Canceling Oil Pan Active Powertrain Noise

ABSTRACT

The powertrain, under urban driving conditions, is one of the major sources for external vehicle noise. The engine's oil pan is a major contributor to the overall powertrain sound emission, particularly during idling and full load acceleration. The oil pan can also contribute considerably to the radiated sound levels of the powertrain. Thermal restrictions and weight reduction can limit the potential effectiveness of passive optimization measures, such as structural optimization and acoustic shielding. Consequently, the EU-sponsored project InMAR studied the possibility of utilizing the Active Structure Acoustic Control (ASAC) method to reduce noise. The method provided proof that vibration induced noise form the oil pan has the potential to be reduced significantly.

This passenger car study conducted within the InMAR project focuses on the installation of an ASAC system based on piezo-ceramic foil technology.

INTRODUCTION

Urban driving conditions have been receiving additional attention, because those conditions have been perceived as being stressful. In addition to the use of secondary noise reduction approaches, primary approaches that concentrate on crucial sources of vehicle noise have great potential.

Especially at idle and during full load acceleration, the oil pan contributes significantly Considerable contributions to the overall powertrain sound emission have been noted with the oil pan, particularly during idling and under full load acceleration. The engine oil pan can also add considerably to the powertrain's radiated exterior noise. Utilization of such passive optimization measures as structural optimization and acoustic shielding can be limited by light-weight designs, integration and thermal constraints. A solution was developed within the EU-sponsored InMAR project to apply the Active Structure Acoustic Control (ASAC) method to the oil pan design process. Fuller /1/ provided the initial research investigation in the principles of ASAC. This methodology has proven to provide considerable noise reduction potential for oil pan vibration induced noise.

POWERTRAIN NOISE SOURCES

The powertrain noise radiation typically is dominated by "thin wall structures" as e.g. oil pan, valve cover and timing drive cover. In Figure 1 the sound power share for a passenger car powertrain is documented as an example.

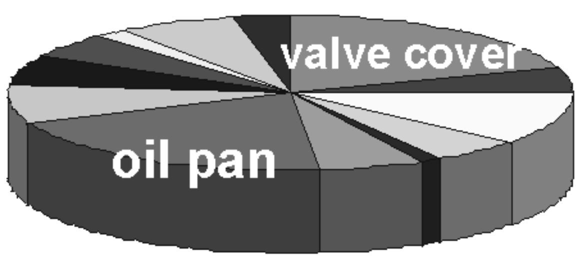

Figure 1: Sound Power Share of a Passenger Car Powertrain

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. Among other things, the amount of oil pan noise contribution depends on the 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, aluminum, plastic, sheet metal, MPM) and connection to the engine (e.g. decoupled oil pan). Typically, most sensitive with respect to noise emission is a structural oil pan made of aluminum.

OIL PAN DEMONSTRATOR

The identification of the most prominent oil pan natural vibration modes under real world engine excitation as well as the definition of sensor and actuator position are performed by means of combined Finite Element (FEM: structural behavior) and Multi Body System (MBS: power train excitation) simulation. After establishing the excitation data at the oil pan flange, an oil pan / steel frame sub-model (steel frame simulates the bottom end block stiffness) was derived from full powertrain FEM model to enable short simulation timings for further ASAC (active structural acoustic control) system development. The simplified model in combination with the real world excitation (obtained by FEM/MBS simulation of full powertrain model) exhibits identical structural behavior as the full powertrain model: The comparison of vibration modes reveals a good correlation between sub-model and full powertrain FE model (Figure 2).

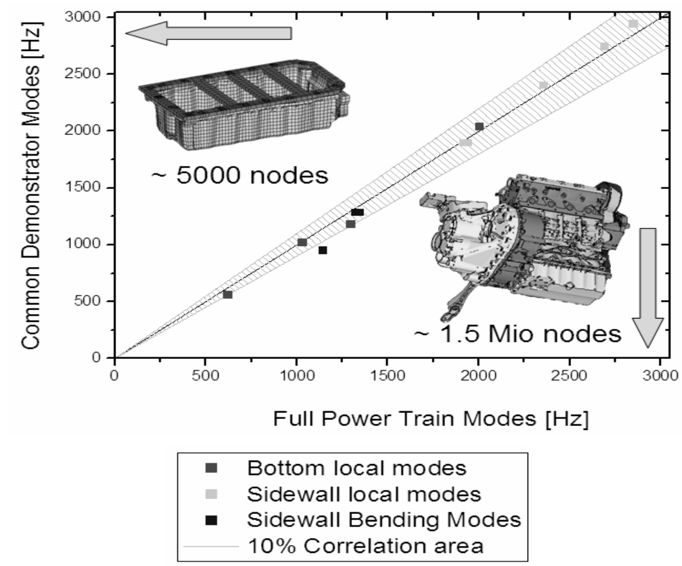

Figure 2: Common Oil Pan Demonstrator Derived from Full Powertrain

IDENTIFICATION OF ACTUATOR POSITIONING

The positions of sensor(s) and actuator(s) as well as piezo patch and specifications were defined based upon FEM/MBS simulation in order to achieve best ASAC system performance. Therefore a speed sweep with the sub-model Oil Pan Demonstrator was performed and the surface velocity levels evaluated. Campbell diagrams, characteristic narrow band spectra and surface velocity distribution were evaluated with the target to identify the most significant modes. The results of this analysis are depicted in Figure 3. The identified natural frequencies correlated well with the corresponding peaks in the measurement.

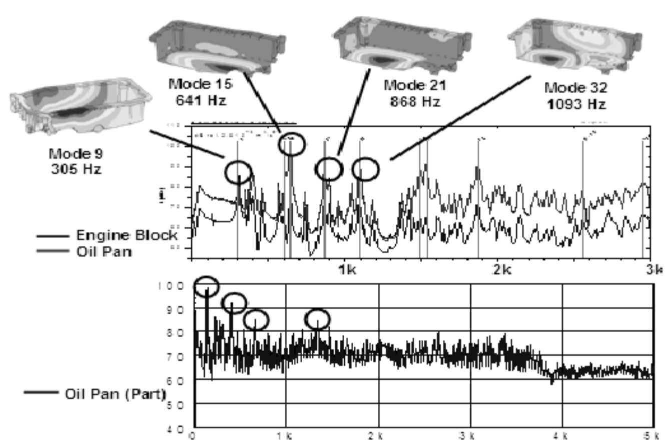

Figure 3: Critical Mode Shapes

SIMULATION OF PIEZO-ELECTRIC ACTORS

Two approaches of simulating the piezoelectric actuators were used for the investigations of the oil pan ASAC system. In a first step the piezo-electric effect was approximated by a thermal elongation of the piezo ceramic. This approach stands out for an easy way of implementation and provides a first impression of the dynamic behavior of the piezo patches. For the dimensioning of the piezo patches this is a sufficient method, but the electrical coupling, i.e. current and tension, can not be simulated by this approach.

The second approach is a coupled field analysis of the piezo elements. This method provides the opportunity to simulate the electrical dimensions which are important for the layout of the control system. On the other hand the modeling effort for a coupled field analysis is much higher than for the thermal approach, which means this method is unsuitable for concept investigations.

POTENTIAL ESTIMATION OF ASAC SYSTEM

The possible acoustic benefit of the ASAC system has been evaluated by calculating the integral surface velocity level over the bottom side of the oil pan. The result of the surface velocity is depicted in Figure 4. In two frequency ranges - around 600 Hz and around 1 kHz - a reduction potential of about 40 dB is visible. In other frequency ranges there is no benefit possible in terms of integrated surface velocity. At 480 Hz for example even a worsening of 20 dB occurs.

The simulation results show a huge potential for noise reduction with an ASAC system on the oil pan. A powerful control has to be set up to avoid the disadvantages at 480 Hz.

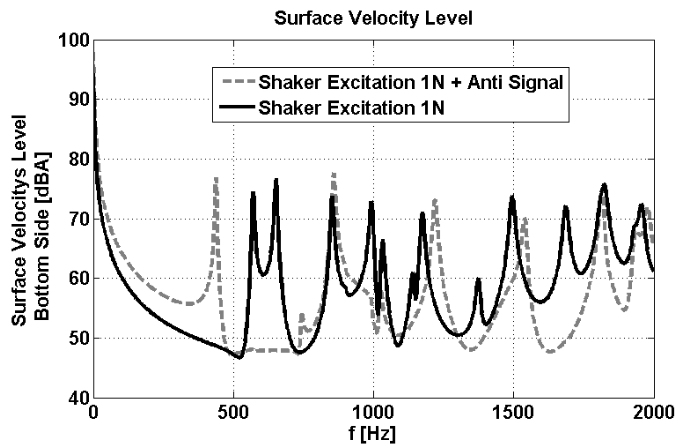

Figure 4: Surface Velocity with and without Anti-signal

IDENTIFCATION OF CONTROL STRATEGIES

The task of the vibration damping system is to reduce resonant response in modes of interest.

Two possibilities of vibration damping were investigated:

1. Semi-passive damping system consisting of passive resonant circuit with a electromechanical transducer-element (C), a coil (L) and a resistance (R) to damp the vibrations in the resonance frequency

2. Active damping system consisting of an actuator, a sensor and a controller.

The objective of the (active) vibration control is to reduce the vibration of the structure by (automatic) modification of the system's structural response [1].

Different control strategies were proved in regard to the present application. At least four different strategies were chosen and investigated, one for the passive and three for the active damping system:

- Semi passive damping,
- Collocated control,
- Adaptive feed-back control and
- Adaptive feed-forward control.

The advantage of the passive resonant circuit in general is that no electrical power input is needed. A disadvantage is the fact that the system only works in the range of one resonance frequency.

Compared to the passive resonant circuit, the active system needs electrical power input. The advantage of the active system is the possibility to reduce the vibrations very efficiently in more than one mode at the same time. For the controller, different concepts depending on the requirements and the condition parameters of the system are possible.

In the following the control strategies are described in detail.

SEMI-PASSIVE DAMPING

The basic idea of semi-passive vibration damping is to transform mechanical energy into electrical energy and its dissipation afterwards. A detailed description and a mathematical formulation of this phenomenon are given in [2]. First of all, the strain has to be transformed from the vibrating structure to the transducer material. The strain energy then has to be transformed into electrical energy inside the active material. The efficiency of damping mainly depends on the effectiveness of this transformation, described by the electromechanical coupling factor k. Piezoelectric materials show a very high coupling factor, therefore they are a very attractive option for damping structures.

Inductive shunting was first proposed in [3]. If the piezoelectric transducer is shunted on a RL circuit so that the natural frequency of the electrical circuit is tuned on the natural frequency of one mode, the system behaves like a tuned mass damper [4]. A serial arrangement and a parallel arrangement of the resistance and the inductivity are possible in principle and show the concept and the block diagram of the passive resonant circuit with resistance and inductance in serial arrangement.

COLLOCATED CONTROL

The collocated control is characterized by a collocated arrangement of actuator and sensor and symptomatic phase characteristics (phase shift of 90° in each eigenfrequency) of the frequency transfer function between actuator and sensor [5]). Different concepts are the velocity feedback or the integral feedback controller.

ADAPTIVE FEEDBACK CONTROL

Adaptive Feedback control is used in cases where the disturbance of the structure cannot be directly observed.

ADAPTIVE FEEDFORWARD CONTROL

Adaptive Feedforward control is used if it is possible to measure the structure excitation and to use this signal as a feedforward signal. Another possibility is to get information about the disturbance source, for example the rotational speed of the engine, to generate synthetic feedforward signals.

SYSTEM IMPLEMENTATION

PIEZOELECTRIC PATCHES

The used piezo patches are high temperature piezoelectric actuator modules [6]. The piezoceramic wafers are provided with uniformly metalized surfaces to operate in the lateral d31-mode. The piezoceramic is embedded between thin layers of insulating material and layers with contacting structures. The contacting layer is made of a conductive flexible material (e.g. copper mesh with a wire diameter of 0.03mm) having the shape and size of the piezoceramic wafer. In case of a break the actuator will still work because the contacting material covers the whole electrode of the piezoceramic so that the broken pieces stay in the electrical field where they can be controlled. The flexibility of the contacting material guarantees a long lifetime and reliability also under dynamical loading.

Two different piezo patches were implemented on the bottom of the oil pan, as shown in Figure 5. The piezo patch 1 is a double layer patch with a capacity of 52,6 nF while patch 2 is a single layer patch with a capacity of 28,6 nF.

TEST SET UP

The oil pan with integrated piezo patches is shown in Figure 5. The oil pan was hung on four rubber wires in a test stand frame. The vibration excitation was provided by an electro-magnetic shaker, fixed to the upper face of the oil pan.

The vibrations of the oil pan were measured with two accelerometers positioned near to each piezo patch and one sensor at the side of the oil pan.

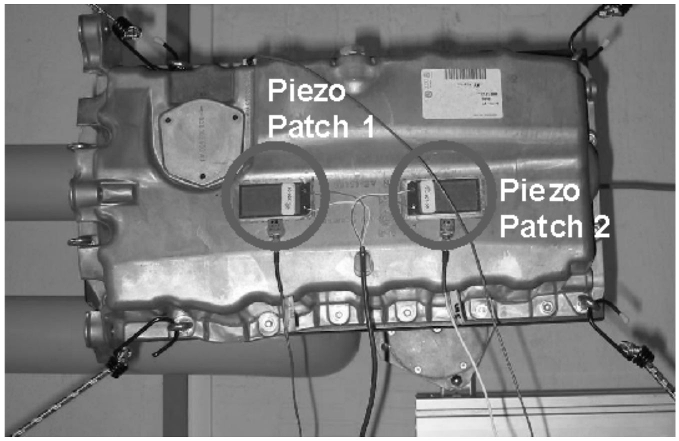

Figure 5: Oil Pan Test Setup with Two Piezo Patches

VIBRATION IDENTIFICATION OF OIL PAN

First of all the frequency response functions between the shaker excitation and the acceleration were measured, as shown in Figure 6. For further investigations especially the modes at 576.6, 655.0 and 902.4 Hz were taken into account.

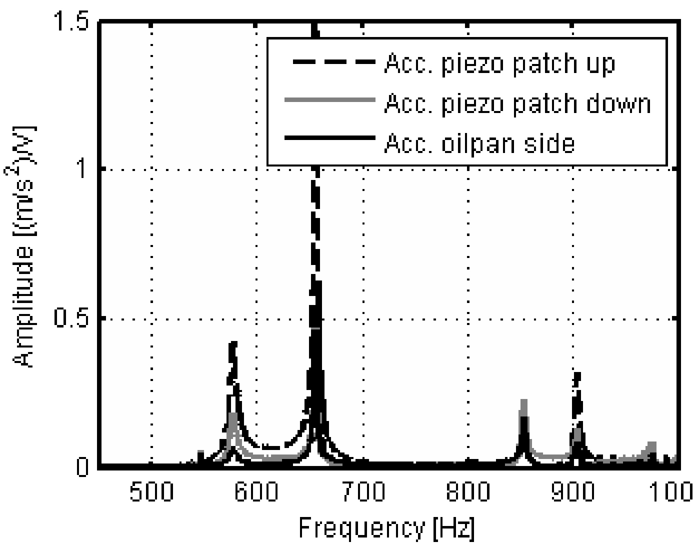

Figure 6: Frequency Response Function of Shaker Excitation to Acceleration Signal for Different Sensor Positions

SEMI-PASSIVE DAMPING

For the passive damping system a resonant circuit in serial arrangement of resistance, inductance and capacitance (piezoelectric actuator) was implemented.

The resonance frequency of the electric resonance circuit is given by $\omega_{D_i} = \sqrt{\frac{1}{L_i \cdot C_i}}$,

with inductance L_i and capacitance C_i.

For the present case the capacitance is given by the capacitance of the piezoelectric transducer with $C_1 = 28{,}6\,nF$ and $C_2 = 52{,}6\,nF$. Therefore, the values of the inductance of the different coils can be calculated based on the relationship to the damped eigenfrequencies. The calculated values are listed in Table 1.

Table 1: Calculated Values of the Inductance for the Different resonant frequencies and Piezo Patches.

	Resonant frequency [Hz]	Piezopatch capacity [nF]	Calculated Inductance value [mH]
Piezo Patch 1	576.60	52.60	1448.46
	655.00	52.60	1122.46
	902.40	52.60	591.37
Piezo Patch 2	576.60	28.60	2663.94
	655.00	28.60	2064.39
	902.40	28.60	1087.62

The optimal values of the resistance were investigated in further experimental tests. In Figure 7 the block diagram of the semi-passive system can be seen.

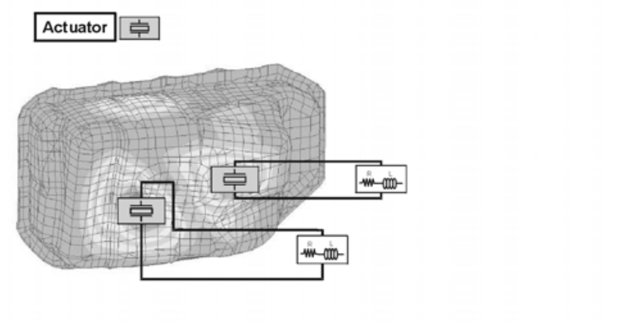

Figure 7: Block Diagram of the Semi-passive Damping System

COLLOCATED CONTROL

As mentioned before the collocated control is characterized by a collocated arrangement of actuator and sensor. In the present application the accelerometers positioned close to the actuator positions were used. The measured frequency transfer function between actuator and sensor is shown in Figure 8.

For the controller a velocity feedback strategy was chosen. The collocated controller was implemented on a Digital Signal Processing System. Due to the high eigenfrequencies of the oil pan the controller has to work with a high sampling rate. Therefore, in this case a sampling rate of 20 kHz was chosen.

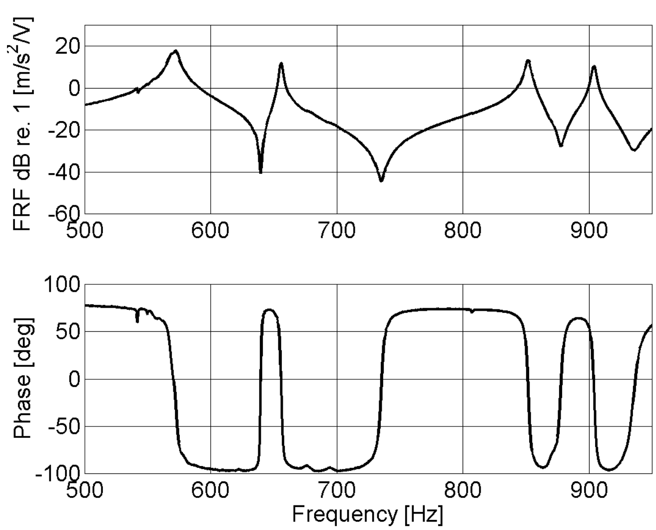

Figure 8: Frequency Response Function between Driving Signal of Patch 1 to Acceleration at Sensor 1

In Figure 9 the block diagram of the collocated controller can be seen. k_M, k_v and k_A are the measurement amplifier, the velocity and the actuator amplifier value respectively.

The Controller Transfer function $G_R(s)$ is given by:

$$G_R(s) = -k_v.$$

The possibility of realization in analogue technique is a big advantage of this controller concept.

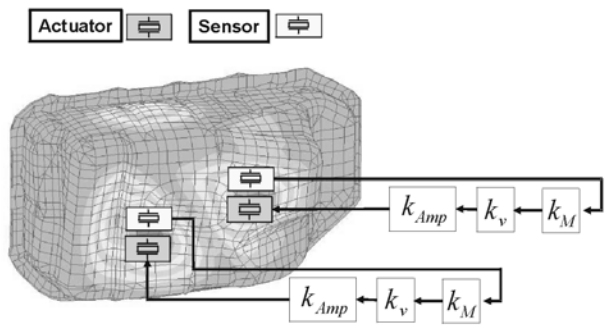

Figure 9: Block Diagram of the Collocated Controller

ADAPTIVE FEEDBACK CONTROL

For the adaptive feedback control a Least Mean Square (LMS) algorithm was implemented. Figure 10 shows the block diagram of the adaptive feedback controller.

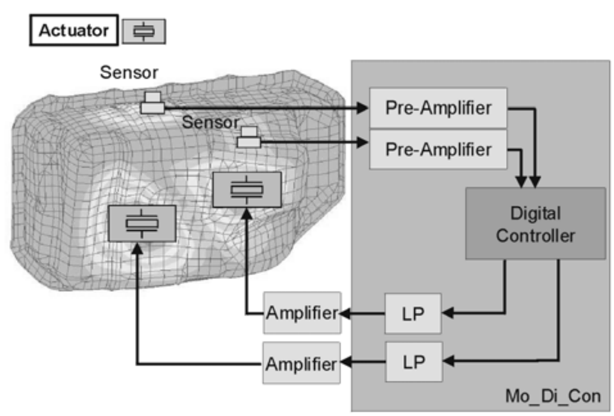

Figure 10: Block Diagram of the Adaptive Feedback Controller

ADAPTIVE FEEDFORWARD CONTROL

Figure 11 shows the block diagram of the adaptive feedforward controller.

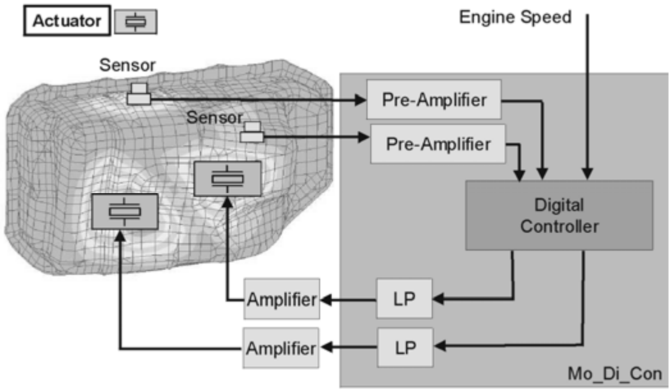

Figure 11: Block Diagram of the Adaptive Feedforward Controller

VERIFICATION MEASUREMENT

SEMI-PASSIVE DAMPING

The vibration behavior of the oil pan with and without semi passive damping can be seen in Figure 12. The vibration reduction is 11.4 dB.

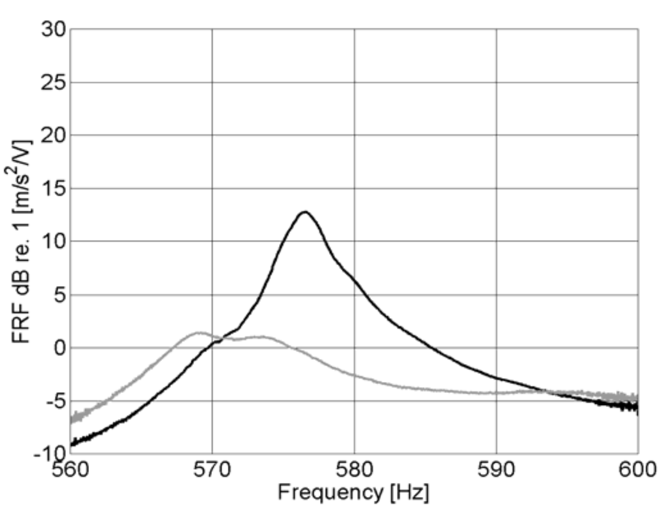

Figure 12: Frequency Response Function of Shaker Input Signal to Acceleration at Patch 1 Position without Semi-passive Damping – Black Curve and with Semi-passive Damping (R=100 Ohm) – Gray Curve

COLLOCATED CONTROL

The vibration behavior of the oil pan with and without collocated control can be seen in Figure 13.

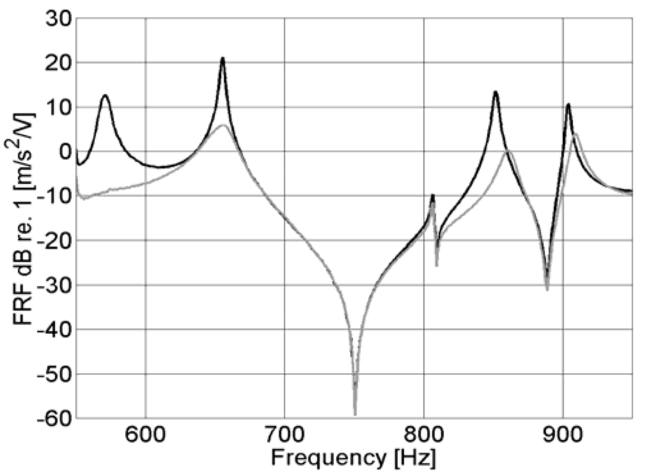

Figure 13: Frequency Response Function of Shaker Input Signal to Acceleration at Patch 1 Position without Control – Black Curve and with Collocated Control – Gray Curve

The vibration reduction in the first two relevant modes is 24 dB and 18 dB respectively. In the next couple of modes the vibration reduction is approximately 12 dB and 6 dB respectively.

ADAPTIVE FEEDBACK CONTROL

The vibration behavior of the oil pan with and without feedback control can be seen in Figure 14. The Vibration reduction in the most relevant eigenmodes in the frequency range from 500 to 1200 Hz is up to ≈ 20 dB.

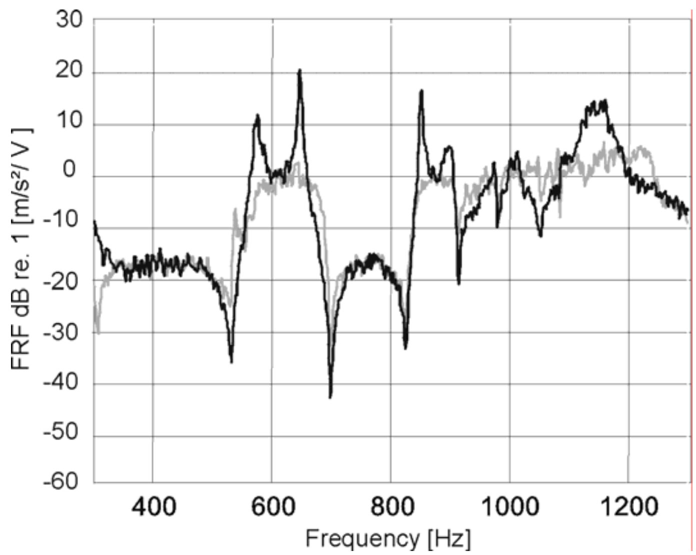

Figure 14: Frequency Response Function of Shaker Input Signal to Acceleration at Patch 1 Position without Adaptive Feedback Control – Black Curve and with Adaptive Feedback Control – Gray Curve

ADAPTIVE FEEDFORWARD CONTROL

The vibration behavior of the oil pan with and with out feedforward control can be seen in Figure 15.

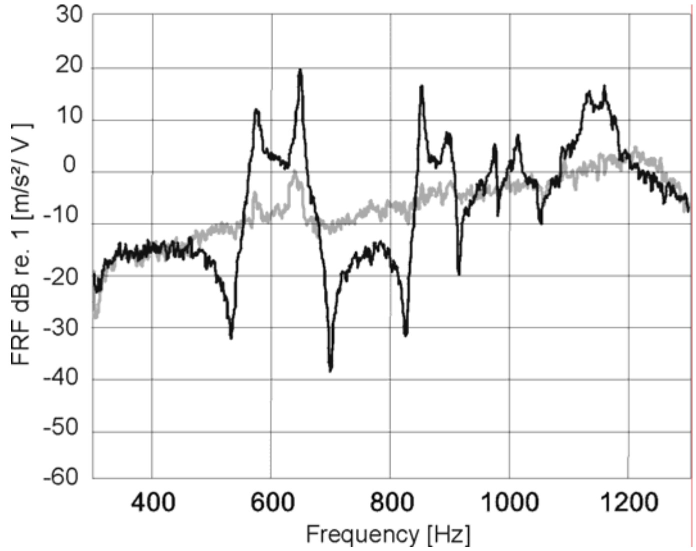

Figure 15: Frequency Response Function of Shaker Input Signal to Acceleration at Patch 1 Position without Adaptive Feedforward Control – Black Curve and with Adaptive Feedforward Control – Gray Curve

With the adaptive feed forward control a vibration reduction of up to 24 dB (acceleration level in peaks) has been achieved.

EVALUATION OF ACTIVE AND PASSIVE OIL PAN NOISE REDUCTION MEASURES

The introduction of an ANR system for automotive powertrain oil pan application in mass production – besides technical and costs challenges - finally depends on the achievable additional customer benefit in comparison to the conventional passive optimization measures.

Table 2 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 2: Comparison of Passive and Active Oil Pan Noise Reduction Measures

Measures	NVH benefit	Costs	Robust-ness	service	package	Fuel consump. (CO_2)	Oil Temp.
Structural Optimisation	base	base	base	base	base	base	base
Material (e.g. magnesium)	+	-	+/-	+/-	+/-	+	+/-
decoupling	+	-	-	-	-	+/-	+/-
Encapsulation (@ P/T or vehicle)	++	- -	-	- -	- -	- -	- -
ANR System (actuator, sensor, control)	++	-	-	-	+/-	+/-	+/-

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

It becomes clear that an ANR system exhibits promising NVH benefit compared to the actual noise reduction measures even the costs of an ANR 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). In principle, the expected promising NVH benefit will be also achievable for other powertrain “thin wall structures” as e.g. valve and timing drive cover. However, the final acceptance in automotive industry will strongly depend on the challenges with regard to costs and robustness.

CONCLUSION

The potential for reducing exterior powertrain noise by reducing engine oil pan noise helped to drive the development work for this study. The prospect of reducing engine oil pan noise through the Active Structure Acoustic Control (ASAC) method was studied, using FEM/BEM simulations of the oil pan. Achieving the optimum ASAC system performance required that the positions of the sensor(s) and actuator(s) needed to be defined. A variety of semi-passive and active control strategies were investigated. Ultimately, four strategies were selected and the vibration damping for each system was evaluated on a test stand. The adaptive feed forward control strategy showed the greatest potential for this application. This strategy achieved a decrease of up to 24 dB in surface vibration levels at critical resonances. These results further highlighted the potential of such strategies. Balancing passive and active noise reduction results, the ANR system showed great promise for reducing potential noise radiated from the oil pan. However, introducing this into automotive mass production primarily depends upon the benefit to the customer. There are challenges considering acceptable system costs and reliability under automotive environment conditions.

ACKNOWLEDGMENTS

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