Vibroacoustic Testing of Technical Plastics Components

Andreas Franck, Jürgen Graf

Institute of Technical Acoustics, RWTH Aachen University, 52056 Aachen, Germany, Email: afr@akustik.rwth-aachen.de

Introduction

Due to their often small stiffness and their associated affinity to vibration, plastics components contribute much to sound radiation in many technical systems. An acoustics-aware design of this components is therefore of special importance.

This paper discusses a measurement approach to assess the accuracy and viability of a full-scale FEM/BEM simulation of sound radiation of plastics components.

Measurement Setup

For the verification of the simulation approach, a series of special test objects (flanges) with different geometry and materials have been manufactured at the Institute of Plastics Processing, RWTH Aachen University. One example of a mounted test object is shown in Figure 1.

The test object is fixed to a custom-made mounting plate and connected to a vibration exciter (shaker) through a push rod made of aluminium.

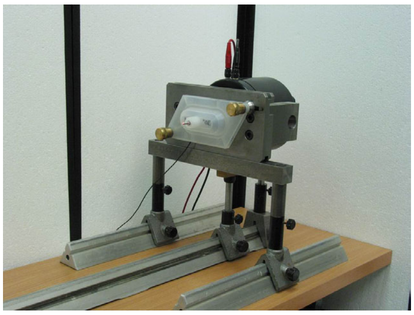

Figure 1: Measurement setup with mounted plastics flange, laser vibrometer not shown

The surface velocity of the test object is measured by a Polytec Laser-Scanning-Vibrometer [1]. Two servo-controlled mirrors allow to set the direction of the laser beam within a range of ± 20° vertically and horizontally. The necessary set of coordinates for scanning a complete surface is obtained from FE meshes.

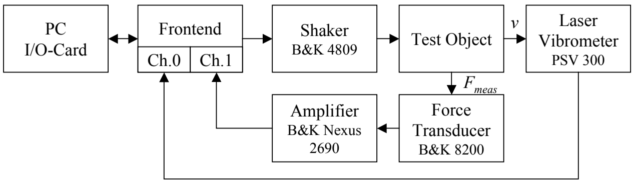

Figure 2: Block diagram of measurement equipment

The force applied to the test object is measured by a force transducer in combination with a charge converter and a conditioning amplifier. Due to the shape of the plastics component, the force transducer could not be connected directly. The push rod was split up and the force transducer was mounted in between. Therefore, the measured force is not exactly the same as the force actually applied to the test object. Detailed research showed its compliance and natural oscillations to be negligible in the considered frequency range. But still the inertial mass of the push rod causes systematic measurement errors that have to be compensated.

Figure 3 shows the equivalent circuit diagram of force transducer, push rod and test object. Force is represented by current, velocity turns into voltage. Masses are replaced by capacities, springs by inductances.

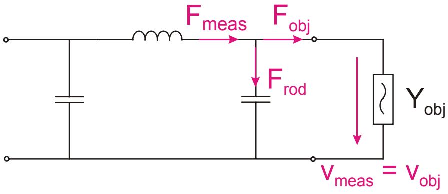

Figure 3: Equivalent electromechanical circuit diagram

To compare the results of measurements and FE-calculations, the measured velocities have to be normalized to a driving force of 1 N at the centre of the test object. Obviously not the whole amount of force measured by the force transducer “reaches” the test object, as the inertial mass of the push rod has to be moved. This contribution has to be subtracted.

The equivalent circuit diagram shows that the force needed to accelerate the mass of the push rod equals

$$F_{rod,load} = \frac{v_{rod,load}}{Y_{rod}} \tag{Eq. 1}$$

where the index “load” means, that a test object is mounted. The admittance of the push rod does not change with load. Thus it can easily be determined under no load condition.

The force F_{obj} can be determined once before scanning the whole surface. Then all velocity data are related to this feeding force. This leads to the so-called “transfer admittance” Y_x at any node position x:

$$Y_x = \frac{v_x}{F_{obj}} = \frac{v_x}{F_{meas} - \dfrac{v_{rod,load}}{Y_{rod,no\ load}}} \tag{Eq. 2}$$

The frequency response of the admittance of the push rod without load represents almost perfectly a concentrated mass. This proves the simple equivalent circuit to be sufficient. Furthermore, measurements with different push rods (material, geometry) lead to the same results after appropriate compensation.

Result Examples

The laser vibrometry measurements were used to obtain the spatial distribution of the normal surface velocity on the test object. The scanning quality was satisfactory for a wide frequency range; two examples (at 269 Hz and 2660 Hz) have been included below in Figure 4 for reference. Blue colour represents the lowest level of velocity, red the highest. At the two mounting holes in the lower left and upper right corners, the model was clamped.

The previously explained measurements have been used to verify the FEM simulations of the object.

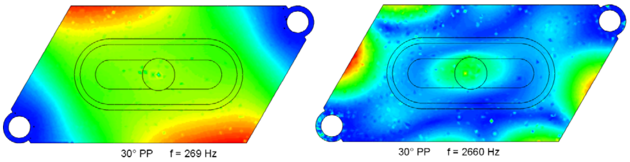

Figure 4: Two examples of laser vibrometry results

The FEM simulations were carried out at the Institute of Plastics Processing, RWTH Aachen University, using specialized custom material models [2] and HKS ABAQUS for the simulation. The results, shown below in Figure 5, are in quite good agreement, with the exception of some frequency shift especially for higher frequencies, due to differences in estimated and real material parameters [3].

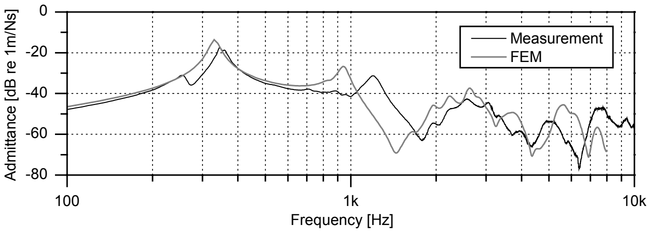

Figure 5: Comparison of FEM-calculated and measured surface velocity at one point of the surface

BEM Simulations

The second major step towards a full-scale simulation of the acoustic behaviour of plastics components is the simulation of their sound radiation. The Boundary Element Method (BEM) is the method of choice for this kind of task.

For this work, LMS Sysnoise [4] was used to calculate the sound radiation from both laser vibrometry measurement results and FEM calculations. To reduce the dimension of the calculated problem and to be able to conduct better comparisons with measurements, the flange model was embedded into a rigid half-space baffle.

The BEM simulations were carried out using LMS Sysnoise 5.6 on a HP C3600 workstation. The simulation time, including data import, was about 27 h for 1000 frequencies between 100 Hz and 8 kHz. After the calculation, the sound pressure was obtained by post-processing on 27 (three planes of 3x3) field points distributed in the space in front of the baffled component. The distribution scheme of one of the field point planes is shown in Figure 6 along with a photograph of the verification measurement setup.

Sound Pressure Measurements

To verify the results of the BEM calculations, some microphone measurements were carried out. Using MF, the transfer functions between the force at the flange input and the sound pressure in a set of field points have been measured. For an assumed harmonic force of 1 N over the whole frequency range, this enables a direct comparison between BEM simulations and measurements. To reduce the sound emitted by the shaker and the back side of the flange, a box housing has been constructed around the setup. For not too low frequencies, this housing is an acceptable approximation of the infinite baffle used in the BEM simulation.

The measurement results have been compared with the two BEM simulation cases (BEM with vibrometry measurement data input and BEM with FEM result data input) in a set of field points. Figure 7 shows this for one of this field points.

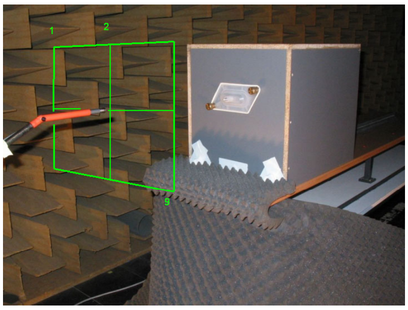

Figure 6: Verification measurement setup with microphone and scheme of a field point mesh plane

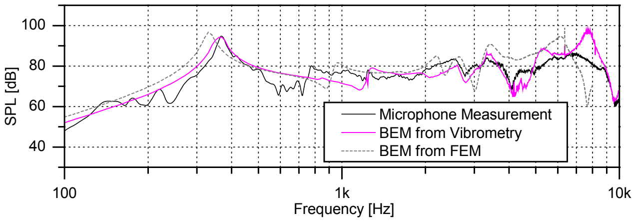

Figure 7: Comparison of sound pressure at a field point, measurements vs. BEM simulations

The agreement between sound pressure measurement and BEM result is acceptable – for low frequencies (below 300 Hz) some differences can be accounted to the measurement setup. The quite large disagreement between 500 Hz and 800 Hz could be explained by internal resonances in the housing.

Conclusions

It was shown that it is possible to model the sound radiation from vibrating plastics components by a full simulation approach. This approach has been validated through comparison with measurements on two stages: the surface velocity and the sound pressure in the field. Possible sources of errors have been identified. Still the simulation quality is good enough over a broad frequency range to even enable some auralization of the component behaviour.

Acknowledgements

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