FEM based Layout of a Piezo Foil Actuator for an ASAC System

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Abstract

Urban road traffic noise comes more in the focus of public discussion, as it is perceived as stressful for human beings. Besides to infrastructure related secondary noise reduction approaches, primary approaches focusing on the vehicle noise sources are most promising.

Under urban vehicle conditions, the powertrain represents one of the major vehicle exterior noise sources. Actual fuel saving strategies like down sizing, cylinder deactivating and high pressure supercharging even increases the powertrain noise. Conventionally acoustic measures are not longer sufficient to guarantee an acceptable vehicle NVH under light-weight aspects.

For this reason active measures become a key feature for future NVH concepts of vehicles. Active engine mounts and active cabin noise control systems are in series production. Current developments for powertrain noise reduction are active structural acoustic control (ASAC) systems applied e.g. to oil pans. A FEM based lay out method of such a system will be described with in this article.

1. Introduction

Noise is more and more in the focus of public discussion as it represents a major source of environmental pollution especially in the cities. In Europe more than 100 million people are affected by noise leading from reduction in cognitive performance up to severe deterioration of human health and an economic loss of more than 10 billion Euros per year is generated (11). Road traffic noise represents for human being the most affecting noise source as documented in Figure 1.
Fig. 1: Road traffic represents the most annoying noise source in environment /1/

Vehicle Exterior Noise Sources

The automotive vehicle incorporates a multitude of possible noise sources contributing to the vehicle overall exterior noise depending on vehicle operating condition, power train speed and load condition, respectively. In urban traffic the power train – as documented in Figure 2 – represents the pre-dominating vehicle noise source, i.e. in the low and mid vehicle speed range especially during vehicle acceleration.

Fig. 2: Urban traffic, separation of power train and tire/road induced vehicle exterior noise /2/

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The noise and vibration generated by the power train affects both, interior as well as exterior noise (see Figure 3). Its contribution to the vehicle exterior noise mainly results from directly emitted power train noise whereas the interior noise is due to both, power train noise transmitted via airborne transfer paths (e.g. firewall) into the cabin and vibration transferred via the power train mountings and radiated by structure-borne noise excited chassis components (e.g. dashboard). Additionally, the interior vibration comfort (e.g. seats, steering wheel, gear shift lever etc.) is influenced by power train vibration, too.

**Fig. 3: Power train noise and vibration transmission in vehicle**

Based upon hybrid simulation tools (combination of analytical and experimental methods) the contribution of the individual power train components and the relevance of noise and vibration transfer paths involved can be analysed for interior noise and vibration (/3/) as well as for vehicle exterior noise. In Figure 4 the simulation of a vehicle pass-by noise test is exemplarily shown. The power train represents the major noise source in mid and high frequency range but especially in low frequency range indicated by firing order related noise contribution.

**Fig. 4: Vehicle exterior noise simulation**
Power Train Noise Sources

The power train noise radiation typically is predominated by “thin wall structures” as e.g. oil pan, valve cover and timing drive cover. In Figure 5 the results of a near-field sound intensity measurement and the corresponding results of a combined Finite Element / Multi Body Simulation are documented for a passenger car power train as an example.

Due to its thin wall structure in combination with relatively high structure-borne excitation by engine block the oil pan exhibits the largest sound power share. The amount of oil pan noise contribution is depending on – among others – bottom end design (e.g. structural oil pan: bolted to engine block and gearbox for stiffening engine/gearbox system), oil pan design, material (cast iron, aluminium, plastic, sheet metal, SDS) and connection to engine (e.g. decoupled oil pan). Typically, most sensitive with respect to noise emission is a structural oil pan made of aluminium.

The valve cover might contribute significantly to the overall sound power of a power train, too, resulting from remarkable structure vibration excitation of the cylinder head induced by the valve train dynamics and injectors (especially for Diesel engines). The amount of valve cover noise emission depends on – besides cylinder head design – cover design, material and connection to the cylinder head (e.g. decoupling).

In /4/ the power train noise reduction required to achieve the future legislative vehicle exterior noise limits for the (modified) pass-by noise test is expected to amount more than 6 dB(A). Taking into account the NVH status of current NVH optimized power train design, this represents a challenging task for the power train NVH engineers.

Fig. 5: Power train component noise contribution; comparison of near-field sound intensity analysis and combined FEM/MBS simulation
Oil Pan NVH Optimisation

Passive (conventional) Optimisation Measures

Structural optimisation by means of Finite Element simulation is an essential task within the development of a new power train. Besides oil pan design (e.g. contouring) and material (e.g. SDS) especially the power train bottom end concept plays a major role for oil pan NVH optimisation. The bottom end concept defines global power train stiffness as well as oil pan layout (e.g. decoupled/not decoupled) and the vibration excitation by the engine block. Several concepts are known (Figure 6: selected concepts) and have to be proven with regard to noise radiation benefit for each individual application. The structural optimisation of the oil pan often is limited by e.g. production line and package constraints, costs as well as required support for global power train stiffness. Therefore, NVH engineers are looking for advanced technologies for oil pan noise emission reduction.

Fig. 6: Examples for power train bottom end concepts

Active Optimisation Measures

With regard to automotive application several active noise reduction (ANR) systems are under development (/5/, /6/, /7/) or even now in series production (e.g. active hydro mounts). Within this paper an ANR system for the active damping of structure-borne induced oil pan vibration will be described in detail. The principle approach consists in monitoring the structural response to the excitation by a sensor and the generation of an opposite-phase actuator displacement for active damping of oil pan vibration resulting in a reduction of structure-borne noise radiation (see Figure 7).

Fig. 7: Principle of oil pan active damping
For the active dampening of the oil pan vibrations, piezo foils are used as actuators. The principle layout of such a piezo module is shown in Figure 3. The modules utilize the so-called 31-effect which causes the modules to contract if voltage is applied. Since the oil temperature varies with speed and engine load, high-temperature piezo modules are used.

**Figure 8: Piezoceramic Foil Modules (/8/)**

**Positioning**

For the system layout, the positioning of the actuators is of decisive importance. The most conspicuous oil pan vibration modes are determined by combining finite element analysis (FEA) and multibody system simulation (MBS) under realistic engine excitation conditions. FEA is used to calculate the structure transfer behavior, while MBS is used to calculate the powertrain excitation. The method outlined here is used as a standard acoustics calculation procedure by FEV /9/. Figure 9 shows the vibration amplitudes of the acoustically relevant vibration modes. The piezos are positioned in such a manner that constitutes an optimal compromise between the number of piezo actuators and the covering of areas with high vibration amplitudes.

**Figure 9: Positioning of the Piezo Patches**

First order mode shapes: 300 Hz - 620 Hz

Second order mode shapes: 990 Hz - 1200 Hz

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Demonstrator

For the further development of the system, a demonstrator with reduced functionality is highly important. It helps to keep experimental and simulation efforts for the numerous necessary studies within reasonable limits. The reduction is divided into two steps: the determination of suitable excitation data and the reduction of the system. In the first step, a simulation model of the entire powertrain is created to determine the vibration signals which in turn are used later as excitation data.

In a second step, the system is reduced to a submodel for further experimental and analytical investigations. The submodel consists of an oil pan and a metal frame as a substitute for the engine block. The frame is designed so as to represent the stiffness of the engine block skirts as accurately as possible. This submodel combined with realistic excitations shows nearly the same structure behavior as the full model of the powertrain, i.e. the comparison of the vibration modes shows an adequate correspondence between the submodel and the full model of the powertrain (Figure 10).

Dimensioning

The dimensioning of the patches is performed by means of the simulation model depicted in Figure 11. Similar to the experimental boundary conditions of shaker investigations, the calculational investigations are carried out with one excitation at only one bolting point. The dimensioning is accomplished by means of a thermic analogy. For this purpose, the shell elements marked red are transformed into multilayer elements. One layer is given the mechanical material properties of the surrounding oil pan structure and one layer is assigned with the mechanothermic properties of the piezo.
The thermic expansion coefficient should be chosen so that the force free actuator expands with a temperature increase of 1°C corresponding to an applied voltage of 1 V. Typical piezo materials show an expansion coefficient of 0.02% at 3/1 operation and 1kV/mm.

![Simulation Model with Piezo Patches](image)

*Figure 11: Simulation Model with Piezo Patches*

This kind of modeling offers the possibility to change the geometry of the actuator rather quick. The thickness can be very easily changed through adjusting the “section-definitions”. The surface can also be modified through selecting or de-selecting elements. However, the magnitude of the electrical response cannot be evaluated with this method, and it does not allow investigations with variable currents and voltages over the frequency.

Due to its advantages regarding geometric variation, the thermic method is used for the rough design of the actuator size. The results are shown in *Figure 12*. It becomes obvious that up to a thickness of about 1 mm the achievable displacements increase linearly with thickness. From 1.5 mm on, the amplitudes decrease with a further increase of thickness. The dependency of the amplitude from the actuator surface shows an increase of 0.5 and is thus not as effective as an increase of thickness. However, actuator thickness is often strongly limited by the maximum permitted voltage, since the piezo electrical effect depends on the specific voltage V/m. Thus, with doubled wall thickness, voltage must also be doubled to achieve the same effect.

![Dimensioning of the Actuators](image)

*Figure 12: Dimensioning of the Actuators*
The actuators are chosen by comparing the amplitudes caused by the operation excitation with the amplitudes achieved by the actuators while considering a maximum voltage of less than 300 V.

**Transfer functions for the Controller Design**

To design the control of the ASAC system, transfer functions are necessary. These transfer functions can only be insufficiently determined by thermic analogies. Therefore, the model is modified to carry out a coupled-field calculation. For this purpose, the patches as well as the supporting oil pan structure are modeled in several layers of solid elements. In **Figure 13** the model and the result of a static simulation are depicted. The local deformation of the patches perpendicular to the surface can be clearly recognized.

**FE-Model of Oil pan demonstrator w/ piezo patches**

**Figure 13: Coupled Field Simulation Model**

**Figure 14: Transfer Functions**
Figure 14 shows some examples of the transfer function calculation. In the left diagram, the vibration levels at the sensor position are depicted for the excitation by itself (without anti-signal) and with the actuator active at the same time (with anti-signal). To this end, the actuator is charged with the quotient of the disturbance function and the actuator/sensor transfer function. It can be seen that the vibration levels are reduced to zero except for numeric noise. In the investigated case, the voltages and currents that occur in this calculation remain below the specifications which validates the adequate dimensioning of the actuator in advance. However, the draw aside of the structure outside the resonance frequencies poses a problem (Figure 14, right hand side). In the diagram on the right, the vibration velocities are depicted summed up over the entire oil pan surface. Aside from significant improvements in the 600 Hz range, an additional peak occurs at 450 Hz. Such peaks need be eliminated through adequate control.

Conclusion

The development and layout of an ASAC system with piezo foil actuators can be efficient supported by FEM simulations. For two different simulation approaches the advantages and disadvantages for different tasks of the system lay out could be shown.

The introduction of an ASAC system for automotive power train application in mass production – besides technical and costs challenges - finally depends on the achievable additional customer benefit in comparison to the conventional passive optimisation measures.

Table 1 reflects some aspects dealing with NVH benefit, cost and other features relevant for oil pan application. Of course, not all aspects are included in this rough overview.

<table>
<thead>
<tr>
<th>Measures</th>
<th>NVH benefit</th>
<th>Costs</th>
<th>Robustness</th>
<th>service</th>
<th>package</th>
<th>Fuel consump. (CO₂)</th>
<th>Oil Temp.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Structural Optimisation</strong></td>
<td>base</td>
<td>base</td>
<td>base</td>
<td>base</td>
<td>base</td>
<td>base</td>
<td>base</td>
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<tr>
<td>Material (e.g. magnesium)</td>
<td>+</td>
<td>-</td>
<td>+/-</td>
<td>+/-</td>
<td>+/-</td>
<td>+</td>
<td>+/-</td>
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<tr>
<td>decoupling</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>+/-</td>
<td>+</td>
</tr>
<tr>
<td>Encapsulation (@ P/T or vehicle)</td>
<td>++</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
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<td>-</td>
</tr>
<tr>
<td>ASAC System (actuator, sensor, control)</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>+/-</td>
<td>+/-</td>
<td>+/-</td>
</tr>
</tbody>
</table>

++ : much better     + : better     +/- : similar    - : worse    - - : much worse

Table 1: Comparison of passive and active oil pan noise reduction measures
It becomes clear that an ASAC system exhibits promising NVH benefit compared to the actual noise reduction measures even the costs of an ASAC system in mass production and its robustness in real world automotive application are not known today. In comparison to the encapsulation benefits are expected especially with regard to package, oil temperature and fuel consumption (due to lower weight). However, the final acceptance in automotive industry will strongly depend on the challenges with regard to costs and robustness.

ACKNOWLEDGMENTS
The present investigations on the active noise control of a passenger car’s oil pan was partly conducted within the EU-sponsored project InMAR (Intel-ligent Materials for Active Noise Reduction).

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