Multi-Domain Simulation for the Assessment of the NVH Behaviour of a Tractor with Hydrostatic-Mechanical Power Split Transmission

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Abstract

This paper presents a methodology for the model-based assessment of the NVH behaviour of a tractor with hydrostatic mechanical power split transmission (PST) within the product development process. The aim of the simulation methodology is the assessment of structure-borne and airborne noise caused by transmission excitations. Due to numerous interactions and couplings within the system, a multi-domain overall system model is proposed. The model combines the aspects of hydrostatic force excitation, structural dynamics and subsequent acoustic response. In this paper, the main components of the simulation model are introduced and its application is demonstrated for a given series tractor with PST. Simulation results are shown and compared with measurement results. The results are discussed and an outlook on future work is given.

1. Introduction

Hydrostatic mechanical power split transmissions (PST) are widely used in tractor drive trains [1]. They combine the customer requirements of a high efficiency, realised by the mechanical transmission path, and a continuous variation of ratio without interruption of traction force, which is realised based on the hydrostatic transmission path. However, customer requirements regarding comfort and Noise, Vibration and Harshness (NVH) behaviour have increased in recent years and have become an important development goal today [2, 3]. Due to high energy density and discontinuous operation of the axial piston units inside the hydrostatic transmission path, PST show strong, tonal excitations. These excitations are transferred from the hydrostatic units via multiple structure-borne as well as airborne transfer paths to the driver's ear inside the cabin. The airborne noise resulting from the tonal characteristic of the excitations in an acoustically relevant frequency range is perceived as particularly unpleasant for the driver and leads to less driving comfort [4].

Furthermore, decreasing development times necessitate methods that help to reduce the probability of acoustic abnormalities in earlier stages of the product development process. Virtual product development methods offer the chance to assess the physical behaviour of the system already prior to prototyping stages [5]. The assessment requires validated modelling methods. As the NVH behaviour depends on the overall system characteristics, validated multi-domain simulation models on system level are required. Examples for multi-domain models, which allow an overall view including excitation, noise transmission and sound radiation up to higher frequency ranges (2-5 kHz) can be found in the automotive sector. For example, a validated methodology for a multi-domain model of an electrified car is described in [6, 7].

Based on the aforementioned methodology from the automotive sector, a validated simulation methodology for describing dominant NVH phenomena for tractors with PST is to be developed. In addition to the known methods, new modelling methods are needed to take the specific characteristics of the tractor into account. This includes especially the different main excitation mechanism as well as fundamental differences in the vehicle structure, such as the cabin suspension system. In addition, due to numerous large and complex components in the tractor drive train and housing structure, a compromise between level of detail and accuracy is to be found in order to create a computationally efficient model.

The derived methodology is based on a combination of the domains of hydraulics, structural dynamics and acoustics. This paper presents the modelling setup and the main components of the developed multi-domain simulation method for a given series tractor with PST. On the basis of a measurement-based system characterisation of the tractor under study, the relevant components for the simulation model are described. In the end, simulation results are shown and compared with measurement results and an outlook on the following steps is given.

2. System characterisation

The modelling method is developed and validated for a given series standard tractor of medium power class with a hydrostatic mechanical PST. The hydrostatic transmission part of the PST consists of two axial piston units that are connected to a closed hydrostatic system. The rotational speeds of the hydrostatic units depend on the specified ratio and on the interconnection structure of the PST drive train.

In a first step, a measurement based system characterisation of the tractor is performed to determine relevant physical effects and relevant transfer paths which have to be taken into account for the simulation model. In order to perform measurements under reproducible conditions, the tests were conducted on a chassis dynamometer in a semi-anechoic chamber, see Fig. 1 left. The measurements were carried out under full load conditions at constant input speed of the combustion engine, while the output speed is varied along linear speed ramps. Fig. 1 right shows the A-weighted sound pressure level (SPL) at the left driver's ear over the ratio, which is proportional to the driving speed.

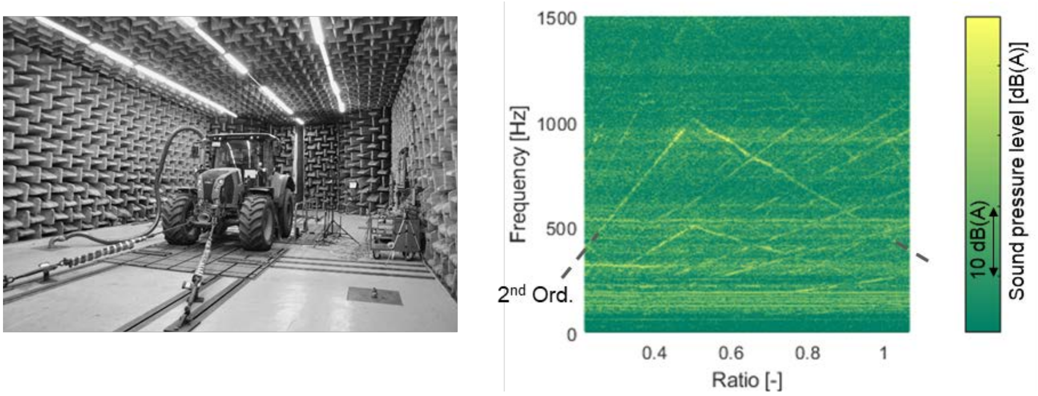

Fig. 1: Test bench in semi anechoic chamber (left), SPL at driver's ear for run up (right)

Under full load condition of the drive train, the hydrostatic units of the PST are the dominating excitation sources. Especially the second piston order, which is marked in Fig. 1 right, is dominating the overall SPL in a wide speed range. In the middle of the forward driving range there is a shifting point, in which two clutches are actuated. At that point, the interconnection structure of the PST is changing. This results in an inversion of the speed ramp characteristics of the hydrostatic units over the ratio, which is visible in the rising and falling frequency characteristics of the hydrostatic orders. The spectrogram also shows resonant behaviour especially in the frequency range up to 1 kHz, which indicates a high relevance of the noise transmission paths and the structural dynamics for the acoustical behaviour.

The sound pressure at the driver's ear is a result of the superposition of excitation, noise transmission (transfer paths) and sound radiation, Fig. 2.

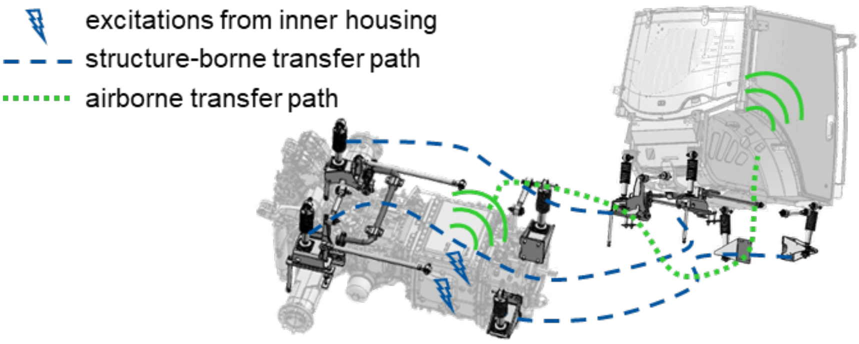

Fig. 2: Noise propagation from hydrostatic excitations to the driver's ear

The hydrostatic units of the PST are mounted in a separate inner housing, that itself is connected to main drivetrain housing via rubber bushings. Structure-borne noise emitted by the hydrostatic units excites the drivetrain housing through these bushings and finally the cabin frame through the cabin suspension. Surfaces in the cabin, for example windows and floor, radiate airborne noise. The gearbox housing itself also radiates airborne sound, which enters the cabin via primary and secondary airborne noise paths, see [8]. For analysing the relevance of the different possible transfer paths into the cabin, an operational transfer path analysis (OTPA) is performed on the tractor. Details are presented in [9]. The results of the OTPA show high contributions of airborne and structure-borne induced noise to the total SPL at the driver's ears. This shows the relevance of both, airborne and structure-borne transfer paths into the cabin, which has to be taken into account for the modelling method.

3. Modelling method

In order to evaluate optimisation measures holistically and to determine the most acoustically and cost-effective measure along the sound transmission chain, a system model is needed that considers the entire machine-acoustic chain. The test bench measurements show the relevance of the operating point-dependent and coupled analysis of the hydrostatic excitations, structural dynamics and acoustics. These domains are combined in a modelling setup that is shown in Fig. 3.

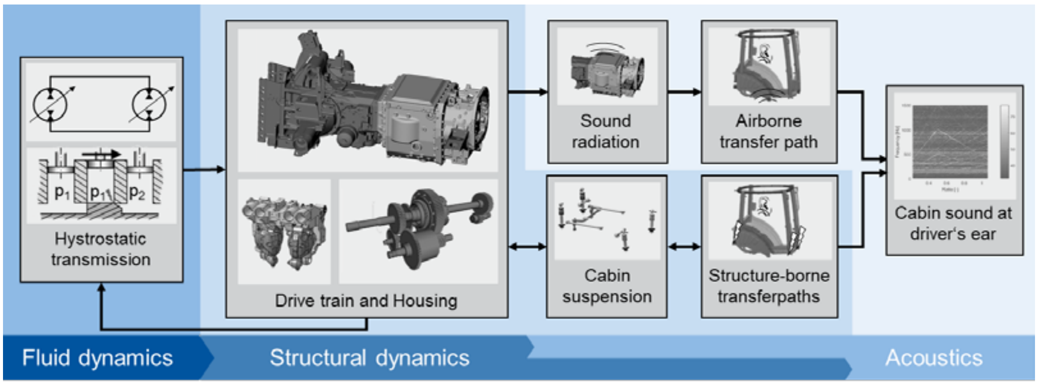

Fig. 3: Multi-domain system model for simulation of structure-borne and airborne noise

The excitation forces of the hydrostatic units are caused by pressure pulsations in the cylinder chambers, especially the so-called compression-induced pressure pulsations [10]. These pulsations result from highly dynamic volume flows caused by pressure equalisation processes between the cylinder chamber and the oil channel after a change of the connected channel [11]. The fluid oscillations can be considered as one-dimensional up to very high frequency ranges (>10 kHz) due to the specified oil channel and cylinder chamber dimensions [12]. Therefore, a one-dimensional fluid simulation is used to calculate the pressure pulsations. Since the pulsations in the piston chambers are also dependent on the connected pipe system, the model also includes the connecting oil channels in addition to the two hydrostatic units with their separate cylinder chambers also.

The pressure pulsations acting on the pistons and chamber walls cause force excitation to the structure. The excitation forces are calculated from the pressure pulsations and transferred via maps into the structural model, where they are transmitted to the structure depending on the operating point. The structural model contains all parts of the inner and outer housings, shafts, gears, bearings and bushings. For efficient calculation of transient processes regarding nonlinearities, the structure is modelled in the elastic multibody simulation (EMBS). All structural components are modelled as flexible bodies in several substructures for the relevant frequency range. The substructures are calculated using the Craig-Bampton reduction in the finite element method (FEM) [13].

As the measurement based system characterisation shows the relevance of the airborne and the structure-borne transfer paths into the cabin, both are taken into account for the model. In order to create a computationally efficient overall model, the cabin is not modelled in a full physical way. Instead, the cabin characteristics are taken into account by measured transfer functions. The transfer functions describe the frequency-dependent transmission behaviour as the ratio of the sound pressure at the driver's ear to the exciting airborne or structure-borne noise. The measured structure-borne transfer paths are coupled directly with the calculated structure-borne noise from of the EMBS model. To include the measured airborne transfer paths, the radiated airborne sound from the outer gearbox housing has to be calculated. Therefore, the calculated vibration velocities of the surfaces are used as an input for the calculation of the noise radiation.

4. Results and Validation

Due to the high system complexity, a bottom-up approach is used for the modelling process.

This includes modelling and parallel experimental validation, starting with the individual structural components up to the overall system. As a result, a high model quality can be ensured despite high complexity [14, 15]. As described above, the structural components are modelled as flexible bodies. The validation for the structural dynamic behaviour is done by comparing the results from experimental and numerical modal analysis using an eigenfrequency and eigenmode comparison. The mode shapes of components are compared using the modal assurance criterion (MAC) [16]. The results for the main gearbox housing assembly is shown in Fig. 4.

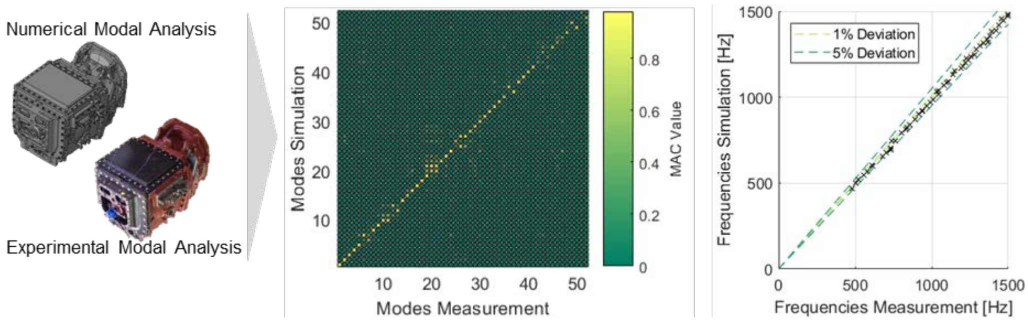

Fig. 4: Comparison of the results of experimental and numerical modal analysis for the assembly of the main gearbox housing

In the frequency range up to 1500 Hz, the gearbox housing shows about 50 eigenmodes. The MAC matrix shows high values on the main diagonal and low on the secondary diagonals, indicating a high degree of conformity of the mode shapes between measurement and simulation. The maximum deviation of the eigenfrequencies is less than 5%, which also shows a high correlation.

The top cover of the gearbox housing is highly relevant for the NVH behaviour due to its large radiating surface and its location directly underneath the cabin floor. Fig. 5 shows the measured and simulated A-weighted structure-borne noise level in vertical direction of a measurement point on the gearbox cover over a wide speed range.

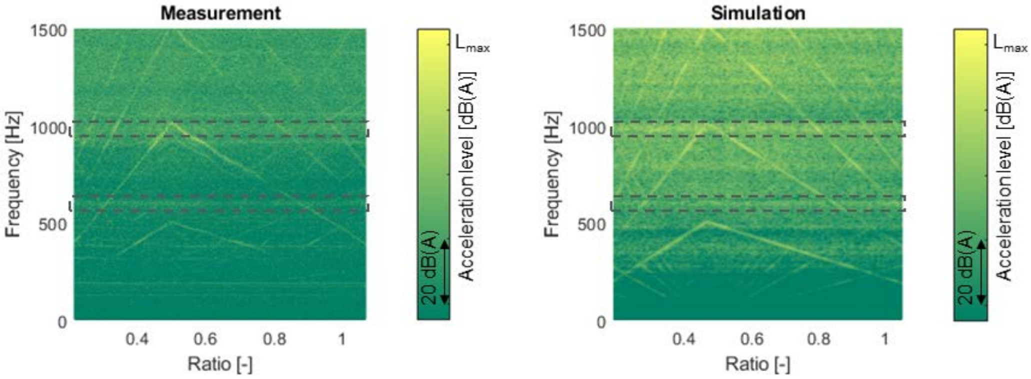

Fig. 5: Measured and simulated acceleration level in dB(A) at the top of the gearbox cover

The results of the measurement and simulation show qualitative a high degree of conformity: In both spectrograms, the excitation orders of the hydrostatic units are clearly visible. At about 600 and 1000 Hz two eigenmodes of the gearbox cover can be identified, which are marked in Fig. 5. The coincidence of excitation and eigenfrequency causes increasing acceleration levels due to resonance effects, which is also visible in both diagrams. Though, the spectrograms show small deviations in the quantitative acceleration levels between measurement and simulation. A reason for this could be neglected damping effects in the simulation model, for example caused by the oil inside of the gearbox housing. The modelling of damping effects with higher accuracy indicates improvement potential for the method.

Overall, the validation of the structural behaviour and the excitations shows good results. The modelling method can be used to identify and optimise abnormalities in the NVH behaviour even before measurements are available.

5. Conclusions

In this paper, a methodology for a holistically examination of the NVH behaviour for tractors with the focus on the excitations caused by the hydrostatic transmission part of PST is presented. On the basis of a measurement based system characterisation, a model chain was built up, which covers the relevant system elements. The individual domains for calculation of excitation forces, structural dynamics and acoustics were briefly explained and first simulation results were compared with measurement results.

Further work is aimed primarily at the development of an efficient method for calculating the radiated airborne sound of the gearbox housing in the shown model chain. For this purpose, spatially highly resolved surface velocities of the housing structure are to be combined with previously calculated acoustic transfer vectors (ATV), which describe the radiation behaviour, also see [17]. Subsequently, the total sound at the driver's ear is to be calculated and the validity of the method is to be assessed.

6. Literature

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