# Development and testing of silencers for a new generation turbocharger for reducing compressor noise on large diesel engines in maritime environments

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## Introduction

The new edition of the "Noise Code" published by International Maritime Organization (IMO) is currently augmenting the significance of the noise issue for applications in the maritime area. As admissible noise levels have no longer just recommendation character but are to be kept mandatory under SOLAS convention, noise levels on board are additionally being limited in customer contracts and regulations in the areas of ship's safety, work safety and emission protection (IMO Noise Code, DNV Comfort Class, etc.). Requirements for large engines, including engine components like the charging system, are derived from these specifications.

In general engine Noise & Vibration topics in maritime applications are divided in different sections: Vibrations are related to the engine including attached components. Target is a low vibration level, as an indicator for reliable engine operation. Structure borne noise above and below the engine mounts is considered because it is transferred to distant ship areas and radiated as airborne noise and as underwater noise. Maintenance work in the engine room requires sufficiently low airborne noise levels, which is significantly influenced by direct and indirect engine noise.

In terms of airborne noise MAN Diesel & Turbo SE (MDT) is required to comply with sound pressure levels (SPL) in the machine room not exceeding a limit value of 110 dB(A) in a distance of one meter to the engine surface at any possible location. In addition to the dominant engine emission, influencing parameters, including the number of engines, room size, wall design and set-up position, are outside MDT's influence. With the latest revision of related IMO code also admissible levels in accommodation areas have been lowered about 5 dB(A). Especially in small ships this is associated with somewhat greater expenses regarding insulation measures in the engine room respectively on the engines itself. In many cases the Turbocharger (TC) noise mainly contributes to measured overall SPL around the engines.

Since overall system responsibility lies with the shipbuilder, the engine manufacturers are more and more forced to reduce noise directly at the sources. TC suppliers are in turn encouraged to lower sound emitted by their products.

## Turbocharger Noise

Looking at the measured noise spectrum of a large-scale TC one quickly realizes that, at mostly relevant part und high load operation, a tonal noise and its harmonics dominates the overall sound. Those frequencies are associated with the so called Blade Passing Frequency (BPF) of the radial compressors. A typical frequency spectrum, measured at the air intake, is shown in Figure 1.

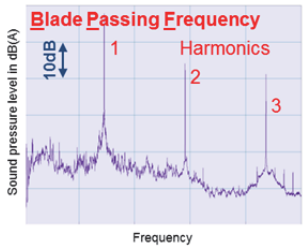


**Figure 1:** Typical frequency spectrum measured at compressor intake

During the last years radial compressor noise was topic of research work focused on understanding the underlying aerodynamic source mechanism including the noise behavior at different operating condition for several configurations. It was noted that the sound field at both, the intake but also the discharge side, is dominated by tonal components [1]. While noise is mainly caused by "rotor-alone-noise" at the compressor intake, the origin of sound generation at the discharge side is additionally based on the rotor-stator and rotor-volute interaction in case of a radial compressor equipped with vaned diffusor and spiral housing [2]. However, due to the fact that the sound field especially on the compressor outlet is highly depending on the mentioned phenomena and propagation to the compressor volute, time- and cost consuming transient 360° simulations are necessary to predict noise using industrial applied CFD-methods. Reducing noise emission furthermore conflicts with other major targets, e. g. compactness, efficiency and structural loads. For those reasons passive measures are still necessary and most commonly used to lower TC noise on engines.

## Intake silencer – Overview

In related marine applications combustion air is sucked from the engine room through filter silencers, which are directly attached to the compressor casing intake.

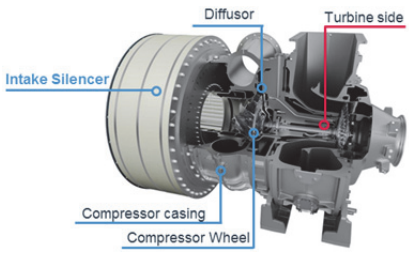


**Figure 2:** Large-scale TC with axial turbine and intake silencer (left)

The general design of large-scale TC is illustrated in Figure 2. Compressor noise measured without silencer at the air intake can easily reach levels of 140dB and above.

Those silencers are working according to the principle of absorption / baffle silencers. Figure 3 shows a part of the cross section through a typical intake silencer. Within given outer dimension, the noise attenuation characteristics can be influenced by the number of damping elements (absorbers + cover sheets), their geometry and used porous materials.

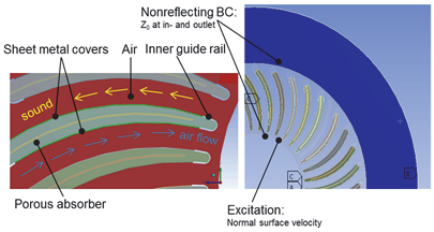


**Figure 3:** Principle simulation setup

After a basic layout was done taking in consideration the general layout parameters, a low-cost simulation is used to predict Transmission Loss (TL, $D_{TL}$) with the aim of optimizing geometry and used materials. In that case a cross section of the silencer is modeled with FEM and a quasi 2D calculation is performed for only one element layer. Figure 3 also contains a description of the schematic model layout. In particular, the inhomogeneous Helmholtz equation is solved and $D_{TL}$ is calculated using sound power at acoustical in- ($P_{In}$) and outlet ($P_{Out}$).

$$D_{TL} = 10\log\left(\frac{P_{In}}{P_{Out}}\right) \qquad [\mathrm{dB}] \qquad (1)$$

In case media are homogeneous the wave propagation in porous materials can be descripted by complex parameters impedance $\underline{Z}_c$ and wave number $\underline{k}$, whereby those values need to be measured according to transfer-matrix approach of Song & Bolton [3] in an extended impedance tube. A very simple way to predict the parameters based on empiric studies was introduced by Delany & Bazley (DLB) [4] and later modified by Miki [5] in terms of improving predicted values for low frequencies in case of materials with high fluid resistivity or multi-layer structures. Only the fluid resistivity Ξ must be known for calculating $\underline{Z}_c$ and $\underline{k}$.

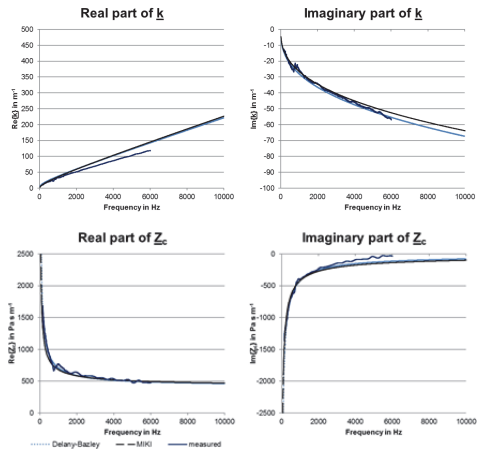


**Figure 4:** comparison of predicted (Delany-Bazley, Miki) and measured characteristic impedance and wave number of a fleece material used in TC intake silencers

To proof validity of DLB respectively Miki-model, complex absorber parameters $\underline{Z}_c$ / $\underline{k}$ and flow resistivity Ξ have been measured for a typical material (see Figure 4). This shows that the acoustical behavior of those materials used in MDT intake silencers can be approximated with good accuracy by using mentioned empirical models.

The modelling of the perforated sheets, which covers the absorbing material with the purpose of protecting it from high flow velocities, short term temperature peaks and dirt pollution, is carried out in an comparable way with the help of analytical models. A 2X2 transfer admittance matrix is used to describe the acoustical behavior of sheet metal plates in the simulation model. The acoustical impedance can be calculated with the help of empirical formulas, which have been proposed my Mechel [7] for example.

Finally the calculated TL was verified by experimental investigations. In particular, the Insertion Loss (IL, $D_{IL}$) of silencer damping elements was determined for two different absorbing materials (different surface weights) using a loudspeaker sound source. The determination of the emitted sound power with ($P_{with}$) and without absorbers ($P_{without}$) was hereby carried out by measuring sound intensity on a certain sector of the air intake area.

$$D_{IL} = 10\log\left(\frac{P_{with}}{P_{without}}\right) \qquad [\mathrm{dB}] \qquad (2)$$

As illustrated in Figure 5 the general noise attenuation behavior and achievable insertion loss is captured very well.

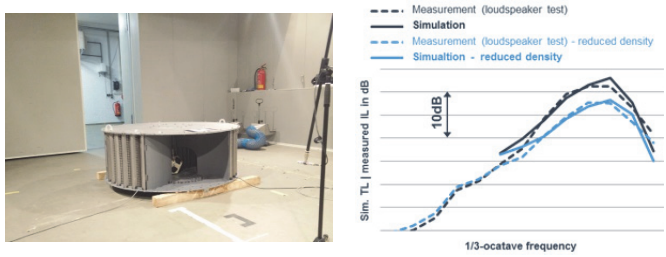


**Figure 5:** Left: measurement setup for determination of IL; right: Comparison of simulated TL and measured IL

## Benefit of the new developed discharge silencer

As already described in the introduction TCs are a dominant noise source especially on large two-stroke engines, which can have a height of up to 15 meters.

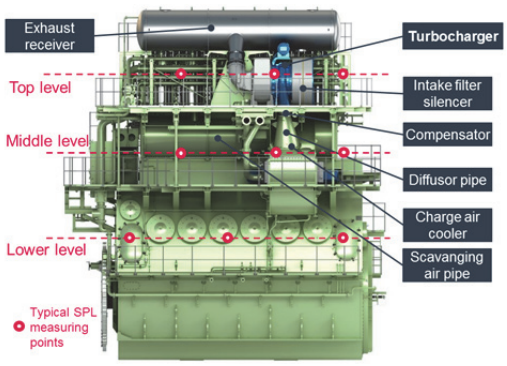


**Figure 6:** Side view and arrangement of a TC on a low-speed two-stroke engine and typical SPL measuring positions at different height levels on the engine

A typical engine with possible, IMO-relevant noise measurement points is illustrated in Figure 6.

In this case high noise levels appear also in the area of the charge air cooler and below. Currently engine builders need to apply additional insulation measures to reduce emission through those parts.

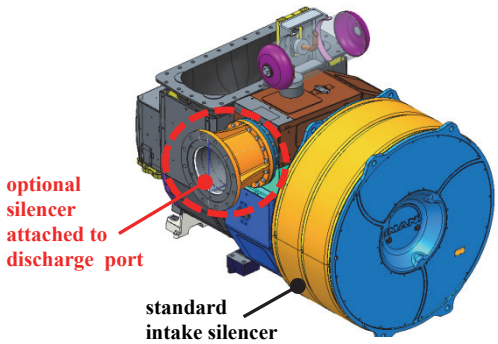


**Figure 7:** New TC-generation with optional silencer applicable at the compressor discharge port

Lowering of compressor noise directly at the discharge port by applying a silencer at the compressor outlet (like illustrated in Figure 7), would lead to the situation, that either such excessive insulation measures become obsolete or the overall noise levels can be further reduced to prevent an exceedance of noise limits in critical applications.

## Development & layout of the discharge silencer

As mentioned before, within the relevant frequency range and dimensions, the propagating sound field at the compressor outlet is characterized by superimposed modes of higher order. Results of numerical and experimental investigations on the modal field at the outlet of a TC radial compressor with vaned diffusor and volute casing showed, that, for the researched operating point, the highest amount of acoustic energy at the 1st BPF is contained in the (-1,0) mode (azimuthal mode order 1 - counterrotating). [2].

For that reason the influence of different acoustical excitation scenarios were investigated for several variants with the help of numerical FE-simulations. A schematic simulation setup can be seen in Figure 8.

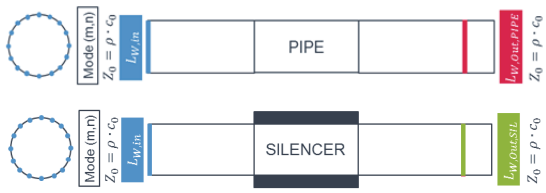


**Figure 8:** Schematic setup of the simulation model

The simulation was carried out for a straight pipe and for each silencer variant with constant excitation at the inlet to predict the noise insertion loss according to formula (2) ($P_{with}$ – with silencer installed / $P_{without}$ – with pipe installed).

In order to excite a prescribed azimuthal mode $m$, a point array at the outer diameter of the pipe was used. An acoustic pressure $p$ was defined depending on the angle around the pipe axis $\phi$:

$$Re\{p\} = \cos(m \cdot \phi); \quad Im\{p\} = sin(m \cdot \phi) \quad [\text{Pa}] \qquad (3)$$

Perforations have been modeled using analytical formulations to predict sheet impedance, like already described in chapter "Intake Silencer". Because Mechels' formulas [7] do not consider the influence of a through-flow, a newer approach was used, which already had been verified by Allam [8] in connection with the development of high frequency resonators for automotive turbochargers.

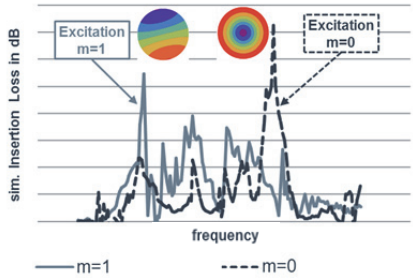


**Figure 9:** Simulation result for two different excitation modes for an identical geometry

An exemplary result of one sample geometry is illustrated in Figure 9. One can clearly see that the achievable noise reduction largely depends on the exciting modal field, which implements that experimental verification measurements are essential. Therefore different general concepts including sub-variants were developed and manufactured for testing.

## Verification of discharge silencer

Three different main concepts had been investigated during a preselection test. Noise IL was measured in a flow rig with the use of an attached loudspeaker and under the presence of flow.

As an exemplary result in Figure 11 the measured IL (using wall-mounted transducers in the pipe) is shown for three sub variants of one resonator concept (perforation geometry was varied for a constant degree of perforation).

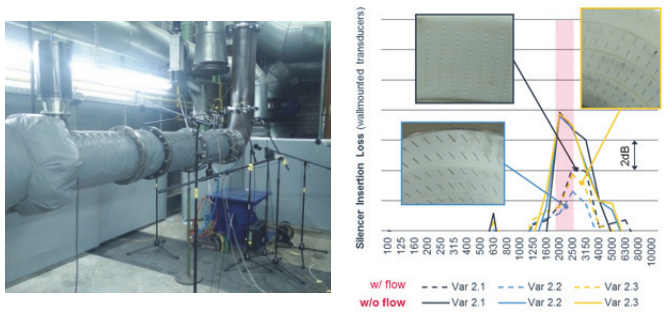


**Figure 11:** Measurement of IL at flow rig – left: test setup; right: exemplary result showing flow influence with different perforation geometries

As might be expected, noise reduction is lowered by the flow. The occurring frequency shift is additionally caused by an increased air temperature due to the screw compressor flow source. The lowest flow influence is achieved by using a simple hole geometry.

Afterwards preselected variants were tested at a fullscale TC on the burner rig test bed. Figure 12 gives an impression of the test setup.

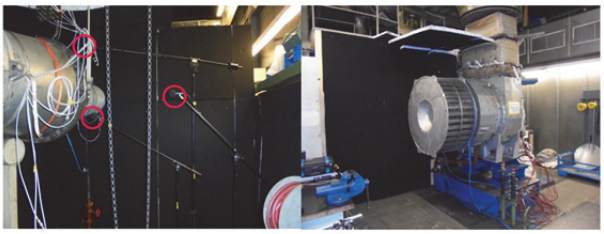

**Figure 12:** Photographs of measurement campaign on TC

With the intention of separating noise emission from the intake side, the test cell was equipped with acoustic barriers and the noise emitted by the connected pipework as well as dynamic pressure fluctuations in the pipe have been measured. The resulting IL (temperature corrected) for measured operating points (OP) on a possible engine operating line and the reduction of first BPF for a high load OP are illustrated in Figure 13.

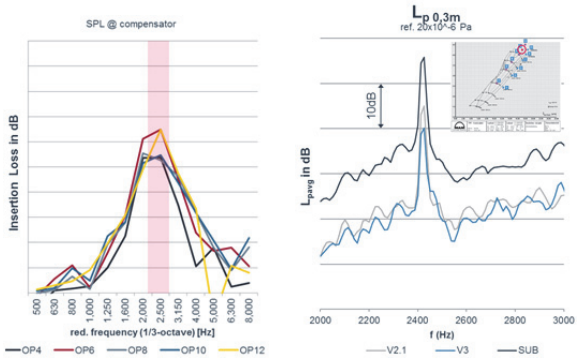


**Figure 13:** Results of burner rig test – left: measured Insertion loss for one variant (temp. corrected) and different operating points; right: reduction of BPF with two variants

This confirms that the tested silencers are working in the target frequency and over the relevant TC operating range. First BPF is lowered at least 10dB at high load.

Finally the effectiveness was successfully proven with a test on a 2-stroke 6S50ME engine, which was carried out in cooperation with MDT licensee Mitsui Engineering & Shipbuilding Co., Ltd. After installing the silencers between the compressor volute outlet and a shortened diffusor pipe, sound intensity measurements at the relevant engine components show a clear decrease in tonal compressor noise emission. A comparison of intensity levels (2.5 kHz, measured with scanning method) is shown in Figure 14.

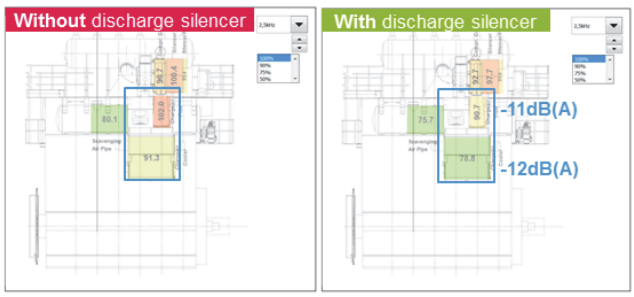


**Figure 14:** Verification on Engine – Surface sound intensity level (2.5 kHz) of noise emitting components (left without und right with mounted discharge silencer)

Looking at the sound pressure levels at typical measuring positions in one meter distance to the engine, a decrease of overall A-rated SPL between 2 and 7 dB(A) has been achieved. The result is shown in Figure 15.

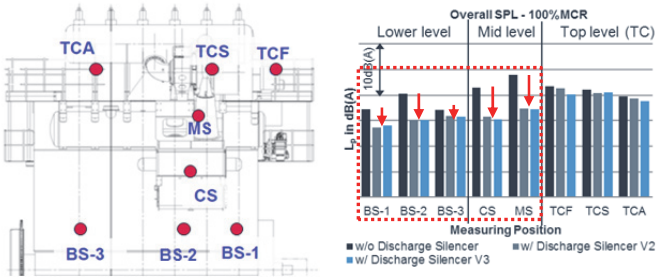


**Figure 15:** Verification on Engine – left: SPL measuring positions; right: overall A-rated SPL without and with two variants of discharge silencers

## Conclusion

With the aim of satisfying stricter noise limits and customer requirements a turbocharger silencer for attachment to the compressor outlet was developed by applying numerical simulations and intensive experimental verification. Measurements conducted at a flow rig, TC burner rig and finally the engine show, that dominating compressor noise at the first blade passing frequency is effectively damped and therefore overall noise levels at the engine can be lowered significantly. Additionally silencers at the TC air intake have been further optimized by adjusting their geometry and choosing suitable absorbing materials, which mainly contributes to lower noise emission of the new MAN Diesel & Turbo turbocharger generations.

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