical surface B = Eh³/12(1-v²)[Nm], p - density of the plate material [kg/m³], h - plate thickness [b],υ - Poisson's ratio.

Reduction in the barrier mass, without deterioration of its insulating power, can be obtained through installation of a multilayer barrier. However, the following conditions must be met in order to obtain the best possible insulating power with a multilayer barrier: number of layers, thickness, resultant rigidity „E” of the barrier, surface mass of the barrier. The barrier should have low-degree rigidity on bending B.

When installing multilayer barriers, it is important that maximal energy suppression should be expected in soft layer (high ηh) through using materials with high factor of internal losses „η”. On the other hand, it is not recommended to maximize layer „h” thickness because this would intensify the barrier rigidity thus causing relocation of coincidence to the band below 5 kHz.

Unlike passive methods of noise minimization, active methods are quite different. Active methods of noise reduction are supplementary to classical (passive) methods. In general, they employ additional (secondary) sound sources to work together with basic (primary) sources. In result, either mutual compensation or destructive interference of primary and secondary waves will occur. In order to obtain maximal (theoretically total) suppression of primary wave, it is necessary that the generated secondary wave should have the same frequency and amplitude as the primary wave but opposite phase. To give an example, in case of harmonic waves, the suppression will be approximately 20dB if the difference between acoustic pressure levels of waves is lower than 1dB and phase displacement does no deviate from 180º by more than 5º. As shown by the example, the secondary source control must obey strict regulations.

Control signal is primary signal detector, e.g. microphone. It must be placed in a different point than observation point, otherwise the system will be unstable, susceptible to self-excitation. Signal from primary detector will travel to the electronic control which in turn will excite secondary source thus changing amplitude and phase of the signal. Therefore it is a filter having appropriate amplitude-phase characteristics.

Summing-up

Main engines and ancillary aggregates used in ships are usually connected with their bodies by ancillary frames. The units are nonstationary and transient when at work. Active or passive methods are used to minimize vibration generated by these machines. Passive methods include vibration dampers, vibroinsulators, multilayer rigid plates to dissipate or partially store the energy. Active methods employ external energy sources to reduce the vibration. These methods are expensive and require complex automatic controls. Their effectiveness may be fairly poor with transient systems.

Similarly, passive methods are used to minimize noise in motor yacht machine-rooms. These methods often include sound-insulating engine casings together with ancillary aggregates or multilayer barriers for the walls.

When using sound-insulating casings it is important to remember that heat and gases must be carried away in order to ensure thermal safety of the place. Gases lighter than air require 60 x h.change of air whereas gases heavier than air require 120 x h. change of air. Correct casings provide reduction of the noise level by 15-20 dB (A). An example of correct casing is shown in fig.5.

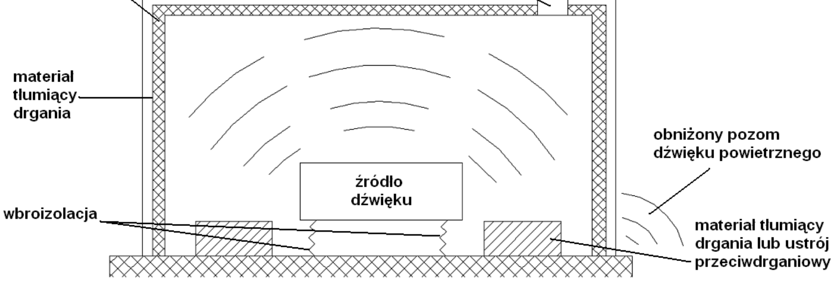

Fig.5. An example of correct sound-insulating casing

Multilayer barriers used for walls are materials of high insulating power. The materials should have the following insulating power: mean sound absorption coefficient $\alpha_{śr}$ ~ 0.6 on the side of the source. Correct insulation of the walls together with their acoustic absorptivity enable reduction of the noise level by more than 20dB(A).

With a broad approach to minimization of noise and vibration in sea motor yachts, it is quite possible to obtain a reduction in noise level by 30 dB(A) using passive methods. Each type of yacht requires an individual, unique approach to select most suitable methods for the case. Changes in acoustic environment can be introduced both for the machine-room and for crew quarters. To obtain maximal results, it is often useful to use passive and active methods jointly because active part operates within low frequencies whereas passive part operates within higher frequencies. Such approach enables obtaining the best acoustic effects.

References

[1] Z. Engel i inni, *Podstawy wibroakustyki przemysłowej*, AGH Kraków 2003 r.
[2] Z. Engels, *Ochrona środowiska przed drganiami i hałasem*, PWN Warszawa 2001 r.
[3] C. Cempel, *Wibroakustyka stosowana*, PWN Warszawa 1989 r.
[4] J. Kowal, *Sterowanie drganiami*, Gutenberg, Kraków 1996 r.
[5] S. Weyna, *Rozpływ energii akustycznych źródeł rzeczywistych*, WNT-Warszawa 2005r.