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Development, simulation and experimental investigation of a function-integrated and foam damped oil pan for a two cylinder diesel engine

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The oil pan is one of the most significant acoustic emitters of conventional internal combustion engines. This is caused by its big surface area, its thin walled structure and its usually stiff connection to the cylinder crankcase. Thus, the acoustic behavior is an important aspect for the development of lightweight oil pans. In this paper an oil pan that integrates functions of cooling-water-oil-heat-exchange and oil filtering is presented at first. Then, stiffness modifications of the structure are investigated to reduce the emitted noise, which result in an acoustic improved structural design. Furthermore, also noise damping effects of different aluminum and plastic foams glued on the surface of the oil-pan are investigated experimentally and as well by simulations. According to the results, the oil pan is optimized regarding noise and weight aspects. Further acoustic experiments show that the different types of foams connected to the vibrating metallic surface increase the energy

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dissipation and also reduce the emitted sound pressure over the frequency range of interest. Finally, the pro and cons of the different approaches are discussed and suggestions for an optimal oil sump design are given.

1 INTRODUCTION

Within the European Union (EU) stricter regulations regarding the vehicle noise will be established within the next years[1]. Thus, passenger car manufacturers are forced to reduce the pass-by noise of a car, which is verified by a test procedure described in the EU-Regulation linked above. The most significant components of the pass-by noise are the rolling noise emitted by the tire-road-contact, the air noise induced by the air flow surrounding the car and the engine noise transmitted to the vehicles structure and radiated into the atmosphere.

The most important radiator of an engine is the oil pan due to its big surface and usually thin-walled structure. There are several possibilities to reduce its sound radiation using structural measures like roll crowns, fins, crimpings which stiffen the structure and avoid the radiation of the dominant frequencies of the first engine orders. A passive measure to stiffen the whole engine structure is a bedplate construction or a bipartite oil pan. Another passive measure is decoupling the oil pan from the crankcase by an elastic ring at the interface between both[2].

The authors of this article investigated another passive measure to reduce the engines' sound radiation significantly. It is a full encapsulation of a whole engine. Its effects are the reduction of the sound pressure level (SPL) about more than 10 dB and a much better thermal behavior reducing the heat losses of the engine and the friction losses due to a faster increasing of the oil temperature after the engines' start[3]. This full encapsulation has the disadvantages to increase the engines weight about at least five kilograms and to increase the package volume of the engine about more than 50 %. Thus, other effective passive measures to reduce the engine's sound radiation are required which are less weight- and volume increasing as an encapsulation, but show its positive effects as well. For this reason, in the paper at hand the effects of stiffening and/or noise-dissipating layers of different metallic and synthetic foams at the surface of an oil pan, which is developed under the aspect of weight reduction with the goal to integrate an oil-water-heat exchanger and an oil filter, are examined. The function integrations are motivated by the aim to reduce the mass of the whole engine by avoiding unnecessary housings and mounting parts.

2 THEORETICAL CONSIDERATION

In order to reduce the radiated sound power of a surface, its influence parameters have to be regarded. According to the basic equation of engine acoustics it is determined by

$$P(f) = \rho \cdot c \cdot \bar{v}^2(f) \cdot A \cdot \sigma(f). \tag{1}$$

In Eqn. (1) $\rho \cdot c$ is the characteristic impedance of the air which depends only on its pressure and temperature and is hence a negligible factor for the development. $\bar{v}^2(f)$ is the squared and frequency dependent particle velocity averaged across the radiating surface. A is the radiating area and $\sigma(f)$ the radiation factor. Consequently, reduction of the engine noise means to reduce these three influence parameters. The particle velocity at the surface of an oscillating plate is determined by the factors

$$\bar{v}^2(f) \sim \frac{F_E^2(f)}{\sqrt{B \cdot m''^3} \cdot \eta(f)}, \tag{2}$$

wherein m'' is the areal density of the plate calculated by $m'' = \rho_{Pl} \cdot h_{Pl}$ and B the bending stiffness. For a rectangular plate B is

$$B = \frac{h_{Pl}^3 \cdot E_{Pl}}{12 \cdot (1-\mu^2)}. \tag{3}$$

This equation describes the influence of the material thickness, the Young's modulus and the Poisson's ratio for the biggest radiating surfaces of the oil pan as well. The third interesting parameter $\eta(f)$ is the dissipation factor. It is defined $\eta(f) = W_v / 2\pi W_r$ which is the ratio of the oscillation energy transformed into heat by dissipation W_v and of the regained energy W_r at one oscillation. Materials as steel and aluminum have low dissipation factors smaller than 10^{-3}. Soft materials as organic and synthetic foams offer higher ones.

The fourth interesting size F_E is the excitation force. It is determined by the combustion pressures and the mass forces of the crank drive. It is dependent on the operating point of the engine and the combustion process and therefore no suitable design variable for an acoustic optimization.

The radiated power is proportional to the second factor, the radiating surface A. So, the aim of the development under acoustic aspects is to minimize A. The radiation factor $\sigma(f)$ is defined as the ratio of the radiated sound power and the maximum power, that can be radiated by a piston radiator with the same area as the radiator at its frequency. A detailed description of the radiation factor and its calculation is given by Kollmann[4]; for the purposes of this investigation it is sufficient to know that it is smaller than one below the bending wave limiting frequency f_g which is determined by

$$f_g = \frac{1}{2\pi} \cdot c \cdot \sqrt{\frac{m''}{B}} \tag{4}$$

and has the value of at least one at higher frequencies[4]. In order to obtain a small radiation factor in a preferably wide frequency range, it is required to increase the areal density and to decrease the bending stiffness i. e. the effective thickness of the plate. Regarding the Eqns. (1) and (2) it is clear that decreasing the bending stiffness would lead to a higher oscillation velocity at the surface of the plate. The positive effect of the acoustical short circuit at lower frequencies that causes the reduction of the radiation factor is accompanied by a higher sound power. Therefore, the effect of the radiation factor is not the point of interest in the development of the oil pan and not further investigated in this article. According to Eqns. (1) and (2) the remaining three ways to reduce the sound power are increasing the bending stiffness of the radiating areas, decreasing the radiating areas itself, and increasing the dissipation factor.

The acoustic improvements at the new-developed, function-integrated oil pan are obtained in four different ways. The first is to stiffen the structure by developing an acoustical optimal, lightweight fin configuration that is able to increase the bending stiffness in order to reduce the surface velocity of the oscillating surfaces. To stiffen the structure results in a shift of its eigenfrequencies. The second tries to achieve a higher stiffness of the structure by adding stiff metal and plastic foams to big surfaces, primarily the oil pan bottom. The third tries to achieve a higher dissipation factor η by dissipating acoustical energy by applying synthetic foams. These foams are not able to increase the bending stiffness significantly, but have approved good absorption coefficients shown in the data sheets of their manufacturers and in Table 1. The fourth way is the reduction of the radiating areas. It results in a diminution of the oil pan and a decrease of the amount of the oil that has a damping effect due to its viscosity.

3 DEVELOPMENT AND EXPERIMENTAL INVESTIGATION

The development starts with the design of the new oil pan that integrates an oil-water-heat exchanger and an oil filter. The oil pan is designed with plane areas to simplify the application of different foams on its surface. Various heat exchanger models are designed and calculated regarding the mass and the heat flow rate at a given temperature difference. The heat exchanger is conceptualized in such a manner that it can be produced by the project partners of the university. The partners take the decision to manufacture an exchanger made of small aluminum tubes with the diameter of 10 mm and a length of 4.60 m. The tubes are finned and thin walled to improve the heat conductivity. The next important task is the design of a lightweight stiffening structure for the biggest sound radiating area of the oil pan: its bottom. Based on already approved fin structures six different fin structures are designed and simulated regarding the vibration amplitudes of the surface. In the simulation the crankcase is modelled with the attached new developed oil pan and a holistic simulation workflow is used[5]. Independently the crank drive is modelled as a multibody system (MBS) and pressurized with the cylinder pressures indicated at the real engine. The calculated reactions in the crankshaft bearings deliver the excitation forces of the crankcase and hence of the oil pan itself. The discretization of the solid and fluid elements is realized with tetrahedrons with quadratic shape functions. The simulation of the sound field is executed uncoupled, i. e. the reacting influence of the air on the structure is neglected. Moreover, only the sector affiliated to the oil pan is modeled for acoustic consideration due to the computational effort. The sound radiation is calculated under free-field conditions. For this, the Sommerfeld's radiation condition is fulfilled by using absorbing boundary conditions. The corresponding finite element model is shown in Fig. 1.

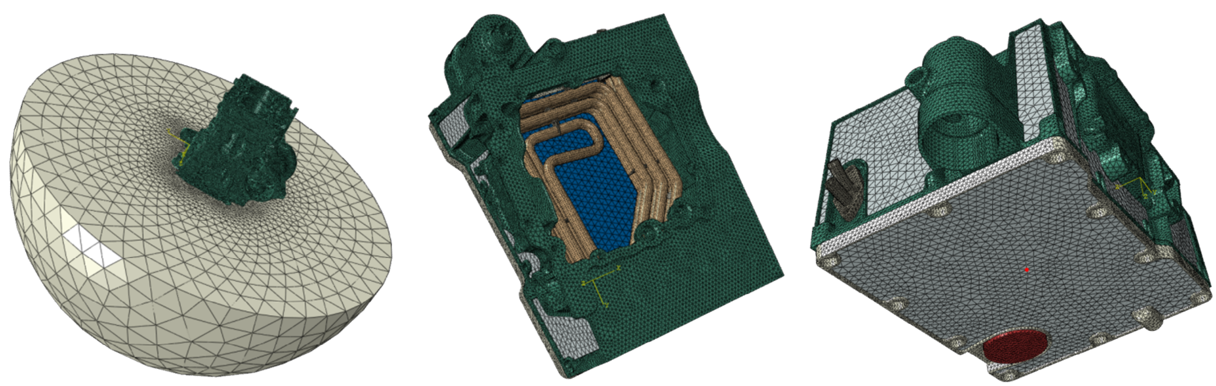

Fig. 1 – Finite-element-model of the crankcase with oil pan and the surrounding air-volume (left) and of the developed oil pan itself (middle, right)

The grey areas at the oil pan shown in Fig. 1 are the foams that are glued on its surface. The oil-water heat exchanger is depicted in brown color at the picture in the middle. There is a blue area underneath the heat exchanger that marks a 20 mm thick foam plate with a smaller density than the oil in the pan used for purposes of heat insulation and mass reduction by oil displacement. The fins of the oil pan bottom are covered by the foam plate at the inside. The aluminum structures with flanges and the oil filter casing are green.

Six different ribbing variants are designed and calculated. Figure 2 shows the simulated vibration amplitudes over all frequencies for five variants. The sixth one is derived from the ribbing variant 4. The difference is that between the fins a lattice of very thin and low ribs is provided with the lattice constant 5 mm. This variant doesn't show a significant difference to the ribbing variant 4 and is hence not depicted in Fig. 2.

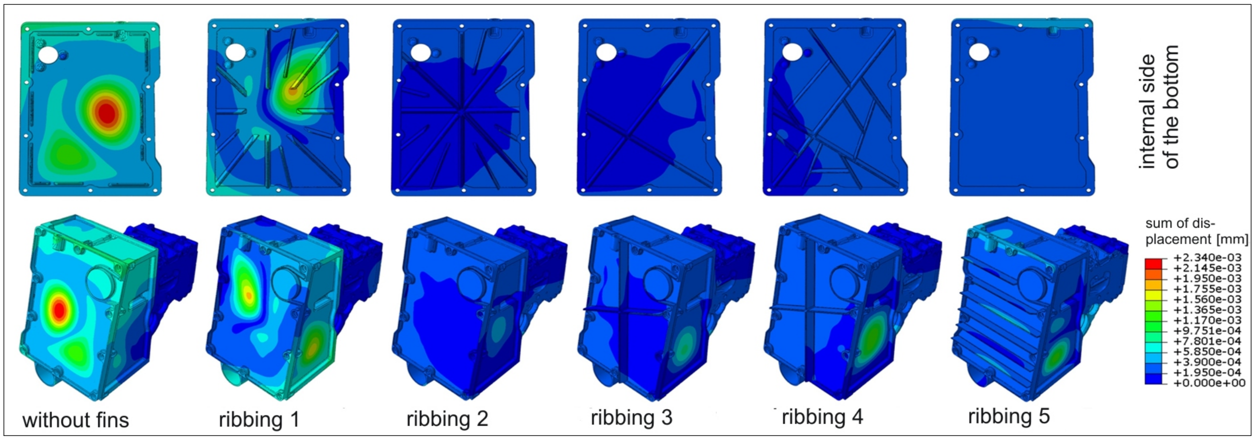

Fig. 2 – Simulated vibration amplitudes at the oil pan at different ribbing variants

As shown in Fig. 2 the ribbing variant 2 is the one showing the smallest vibration amplitudes under the influence of engine combustion. Compared with the variant 3, 4, 5 it has the advantage of a small weight and no fins at the outer surface. So, the whole surface can be "dressed" with foam. It shows a significant improvement regarding the surface oscillation compared to variant 1 that avoids ribbings in the center of the bottom plate. Thus, it can be generalized, that the ribbing of the sound radiating areas should always cover the whole area and preferably should not spare the center of the osciallating area, because this means a significant reduction of the stiffness of the plate and hence higher exitations and surface velocities of the oscillating surface resulting in a higher radiated power.

Consequently, the ribbing variant 2 is chosen and used for the further development of the oil pan. The effect of the ribbing is investigated only by simulations, not by experiments. The results are shown in Fig. 3. Compared with the unribbed variant the ribbed oil pan shows a significant (ca. 30 dB(A)) lowered sound pressure level in the near field than the unribbed one. It is taken into account that the oil pan without fins has a thickness of 3 mm and thus, although the improved acoustic short circuit and the lowered radiation factor at low frequencies, delivers optimal conditions for a high sound power radiation. Because of this result the ribbing of the pan is seen as unavoidable for an acoustic optimal behavior.

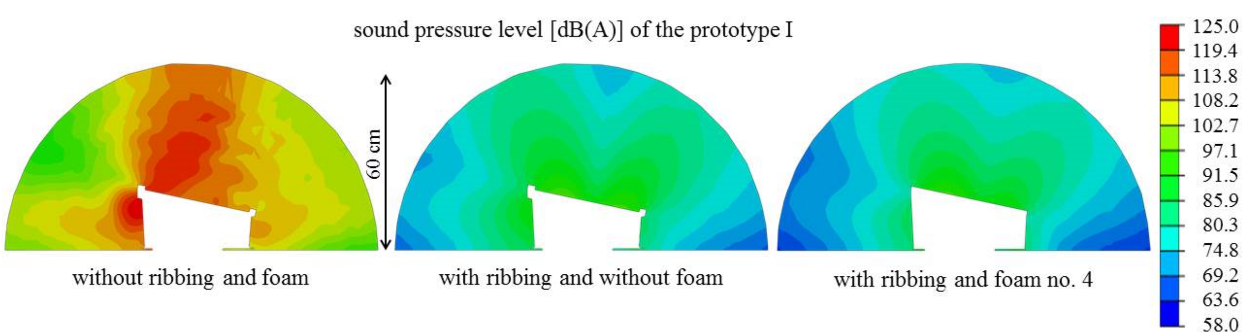

Fig. 3 – Comparison of the acoustic effect of ribbing and using foam at the oil pan with normalized sound pressure levels

The casing with a stiff plastic foam has a quite small influence on the reduction of noise compared to the significant effect of the ribbing in the simulation results. The simulation is carried out with the other stiff foams as well. Since there is no significant improvement or difference in the results observable, it is omitted to present them here.

As outlined, the improvement of the acoustic behavior of the oil pan using foams is basically achievable in two different ways. The first is the additional stiffening of the surface by the foam. The second is the increasing of the dissipation factor, i. e. the causation of dissipation of sound energy at the surface of the foam. The first effect must be gained with stiff foams, the second with soft ones. The simulation results in Fig. 3 showing the effect of a stiff foam at the surface of the oil pan suggest that soft, energy dissipating foams could be a promising alternative. A further interesting variant is the use of multi layered foams and sandwich structures whose influence has been investigated experimentally. The facing of the oil pan with foam in this investigation contains six different structural cases which are sketched in Fig. 4.

In the experiment, all investigated foams are glued at the surface of the oil pan with a hot melt adhesive with a melting temperature of 180°C and a softening temperature of 145°C. At the oil pan bottom the glue covers the whole area with a thickness of 1 mm. Another fixation method using hook-and-loop tapes is experimentally regarded in preliminary testings. It doesen't shows satisfying acoustic improvements in comparison with the glueing method and is as a consequence not used for the oil pan experiments.

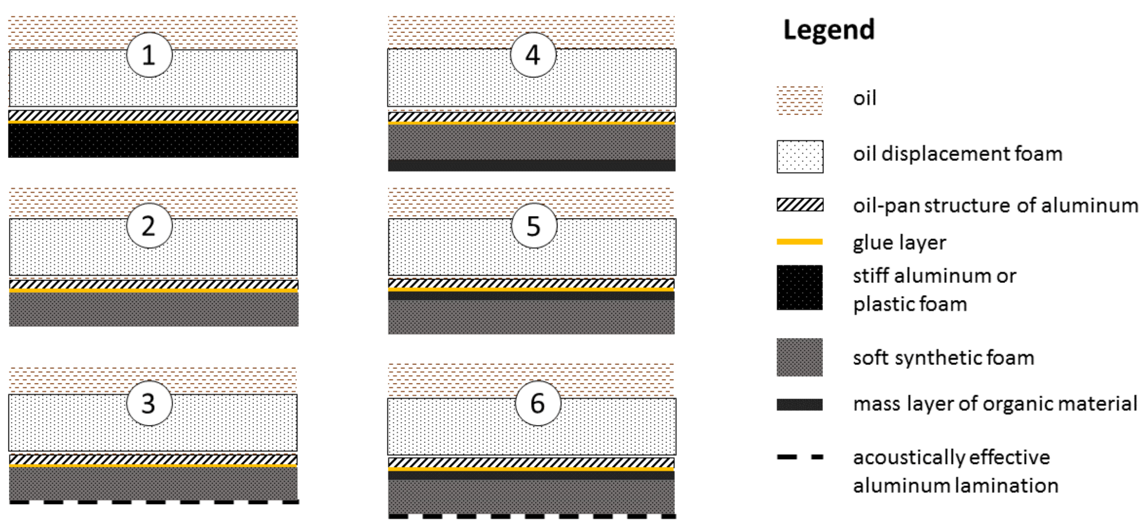

Fig. 4 – Investigated foam structures at the oil pan bottom

The foams that are adhered at the oil pan bottom are described in table 1. For the soft foams the sound absorption coefficient α over the frequency is available from the manufacturers. Usually, foams with noise absorbing properties offer a low absorption coefficient at low frequencies that increases up to a certain frequency about which they remain constantly high. In table 1, this frequency is called f_c. The lower f_c is, the better are the absorption properties at low frequencies.

The first five foams are stiff foams, three of them made of aluminum, two of synthetic material. Foam no. 2 is laminated at both sides with a 0.8 mm thin sheet of aluminum so that it offers a plane surface without visible aluminum foam pores as foam no. 1 and 3. The experimental investigation of the foams is carried out at the running engine on a test bench. The operating point of the engine is identically during all experiments, the operating point settings as well. A microphone array with an area of 350 x 150 mm and a squared grid order of 50 mm is mounted in a distance of constantly 100 mm under the foam surface. The bottom of the oil pan

Table 1 – Investigated foams

foam no.	applied foam structure (see Fig. 4)	thickness [mm]	α [–] at f_c [Hz], testing method	m” [kg/m²]	description
1	1	10	-	4.14	Closed-cell aluminum foam with small pores
2	1	10	-	9.91	Closed-cell aluminum foam with medium pores, double side layered with thin aluminum sheet
3	1	10	-	7.28	Closed-cell aluminum foam with medium pores, inner side layered with thin aluminum sheet
4	1	20	-	2.44	Microporous, stiff polyethylenterephthalat-Foam
5	1	20	-	2.29	Microporous, stiff polymethacrylimide foam
6	6	25	1.05 at 1000 (α-cabin)	2.78	Microporous soft polyurethane foam with stiffening layer and acoustic aluminum foil
7	5	20	1.05 at 1250 (α-cabin)	2.45	Microporous soft polyurethane foam with stiffening layer and a thinner foam layer
8	2	10	0.9 at 4000 (impedance tube)	5.00	Microporous soft polyetherurethane foam with a high density
9	3	10	0.9 at 3150 (impedance tube)	1.60	Microporous soft polyurethane foam with a low density
10	5 and 4	15	0.74 at 1000 (α-cabin)	1.49	Microporous soft, lightweight polyurethane foam with stiffening/mass layer
11	6	15	0.75 at 1000 (α-cabin)	2.05	Microporous soft lightweight polyurethane foam with stiffening layer and acoustic aluminum foil

has dimensions of about 350 x 250 mm. So, the microphone array is slided 150 mm each data acquisition process. The measurement setup is shown in Fig. 5 on the left side. The middle shows the oil pan bottom and its fins from the inner side and without the oil displacement foam mounted onto them. The image on the right hand of Fig. 5 shows the outer side of the bottom with the glued foam no. 8. On the left image, below, a white material is visible (red marker). It is absorption material to avoid sound reflections at the concrete ground of the test bench. This absorption material features full absorption at all frequencies above 100 Hz and consequently only these frequencies are evaluated.

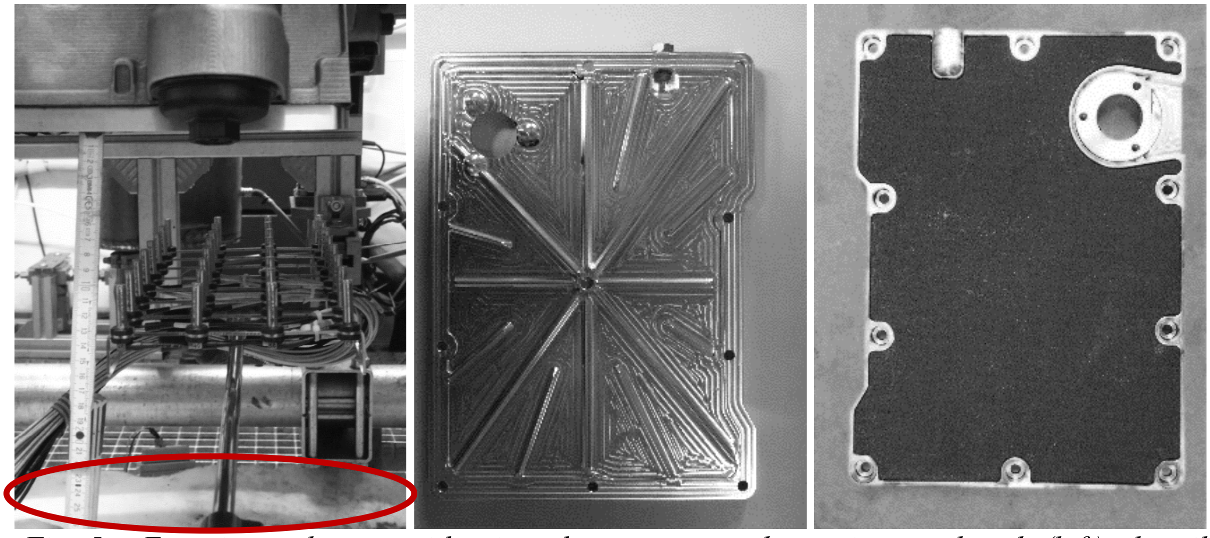

Fig. 5 – Experimental setup with microphone array at the engine test bench (left), the oil pan bottom with fins from the inner side (middle), the bottom with foam at the outer side (right).

The first result of the array measurements are the sound pressure contributions under the oil pan for the different foams in the relevant frequency range from 0.1 to 8 kHz. They are shown in Fig. 6. The upper row contains the results for the stiff foams and the bottom without any foam. The lower row shows the results for the synthetic and soft foams no. 6-11.

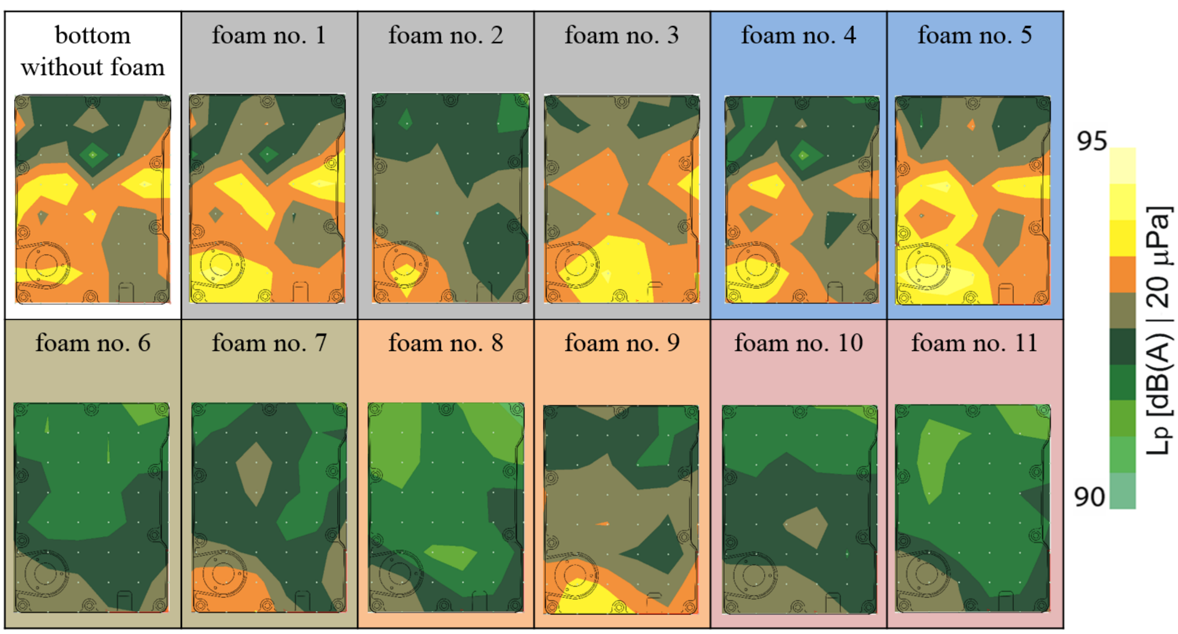

Fig. 6 – SPL contributions at the oil pan bottom at the engine operating point 2000 rpm, 47 Nm in the frequency range 0.1-8 kHz

In accordance with the simulation results (Fig. 3) there is no significant difference in the SPL measured for the stiff foams. The best results are shown by the both-side layered aluminum foam no. 2, which shows an SPL reduction of 0.5 dB(A) over the whole area in comparison to the bottom without foam. The other stiff foams don't manifest such a reduction, the polymethacrylimide foam no. 5 has even a slightly higher SPL than the bottom without foam.

The lowest sound pressure levels are obtained by using the soft, sound dissipating foams no. 6-11. The highest reduction of the sound pressure level is measured at the high-density polyurethane foam no. 8. Here the SPL reduction is 1.3 dB(A). Reducing the density of this foam with foam no. 9 (that is made of the same substances and produced with the same procedure) leads to a significant increase of the sound pressure level in comparison to foam no. 8 resulting in a SPL reduction of only 0.3 dB(A). Thus, for the soft polyurethane foams no. 8 and 9 the areal density seems to play an important role for the reduction of the sound pressure level. This result is confirmed by the effect of the foam no. 6 and 7. The lighter foam no. 7 delivers a higher SPL than foam no. 6. Both, foam 6 and 7 have an identical stiffening layer, but foam no. 6 has a thicker foam layer and additionally an acoustic aluminum foil at its surface so that its areal weight is increased. According to Eqn. (2) the increasing of the areal density m'' has a bigger effect of the oscillating velocity than increasing the bending stiffness B. Nevertheless, the aluminum foams no. 1 and 3, that offer a higher areal mass and stiffness than the foams no. 6 and 7 don't show such an effect. Consequently, for foams applied to the surface a higher areal mass seems to be only acoustically effective if it is combined with energy dissipating effects (that are negligible small in the case of the stiff foams no. 1-5). Regarding the effect of the light foam no. 10, that has the smallest areal density of all investigated foams, it seems that even very light, soft foams may cause a sound reducing effect. Foam no. 10 reduces the sound pressure level about 0.85 dB(A). Its application follows the structure 5 in Fig. 4; the stiff mass layer is adhered to the bottom of the oil pan. Adhering an acoustical effective aluminum foil on the outer surface of foam no. 10 leads to further improvement than foam no. 11 of about 0.3 dB(A). This effect is caused by the additional mass at the surface that transforms the system into a more dissipative effective spring-mass system, close to the structure 4 in Fig. 4. This structure is examined at the weight reduced second prototype of the function integrating oil pan.

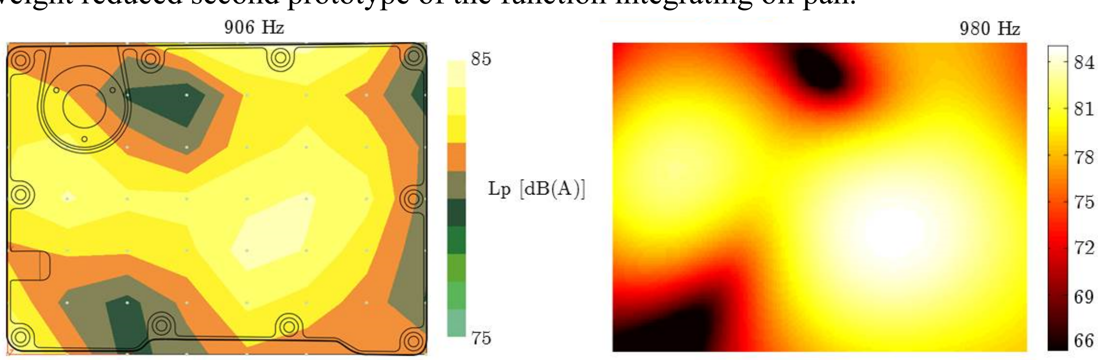

Fig. 7 – Comparison of experiment and simulation at a characteristic bending frequency

After the acoustic measurement the results are compared with the simulation. For this, it is necessary to adapt the material parameters of the foams, because these show a high stochastic uncertainty in their material properties and of course, they have a significant influence on the simulation results. In Fig. 7 the comparison of the real measured values at one characteristic frequency of the oil pan bottom and the correlating simulation results is given. It is obvious, that the results are quite similar. The fact, that the simulation shows the resonance at a frequency being 74 Hz higher than the experiment's one is caused by the oil in the real oil pan, whose influence is not taken into account in the simulation. The mass of the oil causes a shift of the characteristic frequency to low frequencies. This effect is confirmed in a single experiment with an oil pan that is investigated with and without oil inside. It is also known from other publications[6].

With the aim to reduce the weight of the first oil pan significantly a smaller pan is designed, manufactured and applied to the engine. Including the heat exchanger it has a mass reduced about 1.55 kg in comparison to the first oil pan, that has a mass of 7.38 kg. The mass reduction is achieved by decreasing the oil volume and the length of the heat exchanger, that is shortened about 1.0 m and has a length of 3.6 m now. The oil volume is reduced from 6.4 l to 5.0 l. To reduce the weight of the oil pan and the oil it is taken into account that the damping effect of the oil mass is reduced and so higher SPL levels will be obtained compared with the first oil pan. The area of the oil pan bottom is reduced about 25 % and though, according to Eqn. (1), the radiated sound power. The new oil pan mounted at the engine, the new oil pan bottom with its rib structure, the displacement body of the pan and the heat exchanger are shown in Fig. 8.

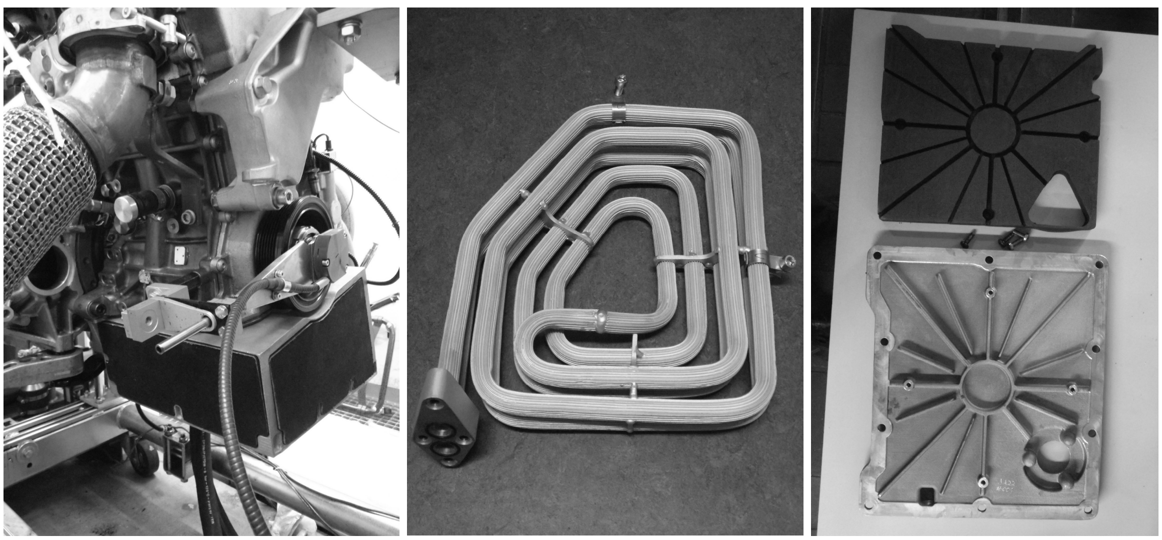

Fig. 8 – New oil pan at the engine (left), new oil-water-heat-exchanger (middle), new bottom with fins and oil displacement plate (right)

Considering the results of the microphone array measurements described above the new oil pan is applied with the structure no. 4 (Fig. 4). The most appropriate foam for that purpose is foam no. 10 (see Fig. 6) that shows good results in the measurements before and has the smallest areal density (table 1). Before the foam is glued at the surfaces, the oil pan is examined experimentally without foam. The acoustical examination is carried out with a single microphone placed exactly 100 mm underneath the oil pan bottom surface as the microphone array before. The settings and the examined operating point of the engine is the same as before. The result of the evaluation of the microphone measurements is depicted in Fig. 9 and compared with the third octave band filter results of the same foam mounted at the first oil pan. The difference of both diagrams indicates the difference of the structural cases 4 and 5 in Fig. 4. At the new oil pan the foam is mounted with the mass layer directed to the outside, at the old one it has been mounted as a stiffening layer to the inner side. Comparing the level of the old and the new oil pan without foam it is obviously that the new one's is bigger. Since there are no changes in the wall thickness of the oil pan bottom and the fins structure remain almost identically, the factors bending stiffness, areal mass or dissipation factor are taken out of consideration being responsible for this change. Thus, the effect has to be explained by the lower amount of oil in the new pan causing a decreased damping effect as outlined before.

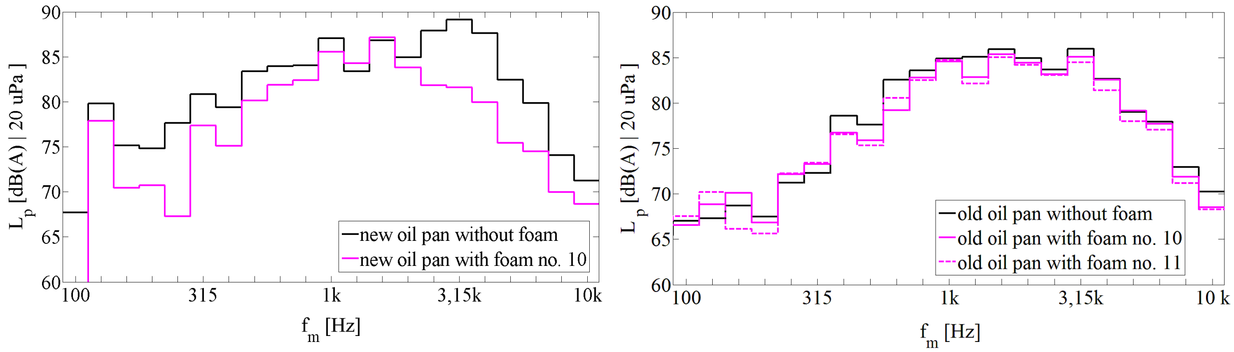

Fig. 9 – Third octave band filter analysis of the SPL at the new and the old oil pan and the influence of foam no. 10 mounted in the structure case 4 at the new oil pan (left) and case 5 at the old one (right). The operating point of the engine at both: 2000 rpm, 47 Nm.

The sound reducing effect of foam no. 10 is quite better at the new oil pan in comparison to the old one. At the new one, a SPL reduction of 3.1 dB (A) is proved in comparison to the bare oil pan. So the structural case 4, in comparison to 5 and the other ones, is the preferable one. At the new oil pan the stiff layer serves as a mass that is beared on the soft foam so that both work as a spring-mass system. Hence, the dissipation factor is increased and more oscillation energy dissipated – not only at high frequencies.

4 DISCUSSION AND CONCLUSIONS

The experimental and numerical development and investigation of a new, function-integrated oil pan leads to the following results.

The acoustic optimal stiffening variant for a big vibrating area as the oil pan bottom is a center orientated regular ribbing spanning the whole surface at one side of the plate, in this case: the oil sump. Irregular ribbings don't show such good results, even if they are applied to both sides of the plate. The ribs don't need to be higher than 10 mm. A parallel ribbing with thin and long fins doesn't deliver such good results. For the center orientated ribbing it is very important, that the center of the plate is not spared out.

A higher oil mass in the pan causes a shift of the characteristic frequencies to lower ones and a damping effect reducing the sound pressure level. That effect can be taken into account for further oil-pan developments, although a mass reduction of the oil has to be pursued to reduce the overall engine weight and to increase the velocity of the heat-up process of the oil during the engine's warm-up.

Using stiff foams to stiff the oil pans sound radiating areas doesn't show an acoustic improvement. Increasing the areal density of the surfaces can show a small acoustical improvement, if the material causes energy dissipation as well. That means it is important for this case that the used materials offer a higher energy dissipation factor than stiff materials such as aluminum foams. This method has the disadvantage to increase the mass of the foam unnecessarily.

The best method to reduce the sound radiation using foams is the mass layer-design, that has a layer with a higher density and stiffness as the energy dissipating foam and is an oscillating mass evoking a damping, dissipating effect at the soft foam. In this investigation the achieved result is a SPL reduction of 3.1 dB(A). This result is lower than the effect of a full encapsulation of the engine that reaches a SPL reducing effect of 10 dB(A) and more[3]. Nevertheless, it is a promising approach to gain audible noise reduction with a lower mass effort than a full encapsulation.

5 ACKNOWLEDGEMENTS

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