Low frequency underwater radiated noise from shipboard machinery

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Introduction

The following report summarizes results from research at DW-ShipConsult in the field of underwater radiated noise.

Besides the propeller, propulsion and power generating machinery is the second contributing noise source. If the propeller does not cavitate, as at low speed, machinery noise will dominate the spectrum of a ship.

Machinery noise can often be separated from propeller noise by its tonal structure, i.e., the radiated spectrum is dominated by single frequencies (tonals) and the multiple (harmonics) thereof. In ships designed to be quiet like research vessels and naval ships, comprehensive efforts are made to limit machinery noise. We observe that often the insulation of machinery to reduce structure-borne noise transmission is excessive which results from large uncertainties in prediction, in particular at low frequencies.

With the combination of available analytical formulas [1] and the application of modern finite element tools a more precise prediction method is developed.

Principles of radiation

In the far field, radiation of structure-borne sound is generally described by

$$P = v^2 \cdot \rho_0 \cdot c_0 \cdot A \cdot \sigma \qquad (1)$$

with

P	radiated power
A	radiating area
v^2	energetic average of structure-borne sound velocity on area A
ρ_0	density of the medium
c_0	speed of sound in the medium
σ	radiation efficiency.

Equation (*1*) does not consider the cause of the sound on the structure. The assumption is that the structure vibrates independent of what causes the vibration. Ship building structures are generally plate-like with stiffeners and finite size. For the simple case of an infinite homogeneous plate the radiation efficiency is unity above the critical frequency and zero below. Critical frequency is defined by sound waves in the medium being the same as bending wavelength on the plate. Below the critical frequency there is no radiation because there is mismatch of the wave in the medium and bending waves on the structure which results in acoustic short circuit: particles in the medium only move in the vicinity of the vibratory motion but not far away from the structure [1].

Infinite homogeneous structures do not exist. If a plate is bounded by a support or by a free edge or stiffened acoustic short circuit cannot be perfect and radiation efficiency takes values between zero and unity approaching unity close to the critical frequency. The radiation efficiency for these cases is approximated by empirical formulas.

The radiating area and the average vibration velocity have to be defined in unison. Estimates are possible based on measurements onboard and finite element analysis.

We have found analysis with this model unsatisfactory in that it predicts too high levels with increasing frequencies. As we found that in quiet ship, machinery noise in underwater radiated noise is from structure-borne noise primarily from low frequencies below about 250 Hz, another model is investigated.

Machinery contributing to underwater radiated noise is usually mounted on the bottom of the ship or at least below the water line therefore within the radiating area. It is obvious that another radiation model available is investigated for better applicability which is that of a force on an infinite plate [1]. The radiated power of one side of the plate is

$$P = \frac{|F_0|^2 \rho_0 c_0 k_0}{4\pi} k \int_0^{k_0} \frac{k_r dk_r}{\sqrt{k_0^2 - k_r^2}\left[\omega^2 m''^2 \left(1 - \frac{k_r^4}{k_B^4}\right)^2 + \frac{\rho_0^2 c_0^2 k_0^2}{k_0^2 - k_r^2}\right]} \qquad (2)$$

with

F_0	Force on plate
k_0	Wave number in medium
k_b	Wave number of bending wave
m‘‘	mass loading of plate
ω	circular frequency = 2πf, f frequency

In underwater noise, radiation is usually not expressed as sound power but as sound pressure on a sphere with a radius of one meter in Decibels with reference pressure p_0 of 1 µPa.

$$p^2 = \frac{P \cdot \rho_0 c_0}{A_0}$$

with

A_0	is the surface area of a sphere of radius one meter = 12.6 m²

and the sound pressure level

$$L = 10log\frac{p^2}{p_0^2}\ re\ 1\ \mu Pa\,.$$

Figure 1 shows an example result with a force F_0 of 1 N.

The vertical yellow line fc marks the critical frequency. As we are considering low frequencies we are on the straight line marked in red --f(F) which indicates the condition that the fluid loading (impedance of the fluid) is much larger than the mass loading (impedance of the mass), hence $\rho c >> m''\omega$.

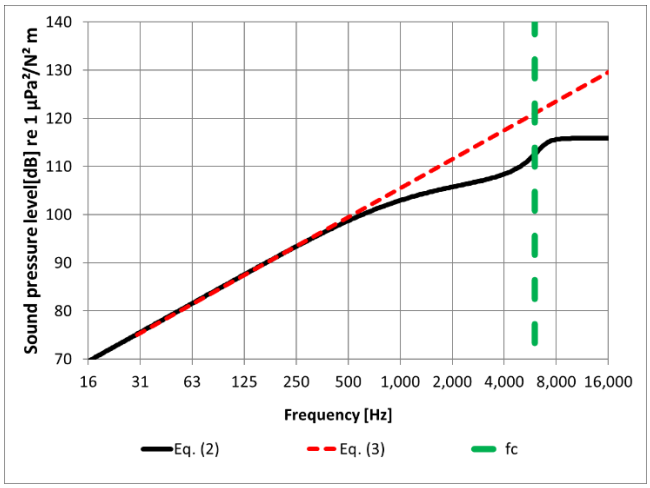

Figure 1*:* Solution of eq. (*2*) and eq. (*3*) for a steel plate of 40 mm thickness

For this conditions eq. (*2*) simplifies to

$$P = \frac{|F_0|^2 c_0 k_0}{12\pi\rho_0 c_0^3}\,. \qquad (3)$$

Note that in eq. (*3*) radiated power does not depend on plate thickness and there is no radiation efficiency [1].

Application of finite element analysis

Eq. (*5*) lends itself as a tool to verify results of finite element analysis. By application of the Ansys tool harmonic acoustics [2], plates of various size on a free water surface were modelled and investigated with regard to radiation. Figure 2 shows results for square plates of size 2x2 to 40x40 m. It can be seen that the numerical solution follows exactly the analytical solution beyond a certain frequency. Below that it deviates to higher values at a lower slope. The frequency at which the deviation starts moves to lower frequencies with the size of the plate. The smaller the plate the higher the radiation which is explained by the radiation of the edges of the plates where bending waves arrive with higher amplitudes than in larger plates. The variation of the level increase with smaller plate size.

The 5x5 m plate is marked with an empirical approximation:

$$L_{source,1N} = 71 + 10\cdot\log(f)\ re\ 1\frac{\mu Pa^2}{N^2}\ m\,. \qquad (4)$$

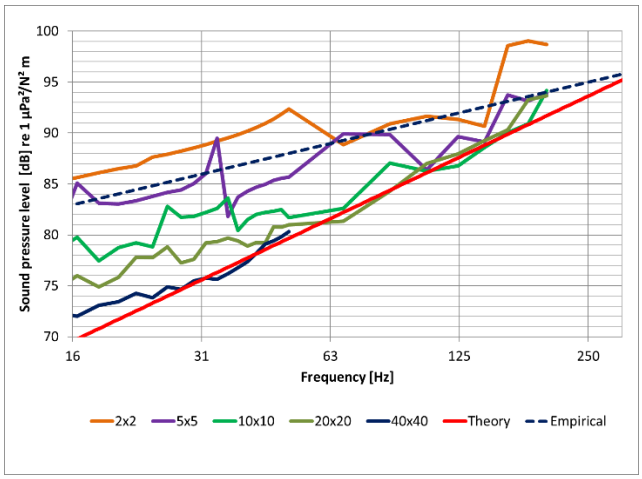

Figure 2: Radiation of steel plates with various sizes

In a next step the same model is applied to a steel ship section of about 20x20 m excited in various positions, Figure 3.

It is found that radiation is within tight boundaries around eq. (*5*) for all structures investigated provided that the input impedance on which the exciting force acts is high enough. This is fulfilled for all normal ship foundations. The green curve in Figure 3 shows the effect of a foundation with too low impedance. The natural frequencies of the structure are accompanied by high radiation.

In all ship structures investigated we found sound radiation of similar levels at a maximum of +5dB above eq. (*5*) provided the input impedance is above a benchmark according to Figure 4. Above that benchmark radiation hardly varies with an ever-increasing impedance.

Above a certain frequency which depends on ship structure, radiation drops which is explained by propagation losses from foundation to hull shell. Below a certain frequency observed to be between 16 and 31 Hz radiation also drops as has been found from finite element analysis of the whole ship. A reason for this effect is still to be found.

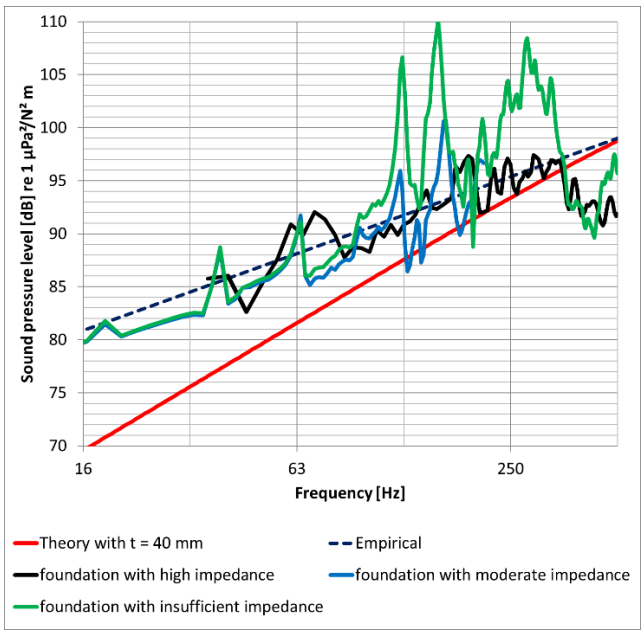

Figure 3: Radiation of various ship structures

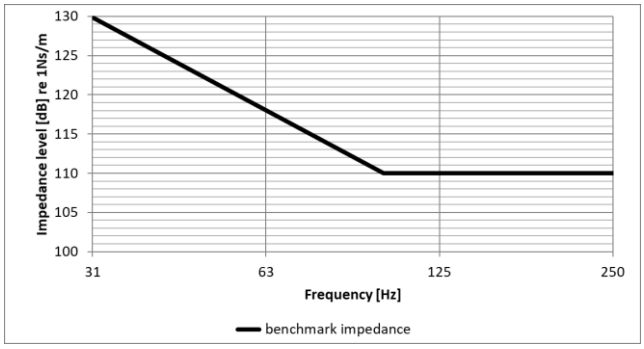

Figure 4: Benchmark impedance above which radiation is about independent of impedance

Determination of exciting forces

High accuracy predictions of underwater radiated noise are primarily sought for ships with high demand in this respect. Dominant noise sources in these ships are resiliently mounted. The complete transmission model for these arrangements is shown in Figure 5. We consider only the part of the path in the blue box which describes the main path and ignores the flanking paths.

For finding the force on the foundation knowledge of the structure-borne noise level and the dynamic mount stiffness of the resilient foundation have to be known. It is further assumed that the input impedance is so high that an attenuation across the resilient foundation of 20 dB is ensured which would typically be the case with the benchmark impedance above. In this case deformation of the top of the resilient mount alone determines the transmitted force. An attenuation of 20 dB means that motion of the foundation has an amplitude of one tenth of that of the machine and can therefore be disregarded when calculating the force.

The force is then simply:

$$F = v_{source} \cdot \frac{k}{2\pi f} \tag{5}$$

with

v_{source} source velocity
k dynamic mount stiffness

The force is to be calculated separately for each mount and each direction. The contribution of all forces can then be added energetically.

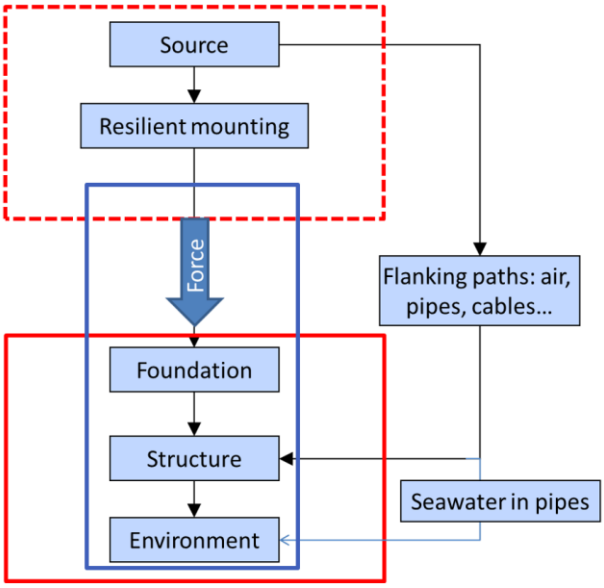

Figure 5: Structure-borne sound transmission model for resiliently mounted equipment

The condition of having a sufficiently high impedance also allows to completely separate a resiliently mounted source from the foundation with the force as the only interface value, i.e. a source measured on its resilient mounting system in the factory is absolutely suitable to determine the forces although it is mounted on a different – yet high impedance – foundation.

Application

A new model was erected based on the above findings and applied to the German Navy's defense research vessel Planet, Figure 6. No finite element analysis was made. The core of the radiation model is eq. (*4*).

As the radiation model yields the monopole level, i.e. the level of an omni-directional source in an unbounded environment at closest point of approach a model representing the measurement geometry (source depth, receiver depth, distance source to receiver) was added.

As the measured level the received average level evaluated between closest point auf approach of the mast and closest point of approach of the stern was chosen.

Figure 6: View of research vessel Planet

The design information of the ship which was commissioned in 2005 was not available to determine the exciting forces. Therefore, the existing structure-borne noise levels on the structure were used to infer the exciting forces transferred to foundations via the resilient mounting systems onboard.

There are high quality recordings of underwater radiated noise from the Heggernes sound range in Norway made at short distance provided by WTD71.

The ship has a diesel electric propulsion system with the diesel driven generators on the platform far above the waterline, double-resiliently mounted and enclosed. No signature of the same could be detected in radiated noise.

Prediction using the model revealed that structure-borne noise transmission prevailed with direct airborne noise transmission through the hull being less significant. With a quiet ship like this the other dominating noise source is the breaking wave system around the ship which is responsible for the levels at higher frequencies.

Figure *7* shows a comparison of calculation and measurement for one speed with excellent match considering the assumptions that have been made to find the exciting forces. Note that no information on structural details and radiation efficiency where used, it was just assumed that noise sources rest on impedances not lower than the benchmark impedance of Figure 4.

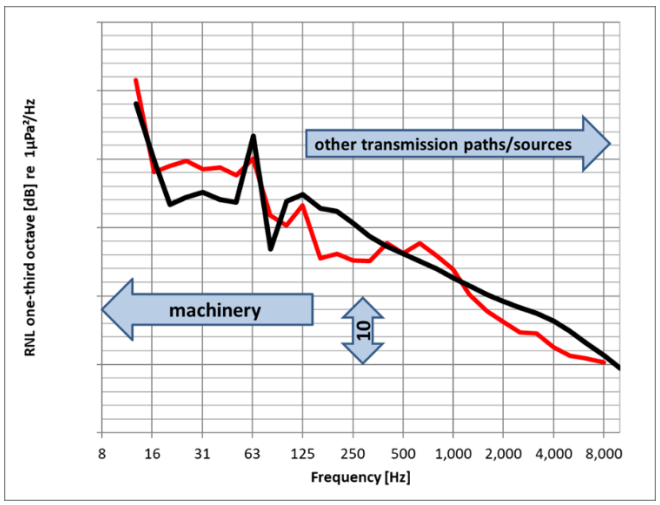

Figure 7: Radiated noise level *RNL* of Planet as measured (red) and as predicted (black)

The new model was applied to other ships of which detailed underwater radiated noise measurements were available. In all cases very accurate predictions could be demonstrated.

There is room for improvement for very low frequencies below about 25 Hz. In this frequency range phase relationships of the excitation become important. Further, measurement of radiated noise at these frequencies are less reliable than at higher frequencies, particularly if they are performed in shallow water.

Acknowledgements

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Literature

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[2] Ansys manual