Surface radiated noise of exhaust systems – Structural Transmission Loss Test rig, Part 1

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Introduction

The radiation of exhaust system can be divided into three categories: Tailpipe noise, transmitted noise through the hanger and surface radiated noise. The radiation from the tailpipe influences the pass-by-test as well as other sources like engine and wheels. The surface radiated noise and the noise transmitted through the hanger regularly do not have so much influence but they are a subjective noise problem.

In these days two rooms are needed to run measurements of surface radiated noise:

- a room to isolate the engine from the exhaust system,
- an anechoic chamber for the noise measurements.

The radiated sound power obtained from these measurements depends on the noise source as well as on the geometry of the surroundings.

This paper deals with a new procedure to measure the radiated sound power irrespective of the surrounding geometry and the source. These measurements still require an anechoic chamber but no extra facilities for the engine. A simple speaker is used as a noise source. Using this method enables to measure the transfer function of the structure, to do noise mapping and to calculate the modal damping. Furthermore, it is possible to separate air-borne resonances from structure-borne resonances. This test rig is not meant as a substitute for the acoustics engine test rig, but as an additional tool.

Test set up

In an early stage of development CAE tools are applied to predict the acoustic behaviour of an exhaust system. The surface radiated noise is modeled using FEM [2]. In a later development phase the structures are investigated using hardware. Since the final design of the engine often does not exist at this development stage, the experiments are made on comparable engines or on synthetic test rigs. One example for a synthetic test rig is described as follows.

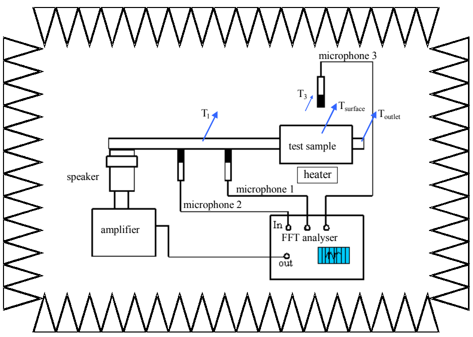

Figure 1: test set up

The measurements are performed in an anechoic chamber. A speaker generates a swept sine in the tube with a sound pressure level of up to 165 dB. The incident wave in the test sample is measured using two microphones which are placed in the inlet tube. The transmitted noise is measured along the structure using one or more microphones. The tailpipe of the test sample is an unflanged open end without any insulation. For measurements under realistic conditions it is possible to heat the test sample. To avoid direct sound transmission from the casing of the noise source to the microphones, the casing is insulated by sand bags.

Incident sound power

The noise attenuating impact of the exhaust system bases on the reflection and absorption of incident sound waves. That means in an exhaust system the waves are a combination of reflected and transmitted waves. Using the two-microphone method based on Chung & Blaser [3] these waves can be separated in a tube to get the incident sound power into a test sample. For that there are two microphones required flush with the tube wall. The distance between the microphones in axial direction is investigated in [1] and [4]. The method is only valid for plane waves where the exact limits of the tube diameter are given, e.g. in [6].

The build up test rig has an upper frequency limit of 3.5 kHz for a muffler system and 16 kHz for a manifold system.

Transmitted sound power

For measurements of the transmitted noise one or more microphones are placed over the structure and the radiated noise is calculated based on the German industrial standard DIN 45635.

Transmission Loss

The frequency-dependent damping function, the so-called Transmisson Loss, is defined as:

$$TL = 10log\left(\frac{incident \quad sound \quad power}{transmitted \quad sound \quad power}\right) \cdot dB \quad (1)$$

The structural Transmission Loss can be used to calculate:

- modal damping
- the separation of air-borne and structure-borne resonances
- noise mapping
- synthetic engine run up
- first statements concerning the ***inner*** acoustic behaviour when the microphone position over the structure is changed to the tailpipe

Modal damping

The modal damping values are required for some FEA calculations. They are obtained from the Transmission Loss based on the 3 dB criteria [7]. Other methods like the Rational Fraction Polynomial Method [5] can also be used, but are not implemented yet.

Separation of air–borne and structure-borne resonances

Resonances are temperature dependent. Air-borne resonances base on the geometry of the exhaust system internals. The frequency shift by temperature is given to:

$$f_{hot} = f_{cold} \frac{c_{hot}}{c_{cold}} \qquad (2)$$

with f: frequency and c: speed of sound.

The frequency shift of air-borne resonances is in dimensions of abundantly more than 100%.

The structure-borne resonance shift by temperature is given by the change of material elasticity:

$$f_{hot} = \frac{f_{cold}}{\sqrt{\frac{\rho_{hot} E_{cold}}{\rho_{cold} E_{hot}}}} \qquad (3)$$

with the density ρ and the Young's Modulus E.

Other effects like changes of structural stiffness caused by unequal surface and baffle expansions are not considered in this context. The experience shows that the shift of structure-borne eigenfrequencies for temperatures lower than 500°C does not exceed 10 %.

The kind of resonances can be investigated by using two methods:

1. heating the structure and investigating the shift of the resonances by equation (2) and (3).
2. adding mass onto the structure and changing in this way only the structure-borne resonances.

Noise mapping

Noise mapping can be done if more than one measurement point is used along the structure. The noise mapping results are useful for adjustment of calculated FEA data. Finding a solution of a structure-borne-noise problem is than consecutively a task of the FEA not of this kind of measurement.

Synthetic source

The transfer function Transmission Loss is independent of the sound source. For more practical investigations a synthetic sound source is added. This synthetic sound source takes into account that the dominating excitation result from gas pulsations from the periodic opening of the exhaust valve, the so-called harmonics and multiplies of it. Higher frequency noise between 1-2 kHz is regularly given by flow noise which increases with increasing mass flow. A typical excitation of an exhaust system is given to:

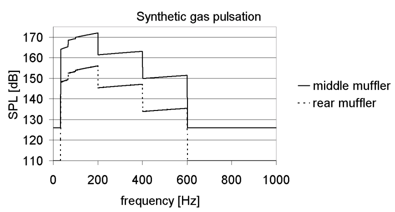

Figure 2: peak hold of a synthetic sound source for a 4 cylinder engine

These synthetic sources are generated based on more than 120 source power measurements on internal combustion engines. More detailed and reworked synthetic sources will be given in Part II.

Synthetic engine run up

The transfer function and the synthetic source can be added to provide the acoustical behaviour of the system under realistic conditions, here for a synthetic run up:

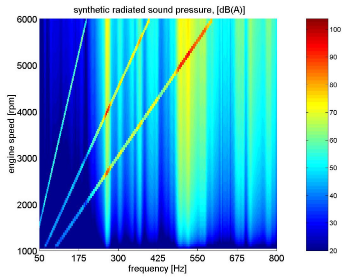

Figure 3: Campbell diagram of a synthetic run up

The example in Figure 3 shows a resonance below 300 Hz which is subjectively disturbing even inside the car.

Conclusions

At early stages of development the shown test rig provides important information on surface radiated noise. In addition to that general acoustical investigations can be performed. The detailed mathematical formulation will be probably presented in part II, at the ISMA 2004 in Leuven, Belgium.

References

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