

Sound Radiation of a Gear Box Using Acoustic Transfer Functions

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1. Introduction

The development of efficient numerical methods in acoustics is still a rather challenging task. In particular, if technical systems need to be optimized with respect to their acoustical behavior long before a prototype exists, computer simulations prove to be extremely helpful.

In the present paper, a procedure is presented in which the usage of so-called acoustic transfer functions (ATF) is discussed with respect to their applicability and accuracy. In particular, the objective is to improve the acoustical behavior of a rather complex practical system, namely a gear box. For this purpose, the results of acoustic measurements of the original gear box have been used as a starting point for the optimization using ATFs. The calculations of the ATFs as well as of the final acoustic response have been performed using the acoustic software LMS/SYSNOISE [4].

2. Measurement and evaluation of the sound radiated from the gear box

The measurement of the sound radiation was conducted in the R&D laboratory of Volkswagen. The sound pressure was analyzed using eight microphones, which were placed in a vertical plane perpendicular to the center axis of the gear box. The radial distance between the axis and each microphone was kept constant. Two run-ups up to 3000 rpm and 5000 rpm were recorded. The measured results have been displayed as waterfall diagrams (Figure 1) and also as polar diagrams.

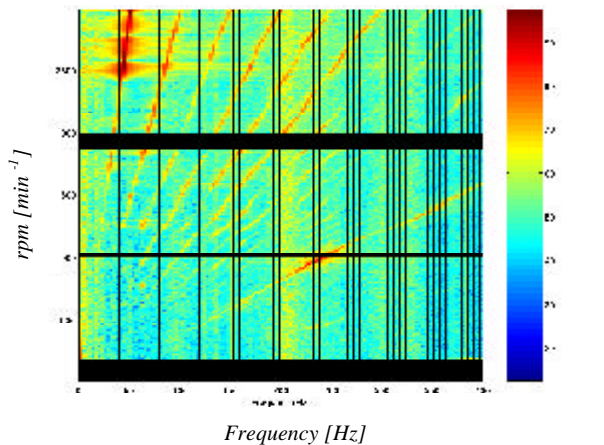


Fig. 1: Waterfall diagram of the mean sound pressure level in the lower frequency range of a run-up.

By means of these diagrams, nine critical operating points – defined as a combination of a frequency and a field point – have been selected, for which afterwards the acoustic transfer functions need to be computed.

3. Acoustic Transfer Functions

The acoustic transfer functions constitute a very efficient tool in numerical acoustics [2-5]. They are frequency-dependent transfer functions that define a relation between the vertical surface velocity of a sound-radiating structure and the sound pressure at a given field point (observation point, microphone).

The ATFs depend on the geometry and the properties of the acoustic medium in which the sound waves are propagating.

The ATFs change when acoustic boundary conditions, e.g., sur-

location of the considered field point. However, the ATFs are independent of the excitation $\{v_n(f)\}$, which makes them so attractive when one needs to consider several load cases. In fact, for each new velocity distribution (new load case), instead of solving the complete BE system of equations [1], only the multiplication

$$\{p_{fp}(f)\} = [ATV(f)] \{v_n(f)\} \quad (1)$$

has to be performed. Here, all ATFs are assembled into the matrix $[ATV(f)]$, and the vector $\{p_{fp}(f)\}$ yields the sound pressure at each field point under consideration.

4. Preparation of the Boundary Element Model

For the investigation of all acoustic models, the indirect boundary element method (IBEM) is used [1, 2]. In Figure 3, the structural system of the gear box as well as its BE discretization are displayed. The preparation of the boundary-element surface mesh turned out to be too time consuming to be done manually. Therefore, the meshing tools implemented in the software Virtual.Lab [5] had to be used.

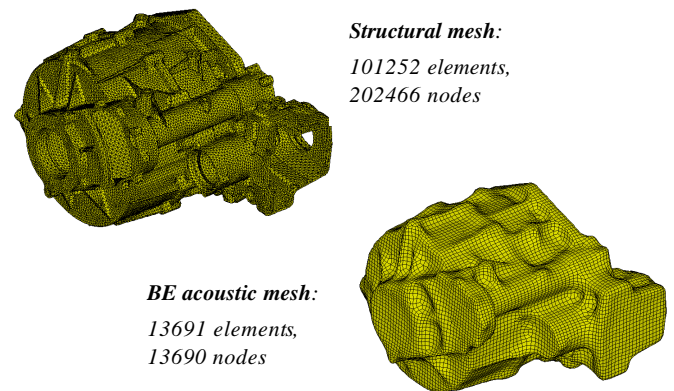


Fig. 2: Structural and BE acoustic mesh of the gear box.

5. Conventional versus ATF-Based Calculation of the Sound Radiation

The objective of the present investigation is the reduction of the radiated sound for nine critical operation points. This shall be reached by a targeted reduction of the surface velocity of the gear box. The problem to be solved, however, is how to find those regions of the vibrating surface that have the strongest influence on the sound radiation at the considered operation points. These regions are found directly, i.e. without a time consuming trial and error procedure.

Basically, two approaches can be used in order to reach the defined goal:

In the **first approach**, only the vibrations of the gear box are considered. In this case, only the normal surface velocities are analyzed and it is assumed implicitly that those regions of the surface where the vibration maximum occurs also are responsible for the high sound-pressure values in the environment of the gear box.

The **second approach** takes into consideration both the surface vibrations and the acoustical behavior of the sound-radiating system. This can be achieved by looking at the acoustical transfer

and boundary conditions. It is assumed that the maximum of the surface velocities as well as the maximum of the ATF have a direct influence on the overall acoustic performance. This influence is most significant if the two maxima are at the same location. One way to investigate the combined influences is to multiply the surface velocities with the corresponding transfer functions and to analyze the surface distribution of this product.

A typical example of this procedure is given in **Figure 4**. For the majority of the nine chosen operating points, the maxima of the product $v \cdot ATF$ occur in locations different from those of the maxima of the ATF and the maxima of the surface velocities alone. An acceptable agreement can be found only for three of the nine operating points which are in the mid frequency range.

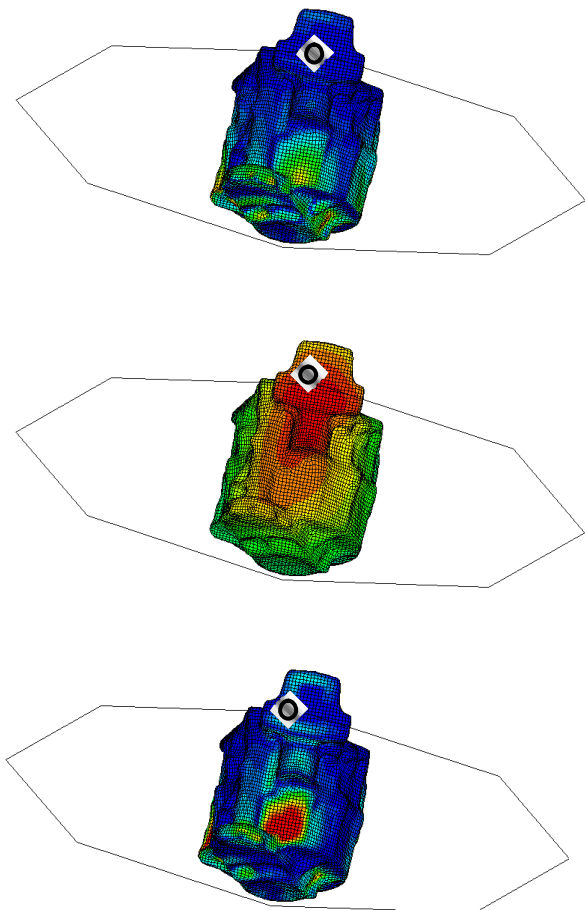


Fig. 3: Distribution of the surface velocities v (top), of the ATF (middle) and $v \cdot ATF$ (bottom) at 157 Hz with respect to the field point \odot .

In order to assess the possible sound reduction at the field point due to reduction of the velocity amplitudes in certain regions of the gear box housing, the following steps are taken. First, on the selected regions, the amplitudes of the surface velocities are reduced artificially up to 75 and 50 percent of their original values. Then the total radiated sound power of the gear box is computed for the original velocity distribution as well as for the two partly reduced variants.

The criteria for the selection of the critical surface regions are defined by the maximum of the surface velocities only (first approach), and by the maximum of the $v \cdot ATF$ distribution (second approach).

For the comparison, the radiated power L defined by

$$L = 10 \lg \left(\frac{1}{N} \sum_{i=1}^N 10^{\frac{L_i}{10}} \right) \quad (2)$$

has been chosen, where L_i is the sound pressure level at the i -th of the $N=8$ measurement points.

To show the obtained result in an easy to read manner, the differences between the sound power L obtained for the original configuration and the models with partly reduced intensity of surface velocity are given in **Figure 4**.

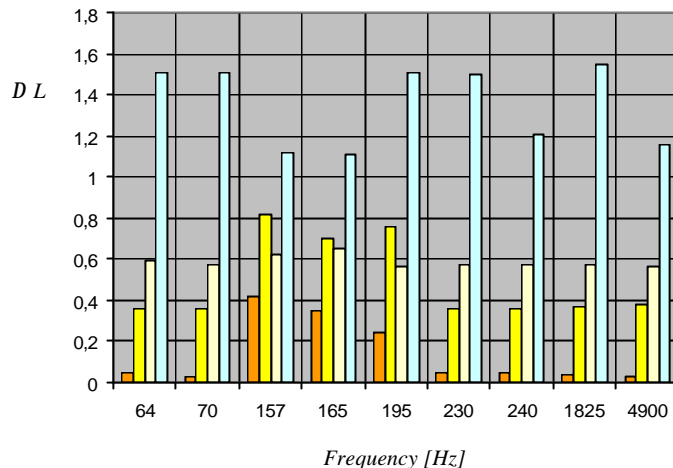


Fig. 4: Reduction of the sound power L after reducing the surface velocities v in regions found via the v -Maxima (to \blacksquare 75%, \blacksquare 50%) criterion and in regions found via the $v \cdot ATF$ -Maxima (to \square 75%, \square 50% of the original v).

It is apparent that choosing the critical surface regions by means of the product $v \cdot ATF$ leads to a more efficient reduction of the radiated sound power than in the case of simply reducing velocity amplitudes on those regions where their maxima occur. The obtained improvement differs by a factor of two to four. As expected, the difference is smaller in the mid frequency range, i.e. between 157 – 195 Hz, as here the distributions of v and $v \cdot ATF$ are very similar. For the other (six) frequencies, a significant improvement of the efficiency can be obtained by using the $v \cdot ATF$ criterion.

6. Conclusions

A new concept has been discussed that makes it possible to determine those parts of a vibrating technical system that contribute in the most significant way to the overall sound radiation. The concept is based on the combined analysis of the surface distributions of the acoustic transfer functions and the normal velocities. The procedure has been applied to a practical problem of sound radiation from a gear box, where significant improvements could be achieved as compared to a standard approach.

References

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