VIRTUAL ENGINEERING APPROACH FOR THE ANALYSIS OF THE ACOUSTIC BEHAVIOR OF AN ENGINE

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Beside the optimization of power and fuel efficiency is the reduction of the noise emission one important objective in the development of combustion engines. The numerical simulation of the noise radiation into the surrounding air volume needs knowledge about the surface velocities of the engine. These surface velocities are received by a finite element based numerical vibration analysis of the engine. Until now the required input of such a vibration analysis is determined by extensive measurements, for example measurements of the bearing forces of the cylinder crank-case. In this paper a holistic virtual engineering approach is presented, which replaces the experimental input by an elastic multi-body simulation. From the elastic multi-body simulation the forces which are exciting the engine can be calculated. These forces are caused by the piston motion and affect the main bearings and the cylinder walls. Based on these excitation forces the vibration analysis of the engine is performed. Subsequently, the noise radiation of the engine in the environment is analysed. The entire simulation workflow is demonstrated for a 2-cylinder diesel engine. With the aid of the presented approach it is possible to investigate the influence of geometric crank drive modifications on the noise emission of the engine early in the product development process. This is the main advantage of a virtual engineering approach, because no input data of real prototypes are required.

1. Motivation

The importance of passenger cars noise emission increases more and more, because it strongly influences the comfort of the vehicle and consequently also the purchase decision of the customers. For that reason, it is essential to be able to predict and evaluate acoustic consequences of modifications of the construction or the engine control early in the development process of passenger cars. Up to now, mostly experimentally determined excitations are used in the analysis of the acoustic behaviour of combustion engines. The challenges in the development process require on the one hand side the minimization of the experimental effort but on the other hand measurements are not possible due to the non-existing prototypes in early development stages. Hence, a holistic virtual engineering approach for the acoustic analysis is presented in the current paper, which uses a precedent elastic multi-body simulation instead of extensive measurements to obtain the vibration excitations of the combustion engine. Additionally, changes in the acoustic behaviour, which for example are caused by a modified combustion process or modifications of the crank drive, can be calculated by this way with the help of the holistic simulation workflow. A further advantage of a previously executed elastic multi-body simulation is the possibility to consider the deformations of the single cylinder walls. These deformations result from lateral and tilting motions of the piston due to the combustion process respectively the thereby generated gas forces. The resulting superposed load on the cylinder walls is in contrast to, for example, the bearing forces not measureable. Consequently, the presented approach provides on the one hand the opportunity to substitute the experimental determination of the excitation of the engine structure and on the other hand it enables the consideration of a more realistic vibration excitation.

2. Holistic virtual engineering approach

In the paper at hand the analysis of the acoustic behaviour of combustion engines is carried out with the help of a holistic virtual engineering approach, which is illustrated in Fig. 1. At first, the forces on the bearing blocks and the cylinder walls are determined, which excite the cylinder crankcase and in turn the whole engine. These forces are caused by the combustion process and the piston motion. Both are calculated by an elastic multi-body simulation. The only necessary input of the elastic multi-body simulation is the gas pressure curve of the combustion process. The necessity for this input parameter is not in conflict with the aim to substitute the experimental input of the acoustic analysis by the elastic multi-body simulation. The gas pressure curve hasn't to be obtained by measurements of the current engine prototype. It can also be taken out of a representative data base of previous measurements. Another possibility is the usage of the gas pressure curve, which is defined as aim of the combustion process design. This provides the best option, because this curve is available very early in the development process and furthermore virtually coincides with the real gas pressure curve of the final design.

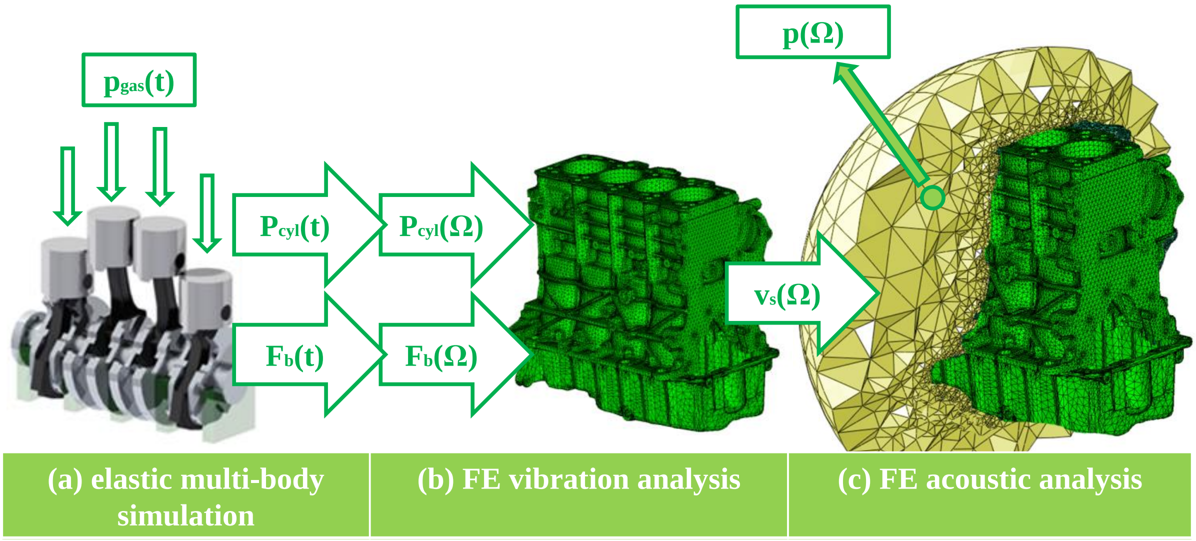

Figure 1. Holistic virtual engineering approach for the acoustic analysis of a combustion engine

The excitation of the cylinder crankcase is mainly caused by the internal cylinder pressure, the forces in the crankshaft main bearings and the secondary motion of the piston. The calculation of the piston lateral motion and the piston tilting requires the consideration of the hydrodynamic fluid film reaction as well as the solid contact of piston and cylinder [1]. A multi-body simulation (MBS) is carried out for five operating cycles of the engine, from which only the last one is taken into account as input for the subsequent vibration analysis to avoid initial disturbances in the calculated time signals of the excitation loads. The resulting signals are periodical. For this reason, it is possible to extend the time signals by repeating the representative excitation from one operating cycle into the next. A longer time signal is advantageous for the Fast Fourier Transform (FFT), which is necessary to transform the various excitation sources into the frequency domain. Both contact parts (piston and cylinder) are modelled as elastic bodies in the MBS. Therefore, they are able to capture local deformations, which result from the elasto-hydrodynamic contact [2]. The main bearing forces are received by the solution of the Reynold's differential equation. Finally, the gas forces, the contact forces and the main bearing forces are applied as loads of the cylinder crankcase model (cf. Fig. 1a, b). The feedback of the crankcase vibrations to the crankshaft and the piston motion can be neglected, as shown in [3].

The finite element method (FEM) is used for the subsequent vibration analysis of the engine. It is executed exclusively in the frequency domain due to the computational costs as discussed in [4]. The bearing reactions are considered as forces and the excitations of the cylinder walls are considered as pressures (cf. Fig. 1a, b) to facilitate the application of the loads to the FE-mesh. The discretizations of the cylinder walls in the MBS and the FE-analysis are not coincident. The nonlinear elastic MBS of the crank drive requires the minimization of the number of degrees of freedom as far as possible due to computational costs. Therefore, in the MBS a rough discretization of the cylinder walls is used. In contrast, in the vibration analysis a detailed model of the whole cylinder crankcase and oil pan is necessary. Thus, a much finer discretization is required. Consequently, a coincident discretization of the cylinders in the MBS and vibration analysis has to be omitted. The resulting forces of the MBS have to be transformed to the nodes of the discretization of the vibration analysis. To create a static equivalent load by nodal forces on a finer discretization is simple, but an energetic equivalent load requires much more effort. In the current paper pressure values are used as surface loads to create the equivalent load for the vibration analysis of the engine. In the vibration analysis model this surface load is defined at the midpoints of the finite element surfaces, which form the inner contour of the cylinders. The amplitude of each surface load is obtained by the elastic MBS with the help of the shape functions used for describing the pressure distribution. Therefore, the midpoints of the element surfaces of the vibration analysis model are provided for the elastic MBS, where the shape functions of the pressure as primary variable are used to calculate the pressure values at these points. Hence, the applied loads of the vibration analysis model match exactly with the calculated loads of the elastic MBS without an additional error by the transformation between the different discretizations. The transformation of the excitation of the cylinder walls in the frequency domain is executed after the calculation of the pressure values at the midpoints of the element surfaces of the vibration analysis model.
Subsequently, the vibration analysis is executed with the help of this excitation to get the surface velocities of the whole engine (cf. Fig. 1b). These surface velocities are required as excitation of the surrounding air volume in the following acoustic analysis (cf. Fig. 1c). An uncoupled acoustic analysis is implemented, neglecting the feedback of the air volume to the vibrating structure [5]. This is a common assumption, because the engine is made of aluminium and consequently much stiffer than the air. The excitation of the air volume is applied as boundary condition in the calculation of the sound radiation. The degrees of freedom of the structure (displacement) and the fluid (sound pressure) are coupled by special interface elements. These elements are shell type elements without stiffness or mass. They require a coincident discretization of the engine and the air volume at the fluid structure interface. As already mentioned, the previously calculated surface velocities of the engine are applied as boundary conditions at the nodes of the interface elements. The spherical air volume is modelled with a discretization, which becomes coarser to the periphery due to the computational costs (cf. Fig. 1c). Generally, tetrahedrons with quadratic shape functions are used to discretize the whole engine and the air volume. The splitting of the large multi-physics system of equations into smaller problems causes a significant reduction in the computational effort. This is due to the fact that the computational effort required increases nonlinearly with the number of degrees of freedom considered.

In automotive applications the engine sound is dominated by multiples of the half or full so-called engine orders [6]. Thus, only these frequencies are taken into account in the analysis due to the computational costs. Furthermore, the frequency range is limited to the human hearing.

Differences in the combustion process are not considered as it is assumed that the differences between two sequenced combustions are negligibly small [3]. The actual load on the engine (idle, part load, full load) is taken into account by the gas pressure distribution as input parameter of the MBS. Furthermore, a warmed up engine is assumed in the consecutively calculations. Otherwise, the temperature changes and the corresponding interactions have to be taken into account in the simulation. This would lead to an enormously increase of the complexity of the problem and the required effort. It is preferential to analyse the sound radiation under stationary operating conditions. Therefore, the present paper focuses on this aspect and enables the opportunity to calculate the acoustic behaviour of combustion engines early in the development process.

3. Modal reduction in the vibration and the acoustic analysis

In multi-body analyses it is common to use modally reduced FE-models for all elastic bodies in order to reduce the computational effort. For this reason, the applicability of modally reduced models in the vibration analysis is investigated in this section. A modal reduction of the air volume in the acoustic simulation is not possible, because standing waves and eigenmodes of the fluid don't exist under free field conditions. Therefore, the influence on the results of the vibration analysis is explored and in turn, the resulting sound pressure radiation into the surrounding air. The results of the presented approach with and without a modal reduction of the FE-model in the vibration analysis are compared in Fig. 2. Next, the influence of the number of eigenmodes considered in the reduced model on the final results is investigated. The excitation is obtained from the MBS for a fixed operating point (2500 rpm, 47 Nm) of the engine in all of the following cases. Fig. 2 shows the distributions of the overall sound pressure level on a plane cut through the centre of the spherical air volume and takes into account all multiples of the quarter engine order up to a frequency of 2 kHz.

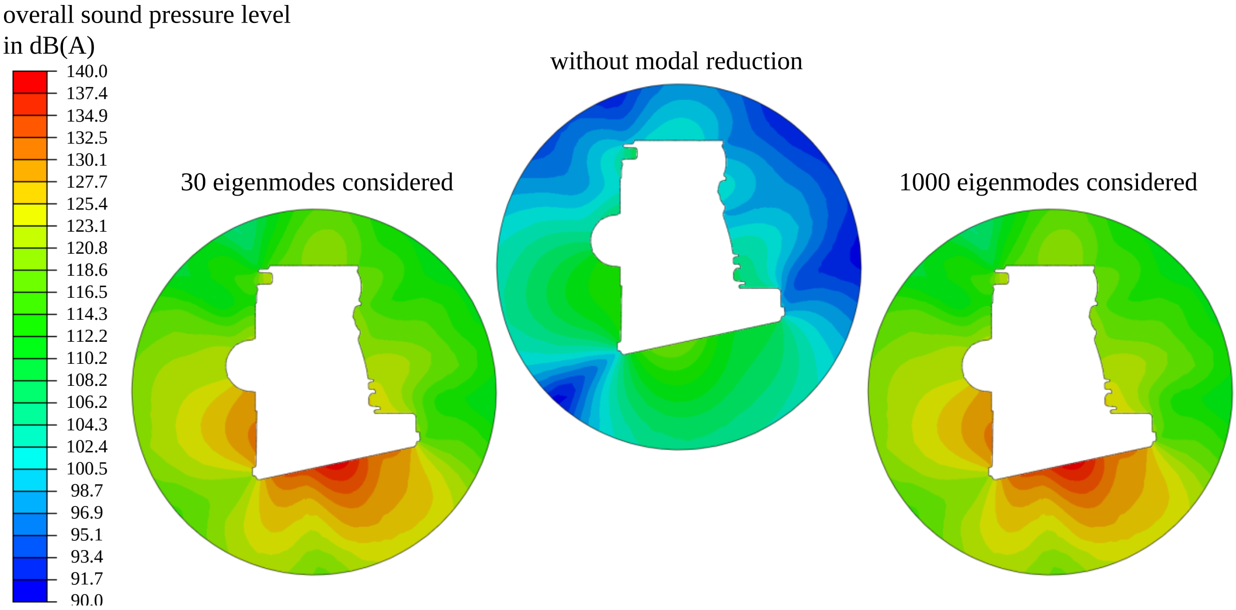

Figure 2. Influence of a modal reduction on the resulting sound pressure distributions

The comparison of the sound pressure distributions in Fig. 2 shows that the usage of a reduced FE-model in the preceding vibration analysis leads to much higher vibration amplitudes of the structure, which also cause much higher sound pressure levels. The results with and without modal reduction are qualitatively similar but are unsuitable for a quantitative evaluation.

A convergence study of the required number of considered eigenmodes was executed to ensure that a sufficient number of eigenmodes is taken into account. The results with consideration of 30 and 1000 eigenmodes are compared in Fig. 2 and there are no notable differences between these results. The 30th eigenfrequency is 3218 Hz and is already 1.5 times higher than the analysed frequency domain. Consequently, the conclusion can be drawn that eigenmodes higher than the 1000th eigenmode have a non-negligible influence. This problem could be solved by two different methods. On the one hand, it is possible to consider more than 1000 different eigenmodes and on the other hand, it is possible to consider only the eigenmodes which have an important share of the entire vibration. These modes can be found with the help of the approach in [7] by modal participation factors. Unfortunately, the model of the investigated engine in the current paper possesses approximately 2.7 million degrees of freedom and consequently just as many eigenmodes. Due to the required effort, both options are not very expedient for the presented virtual engineering approach of this paper.

As a result, the applicability of higher damping is investigated to improve the agreement between the results of the modally reduced and the non-reduced model. Hence, a comparison of the response functions of the different models is shown in Fig. 3. The solid black line shows the response function of the model without modal reduction of Fig. 2 and the solid grey line shows the response function of the modal reduced models of Fig. 2. It is obvious that these response functions agree very well in the frequency domain below 600 Hz, but above 600 Hz the amplitudes of the modal reduced model are in some cases up to 20 times higher than the amplitudes of the non-reduced one. For this reason, an unrealistically high modal damping is applied additionally to the modal reduced model in order to achieve better matching results (see Fig. 3 dashed green line). The conclusion that can be drawn from Fig. 3 is that the application of high modal damping seems to lead to a very good quantitative agreement and provides a useful opportunity to solve the problem. In contrast, the overall vibration behaviour of the engine shows a completely different situation. In Fig. 4 the results of the vibration analysis at 1698 Hz (see dashed grey line in Fig. 3) are shown for the configurations of Fig. 3. It becomes clear that the additional high modal damping increases the agreement between reduced and non-reduced model, but it is still not sufficient. The shape of the vibration modes as well as the corresponding amplitudes display non-negligible differences. For this reason, adapted modal damping parameters are defined over the whole frequency range of interest. These adapted damping coefficients are determined by approximating the corresponding Rayleigh damping function of the non-reduced model. These damping factors are applied for a modal reduced model with consideration of 30 and 100 eigenmodes as a test to ensure that the consideration of 30 eigenmodes is now absolutely sufficient. From both Fig. 3 and Fig. 4 it is obvious that the results of the reduced model with the adapted damping parameters match perfectly with the results of the non-reduced model.

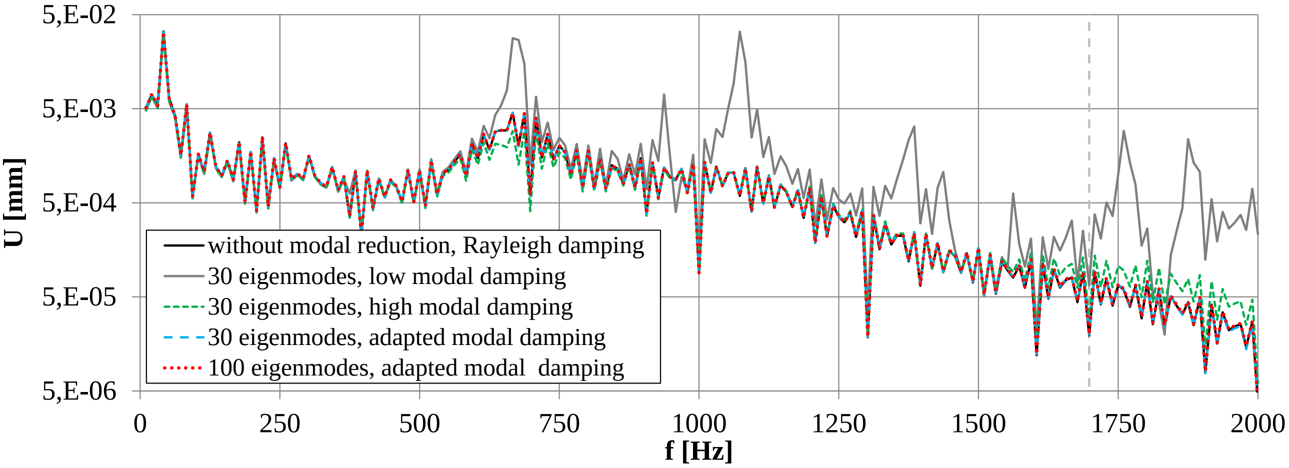

Figure 3. Response function of a point in the middle of the outer surface of the oil pan bottom without and with modally reduced FE-models with different damping factors

The results in Fig. 4 show that the observations of Fig. 2, which were that the reduced model only shows a similar behaviour from a qualitative point of view but leads to much higher amplitudes, are not exactly representative if adapted modal damping coefficients are used. A comparison of the required calculation time shows the large advantage of the modally reduced models in terms of their computational costs. The consideration of 30 eigenmodes in the vibration analysis needs 17 minutes of computational time, 1000 eigenmodes needs 8 hours and 28 minutes and a non-reduced model needs 2 days 3 hours and 36 minutes on the same computer under identical conditions. Therefore, a modal reduction is suggested for the vibration analysis, if adapted modal damping is used.

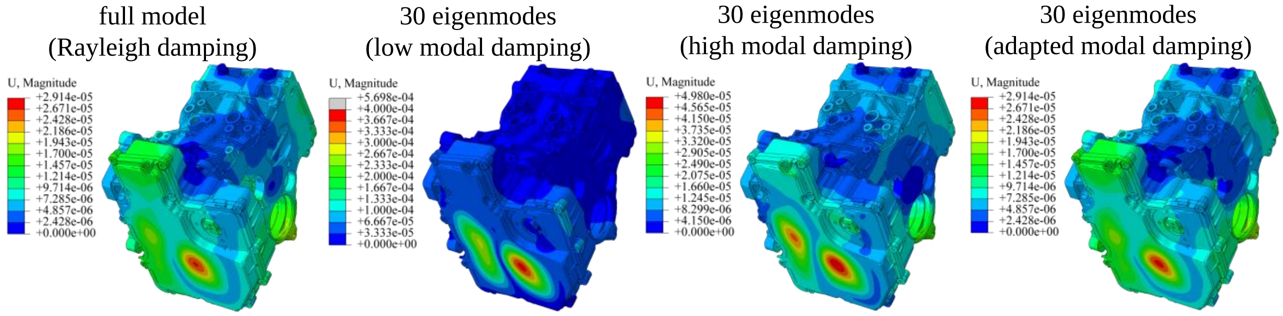

Figure 4. Results of the vibration analysis without and with modally reduced FE-models with different damping factors at 1698 Hz (see dashed grey line in Fig. 3)

4. Results

Up to now, the cylinder crankcase respectively the engine in an acoustic analysis is excited exclusively by the reaction forces in the main bearings of the crank shaft, which are caused by the crank drive motion (cf. Fig. 5a). Moreover, in general, measurements of these forces are used as input of the vibration analysis. That is why acoustic simulations typically are not possible early in the development process, because real prototypes are required to carry out the measurements of the bearing forces. Additionally, the build-up of the prototypes and the corresponding test stations and their equipment leads to an increase of the financial effort and is also time consuming. In contrast, the presented virtual engineering approach provides the possibility to analyse the acoustic behaviour of the engine early in the development process. Furthermore, the substitution of the measurements by simulations avoids costs. Moreover, due to the easy adjustable simulation model it is possible to investigate more design variations with just a marginal increase of the financial effort. In contrast, experimental design variations require all variants as prototypes, such causing significant increase of costs and time.

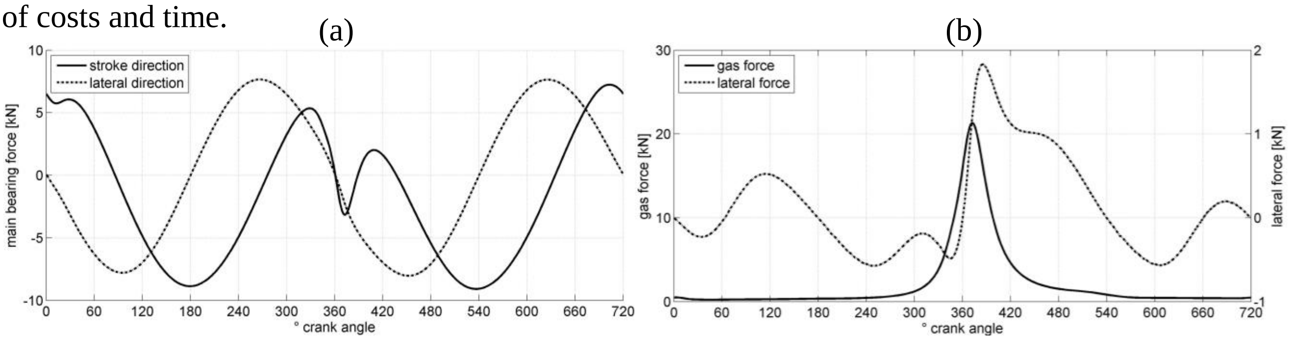

Figure 5. Forces for one working cycle: (a) main bearing reaction forces caused by the crank drive motion, (b) forces on the piston

Fig. 5b visualizes the forces, which are acting on the piston during one working cycle. The gas force results from the gas pressure and has its maximum briefly after the ignition-TDC (top dead centre), which occurs at a crank angle of 360°. The lateral forces arise by the contact of the piston and the cylinder. Dependent on the connection rod angle the lateral forces rises to its maximum of 1.8 kN at a crank angle of 387°, which occur after the maximum of the gas force. The lateral force is smaller than the reaction forces in the main bearings as shown in Fig. 5a, but its amplitudes achieve round 20-25% of the maximum value of the main bearing reaction forces. The peak of the impulse of the gas force during the ignition is even much higher than the maximum value of the main bearing reaction forces. Consequently, it is not reasonable to neglect these important excitations. Indeed, the lateral force is not measureable with the help of common test bench systems. Thus, it becomes clear, why the lateral force is up to now mostly not considered in acoustic simulations. As previously mentioned, measurements are generally used to receive the excitations of the engine. This emphasizes the necessity of the precedent MBS in the presented workflow, which is able to determine this important excitation source.

Fig. 6 visualizes the resulting sound pressure distribution on a plane cut through the centre of the spherical air volume, which is orthogonal to the crank shaft axis. The results of the acoustic simulation with and without consideration of the excitation of the cylinder walls are compared in Fig. 6 during a fixed revolution speed of 2500 rpm and the momentum load of 47 Nm. For this, the multiples of the half engine order up to 5 kHz are calculated.

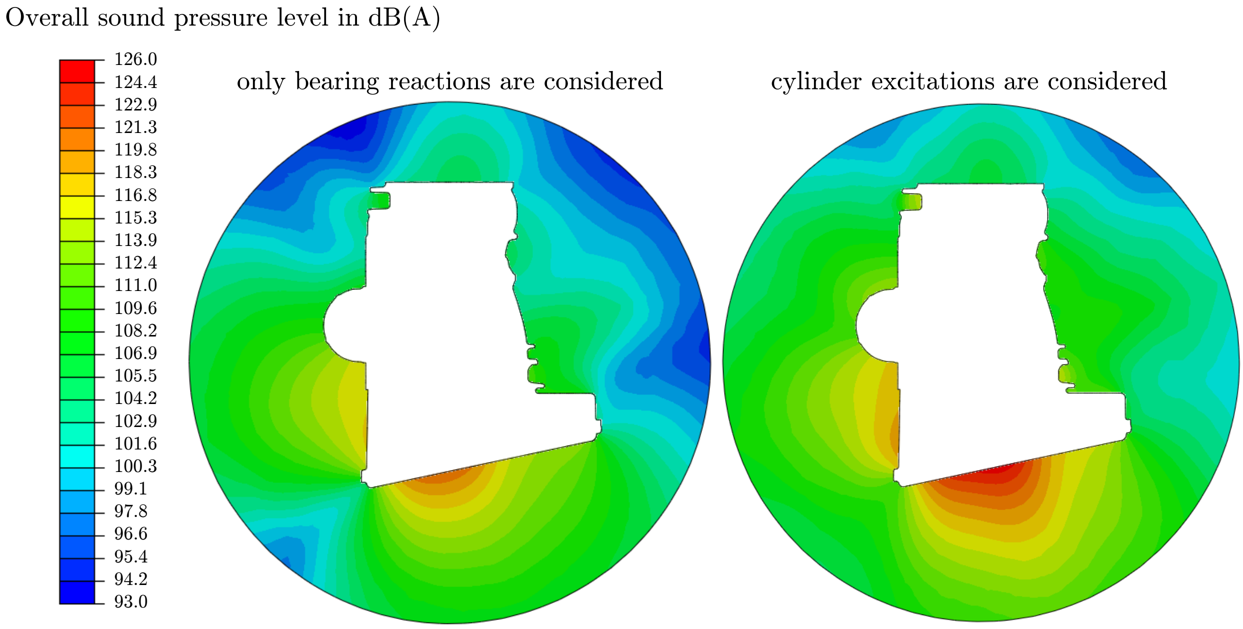

Figure 6. Sound pressure distribution without (left) and with (right) consideration of the cylinder forces

In Fig. 6 it is clearly visible that the additional consideration of the cylinder wall excitation has a big influence on the acoustic simulation. The oil pan is still the dominant radiator, but the sound pressure distributions differ with respect to the spatial distribution and the amplitudes. This proves that the resulting forces of the lateral and tilting piston motion and also the resulting forces of the combustion process are definitely non-negligible as excitations. They have to be considered in the vibration analysis as well as in the acoustic analysis. The presented holistic virtual engineering approach provides this opportunity for the acoustic analysis of combustion engines.

As was expected, it is clear that the cylinder crankcase isn't a good radiator from an acoustical point of view, because of its complex geometry with only small plane areas and its large wall thicknesses. Therefore, amplified vibrations of the cylinder crankcase are less important for the noise emission. In contrast, the additional energy, which is introduced at the cylinder walls, is transmitted by structure-borne noise paths in regions of higher radiation factors and increases significantly the sound radiation in these regions of the structure. The oil pan is one example of such a critical region. The results presented in Fig. 6 show that the increase of the lateral sound radiation of the cylinder crankcase by the consideration of the cylinder forces is less critical than the resulting increase of the sound radiation of the oil pan bottom.

The main point that should be taken from our investigation is not the fact that the lateral sound radiation of the cylinder crankcase increases by the additional consideration of forces acting on the cylinder walls, but that these forces have to be included in the numerical model as an important additional excitation source.

5. Conclusion

In this paper a holistic virtual engineering approach is presented, which is able to substitute the experimental determination of the input of the vibration analysis on the one hand and to consider the excitation of the cylinder walls resulting from the piston motion and the combustion process on the other hand. Moreover, it was observed that an application of a modal reduction in the vibration analysis can be recommended, if suitable damping parameters are used. The cylinder gas pressure distribution of the investigated combustion engine is the only required input of the presented virtual engineering approach.

With the help of simulation results it is shown that the via MBS calculated excitations of the cylinder walls have to be taken into account for the analysis of the acoustic behaviour, because the conventional consideration of the bearing forces as the only excitation of vibrations is not sufficient.

Additionally, acoustic consequences, which are caused by modifications of the crank drive, can be calculated and evaluated by the presented approach. Potential influence parameters for improving the engine acoustics as the bearing geometry, the piston fine geometry, the pin-offset of the piston, the piston skirt stiffness and the cylinder crank case geometry can be investigated in detail by the presented virtual engineering approach in future work. First investigations of the influence of the pin-offset of the piston were already carried out in [8]. Furthermore, the applicability of the presented approach to psychoacoustic problems was meanwhile also investigated in [9]. In future, automated optimizations will be executed based on the presented holistic simulation workflow.

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