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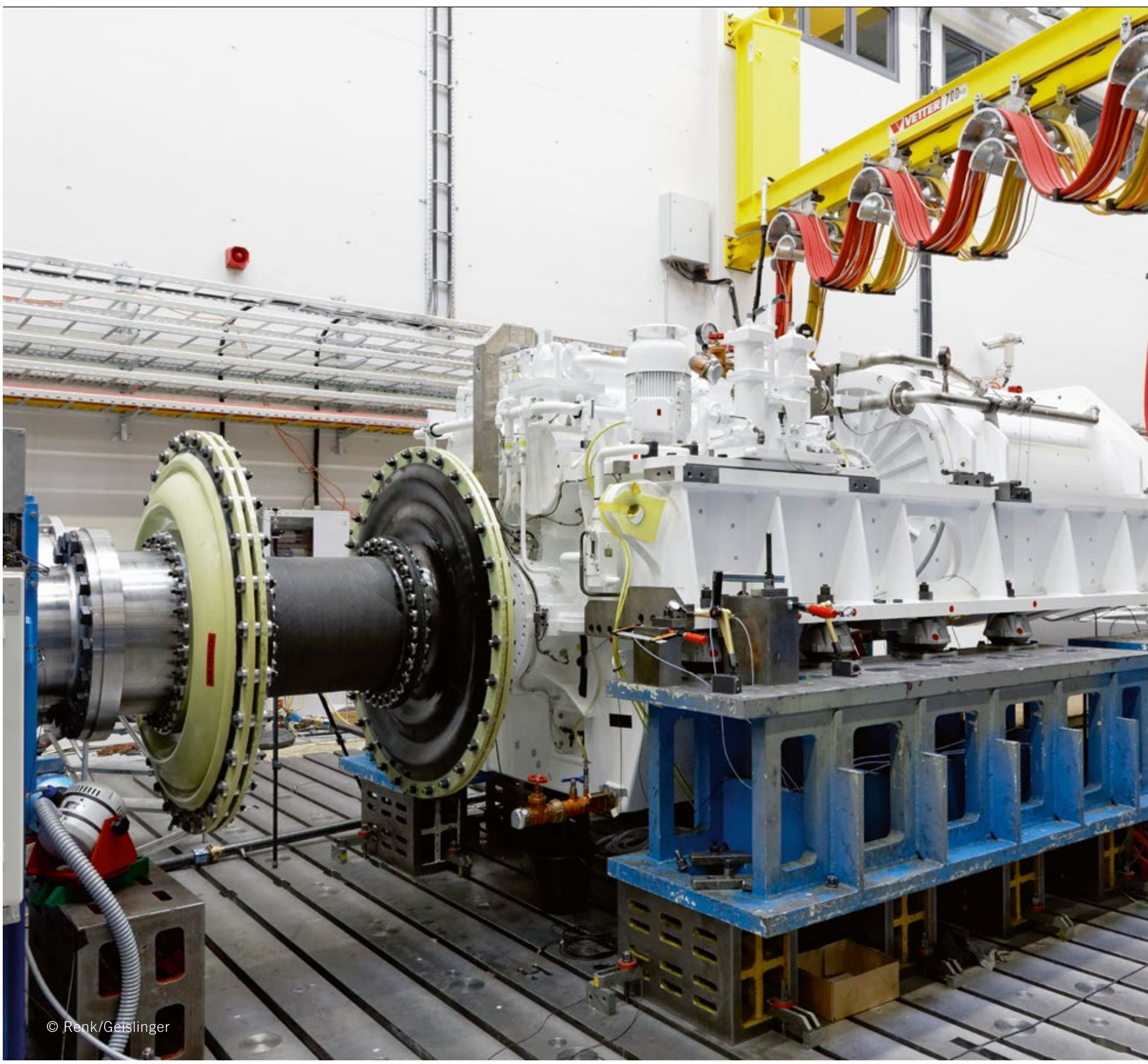
Vibration Isolation of Large Machinery

Machinery generating or transmitting torque can contribute significantly to the noise emissions of a mechanical system. To assess potentials for reducing acoustic signatures in marine propulsion systems, Renk and Geislinger have conducted tests on a Renk Advanced Electric Drive and a Geislinger Silenco coupling. This topic will be presented at the 2nd Torsional Vibration Symposium, 17th to 19th May 2017, Salzburg, Austria.



1 MOTIVATION

Continued and increasing demand for quiet propulsion machinery exists in the shipbuilding industry. For research vessels the aim is to improve fish avoidance behaviour and signal-to-noise ratios for acoustic surveying. For yachts, the end-customer expectation may be shaped by onshore building noise levels, while for navy vessels low detectability is the major consideration.

Major sources of noise in ships are breaking surface waves, vibrations excited by the hydrodynamic interaction of the propeller and the noise of machinery transmitted and radiated by the ship's structure. The latter can be minimised by reduction of the generated noise at the source, and by reduction of the noise transmitted to and within the ship structure. Usually the engine is a major source of noise, and therefore is of primary interest in source level reduction.

For a gearbox many options exist for noise source reduction, reaching from consideration of acoustic demands at the project stage to optimised bearings, gear flank micro-geometry optimisation and double-helical gearing. Vibrations from the machinery to the ship are primarily transmitted as structure-borne noise. Acoustic isolation of this noise path can be achieved by resilient mounting of the source component. Typically, this is carried out with elastomer elements between the component and the machinery foundation as well as elastic misalignment couplings connecting the machinery foundation and the propulsion component in the ship.

2 THEORETICAL BASIS

The resiliently mounted system has the characteristics of an oscillator. For explanation of the underlying principle and its application to noise isolation, the simplified case of a one-dimensional mass-spring oscillator is considered first.

A mass suspended by a spring has one natural frequency. The reaction of the system on force excitation at the mass is strongly dependent on frequency. Static and low frequency excitations are transmitted through the spring with no or only minor amplification. Close proximity of the exciting and natural frequency lead to resonant behaviour, where resulting force amplitudes in the spring can be significantly higher than the excitation force amplitudes.

This behaviour is inverted for an excitation frequency distinctly above the natural frequency, where the force transmitted in the spring is lower than that of the excitation. The vibration isolating effect is frequency-dependent, thus the transmitted vibration decreases with 12 dB per octave above the natural frequency, as shown by the black curve in **FIGURE 1**. This principle is applied in resilient mounting systems.

The natural frequency of the system has to be significantly lower than the noise frequencies to be isolated from the ship structure. But in practice it is a system with higher harmonics, which may cause additional resonances, as shown in the blue curve in **FIGURE 1** [1].In applying this principle, structure-borne noise originating in machinery can be attenuated by a resilient mounting

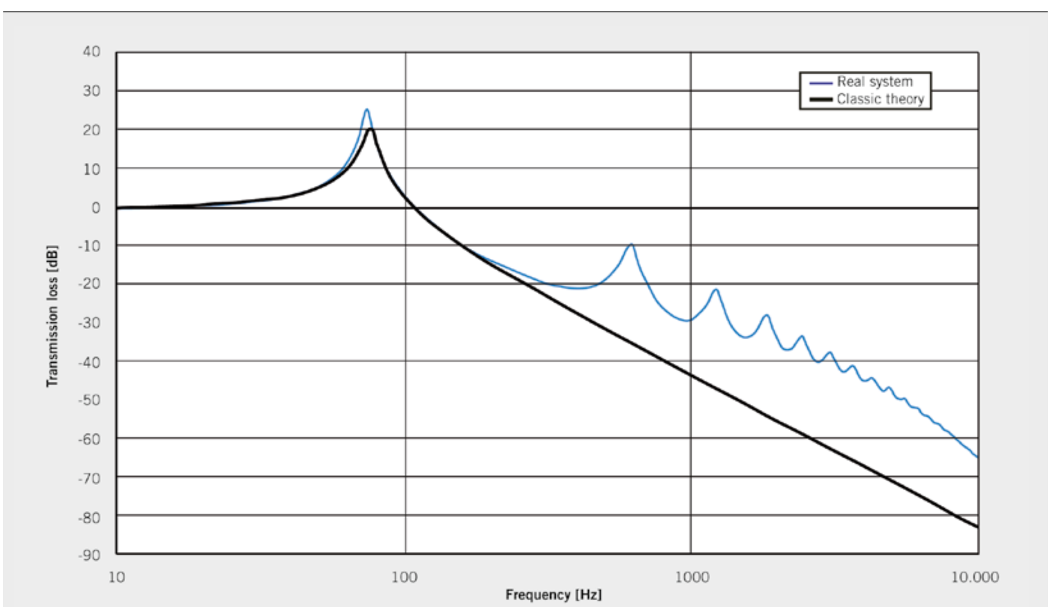


FIGURE 1 Transmission loss of mass-spring oscillators (© Renk/Geislinger)

system with a natural frequency significantly below the relevant excitation frequencies.

3 CHALLENGES IN APPLICATION

In reality, the theory of the mass-spring oscillator finds its limits where the assumption of the underlying simplified model is no longer valid, as suggested by the blue curve in **FIGURE 1** [1].

Further, a resiliently-mounted rigid object has six degrees of freedom (three translational, three rotational), resulting in six natural frequencies.

For ship propulsion systems several major force, torque and imbalance excitations have to be considered. These usually are the diesel engine ignition frequency with harmonics, engine and propeller shaft revolution frequencies, and propeller blade passing frequencies with harmonics. All these frequencies change proportionally with operating speed, which for the typical ship has a range of 40 to 100 % of nominal speed.

Thus, in application of the characteristics of the mass-spring-oscillator, the six natural frequencies are usually chosen to reside above the frequencies of the described excitations to avoid resonant operating conditions, but far enough below the frequencies of the acoustics-relevant excitations to be isolated.

Every connection of the machinery to its surroundings provides a transfer path for structure-borne noise and vibration. The full potential of a resilient mounting system is achieved only if all other transmission paths are also considered.

The way that noise is generated, transferred through a structure and finally radiated into the surrounding area can be described with the general equation of machine acoustics [2], as shown in **FIGURE 2** and Equation 1:

Eq. 1 $$P_i(f) = \tilde{F}_i^2(f) \, \frac{1}{Z_{E,1}^2(f)} \cdot T_{v,i}^2(f) \cdot S \cdot \sigma(f) \cdot (\rho \cdot c)$$

The equation includes the following parameters:

P = acoustic power
F = excitation force
Z_E = input impedance
T_v = transfer function of the structure-borne noise
σ = radiation factor
$(\rho \cdot c)_{Air}$ = impedance of the surrounding media (usually air)
S = radiating surface.

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Typically, there are two critical transfer paths of the structure-borne noise from the engine into the ship's hull, where the structure-borne noise is also the source for airborne noise transferred into the ship's cabins and for underwater sound radiated to the environment. The primary transfer path goes via the engine mounts and the gearbox into the ship's structure, as shown by the red lines in **FIGURE 3**, and the secondary path goes along the powertrain: via coupling, gearbox, gearbox mounts, shaft and its bearing into the ship's hull, as shown by the blue lines in **FIGURE 3**. Solutions to minimise the sound-transfer along the primary path are for example soft, optimised mountings as described above or active systems [3]. For the secondary path, which has come to be of interest in recent years, an elastic coupling absorbing vibrations and compensating for misalignment is the best solution for noise reduction.

Geislinger's contribution to this challenge was the development of the new lightweight Silenco coupling. To minimise the transfer of structure-borne noise for components from the modular system of the coupling with different impedances have been tested and optimised. They are made of selected composite materials and steel with the best acoustical performance. In addition, special flanges made of a combination of composite material, rubber and steel have been developed. Designed to avoid the resonance effects of its components, the coupling ensures a broad-band reduction of the transfer of the structure-borne noise.

4 SPECIFICS OF LARGE PROPULSION SYSTEMS

The mass of the resiliently-mounted machinery can typically not easily be modified, so only stiffness and location of the spring ele-

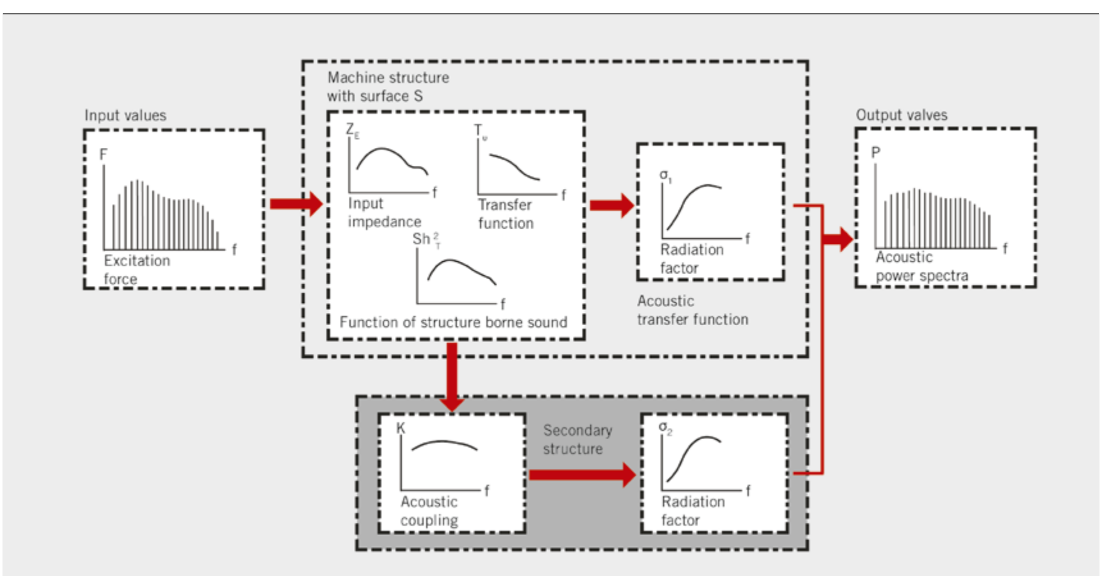

FIGURE 2 An illustration of the factors of the general equation of machine acoustics (© Geislinger)

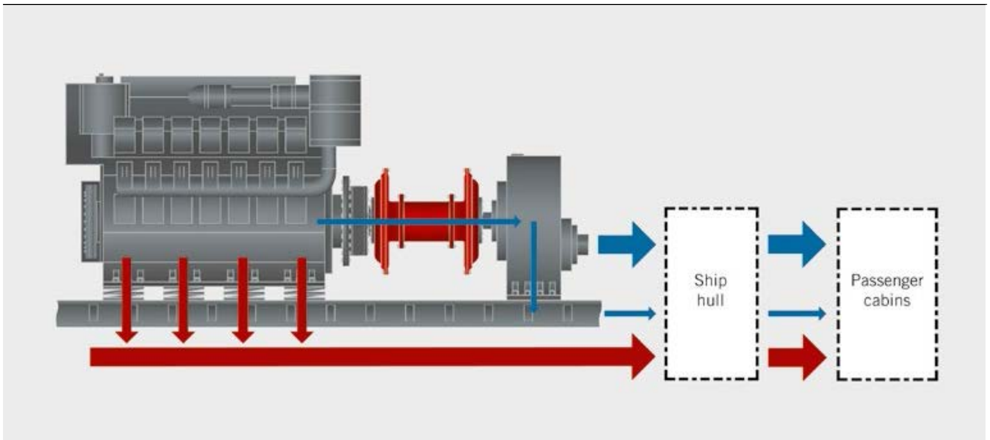

FIGURE 3 The primary (red) and secondary transfer path (blue) of the structure-borne noise (© Geislinger)

ments remain as adjustable parameters for the optimisation of the resilient mounting system. Since the mounting locations are usually fixed early in the design process, the only free variable to influence the natural frequencies is the stiffness of the elements.

Natural frequency results from mass and stiffness. With the mass and geometry of the machinery fixed, the stiffness of the resilient mounting elements is a direct result of the targeted natural frequencies. To achieve a specific frequency, the stiffness has to increase proportionally with the mass. Thus, for heavy machinery units, proportionally stiff mounting elements are required.

At the same time, large structures can no longer be considered as rigid, but possess flexibility and low-frequency modes themselves. This is the case for larger propulsion machinery as well as for ship structures.

In sum, for heavy machinery the model of a rigid mass connected by a relatively soft spring to a rigid foundation is no longer valid *a priori*. Instead there exists potential for dynamic interaction between the shipside machinery foundation of limited stiffness, the resilient mounting system and the stiffness and vibration characteristics of the machinery itself. These have to be taken into account to achieve optimal vibration isolation with heavy machinery.

In general, for larger structures and components, rigid body assumptions approach their limits, while internal flexibility and vibration modes need special consideration. Possible solutions are specific stiffening of propulsion components and ship structures close to the mounting points, optimisation to avoid natural frequencies at known excitations by FE methods, and location of vibration nodes close to mounting points.

5 CHALLENGES IN IMPLEMENTATION

The principles described have to be applied in the design process to achieve a quiet propulsion system. However, their application is necessary, but not sufficient.

If intrinsic generation of vibration and noise is avoided by appropriate design measures, remaining noise excitation levels are largely dependent on tolerances and manufacturing quality.

Natural frequencies within structures and of resilient mounting systems can be determined and optimised by the application of FE and MBS methods. However, some material and virtually all damping parameters are usually not well known. In this respect, the elastomer material of the mounting system poses a major challenge for predicting the actual vibration and acoustic characteristics of a propulsion system. In addition, the single component with the lowest vibration isolating behaviour determines the overall effectiveness of a resilient mounting system.

To summarise, the acoustic performance of a propulsion component and its isolation is dependent not only on design, but equally on material and component characteristics, as well as close attention to detail during manufacturing. In this way a specific quietness level cannot be achieved solely from theory and its application in design but is dependent on optimisation with prototypes, experiments and measurements. Likewise, measurements with scaled-down models are not enough since these will not show the characteristics pertinent to large and heavy machinery.

6 EVALUATION OF A SYSTEM DESIGNED FOR QUIETNESS

Renk has developed the Advanced Electric Drive (AED), a propulsion unit consisting of an electric motor and a gearbox on a common steel raft. The design is optimised for low noise emissions based on the theoretical principles outlined above. For isolation of the structure-borne noise, the AED propulsion unit is equipped with a resilient mounting system. A remaining path for structure-

FIGURE 4 Acoustically optimised design of the Geislinger Silenco coupling (© Geislinger)

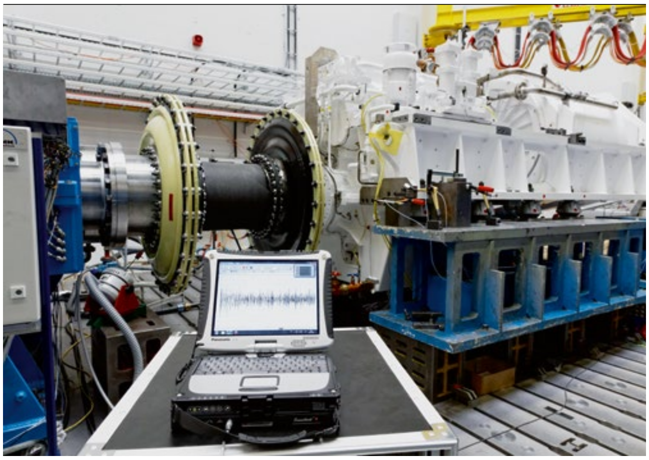

FIGURE 5 Test of the AED system including its acoustically optimised coupling at the Renk test facility (© Renk/Geislinger)

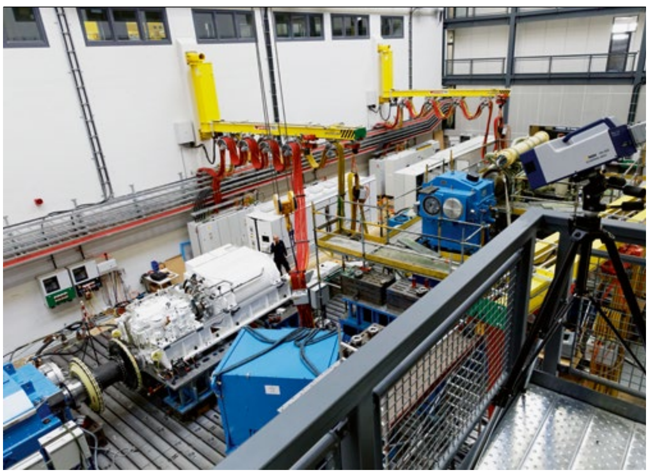

FIGURE 6 Measurement of surface velocities using a Laser-Scanning-Vibrometer (© Renk/Geislinger)

borne noise is the connection of the powertrain and the propeller shaft, through which the resilient mounting isolation can be circumvented. To avoid this problem, Geislinger has developed the Silenco coupling which combines noise isolation between the output flange and the propeller shaft with compensation of the misalignment and the movements of the AED frame under load. Depending on the acoustical needs and the required torque (up to 186 kNm), different versions of the components (flanges, membrane and shaft) are available. In this way, this new and highly flexible, lightweight acoustic coupling sets higher standards for the acoustic attenuation of powertrains. The design of the lightweight coupling is shown in **FIGURE 4**.

The design of the AED system as well as the adapted coupling was subject to extensive testing at the new Renk test facility in Augsburg, Germany. With 1,250 m² floor space and up-to-date equipment, the facility provides testing capability up to 12 MW power at up to 11,000,000 Nm torque and is thus suitable for full-scale experiments and tests on large machinery. The set-up is shown in **FIGURE 5**. The measurement programme included operating tests up to full power (4 MW) using laser-vibrometry as well as classic acceleration sensors for data acquisition. In addition, transmission loss measurements were made using an electrodynamic shaker.

Laser vibrometry was applied to identify the frequencies and mode shapes of the resilient mounting system and of the AED frame, as shown in **FIGURE 6**. A laser Doppler vibrometer is a scientific instrument which is used to take non-contact vibration measurements from a surface. Its laser beam is directed at that surface, and the vibration amplitude and frequency are derived from the Doppler shift of the reflected laser beam frequency due to the motion of the surface. The output signal is directly proportional to the target velocity component along the direction of the laser beam.

In the left and right parts of **FIGURE 7**, two of the major operational modes of the AED system are shown. The intensity of the red and green colours indicates the surface velocity towards or away from the position of the laser. In addition, the illustrations show the shape of the modes as a 3-D offset from the measured structure. The measurement was done under full load (4 MW at 1800 rpm).

The structure-borne noise levels were measured during AED operation above and below the elastic mounts. The resulting graphs are plotted in **FIGURE 8**.

The gear meshing frequencies with first harmonic are designated. Although, as expected, they can be recognised in the narrow band graph above the resilient mounts, the success of application of the principles of noise source reduction is easily visible in the corresponding 3rd octave bands graph. While the gear meshing contributes to the overall noise level, it is not the single dominating source component.

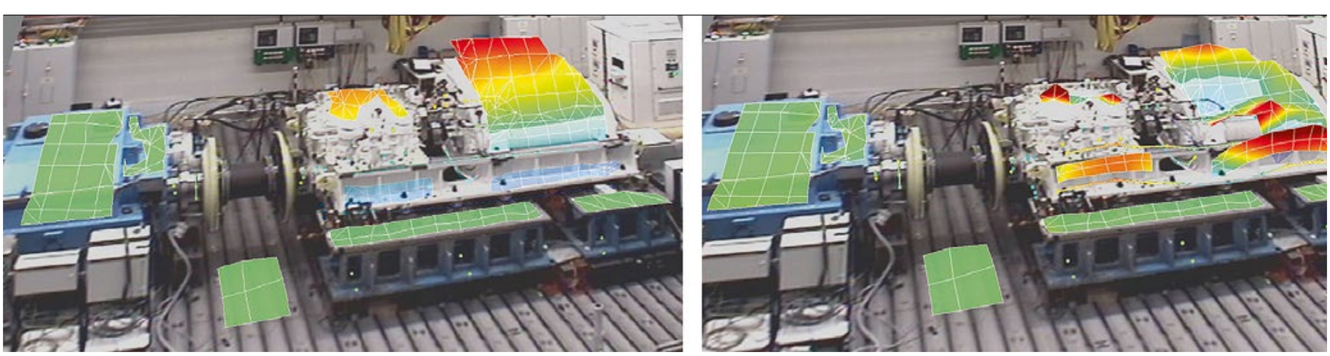

FIGURE 7 Operational modes in the in the 16 Hz (left) and 250 Hz octave bands (right) under full load (4 MW, 1800 rpm) (© Renk)

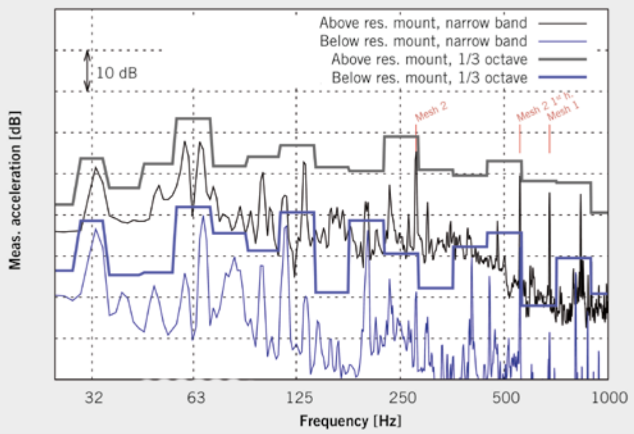

FIGURE 8 Measured structure-borne noise levels for the AED (© Renk/Geislinger)

Broad band noise and various additional frequencies are present in the measurements above and below the elastic mounts. Nonetheless, evaluation of the vibration isolating performance of the resilient mounting system is possible from excitations known to have a physical origin at the AED. For the designated gear mesh frequencies, at about 300 and 700 Hz a reduction in structure-borne noise levels of more than 30 dB is evident.

The combination of low source noise levels originating from the Renk AED, together with high attenuation by elastic mountings and the Geislinger Silenco is thus able to significantly reduce the amount of structure-borne noise introduced from the propulsion system into the machinery foundation and ship structure.

7 SUMMARY

While the theoretical principles are well known, in practice optimal vibration and noise control is only achieved through a combination of the application of the principles of silent machinery during design, together with measurements, experiments and iterative optimisations using full-size components. This is especially important for large machinery units for which the assumption of spring-mounted rigid bodies comes to its limits. New and established techniques for running prototypes under realistic load and mounting conditions, used at the Renk test facility in Augsburg, Germany, in combination with state-of-the-art measurement equipment like laser vibrometry, allow research to take a major step forward. Combining simulation and real-life tests in such a way, broadband noise and vibration isolation of heavy machinery is improved significantly.

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