Structure-borne Sound Propagation in the Crankshaft Drive

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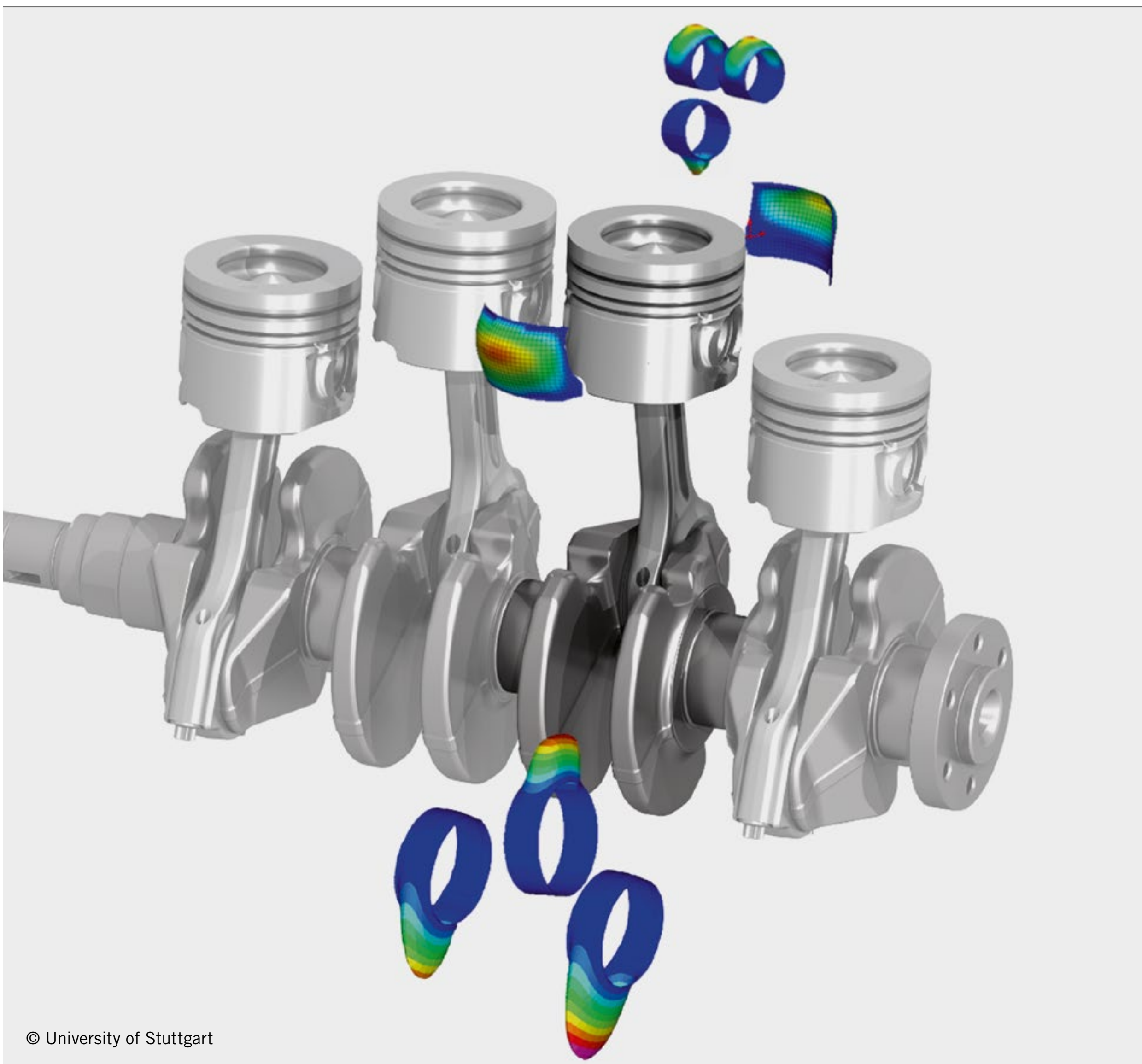
To improve existing simulation tools regarding the emitted airborne sound of internal combustion engines the emergence and the transmission of structure-borne sound inside the crank drive offers enormous potential. By using specifically designed in-situ measurement techniques, the Institute of Internal Combustion Engines and Automotive Engineering of the University of Stuttgart produced previously unavailable values which are used for an improved representation of dynamic processes inside the hydrodynamic bearings of a thermo-elasto hydrodynamic lubrication simulation model running at the Institute for Powertrain and Vehicle Technology, Machine Elements and Tribology of the University of Kassel.



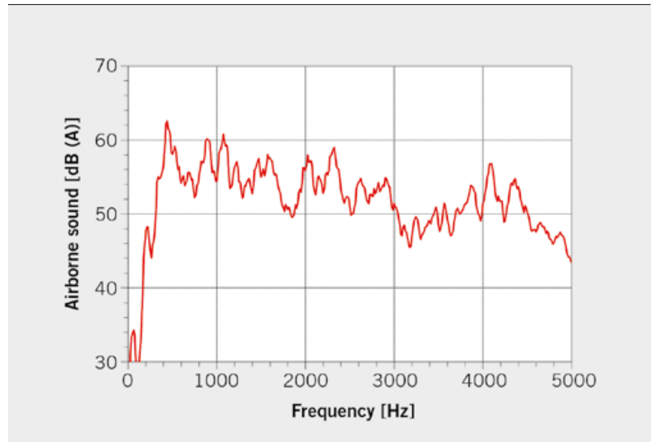


FIGURE 1 Airborne sound at 1 m distance on cold engine side (© University of Stuttgart)

1 MOTIVATION

Despite the growing popularity of Battery Electric Vehicles (BEV), Internal Combustion Engines (ICE) will be the propulsion unit for passenger cars over the next decades. To improve their Noise, Vibration and Harshness (NVH) behavior the used simulation tools have to represent all noise mechanisms inside the engine. In particular, the contribution of the inner transfer path of structure-borne sound, which causes a substantial share of the noise emitted, offers the biggest potential for improvements [1, 2].

High conversion rates of the combustion pressure act as a broadband excitation that causes natural vibrations within the coupled crank drive components piston, connecting rod (conrod) and crankshaft. Transmitted to the main bearings the crankcase and the oil pan represent the main emission surfaces of airborne sound. The contribution at 1 m distance to the used 1.5-l four-cylinder inline diesel engine with Direct Injection (DI) on the cold side is characterized by the rising levels between 3 and 4.5 kHz for the parts of the internal structure-borne sound conduction path, **FIGURE 1**. Further investigation shows that a natural vibration of the conrod in longitudinal direction is responsible for this [3, 4]. Identification of these vibrations in combination with an improved description of dynamic processes inside the hydrodynamic bearings during engine operation gives the opportunity to refine present simulation techniques and to develop new approaches to the NVH behavior of Multi-body Simulations (MBS).

2 MODAL ANALYSIS

At the beginning of the investigations, a modal analysis of all crank drive components and of an assembly of piston, piston pin, conrod, crankshaft and flywheel of the third cylinder in the Top Dead Center (TDC) position was carried out using Experimental Modal Analysis (EMA) and Finite Element Analysis (FEA). Using damping values from the EMA, the complete modal behavior can be embedded within the simulation model. The quality of the modal analysis is assessed by using the Modal Assurance Criteria (MAC) between EMA and FEA. For modes with global mode shapes all MAC values of the assembly are shown in **TABLE 1**. Mode number 10 represents the significant mode according to [2–4] with a resonance frequency of 4538 Hz and a MAC value of almost 1, **FIGURE 2**. The mode shape shows a longitudinal vibration of the conrod.

MAC comparison assembly from EMA to FEA		
Mode number	f_{FEA} [Hz]	MAC [-]
1	192	0.96
2	246	0.99
3	364	0.85
4	736	0.9
5	1486	0.89
6	1854	0.97
7	2730	0.87
8	3387	0.88
9	4241	0.81
10	4538	0.98

TABLE 1 Comparison between EMA and FEA in assembly configuration (© University of Stuttgart)

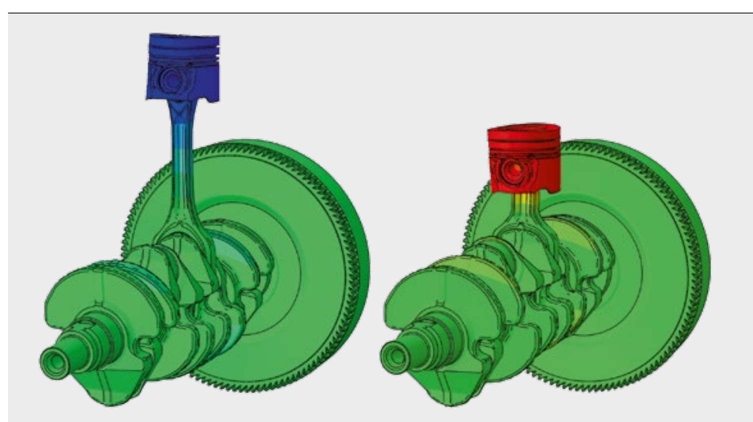


FIGURE 2 FEA result of the longitudinal natural vibration of the conrod in assembled conditions (© University of Kassel)

3 VIBRATION MEASUREMENT INSIDE THE OPERATING ENGINE

To capture the vibrations directly at the shaft of the conrod Semiconductor Strain Gauges (SSG) are used because of their minimal weight and excellent sensitivity. Four strain gauges were placed in longitudinal direction of the conrod shaft to allow a comparison with acceleration signals captured at the main bearing in stroke direction. Due to the higher temperature dependency of SSGs, the usage of specially developed temperature compensation was necessary. Through this developed compensation, which eliminates existing drift within the gas exchange loop, a frequency analysis of the high pressure part was possible without restriction. Signal transmission out of the engine was realized via a cable linkage which was attached at the big end of the conrod and the crankcase. A cylinder pressure indication system completes the metrological presentation of the inner transfer path inside the engine. By varying the injection timing under constant operating conditions, and the mechanical weakening of the connecting rod shaft, a validation of the simulation model with regard to excitation strength and connecting rod stiffness has been achieved. A simulation of the full engine with and without cable linkage resulted in a limited influence of the strain signals at frequencies below 1600 Hz.

FIGURE 3 shows a wavelet analysis of the configurations described. Since wavelets are not subject to the time-frequency uncertainty principle a very high time resolution of the signals is possible. The upper row shows the cylinder pressure and pressure gradient with respect to time. The follwing three show the Morlet wavelet time frequency analysis of the cylinder pressure, the time-frequency analysis using Morlet wavelet of cylinder pressure, conrod strain and acceleration at the main bearing in the stroke direction. The first two columns correspond to the serial conrod with minimum (1.7 bar/°CA, base calibration) and maximum pressure gradient (7 bar/°CA). The right column shows the maximum pressure gradient with the mechanically weakened conrod. It can be seen that the vibrations at the conrod and at the main bearing depend directly on combustion time without having comparable frequency characteristics in comparison with the cylinder pressure. Comparing the minimum pressure gradient (left column) with the maximum pressure gradient (middle column) in serial configuration, it can be seen that both vibration levels increase rapidly with rising pressure gradients. The right column shows the vibration shift towards lower frequencies due to the decreasing stiffness of the conrod shaft. In particular the frequency levels around 4.5 kHz are affected.

4 SIMULATION MODEL

To analyze the elastic and lubricated component behavior of the test engine, an Elasto-hydrodynamic Lubrication Multi-body System (EHL/MBS) simulation model of the complete engine was built.

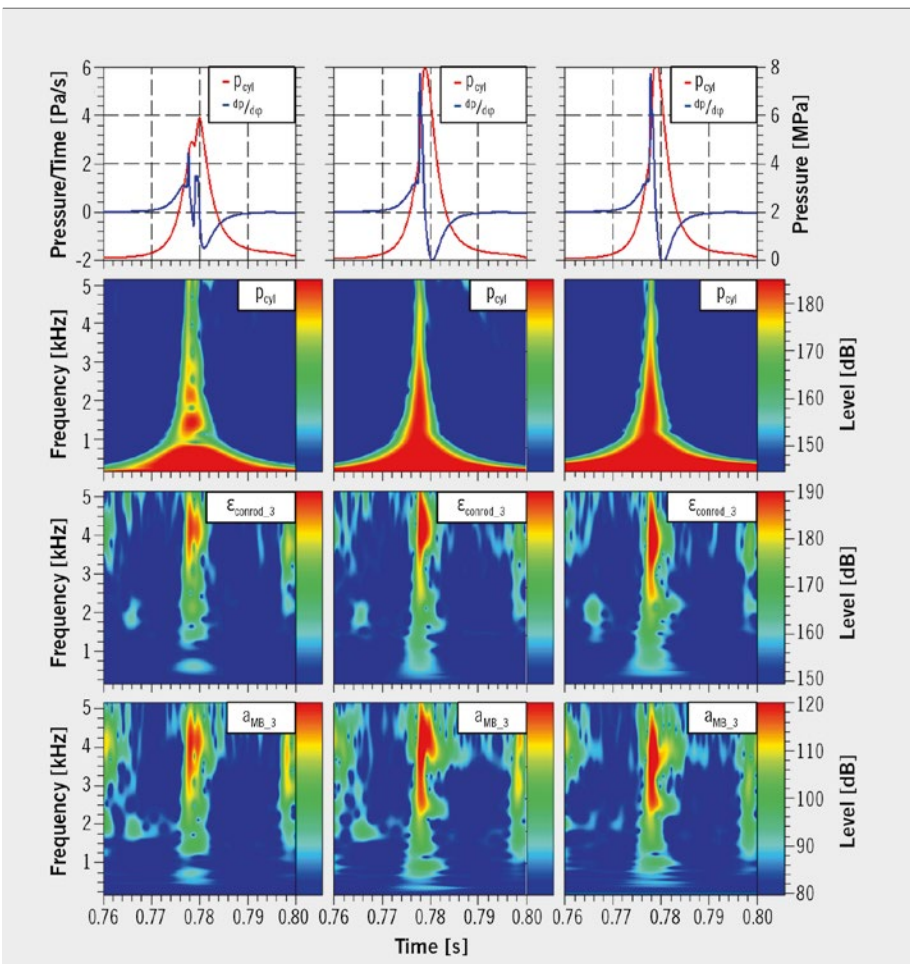


FIGURE 3 Wavelet analysis of cylinder pressure (p_{cyl}), conrod elongation (second derivate of time (ε_{conrod_3}), and acceleration at main bearing in stroke direction (a_{MB_3}) at 1500 rpm and 7.2 bar indicated mean effective pressure (left: serial conrod and pressure gradient, middle: serial conrod and maximum pressure gradient; right: weakened conrod and maximum pressure gradient) (© University of Stuttgart)

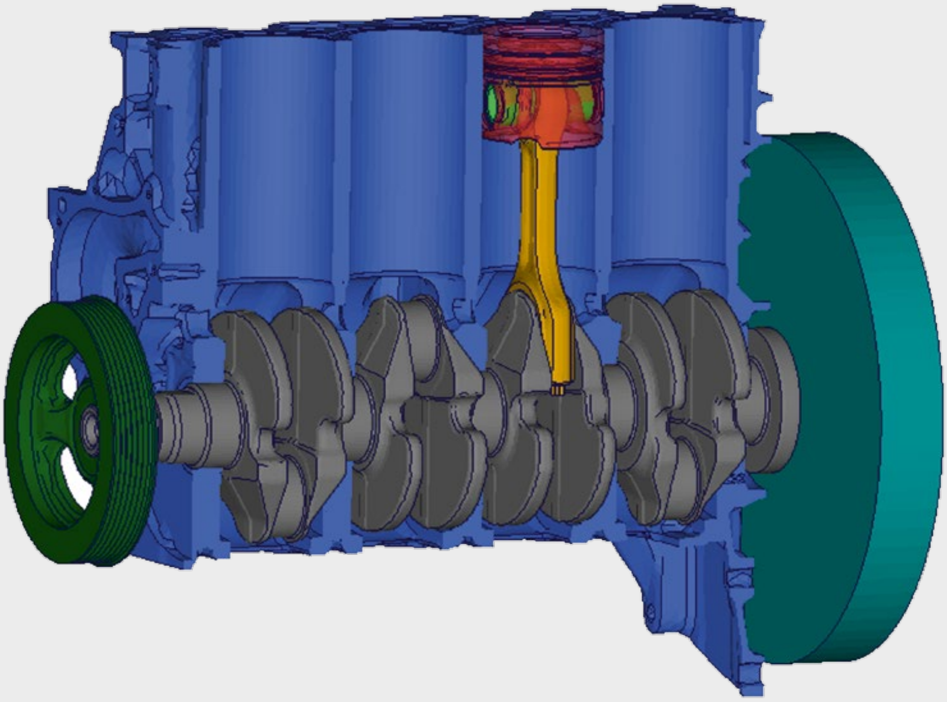


FIGURE 4 Simulation model of the 1.5-l Euro 6 diesel engine (© University of Kassel)

Based on the integration of Newton's equation of motion in the time domain, the dynamics of the components can be calculated by using the finite element method. Furthermore, it is possible to investigate nonlinear effects that occur in lubricated contacts. To get acceptable calculation times, a reduction of the degrees of freedom of all elastic crank drive components was necessary. The reduction was executed using the Craig-Bampton method, in which all important deformation properties of the structures are considered. **FIGURE 4** shows the simulation model, in which the crankcase and the crankshaft as well as all components of the third cylinder are modeled as elastic structures. To keep the calculation time as low as possible, pistons, piston pins and conrods of the other cylinders are considered by using an analytical crank drive (not displayed).

To enable a more accurate design of transient tribological systems, the new Thermo-elasto-hydrodynamic Lubrication (TEHL)-functionality was used [5]. The thermal effects of a tribological contact are integrated into the simulation model. Since the temperature-induced change of the oil viscosity and the gap width strongly

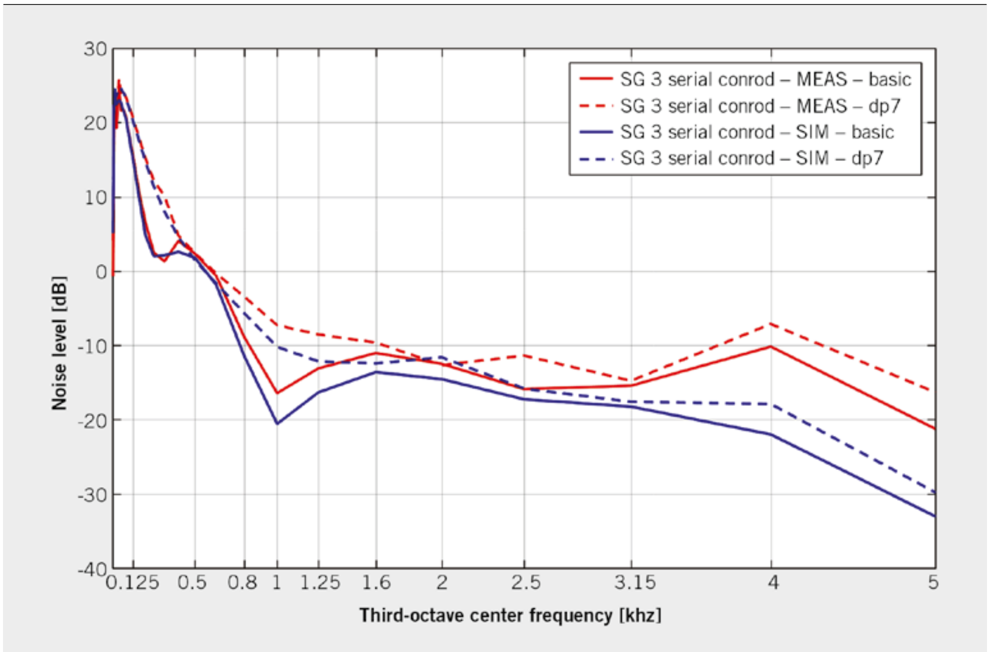


FIGURE 5 Comparison of measurement and simulation for two different combustion excitations and serial conrod (© University of Stuttgart)

interacts with the hydrodynamic pressure, the friction power and the fluid velocities, the EHD simulation model was extended to a TEHD model to represent the correct tribological state. The temperature distribution in the lubricant film within the Reynolds equation, which is supplemented by the so-called Dawson integrals, is also taken into account as well as the thermal deformation of the elastic FE structures.

5 CORRELATION BETWEEN SIMULATION AND MEASUREMENT

In order to ensure an exact comparison with the measurement signals, the four SSGs at the conrod shank are mapped in the simulation model and the elongation calculated. **FIGURE 5** shows the third-octave spectra for the serial conrod of the measured values in red, the simulation results for serial conrod in blue, the base calibration in solid and the maximum pressure gradient in dashed lines. Slight deviations below 1.6 kHz are due to the fact that the linkage system is not shown in the simulation. Above 1.6 kHz the influence is negligible, which can be seen in the following similar process between the measurement and simulation up to 3.15 kHz. Above 4 kHz, the deviation increases with continuing qualitative characteristic.

6 SUMMARY

The investigations carried out show that combustion-induced natural vibrations in the crank drive, in particular in the conrod, play a significant role in the indirect combustion noise and thus in the overall engine noise. A complex MBS model was created for the numerical calculation and validated by modal comparison with the measurement data from the EMA. The use of HL strain gauges on the conrod shaft delivered the hoped-for gain in signal bandwidth and enabled optimum recording of the natural oscillations that occur in engine operation and thus a suitable validation variable for transient simulation. The TEHL methodology has shown that the consideration of all thermal effects and their interactions, which occur in the mixed friction range of tribological contacts, leads to a better imaging accuracy of the system.

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