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SOME REMARKS ON LOCAL STRUCTURAL VIBRATION AND STRUCTURE-BORNE SOUND PHENOMENA CAUSED BY MARINE PROPELLERS

Summary

Increasing of propulsion power and keeping or even reducing vibratory and noise level requires, particularly with passenger vessels, a more careful hydrodynamic design of hull's after-body and, likewise, a more careful structural design of the hull section close to and above the propeller. Installing specifically designed stern bulbs and making use of the advanced blade configurations, vibratory and noise output of the propeller can be sufficiently reduced at present. On the other hand, structural improvements in order to reduce vibratory and noise input and to minimize the vibratory and noise energy scattering within the structure and its travelling to the upper decks are from the viewpoint of the speaker not sufficiently investigated yet up to now. It should be considered, that the periodically varying propeller pressure field causes periodically varying deflections of the hull plating, thus generating flexural and compression waves. As an example, the experimentally found distribution of local vibratory amplitudes in several upper decks in a medium-size passenger ship which are above the prediction are investigated by means of a simple multi-mass system. Results show the importance of the compression waves for the local structural vibrations.

Key words: structural vibration and structure-borne noise, propeller pressure, energy waves

NAPOMENE O LOKALNIM STRUKTURNIM VIBRACIJAMA I ŠIRENJU BUKE PROUZROČENIH RADOM BRODSKOG VIJKA

Sažetak

Povećanje pogonske snage i zadržavanje pa čak i smanjivanje dozvoljenih razina vibracija i buke zahtijeva, naročito kod putničkih brodova, pažljivije osnivanje oblika krme brodskog trupa s hidrodinamskog stanovišta i, jednako tako, pažljivije osnivanje structure presjeka brodskog trupa u blizini i iznad brodskog vijka. Poboljšanja structure u svrhu redukcije vibracija i buke kao i minimiziranja širenja njihove energije kroz strukturu sve do gornjih paluba nisu po mišljenju autora još dovoljno ispitana do sada. Treba uzeti u obzir, da periodski promjenljivo polje tlakova od vijka stvara periodski ppromjenljive deformacije oplate trupa kao i periodski promjenljive strukturne pomake elemenata povezanih s oplatom trupa, a time i razvijanje savojnih i tlačnih valova. Kao primjer, mjerenjem određena raspodjela lokalnih amplituda ubrzanja na nekoliko gornjih paluba putničkog broad srednje veličine, koje su bile iznad predviđenih vrijednosti, istražene su pomoću jednostavnog sustava s više masa. Rezultati istraživanja pokazuju značaj tlačnih valova za lokalne strukturne vibracije.

Ključne riječi: strukturne vibracije i buka, tlakovi od vijka, energetski valovi

1. Introduction

In order to achieve sufficient low vibratory amplitudes in the structure, following facts are to be considered:

1. Low vibratory hull pressures caused by propellers.
2. Low vibratory input into structure.
3. Effective structural attenuation to reduce the proceeding of vibratory power and low frequent structure-borne sound power.

In the past good results about low vibratory hull pressures were obtained by means specificly designed stern bulbs, as shown in Fig.1a.

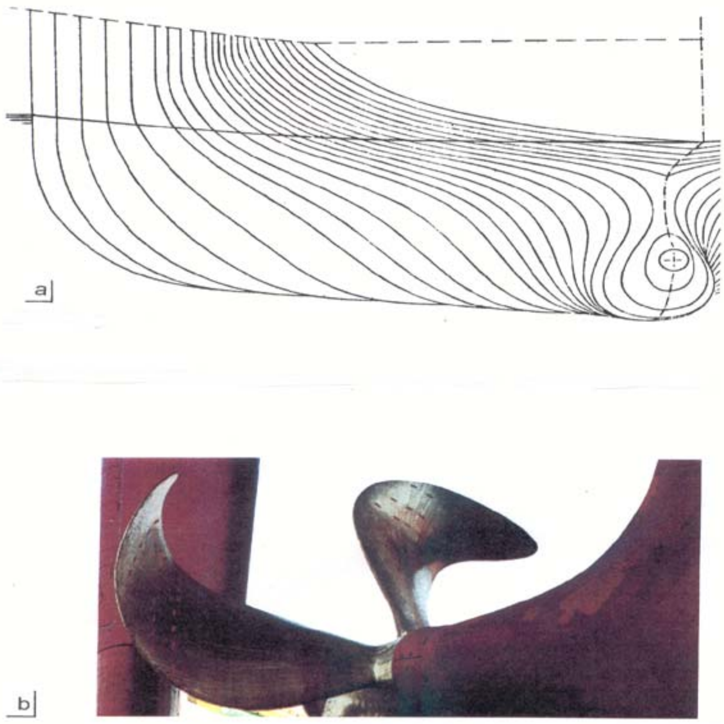


Fig. 1 : Stern Bulb Design (a), Kappel-Propeller [1] (b)

Fig. 1 Stern Bulb Design (a), Kappe-vijak [1] (b)

Slika 1. Nacrt krmenog bulba (a), Kappe-propeler [1] (b)

Obviously some information on stern bulb design has been lost during the last period of time. Consequently, at present the stern bulbs are more or less out of date and vibratory levels at single-screw ships are increasing again. In case of twin-screw ships optimal shaping and favourable positioning of shaft brackets and bossings reduce vibratory pressures, what is well known. Finally, newly developed propellers, for example of Kappel-Type (Fig. 1b, [1]) with single-screw ships and pod-propellers with twin-screw ships, will reduce vibratory hull pressures likewise. Unfortunately, even with minimized nonhomogenity of the propeller inflow velocity, thus avoiding unsteady blade loading as well as unsteady cavitation and vibratory excitation by propeller shaft bearing loads, vibratory hull pressures will be produced by pure thrust generation. The vibratory energy of this pressure will be partly reflected at the shell plating and partly transmitted into the structure. The frequency band of interest is between 0 and about 100 Hz.

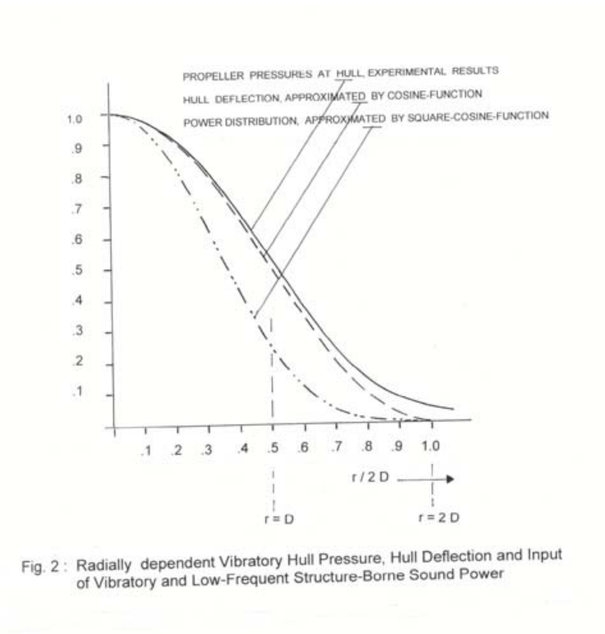


Fig. 2 Radially dependent vibratory hull pressure, hull deflection and input of low-frequency structure-borne sound power

Slika 2. Radijalno ovisni vibracijski tlakovi na trupu, pomaci trupa i ulazne jakosti nisko-frekvencijskog strukturnog zvuka

Low vibratory input into the structure requires sufficient stiffness, what means good reflecting properties from the standpoint of acoustics. Experimental investigations have shown, that at a spatial distance of 2 times diameter (2-D) from the propeller centre, vibratory hull pressures are less than about 5% of maximum pressure at the hull point above the propeller. Pressures between maximum and about 50% of maximum occur in 1-D area. Figs. 2 and 3 show this situation. Considering the weight of the steel, it should be sufficient to fix additional stiffeners between frames only in 1-D area in such a manner, that plate fields are more or less quadratic. Quadratic plate fields yield minimum deflections, and consequently minimum vibratory power input, what is generally known.

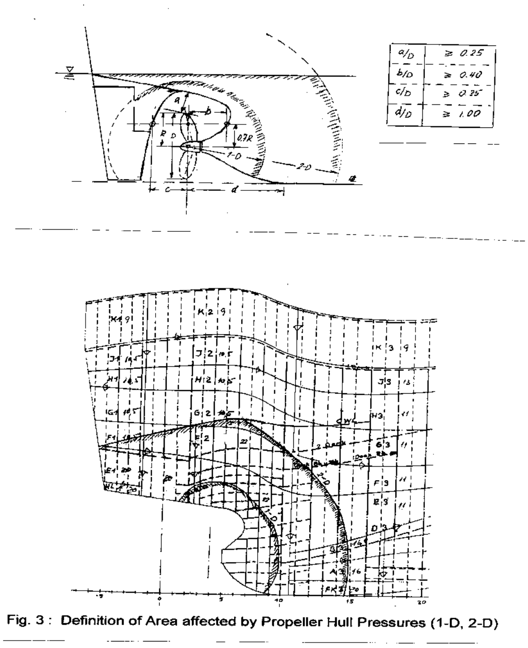


Fig. 3 Definition of area affected by propeller hull pressures (1-D, 2-D)

Slika 3. Definicija površine na koju djeluju tlakovi od vijka na trup (1-D, 2-D)

The problem is now, how to reduce the local vibratory amplitudes. With ships fully equipped and already in service possibilities are extremely limited. Extensive constructional measures cannot be executed in normal case; local stiffeners will shift vibrations to other locations, in general without any reduction; fixing locally placed blocking masses obviously cannot be allowed in upper decks because of small spare stability particularly with passenger ships. Consequently, the propagation and scattering of vibratory and low frequency structure-borne sound power input through-out the structure should be investigated, based on physical fundamentals. Considering statistical energy methods or noise-FEM [3] to investigate local vibrations is very time and money consuming. Therefore, in this paper a new „philosophy“ is outlined to solve these problems.

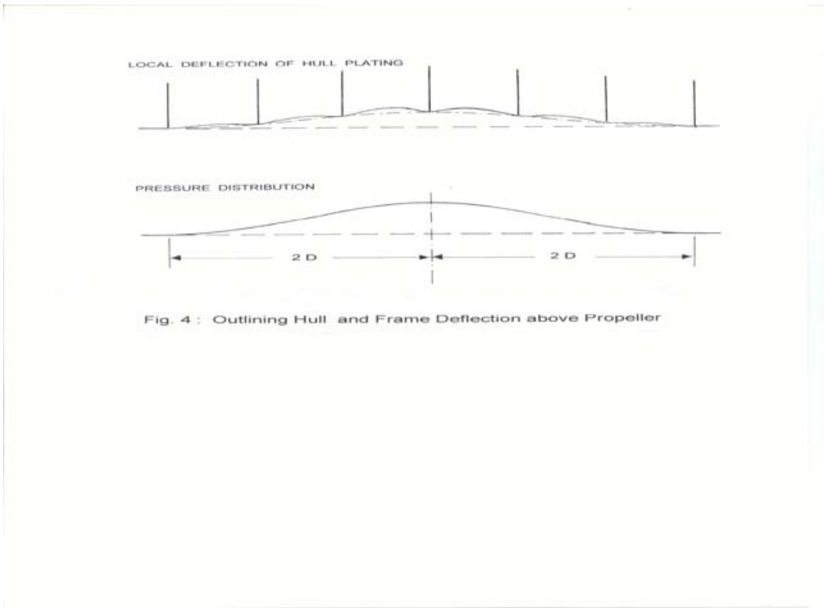


Fig. 4 : Outlining Hull and Frame Deflection above Propeller

Fig. 4 Outlining hull and frame deflection above the propeller

Slika 4. Prikaz pomaka na trupu i rebru iznad vijka

In order to estimate this power input, the deflection of the hull plating, the floor frames, etc. caused by the propeller pressure field is to be considered. Within 2-D area both the pressure distribution and the deflection of the frame system can be approximated by cosine-functions (Fig. 4). The local deflection of the hull plating between the bottom frames and stiffeners can be approximated by the well-known basic-to-strength calculation methods. For the given hull pressure amplitude, given frequency and given structural design the dynamic deflections of plating and frames can be calculated and hence the input velocities. Obviously, the input velocity and power input will decrease when the stiffness is increased.

Now mechanism of transport of vibratory energy and of low-frequency structure-borne sound energy within the ship structure will be considered. Because of the time dependence of the power input, internal waves will be generated by the deformation of the hull plating and adjoining structure. Two types of waves are dominant, those are the Extensional Waves (quasi-longitudinal waves) and the Bending Waves (pure bending waves). According to [2] for the extensive structural elements (plates, beams, pillars,etc.), what can be assumed for conventional ship structures, the following formulations for the vibratory power are in use:

extensional waves, wavelength is not frequency-dependent

$$P_E = \rho \cdot c_D \cdot A \cdot v^2_{Eeff} \quad \text{with} \quad c_E = \sqrt[2]{\frac{E}{\rho \cdot (1-\mu^2)}} \ \text{- Plates} \quad c_E = \sqrt[2]{\frac{E}{\rho}} \ \text{- Beams}$$

bending waves, wavelength is dependent on the frequency and on the plate thickness or beam dimensions respectively $P_B = 2 \cdot \rho \cdot c_B \cdot A \cdot v^2_{Beff}$ with $c_B = \sqrt[2]{2\pi f} \cdot \sqrt[4]{\frac{E}{\rho \cdot (1-\mu^2)}} \cdot \frac{h^2}{12}$ - Plates

$$c_B = \sqrt[2]{2\pi f}\cdot\sqrt[4]{\frac{E\cdot I}{\rho\cdot A}}$$ - Beams, where:

ρ: Density of material, A: cross-sectional area direction of wave propagation

I: second moment of area, c_E: phase velocity, extensional waves

v_E: particle velocity, extensional waves, v_{Eeff} : effectiv<u>e</u> particle velocity, $v_{Eeff} = \frac{v_E}{\sqrt{2}}$

c_B: phase velocity, bending waves, v_B: particle velocity, bending waves

v_{Beff}: effectiv<u>e</u> particle velocity, $v_{Beff} = \frac{v_B}{\sqrt{2}}$

In the ship structure the transport of vibratory power and structure-bound sound power occurs particularly by plates. The transport velocity for extensional waves in steel plates is the phase velocity c_E = 5439 m/s. When transporting vibratory power by bending waves in dry steel plates is considered, for example the bulkhead (plate thickness 6.0 mm, frequency 14 Hz), the phase velocity reads c_B = 28.6 m/s (Fig. 5). In the bottom shell plates the phase velocity is reduced due to the effect of fluid inertia.

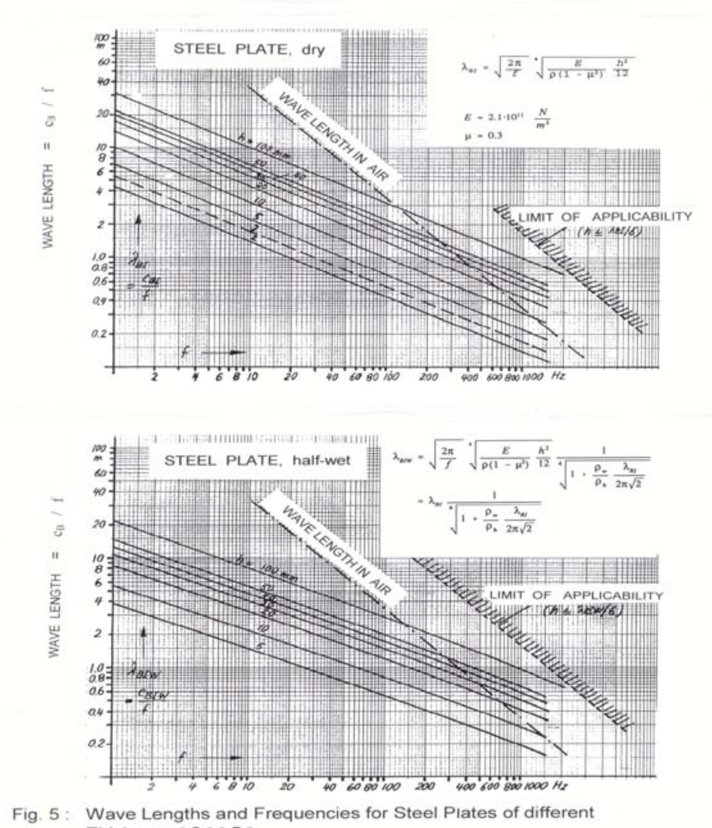


Fig. 5 : Wave Lengths and Frequencies for Steel Plates of different Thickness [2],[5]

Fig. 5 Wave lengths and frequencies for steel plates of different thickness [2], [5]

Slika 5. Duljine i frekvencije valova za čelične ploče raznih debljina [2], [5]

Local periodical deformation of the structure close to the propeller as sketched in Fig.4 results in the generation as well of extensional waves as of bending waves. Extensional waves will propagate in floor frames, whereas bending waves will propagate in the hull plating. On it's way to the upper decks the waves have to pass a great number of structural discontinuities caused by frames, decks, bulkheads, etc. At these discontinuities the vibratory and structure-borne sound power will be transmitted, reflected or even converted from the bending wave power to the extensional wave power and vice versa. The conversion from the bending waves to extensional waves, because of structural elasticity is outlined in Fig.6.

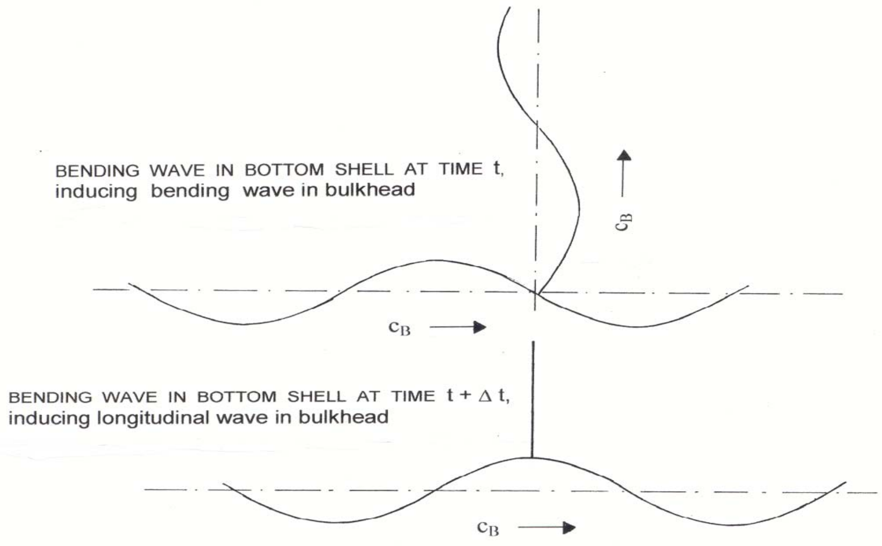


Fig. 6 : Outlining Generation of Bending Waves and Conversion from Bending Waves to Extensional Waves at Branch Point

Fig. 6 Outlining generation of bending waves and the conversion from bending waves to extensional waves at the branch point

Slika 6. Prikaz generacija savojnih valova i pretvorbe savojnih u vlačno-tlačne valove u točki grananja

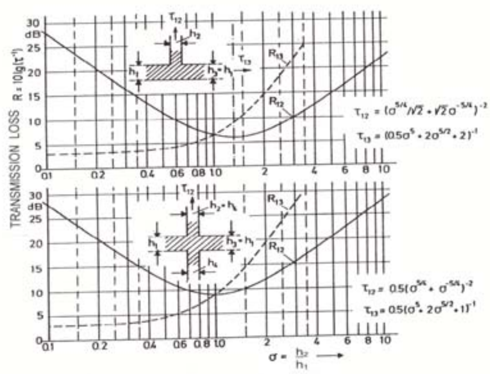


Fig. 7 : Effect of Thickness Ratio on Branch Point on Flow of Bending Wave Power [2]

Fig. 7 depicts two typical discontinuities (branch points) having effect on the flow of the power transported by bending waves [2]

Slika 7. Učinak omjera debljine u točki grananja na tok jakosti savojnog vala [2]

Figure 7 Effect of the thickness ratio at the branch point on the flow of the bending wave power [2] The configuration and thickness ratio at the branch point define the magnitude of the vibratory or structure-bound sound power in the branches. The transmission loss occurring at the branch point when affected by the bending waves is given by [2] such as

$R = 10 \cdot \log(\frac{P_{B1}}{P_{B2}})$ where P_{B1}: vibratory power input into the branch, P_{B2}: vibratory power output behind the branch

Branch points of this kind exist on junctions of shell plating and bulkheads and of deck plating and bulkheads. Further discontinuities of importance are junctions of stiffeners at plates, such as the frames at shell plating or beams and girders at deck plating or stiffeners at bulkheads. Because of the uniform distance of stiffeners in general, these units can be considered as "spatially periodic structure", or "cascades", or "chains". In Fig.8 the results of investigations for the extensional chain and the flexural or bending chain are presented ([2], [4]).

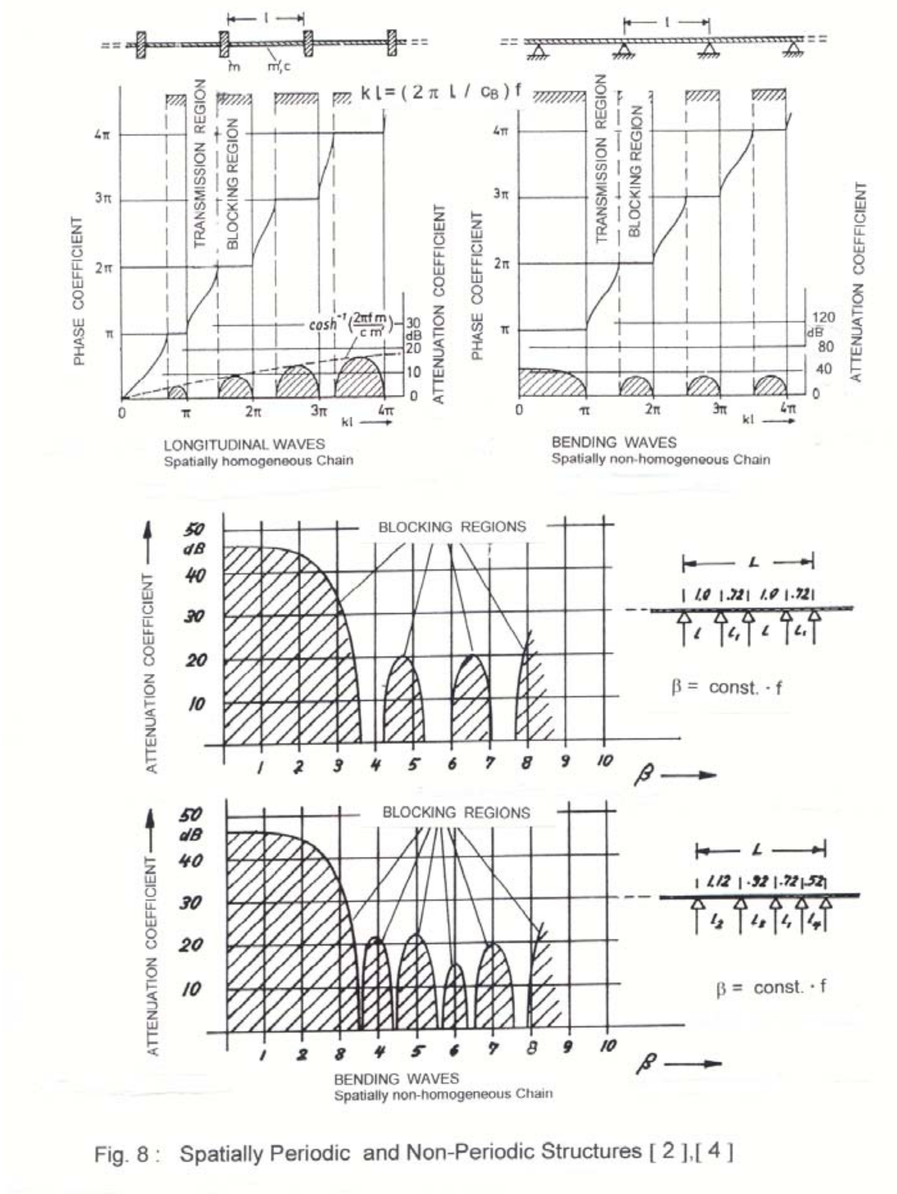


Fig. 8 Spatially periodic and non-periodic structures [2], [4]

Slika 8. Prostorno periodske i neperiodske structure [2], [4]

The presented results are obtained using a high degree of abstraction, so that only general conclusions can be drawn. Regarding the flow of power in a cascade, transmission regions and attenuation regions are to considered. Transmission region means that the power will not be reduced when travelling through the chain, blocking region means that the power is reduced to a certain extent, when travelling through chain. Attenuation of -6 decibels means reduction to 0.25 of input power. Parameter is kL, where frame spacing: L, wave number: $k = \frac{2\pi f}{c}$, wave length: $\lambda = \frac{c}{f}$

As mentioned above, results in Fig. 8 are highly idealized. Among others, the extension at both sides is endless, and in case of the bending chain, supporting points are totally fixed. In reality, this is not correct, what means that in reality the transmitting region and blocking region are not strictly defined. It should be of interest, that non-homogeneous chains, what means chains with unequal plate lengths between stiffeners or decks, yield more blocking regions and smaller transmission regions (Fig. 9). The attenuation effect within a bending chain is caused by the reflections at the stiffeners and is mainly dependent on the length of the plate fields and wave length, i.e. frequency of the power flow. For practical application the “finite” chains with equal and non-equal plate lengths are to be used.

3. Illustrative example

As an example, a medium-sized cruise vessel is considered, having length of about 170 m, 10 decks, twin-screws, cruising speed 16 to 17 kn at 168 propeller rpm. Diameter of propellers (CPP) 4 m, tip clearance 1.25m (= 0.32 D). Vibratory and noise behaviour of the vessel at cruise speed is satisfactory. At speed 18 kn in some decks local vibrations close to or slightly above limit were observed at propeller blade frequency (14 Hz). In Fig. 9 the side view of the aft-ship is depicted. The numbers given on Deck 4, 5, and 6 correspond to the results of experimental investigations (mm/s).

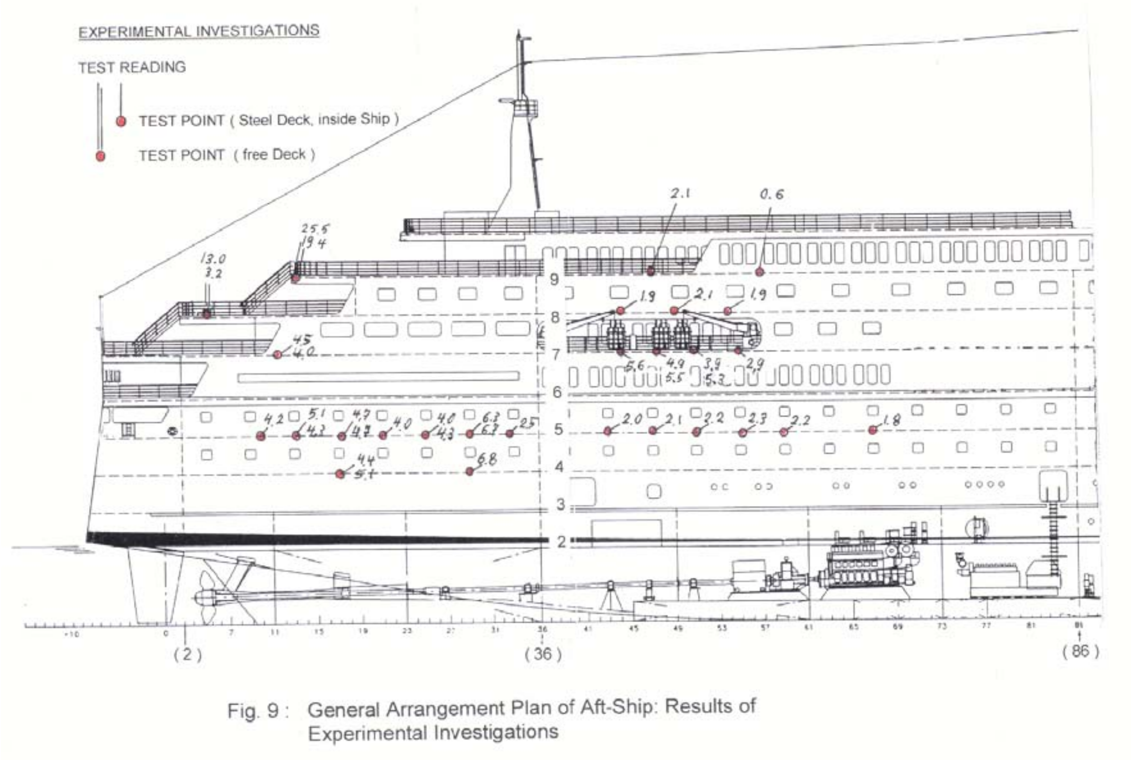


Fig. 9 General arrangement plan of the aft-ship. Results of experimental investigations

Slika 9. Opći plan krmenog dijela broda. Rezultati pokusa

Pressure measurements inside the shell above the propeller showed maximum pressure 1.2 kPa at propeller blade frequency 14.0 Hz. For single-screw cargo ships or container ships the upper limit of 4 kPa is recommended. In case of passenger ships a limiting pressure amplitude not above 1.5 kPa can be recommended, under the condition that the stiffness of the structure in 1-D area is sufficient. In our case this is not true, because of missing transverse stiffeners in between longitudinal frames in this area, what is the author’s opinion. Measurements among others performed in the plane of propeller, at the bottom of the structure at centre line of ship, proved a maximum vibratory velocity of 0.32 mm/s at 14.0 Hz and an appropriate maximum vibratory deflection of 0.0036 mm. At the structure above the propeller, vibratory velocity 0.67 mm/s and vibratory deflection 0.0077 mm were found. As mentioned afore (Fig.2), available experimental results prove that vibratory hull pressure of the propeller within 2-D area can be approximated by the cosine function. Likewise, vibratory structural deflection in 2-D area as a consequence of vibratory hull pressure can be approximated by the cosine function. Considering the approximations and the data from above, input of vibratory power generated by propellers at the propeller blade frequency can be calculated. For both propellers the input of vibratory power as a consequence of vibratory hull pressures results approximately:

$P_{VIBRATORY} \cong 5$ kW

This power can be split up into

$P_{DEF} =$ 4.42 kW, periodic deformation of aft structure and, presumable to certain extent, transformed into several kinds of structural waves.

$P_E =$ 0.02 kW, transformed into extensional waves in plates and bottom frames, and bottom structure.

$P_B =$ 0.56 kW transformed into bending waves in plates and in longitudinal frames

Compared with the propulsion power (6500 kW), the vibratory power input is extremely small (< 0.1 $^{o}/_{oo}$). At the bottom structure near the propellers the measured vibratory velocity induced by both propellers was found to be 0.32 mm/s. Consequently, on the path from the input location up to the upper decks, the magnifications, caused by light-weight structure and/or by local resonances, result in increased vibratory velocities. In this connection it should be mentioned, that even small amounts of vibratory power could produce considerable vibratory velocities. In connection to this, the following example is given:

A dry steel plate with sufficient length, 5 m in breadth and 6 mm thickness will be excited with 14 Hz by bending waves with a vibratory velocity amplitude of 4mm/s. This velocity amplitude is in accordance with the limiting value for vibration on board ships in superstructure. The speed of propagation comes to 28.56 m/s (wave length: 2.04 m). The amount of vibratory or low-frequent structure-borne sound power input will be calculated by means of formulae as given above, yielding P_B = 0.11 W. In case of excitation by the extensional or longitudinal waves, speed of propagation comes to 5439 m/s and the power amounts to P_E = 10 W.

In the ship's general arrangement plan, Fig. 9, on several decks in the aft part of the ship, vibratory velocity amplitudes at propeller blade frequency 14 Hz obtained by measurements are plotted. As can be seen from drawing, on lower accommodation deck 5 between bulkheads (2) and (36), the velocities of magnitude 4 mm/s and higher exist, what is too much for passenger decks. In the same deck, between bulkhead (36) and (86), velocity amplitudes of magnitude around 2 mm/s were found, what is acceptable. From the steel drawings it can be concluded, that deck 5 from aft-end to amidship is approximately of the same „structural quality". So, different vibratory behaviour should be a consequence of a different power input. On deck 4, in the aft region, local vibratory velocities of magnitude up to 6.8 mm/s were found, what could be caused by a local resonance. In upper deck 7, between bulkheads (36) and (86) (midship section is at frame 102), vibratory velocities from 2.9 mm/s to 5.6 mm/s were found. Finally, on decks 8 and 9 between bulkheads (36) and (86) velocities approximately 2 mm/s were found. On decks 4 to 9, close to bulkhead (36), vibratory velocities of magnitude approximately 2 mm/s in front could be measured. The latter indicates an almost constant input of vibratory power from bulkhead (36) into decks 4 to 9. Calculations to be reported below proved, that as well in case of finite bending chains as in case of finite extensional chains the output at branch points is constant. Obviously, no clear dependence of vibratory velocities on the distance of decks above the bottom line exists.

Now, bulkheads (2) and (36) are considered. Excitation is caused as well by extensional waves as by bending waves proceeding in the bottom shell plates. Starting from the area of power input by propeller pressure field (2-D area), vibratory and sound power level and resulting vibratory and sound velocities in more distant sections of ship structure can be determined by means of well-known rules for the propagation of the structure-borne sound.

At bulkhead (36) (Fig.10) the vibratory excitation is caused by bending waves arriving from the area of excitation in the bottom shell plating (2-D area). At the branch point the vibratory energy power will be partly transmitted into the bulkhead. To some extent at the branch point the bending waves will be converted into extensional waves, which transport the vibratory and low-frequent structure-borne sound power into the bulkhead and further into the decks.

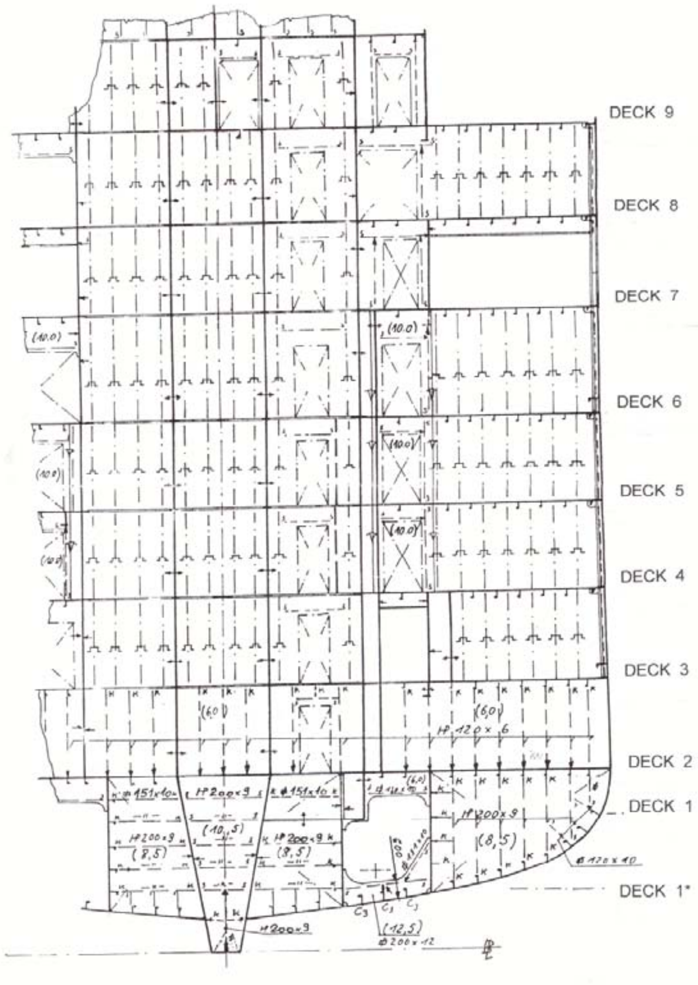
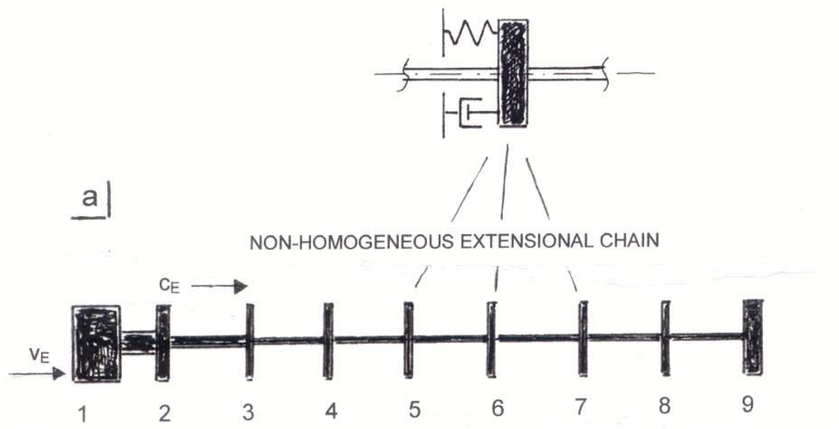


Fig.10: Bulkhead (frame 36), Drawing

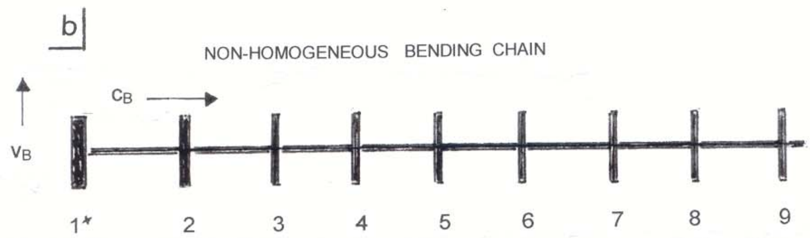
Fig. 10 Bulkhead (frame 36), drawing

Slika 10. Pregrada (rebro 36), nacrt

Bulkhead (36) can be physically modelled as a finite non-symmetrical extensional chain (Fig. 11a) and as a finite non-symmetrical bending chain (Fig. 11b).



v_1/v_0	v_2/v_0	v_3/v_0	v_4/v_0	v_5/v_0	v_6/v_0	v_7/v_0	v_8/v_0	v_9/v_0
1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0



v_1/v_0	v_2/v_0	v_3/v_0	v_4/v_0	v_5/v_0	v_6/v_0	v_7/v_0	v_8/v_0	v_9/v_0	f Hz
./.	0.014	0.147	0.149	.0148	0.147	0.146	0.150	0.144	14
./.	-.18.5	-8.3	-8.3	-8.3	-8.3	-8.4	-8.4	-8.4	dB

Fig.11: Bulkhead (frame 36), Physical models
a : Non-Symmetrical Extensional Chain
b : Non-Symmetrical Bending Chain

Fig. 11 Bulkhead (frame 36), physical models, a: Non-symmetrical extensional chain, b: Non-symmetrical bending chain

Slika 11. Pregrada (rebro 36), fizikalni modeli, a: nesimetrični vlačno-tlačni lanac, b) nesimetrični savojni lanac

Calculations prove that for the propeller blade frequency of 14 Hz, in the bending chain the vibratory velocity is reduced to 0.1444 at all branch points when travelling through the chain, what means a vibratory power reduction to 0.010. Analogous calculations with the extensional chain show no reduction of power at the branch points. From this it can be concluded, that the power transport in the bulkheads is primarily produced by extensional waves.

At bulkhead (2), placed in 1-D area, the vibratory excitation and power input occurs directly from the bottom structure by extensional waves and by bending waves. Bulkhead (2) can be just as bulkhead (36) physically modelled as finite non-symmetrical chains.

Measurements of the vibratory velocity on deck 5 between bulkheads (2) and (36) show about twice magnitude of velocity as between bulkhead (36) and (84). This might be caused by the power input into deck 5 from both sides. Because of the larger distance of bulkhead (86) from 2-D area, the vibratory and sound power in this bulkhead is relatively small, so no considerable power input in deck 5 will exist.

Principally, from about frame (20) towards fore-ship the transport of vibratory and sound power in the bottom shell plating causes vibration and structure-bound sound phenomena inside the ship. Side shell plating is for the transport of the power in the ship's length direction of minor importance because of the power reduction at several branch points. In 2-D area the bending waves will be excited proceeding thwart-ships. Because of the longitudinal frames and decks „ the bending-chain" effects will occur. By this, the vibratory and sound power will be sufficiently reduced and the vibratory and structure-borne sound phenomena will be avoided in the upper decks.

When executing the vibration measurements on free decks, in the aft-ship at some points unusual high vibratory velocities have been observed. Possible reasons might be the local resonances due to local failure of structural design or by special effects at "free" end with finite extensional chains.

4. Conslusions

On the basis of foregoing remarks, two methods to avoid or at least to reduce vibratory and structure-borne sound phenomena in ships under construction should will be recommended.

1. Reducing the size of plate-fields between frames by mounting additional stiffeners at least in 1-D area. By this, the input of the bending wave power will be reduced.
2. Reducing the power input from the bottom shell plating into the bulkhead plating by doubling the bottom shell plating in the front and rear bulkheads (length: 1 frame-distance, breadth: beam of the ship). By this, the transmission loss at the branch points will increase, what means less power input into the bulkhead.

Finally, it can be stated that, without regard to weighty and expensive additionally installed insulation against vibration and noise radiation, merely preventive measures are practicable to improve local vibratory and structure-borne sound behaviour.

The author is experienced in this field since many years.

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