

## Structure-borne noise reduction of gearboxes in maritime application

Robin D. Seiler<sup>1</sup>

<sup>1</sup> Hackforth Holding GmbH & Co. KG, Germany

### ABSTRACT

The reduction of noise and vibration levels on board ships offers not only occupational safety for the crew, but also increased comfort for passengers and less impact on the environment. In order to meet defined noise and vibration requirements, an effective noise control concept must integrate a variety of aspects and includes several coordinated counter measures. The foundation for this is laid in the engine room, because the excitement of the ship's structure in the engine room decisively shapes the vibro-acoustic situation on board. Here you can "gain" the most decibels. Single or double-elastic suspensions for ship engines already show high transmission losses, when rubber mounts are used. As a result, the structure-borne noise (SBN) via gearboxes becomes a crucial sound transmission path.

Against this background, this study investigates the central role of gearboxes in powertrains of ships. The presented countermeasures include different drive train configurations:

- 1.) Highly flexible couplings with low torsional stiffness between engine and gearbox,
- 2.) Propeller shaft couplings in combination with very soft gearbox mounting systems,
- 3.) Active vibration controls to reduce velocity amplitudes in the lower frequency region on the gearbox foundations.

The effects and potentials of the outlined solutions on the reduction of airborne noise (ABN) levels in receiving rooms on board are discussed.

Keywords: Gearbox, flexible coupling, resilient mounts

### 1. INTRODUCTION

The reduction of noise and vibration levels for work, navigation and accommodation spaces is beneficial for safety and health of seamen on board ships. Furthermore, low levels are demanded to meet up with permanently growing passenger expectations toward acoustic privacy and comfort.

In recent years, stricter limits for noise and vibration levels have been defined and published in international regulations and standards [1], [2], [3]. Parts of these limits have been integrated into the SOLAS convention (Safety Of Life At Sea) and have become mandatory for new ship construction since 2017.

While regulations, standards and the phenomena related to human performance and perception of noise and vibration request lower levels, the technical and design trends are heading in the exact opposite direction. For instance, more powerful engines and components that are optimized regarding performance emit higher noise and vibration levels. Lightweight designs react more sensitive to applied forces and accelerations. Cost pressure leaves little room for extensive secondary counter measures along the noise transmission path.

The shipbuilding industry is aware of these apparently opposite goals and many ship yards have started to integrate aspects concerning vibro-acoustics in their design processes. In order to be compliant with defined noise and vibration requirements, an effective noise control concept must integrate a variety of aspects and includes several coordinated counter measures. This survey deals with the possibilities of reducing structure-borne noise transmission of gearboxes by the use of resilient connections.

---

<sup>1</sup> robin.seiler@vulkan.com

## 2. METHOD

### 2.1 Noise Reaction Chain

The structure-borne noise produced in the machinery room is responsible for a significant portion of noise development on board. The foundation for planning noise insulation and vibration isolation concepts is laid in the machinery room. Here you can "gain" the most decibels. The majority of primary sources of noise and vibrations on ships - excluding the propellers - are located here, the main engines, auxiliary engines and gearboxes. Figure 1 illustrates an example of the reaction chain for SBN on board of a pleasure boat.

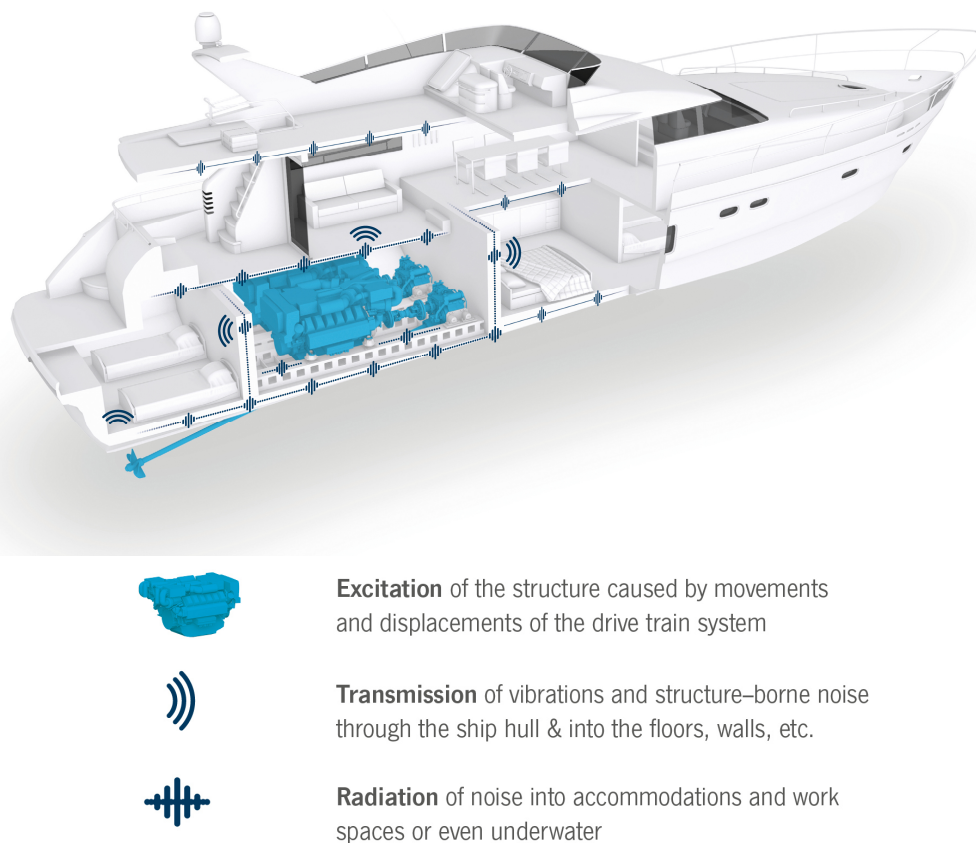


Figure 1 – Example for structure-borne noise reaction chain on pleasure boats

### 2.2 Structure-borne Noise of Gearboxes in Maritime Applications

The spectral analysis of measurement data, acquired on the gearbox shows that the sound typically consists of two major factors. The lower frequency range, particularly up to 300 Hz is dominated by spectral components that result from the transmission of engine orders and their harmonics. These could be torsional vibrations, which are converted into linear vibrations due to internal processes in the gearbox. In this case, the gearbox acts as a connecting link from engine to gearbox foundation / ship structure (see Figure 2).

Secondly, there is a tonal component in the mid-frequency range resulting from the meshing processes of the gear teeth. The so-called gear meshing frequency and its harmonics can be detected easily in the frequency spectrum, as exemplarily depicted in Figure 2 on the right side.

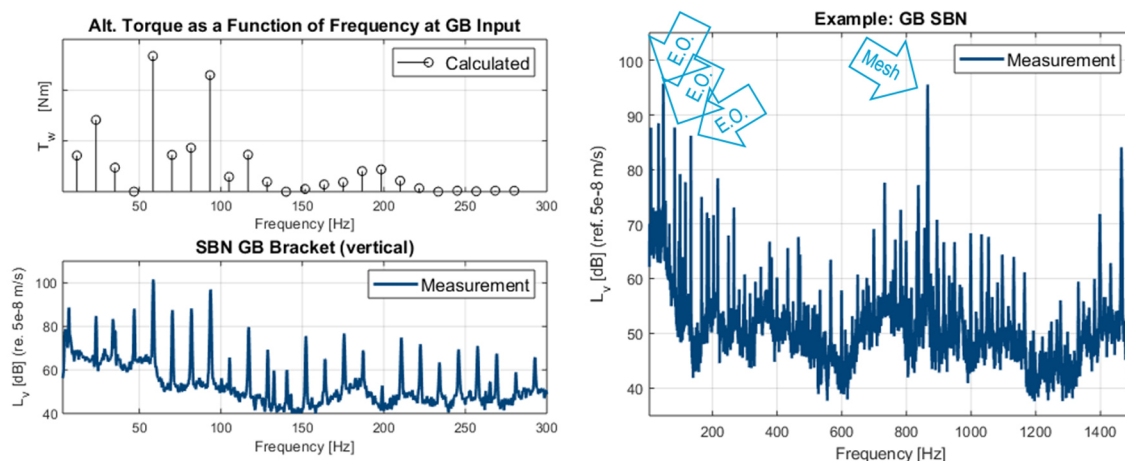


Figure 2 – Left side: Functional relationship between torsional and linear vibrations. Alternating torque at gearbox input (above) and velocity levels at gearbox bracket (below).

Right side: Velocity levels at gearbox brackets (E.O. - Engine orders, Mesh - Gear teeth meshing frequency)

## 2.3 Internal Mechanisms

For the sake of completeness, the essential aspects of internal measures for noise reduction are briefly mentioned. To reduce noise emission of gearboxes, primary counter measures that focus on internal mechanisms are first and foremost taken into account by the manufacturers of marine gearboxes. These measures aim mainly to affect the tooth mesh process as a significant source of noise.

The key points are the choice of the right main geometry of the gearing to optimize stiffness and variation over the tooth meshing. The choice of the right gearing correction to influence the displacements and deformations under load. The choice of the right gear quality to reduce the negative influence of manufacturing deviations, such as division errors, profile deviations or flank line deviations. Furthermore, the realization of low surface roughness by appropriate grinding procedures and grinding parameters as well as additional measures such as barrel finishing.

In addition to the noise optimization of the gear meshing effects, the use of plain bearings compared to roller bearings tends to yield positive results.

## 2.4 Approach

Apart from the statements in the previous section, a reduction of velocity levels at the gearbox foundation, can be achieved either by the help of highly flexible couplings, that reduce the transmission of vibrations and low frequencies into the gearbox or by structure-borne noise insulation of the mounting system which connect the gearbox and foundation. The main direction of transmission is the torsional direction for couplings (rotation around x-direction) and the vertical direction for resilient mounts (z-direction).

## 2.5 Limitations

The reduction of SBN levels at gearbox foundations, can be (but is not necessarily) measured / perceived by a reduced A-weighted sound pressure level in cabin, salon or workplaces on board ships, for instance. The extend of success of the applied countermeasure (section 2.5) is determined by a variety of factors.

The basic conditions that have to exist on board are the following: The structural damping of the ship structure from the point of excitation to the sound radiating structure of the receiving room has to be low. The proportion of the SBN path to the overall noise level in the receiving room of interest

has to be high and the contribution of other SBN is not dominant in the same frequency range. The implemented countermeasures have to be very effective at the gearbox foundation. The transmission of ABN generated by the engine and the gearbox into the areas of interest is negligible. Local ABN sources are not masking the achieved improvements. The benefit of the applied countermeasures can be controlled and evaluated at four stages along the transmission path: The gearbox, the gearbox foundation, different spots of ship structure and the receiving rooms. The causality is depicted schematically in Figure 3.

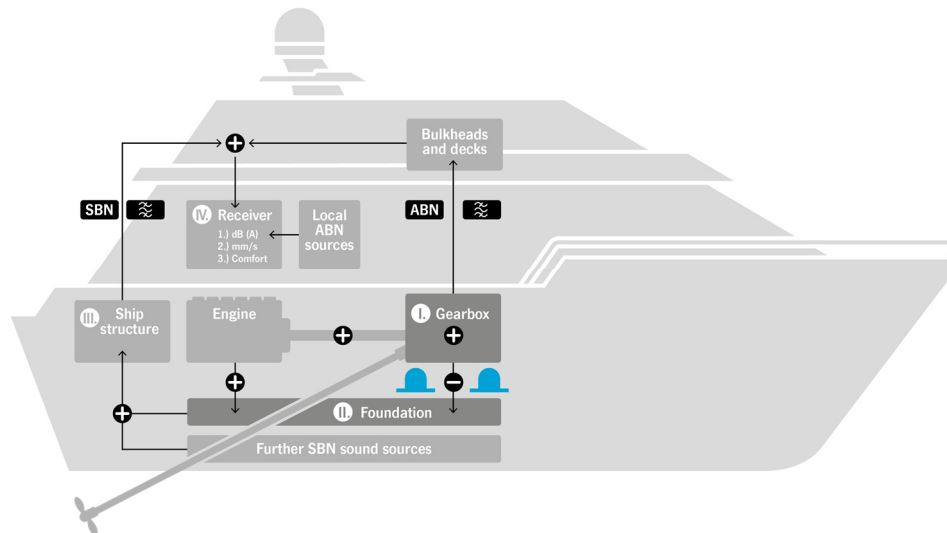


Figure 3 – Sound sources added along the transfer path

In the following section, two examples for a successful modification of drivetrain configurations by replacing resilient connections with more suitable solutions in terms of vibro-acoustics are given. The third example introduces an active vibration control device in combination with rubber mounts.

## 2.6 Three Scenarios

Case 1.) In the first example, the acoustic quality was significantly improved by exchanging and modifying the highly flexible coupling between engine and gearbox. Thereby, the original coupling was replaced by a product with lower torsional stiffness. By adding an additive intermediate mass, the coupling was tuned to achieve optimal results.

Case 2.) Good structure-borne sound isolation is achieved by mismatching the impedances between the bearings and the foundation. Soft resilient mounts on stiff foundations usually lead to good results. However, this type of execution is not possible if the propeller thrust has to be accommodated by the resilient mounting system. The second study outlines the benefit of a configuration consisting of a propeller shaft coupling with integrated thrust bearing in combination with a soft elastic gearbox mounting in contrast to a simple configuration with semi-elastic gearbox mounting only.

Case 3.) A promising option to reduce the transmission of induced vibrations and low-frequency sound components is the use of combined passive and active components. The active system acts by a destructive superposition of sound waves in the lower frequency range. The high-frequency vibration components can be effectively decoupled from the ship structure by semi-elastic rubber mounts. The concept of the interaction of both systems is shown schematically in Figure 4.

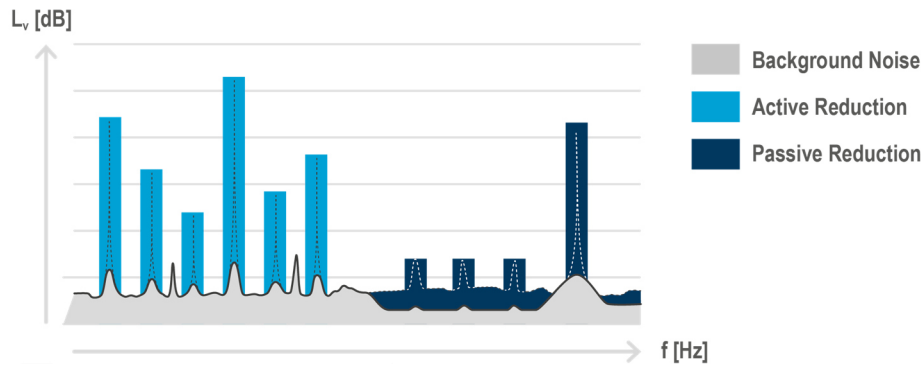


Figure 4 – Areas of influence for the active & passive systems as described for case 3

### 3. RESULTS

#### 3.1 CASE 1 - VULKARDAN E (HIGHLY FLEXIBLE COUPLING)

A convincing example for possible improvements of vibro-acoustic quality by the choice of the right products with suitable vibro-acoustic properties is summarized in the following. Lloyds Register performed the measurements and calculations presented in this section (3.1). The findings presented here have been released in [5] originally.

The cause for the investigations and the latter replacement of the flexible coupling was given by the exceedance of comfort limits at lower cruise speeds of a 70m yacht. Vibro-acoustic measurements have revealed that the noise problem was related to torsional vibrations and that the levels on the gearbox foundation were higher than levels below the engine. As the installed 3-row coupling was identified to cause resonance with the 4<sup>th</sup> engine order at lower cruise speeds, a high flexible couplings with lower torsional stiffness was installed.



Figure 5 – Drivetrain modification case 1: Installing a two-row coupling with low torsional stiffness and additive intermediate mass

In a second step, an additional intermediate mass was added to ‘detune’ the propulsion system. The A-weighted sound pressure levels and vibration RMS values of the three configurations are shown in Figure 6.

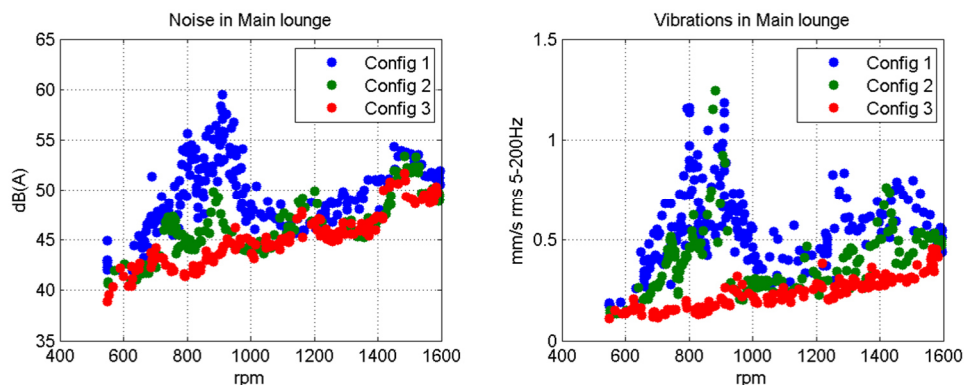


Figure 6 – Left side: ABN Levels in the main salon for different configurations. Right side: Vibrations measured in the main salon for different configuration

Herein, configuration 1 is the original configuration with the 3-row coupling. Configuration 2 is the one with the two-row coupling with low torsional stiffness. Configuration 3 represents the final solution with an added intermediate mass. The acoustic quality in the main salon could be improved significantly, especially in the cruise range between 700 and 1100 RPM. A maximum of 14 dB reduction of the overall noise level could be achieved as depicted in Figure 6.

### 3.2 CASE 2 - PROPFLEX (PROPELLER SHAFT COUPLING AND THRUST BEARING)

This case study was carried out by comparing two yacht designs, which – beside the variations in drive-train systems shown in Figure 5 - are identical in terms of basic design.



Figure 7 – Drivetrain modification case 2: Replacing semi-elastic gearbox mounts by a soft gearbox mounting systems in combination with a propeller shaft coupling and an additional thrust bearing.

While the drivetrain of yacht A illustrated on the left hand of the Figure 7, is equipped with semi elastic gearbox mounts, yacht B is outfitted with a more complex set up consisting of a flexible propeller shaft coupling and elastic gearbox mounts. In comparison to the semi-elastic mounts, the elastic mounts show a higher transmission loss, especially in the lower frequency range. The improvement of acoustic quality, in this case can be understood as a reduction of the A-weighted sound pressure level in different receiving rooms, is depicted in Figure 8.

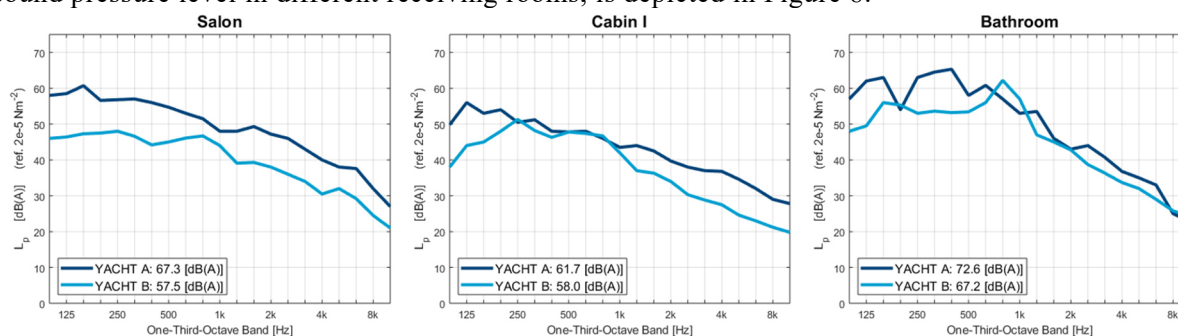


Figure 8 – Airborne noise levels in different receiving room at 1600 RPM

The overall A-weighted ABN levels reduction that could be achieved, averaged over receiving rooms are 4,5 dB at 1600 RPM, 3 dB at 1800 RPM and 4 dB at 2100 RPM [4].

### 3.3 CASE 3 – VOICE COIL ACTUATOR SYSTEM (ACTIVE VIBRATION CONTROL)

Depending on the design and the tuning of the coupling, certain vibrations are transmitted from the engine to the gearbox. The correlation between engine orders and linear vibrations has been depicted in Figure 2 already. As the lower frequency range is dominated by tonal components, the peaks can be minimized by the use of active vibration control. By the combination of active control devices and semi-elastic gearbox mounts – that are able to absorb propeller thrust – the effective noise insulation is extended towards lower frequencies.

The destructive superposition of additionally injected, phase inverted signals, aims to reduce the amplitude peaks in the spectrum that are resulting from the excitation by engine. The actuators used for this purpose are mounted to the gearbox on lateral flange plates. The force inducing assembly consists of voice coil actuators of cylindrical design. The basic operating principle corresponds to that of the plunger coil, where an alternating linear movement of a magnetic rotor is generated by applying an alternating magnetic field with the help of a coil surrounding this rotor [6].



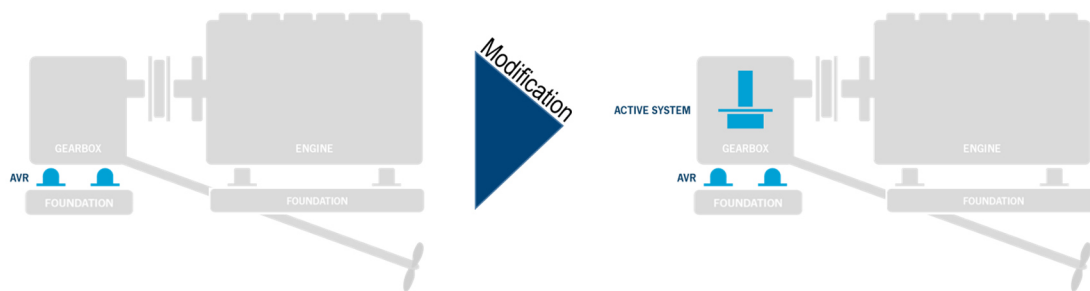


Figure 9 – Drivetrain modification case 3: Active vibration control on gearbox flange

Figure 10 shows the results measured at the first on board tests.

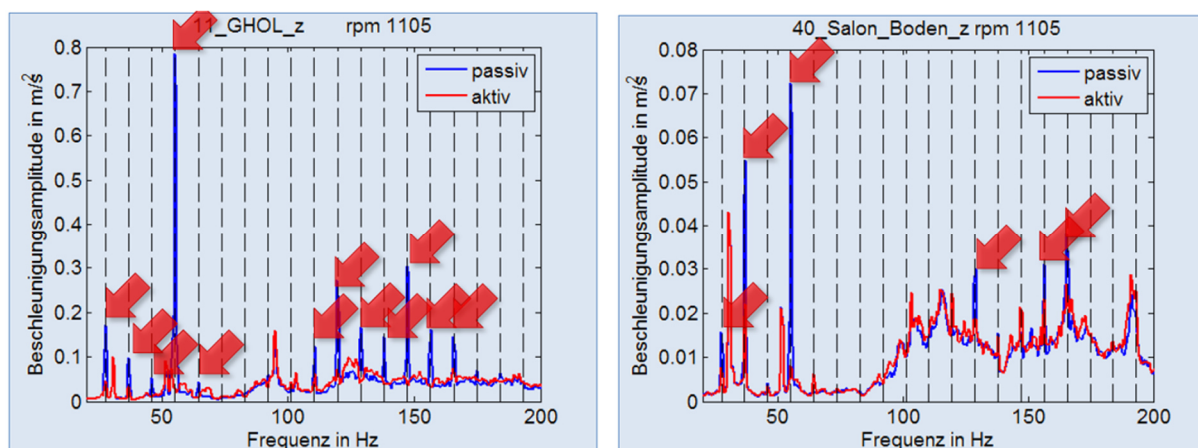


Figure 10 – Comparison of acceleration amplitudes: Active (red) vs. passive (blue)

The blue curves in the figure above represent measurement data acquired when the actuators are switched off. The red curves show the achieved improvements by using active vibration control. The red arrows highlight the harmonics of the engine orders that have been allocated by the control algorithm. The left diagram displays data that were measured on the gearbox flanges. The right diagram shows data that were measured at the floor of the salon above the machinery room, both at approximately 1100 RPM.

#### 4. DISCUSSION

The dynamic transfer stiffness of rubber couplings – and resilient rubber elements in general – is influenced and dependent on a vast number of parameters, such as temperature, frequency, amplitude and preload. For the example, as described in section 3.1, the calculated results meet the measurement data, if the stiffness is factorized by 2.7 for the calculations [4]. This value was found with the help of reverse engineering.

For the selection of couplings in acoustic sensitive applications as well as for the prediction of noise levels, it is advisable to give more importance to the stiffness values, in order to avoid unwanted resonances. This is especially recommended for a frequency range lower than 300 Hz. A safety margin of  $\sqrt{2}$  is a practical approach. In higher frequency regions, the amplitude response shows more complex behavior due to the phenomenon of standing waves in the rubber material. Resonances cannot be avoided throughout the whole frequency range. The noise issues that occurred on the 70m yacht could be clearly assigned to a resonance with high excitation levels. By detuning the systems with the help of a highly flexible coupling with lower torsional stiffness and additional tuning mass, a significant reduction of noise in the salon above the machinery room could be reached. Up to 14 dB improvement of A-weighted sound pressure levels for certain cruise speeds could be realized with low cost efforts. Improving this order of magnitude can only be a correction.

By installing a propeller shaft coupling and thrust bearing combination, the use of a soft mounting system was made possible as depicted in Figure 7 for the second example. Thereby, the overall noise levels could be reduced throughout the entire ship. This is highly welcomed as the noise levels found on the yacht A (equipped with a semi-elastic mounting system for the gearboxes) range from 54 dB(A) to 80 dB(A) for accommodation spaces and can be considered to be quite high for a yacht. The measurement data for yacht B (equipped, as describe above) range from 48 dB(A) to 73 dB(A). When considering individual receiving rooms, improvements from 1 up to 10 dB have been recorded. Although the modifications can be evaluated positively, further countermeasures to increase acoustic comfort appear mandatory. As already mentioned, the measurement data have been acquired during sea trials. Therefore, a slight influence of weather conditions on the measurement results cannot be excluded as the sea trials were carried out on different days.

For case 3 it can be stated that the dominant acceleration peaks in the amplitude spectrum, which have been selected to be controlled by the algorithms functional pattern, could be minimized locally up to a frequency of approximately 200 Hz by the active system. Single peaks could be reduced up to 30 dB and more. The overall level for the allocated frequency range could be reduced up to 10 dB. The potential of the active control is even higher but the results were limited due to the high intensity of the existing background noise floor. This regulation was observed on the gearbox flange and on the gearbox foundation. Hence, coherent signal components measured on different areas of the ship structure also show significant reduced velocity peaks. This effect complies with the subjective perceived changes on board when the actuator system was turned on/off during the sea-trial.

Further development potential for improvement is expected to be found by optimizing the controller strategies and in the performance of the frequency transition between the active and the passive noise isolation systems.

## 5. CONCLUSIONS

Three different drive-train modifications have been introduced in this work. All of them aim to reduce vibrations and the structure-borne noise transmission from gearboxes into the ship structure by the use of resilient connections. The effects and potentials of the outlined solutions on the improvement of airborne noise levels in receiving rooms on board have been discussed. It has been revealed that minor changes can have quite an influence on ABN and improve acoustic quality and comfort. The approach implies that a series of basic requirements have to be met. Nevertheless, building silent ships without effective isolation of the highlighted transmission path appears to not be feasible. It should be noted that the relations between the torsional system and its implications on linear vibrations and structure borne noise excitation of the ship structure is a crucial aspect.

Improvements of vibro-acoustic properties represents great development potential for the shipbuilding industry, particularly since these properties are factors that significantly determine the serviceability of the overall product.

## ACKNOWLEDGEMENTS

The data provided by Mr. Per Tellefsen, Mr. Roberto Masarino, Mr. Robert Wimmel and ERAS, form the basis for the present document. Their cooperation and support is gratefully acknowledged.

## REFERENCES

1. International Maritime Organization. Code on noise levels on board ships. MSC 337 (91): 2012.
2. International Standard Organisation. Guidelines for measurement, evaluation and reporting of vibration with regard to habitability on passenger and merchant ships. ISO 20283-5: 2016.
3. International Standard Organisation. Guidelines for measurement, evaluation and reporting of vibration with regard to habitability on specific ships. ISO 21984: 2018.
4. Masarino R. EMS Measurement report - Confronto rumore. Novi Ligure, Italy: unpublished.
5. Tellefsen P, Cunningham K. Noise and vibration challenges due to torsionally excited gearbox vibration. Torsional Vibration Symposium; 21-23 May 2014, Salzburg, Austria.
6. Ickler A, Bokemeyer J. Abschlussbericht VGM IIb. Göttingen, Germany: unpublished