

In-Situ Measurements of the Piston and Connecting Rod Dynamics Correlated with TEHL-Simulation Techniques

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ABSTRACT

High combustion pressure in combination with high pressure gradient, as they e.g. can be evoked by high efficient combustion systems and e.g. by alternative fuels, acts as broadband excitation force which stimulates natural vibrations of piston, connecting rod and crankshaft during engine operation. Starting from the combustion chamber the assembly of piston, connecting rod and crankshaft and the main bearings represent the system of internal vibration transfer. To generate exact input and validation values for simulation models of structural dynamic and elasto-hydrodynamic coupled multi-body systems, experimental investigations are done. These are carried out on a 1.5-l inline four cylinder Euro 6 Diesel engine. The modal behaviour of the system was examined in detail in simulation and test as a basis for the investigations.

In an anechoic test bench airborne and structure-borne noises and combustion pressure are measured to identify the engine's vibrational behaviour. To understand the behaviour of the connecting rod as the key component in more detail its elongation, using semiconductor strain gauges at the connecting rod shank and a linkage system, is also measured. Furthermore temperature measurements of piston and liner under fired conditions, allow the determination of their warm contours for a better geometrical description. Variation of injection timing and connecting rod stiffness allows the basic adjustment of the simulation model. In a further step the simulation model is calibrated by the results of the measurement of the qualitative oil content in the gap between the liner and the piston. For this purpose a combined gap measurement system consisting of eddy-current and capacitive type sensors was specially developed. To adapt sensor and liner surface these are honed together after sensor installation. Finally the elasto-hydrodynamic simulation model is extended by a new software component to a so called thermo elasto-hydrodynamic coupled multi-body system. This leads, especially on Diesel engines, to an improved robustness of the simulation of NVH behaviour.

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INTRODUCTION

Upcoming passenger cars have to achieve the target of 95g CO₂/km in 2020 [1]. Therefore the actual development of DI-Diesel and DI-Gasoline engines will result in increasing combustion pressures, which acts as a broadband excitation force inside the combustion chamber. The transmission of the vibrations of piston and connecting rod via main bearings, crankcase and oil pan into airborne sound

represent the system of internal vibration transfer which is responsible for the majority of the structure-borne sound emitted [2], [3], [4], [5]. The consideration of those mechanisms in the digital development process of engines requires detailed information of dynamic contact forces and lubrication film characteristics of all crank train components to generate a convenient simulation model.

To achieve this objective experimental and simulative investigations are carried out on a 1.5-l inline four cylinder Euro 6 Diesel engine. First the generated structural and elasto-hydrodynamic coupled multi-body system simulation model is validated by experimental and simulative modal analysis of piston, connecting rod, crankshaft and an assembly configuration. To take the geometrical shape of piston and cylinder during operating conditions into account, their temperatures are measured to determine the heat influenced deformation. In further steps various measurement techniques are implemented in the test engine such as strain measurement at the connecting rod shank and a system for measuring oil content in the lubrication gap between piston and liner.

NOISE IDENTIFICATION IN THE INTERNAL TRANSFER PATH

For the following investigations the engine is operated in an anechoic test bench with the operating point at 1500rpm and 79Nm corresponding to 30% of the maximum torque at this engine speed.

The test engine is basically instrumented with a three-axis accelerometer at the third main bearing and with six free-field microphones around the engine. The characteristic engine noise is shown in [Figure 1](#). The measured airborne noise level on the right side of the engine and the acceleration in stroke direction at main bearing three shows a rising level around 4kHz.

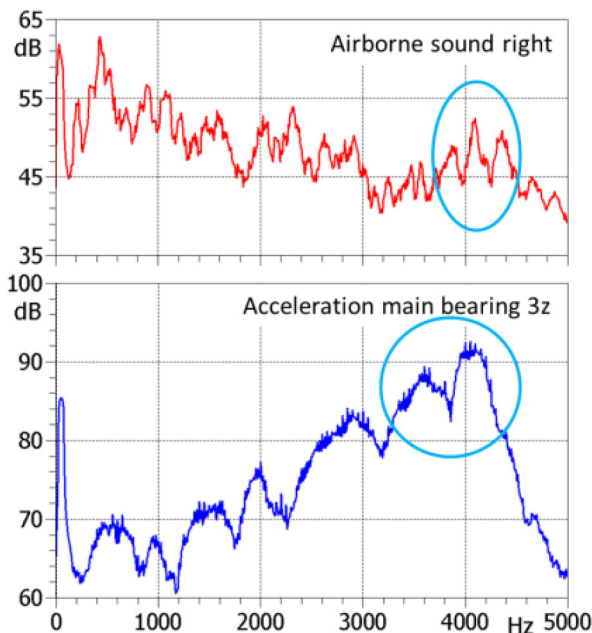


Figure 1. Spectrum of airborne sound right (top) and acceleration at main bearing in stroke direction (bottom)

Modal Analysis

To yield an optimized behaviour of the simulation model a modal analysis is required. Therefore the Finite Element Analysis (FEA) is correlated by an experimental modal analysis (EMA). Both analyses are carried out for the single components piston, connecting rod and crankshaft as well as for an assembly of piston, piston pin, connecting

rod, crankshaft and flywheel. The identified modes of this configuration, which represent the operating conditions inside the engine as well as possible, are shown in [Table 1](#).

Table 1. Identified modes with Eigen frequencies and MAC-Value of the assembly configuration

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

In [Figure 2](#) the assigned mode of the FEA analysis of the assembly is shown. The mode shape is comparable to a longitudinal vibration of the connecting rod, as it is also identified in [\[5\]](#), [\[6\]](#).

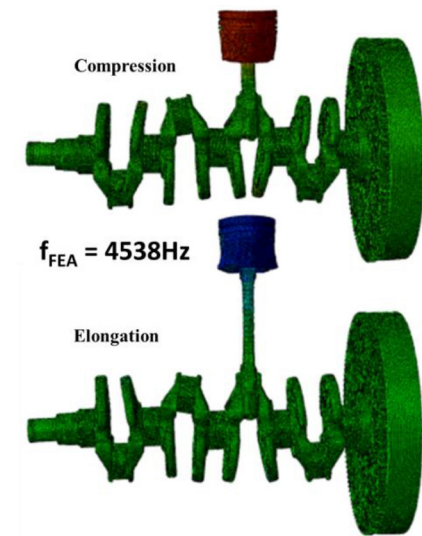


Figure 2. Natural longitudinal vibration of connecting rod as a result from the FEA modal analysis in assembly configuration

In-Situ Strain Measurement at Connecting Rod

Measurement Technique and Implementation

To capture the behaviour appropriate to [Figure 2](#) the connecting rod of cylinder 3 is implemented with four semiconductor strain gauges. The signal transmission is performed by a cable linkage system which is adapted at the big end of the connecting rod. The technical specification of the semiconductor strain gauges are shown in [Table 2](#).

Table 2. Technical specifications of the used semiconductor strain gauges

manufacturer	name	type	R_0 [Ω]	k	grid [mm]	base [mm]
KYOWA	KSP-1-350-E4	p-Si	350	164	1x0.25	6.6x4

The analysed frequency range is up to 5kHz and thus 50kHz carrier frequency amplifiers are used, which specifications are shown in [Table 3](#).

Table 3. Technical specifications of the used carrier frequency amplifier

manufacturer	name	f_{carrier} [kHz]	U_B [V]	f_{signal} [kHz]	$\varepsilon_{\text{Max}}(k=164)$ [$\mu\text{m}/\text{m}$]	linearity [%]
HBM	KWS 3085	50	5	10	± 1220	<0.1

The strain gauges are connected in a Wheatstone half bridge circuit whereby one resistor is replaced with active temperature drift compensation, so no signal overload can occur. Figure 3 shows the application design on the left side and the positions at four different positions on the right side. The measuring grid of each strain gauge is exact at the centre line of the connecting rod and the measuring direction is longitudinal.

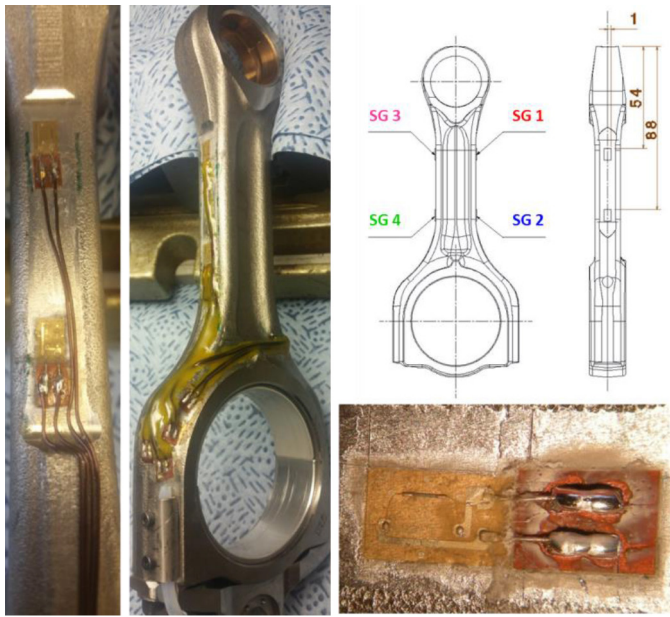


Figure 3. Positions (top right) and application design of all four semiconductor strain gauges

Signal Analysis and Results

Figure 4 shows the time signals of all four strain gauges and the cylinder pressure averaged over 375 four-stroke cycles as a function of crank angle. It is obvious that the strain follows the cylinder pressure during combustion due to high loads. Beyond the combustion a bending of the connecting rod is evident, which can be assigned to the linkage influence by comparing the simulation results with and without linkage. Figure 5. For this reason and consideration of the calculation time all described EHL and TEHL simulation models are without linkage.

Measurements undertaken include variations of the injection timing to generate different pressure gradients influence the shape and frequency content of the excitation force at identical engine torque. As depicted in Figure 6 the maximum value of the pressure gradient is incremented with 1 bar/°CA in the range from 1.72 up to 7bar/°CA. This corresponds to a maximum cylinder pressure between 58 and 80bar. It can be shown, that the two variations with the lowest gradients reach higher gradients due to pre-injection than during main

injection and that the locations are very similar to those of 6 or 7bar/°CA. So the geometrical positions of the crank drive components are also very similar and therefore well comparable.

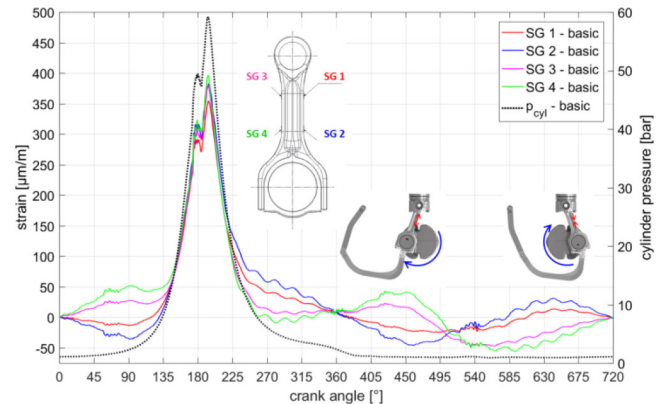


Figure 4. The four strain signals within four-stroke cycle show the influence of the cable linkage beyond the combustion sector

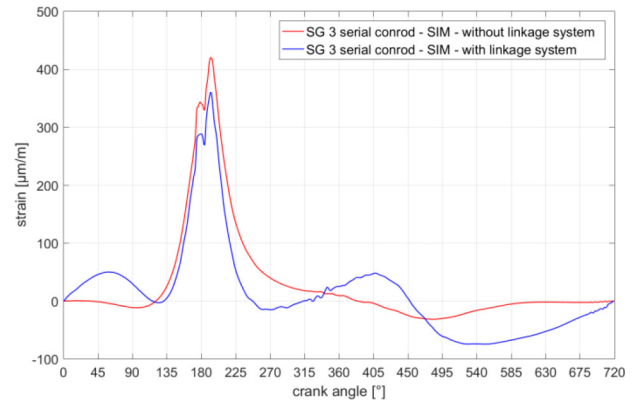


Figure 5. Simulation results of connecting rod strain without (red) and with (blue) linkage system

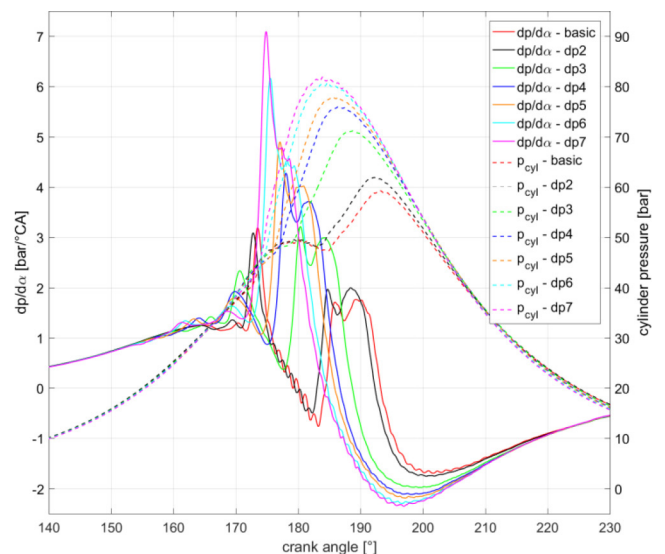


Figure 6. Pressure gradient variations (left axis) and resulting cylinder pressure curves (right axis) at 79Nm torque