Multi physical domain simulation of a NVH reduction system for a generator-electric vehicle

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ABSTRACT
The limited range and high costs of battery-electric vehicles currently slow down their market impact. A concept for a new low emission generator-electric vehicle (GEV) has been developed and realized that combines transverse flux electric drives and a highly efficient power generator. The power supply works continuously and much more efficient than in traditional power generation industries, delivering unmatched driving distances and carbon footprint. The modified drive causes a significant change of the NVH behavior of the vehicle. In this paper passive and active measures to increase comfort and reduce airborne as well as structure borne noise are considered. A systematic approach to model, simulate and assess multi physical domain systems is presented. In system simulation, the excitation, mechanical structures, sensors, signal processing and actuator systems are included. The interaction between mechanical, electrical and acoustic domain is taken into account and the performance of vibration reduction measures is evaluated. Diverse modeling approaches, such as order reduced finite element modeling, system identification and analytical description are combined. The model of the generator-electric vehicle is tested and simulation results are presented.

Keywords: system simulation, hybrid drive, vibration reduction

1. VEHICLE CONCEPT
Hybrid powertrain concepts have a difficult time due to matching specific challenges in terms of costs, packaging and noise requirements. The existing approaches do not pursue the "hybrid idea" consistently because at least one of both drive components on its own can act as a complete powertrain. The range extender of the BMW i3 has no effect on the size and the weight of the battery system what makes it expendable for driving actions. Leaving the Opel Ampera with a completely discharged traction battery results in driving the car as a conventional vehicle with internal combustion engine what finally causes a high retail price. Therefore, in order to drive with (local) zero-emission, the current concepts of partially- or fully-electric cars have the disadvantages of high costs due to the complex technology and high weight due to additional components.

The concept presented in this paper brings significant advantages over currently available hybrid vehicles on the subject’s weight and costs. The new Generator-Electric-Vehicle is set up featuring a 1-cylinder CNG engine in combination with a transversal-flux generator thus, providing energy for electric traction and constantly procures the power to enable the demands on average speed and acceleration. A small battery system with intrinsic safe lithium-iron-phosphate round cells buffers increased power requirements at demanding acceleration procedures and at high speed. It is charged during recuperation or underperformed power demands.

By knowing the average power requirement of 18 kW to 22 kW for realistic driving cycles based on service load measurements at the Fraunhofer LBF research fleet, at the first development stage the...
power- and thereby cost-reduction of an internal combustion engine is realized by radically downsizing to a single cylinder. By the strong limitation of the rpm range and the shift to an optimal operating point, the improvement of the efficiency to about 35% was achieved. Internal gasoline combustion engine applications usually achieve a temporally averaged range below 25%. With discrete and direct coupling to the drive shaft, this steady-state operation would result in a limitation of driving demands to a very small power range. The next development step, therefore, is the decoupling of the internal combustion engine from the drive by constructive and functional realization of an electric traction. Furthermore, with the integration of a compact high-power battery with relatively low capacity, the engine runs completely independent of the existing demands on dynamic acceleration and speed.

In terms of weight and cost, this unique concept brings a decisive edge on current hybrid concepts but has new special requirements in terms of NVH issues.

2. IMPACT OF OPERATIONAL CONTROL STRATEGY ON NVH COMFORT

The vehicle’s drivetrain topology is based on a power unit (PU), which consists of a single cylinder internal combustion engine (ICE) coupled with a permanent-magnet synchronous machine (PSM), and two additional PSM traction drives for propulsion. In terms of energy efficiency, the PU operates at a favorable operation point of 5000 rpm, causing increased airborne as well as structure-borne noise. Activation of the PU takes place if one of two conditions is satisfied. The first condition occurs if the battery’s state of charge (SOC) falls below a value of 20%. In this case the PU starts charging the battery until a SOC of 80% is exceeded. The second condition for activation is based on the average power request of the traction drives. If the request exceeds an average value of 20 kW over a period of 60 s, the PU starts to supply energy to the high voltage (HV) circuit until a SOC of 80% is reached. This operation strategy has been proofed functional by numerical simulation and experimental testing for several driving cycles. Figure 1 and Figure 2 show actual, average and threshold values over time for the power request of the traction drives as well as SOC of the battery system and the supplied power of the PU during two different driving cycles. During the artificially designed New European Driving Cycle (NEDC), an activation of the PU takes place only once at the end of the drive cycle due to the average power transgression (Figure 1). Although for most of the time during NEDC the vehicle is operating in electric vehicle mode, a negative impact on the NVH comfort can be observed at the end of the cycle when the ICE is running at 5000 rpm to produce 20 kW of electrical power.

The LBF-traction cycle simulates a real world driving scenario. In this driving cycle the actual and average power request is far more often higher than the defined threshold value. Furthermore, the PU is activated during this cycle twice for approx. 1000 s each time as Figure 2 illustrates. This corresponds roughly to a third of the whole cycle duration. In this case it can be considered that in real world driving scenarios a negative impact on NVH comfort is predominant for a significant amount of

Figure 1 GEV|one operation strategy during NEDC

![Figure 1](image_url)
time. Thus, it is necessary to integrate vibration optimization measures for comfort improvement. The methodological model-based development of suitable measures is described in the next chapters.

3. MODEL-BASED SYSTEM DEVELOPMENT

An efficient modeling strategy frames the development process of vibration optimization measures in an essential manner. Using a model-based system development, different measures can be compared with respect to their capability. Once a promising concept is defined, passive or active measures, control concepts or other components can be designed and proven virtually. Therefore a system simulation model that can be adapted to the different development stages is of great benefit (1).

The consideration of active components, as actuators and sensors, strongly suggests the usage of an impedances-admittances structure of the system simulation model. The advantages are an easy exchangeability of single system components due to standardized interfaces and the possible combination of mechanical, electrical and hydraulic components (2, 3). Exemplarily Figure 3 illustrates the impedance-admittance formulation of a mechanical system, consisting of two bodies (e.g. car body and combustion engine) combined with a juncture element (e.g. engine mount).

Figure 2 GEV operation strategy during LBF traction cycle

Figure 3 Scheme of the impedance-admittance formulation of two bodies connected by a juncture element
While bodies are simulated as admittances with forces \((f_1, f_2)\) respectively) as input and velocities \((v_1, v_2)\) as output, junctures are modeled as impedances with relative velocities \((\Delta v)\) as input and forces \((f_1, f_2)\) as output. According to this system simulation strategy, models consisting of multiple bodies and junctures can be build up using mechanical, electrical, hydraulic or mixed components. Doing so, electrical components as actuator amplifiers, sensors or controllers can easily be connected to electromechanical transducers as e.g. electrodynamic actuators. Components themselves are represented at different detailing levels, thus defining a certain detailing level of the whole system (1).

Figure 4 gives an overview over the different detailing levels, the related modeling depth of the system and the obtained benefit during the particular stage of system development.

<table>
<thead>
<tr>
<th>level</th>
<th>modeling</th>
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<td>level 1</td>
<td>Idealized Components</td>
<td>Determination of promising concepts</td>
</tr>
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<td></td>
<td>E.g. ridged bodies, stiffness values</td>
<td>Determination of general parameters</td>
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<td>Comparison of system simulation model with measured data</td>
<td>Evaluation of optimization approaches for further development tasks</td>
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Figure 4 Detailing levels of the (electro-)mechanical system

At 1st level, idealized components as rigid bodies, simple stiffness and damping values or ideal actuators can be used in order to determine promising vibration optimization concepts. Once a concept is defined, the model can be used to derive first parameters or validate control strategies. At 2nd level, existing or set up finite element models (4) can be used to get a dynamic description of single components. By this virtual system assembling, concepts can numerically be proven considering elastic dynamic behavior. Possible unexpected effects that might occur in a hardware realization can be investigated. At 3rd level, single components are substituted by a model based on measured data. Thereby, its realistic dynamic behavior can be introduced and its interaction can be evaluated in system frame. At the 4th level, the system simulation model can finally be compared to the hardware realization in order to evaluate optimization approaches for further or following development tasks.

The different component models required at each detailing level can be substituted without any further changes in the system simulation model. Therefore relatively automated reduction procedures are defined. The exemplary derivation scheme of a reduced body that can be used in system simulation is shown in Figure 5.
In order to ensure an easy exchangeability of single components in the system simulation, the format of the reduced model has to be independent from the data used for its derivation. This is realized by the chosen admittance state-space formulation with the force vector $\vec{f}_{n}$ at $n$ defined coupling points $P_{n}$ as input and the velocity vector $\vec{q}_{P_{n}}$ as output (5).

The states of the state-space formulation of a rigid body

$$
\begin{bmatrix}
\dot{\vec{q}}_S \\
\ddot{\vec{q}}_S
\end{bmatrix} = \begin{bmatrix}
\vec{A} & \vec{B} \\
\vec{C} & \vec{D}
\end{bmatrix} \begin{bmatrix}
\vec{q}_S \\
\dot{\vec{q}}_S
\end{bmatrix} + \begin{bmatrix}
\vec{B} \\
\vec{D}
\end{bmatrix} [\vec{f}_{P_{n}}]
$$

(1)

are the displacement vector of the rigid body degrees of freedom $\vec{q}$ and its derivatives. $\vec{B}$ includes the inertia tensor and the transformation of the forces acting at the coupling points into the equivalent forces and moments acting at the center of gravity. The contrary transformation of the velocities is included in $\vec{C}$.

For the elastic model achieved by numerical and experimental results, the states of one possible state-space formulation

$$
\begin{bmatrix}
\dot{\vec{p}} \\
\ddot{\vec{p}}
\end{bmatrix} = \begin{bmatrix}
\vec{A}_m & \vec{B}_m \\
\vec{C}_m & \vec{D}_m
\end{bmatrix} \begin{bmatrix}
\vec{p} \\
\dot{\vec{p}}
\end{bmatrix} + \begin{bmatrix}
\vec{B}_m \\
\vec{D}_m
\end{bmatrix} [\vec{f}_{P_{n}}]
$$

(3)

are the displacement vector of the single mode shapes $\vec{p}$ and its derivatives. $\vec{A}_m$ includes the eigenvalues and the modal damping ratio and both $\vec{B}_m$ and $\vec{C}_m$ include the eigenvectors. The
state-space formulation based on numerical results can also have a different structure depending on the type of numerical analysis. Similar model reduction procedures exist for the various types of juncture elements.

Using the modeling strategy, complex electro-mechanic dynamic systems can be consistently built up in system simulation in dependence of the particular stage of development.

4. MODELING AND SIMULATION OF AN NVH REDUCTION SYSTEM

The original setup of the engine mounting system consists of an isolation subframe that is located between the engine and the car body. The isolation subframe is connected to the engine and the car body by elastic components. Besides the design of the subframe mounts, the integration of a vibration absorber, tuned to the relatively constant operation speed of the engine seems promising in order to optimize the NVH behavior of the concept car (Figure 6).

![Figure 6 Engine compartment of the concept car](image)

To investigate the potential of an NVH reduction system, an impedance-admittance model of the engine, its mounting structure and the attached car body is built up. The mechanical model structure is given in Figure 7. The system consists of the components engine, engine excitation, engine mounts, vibration absorber, isolation subframe, subframe mounts, car body and transfer path to the passenger’s ear.

![Figure 7 Mechanical structure of the system simulation model](image)
Depending on the stage of development of the real system, the single components are modeled at different detailing levels:

- **Engine**: Rigid body model based on CAD-data; three coupling points at the engine mounts and one for the force excitation
- **Engine excitation**: Force signal based on the indicated pressure and the engine geometry (lever); Acting on the engine
- **Engine mounts**: Fractional model, based on given dynamic stiffness of the elastomer; three engine mounts placed between the engine and the isolation subframe
- **Vibration absorber**: One-dimensional mass, stiffness and damping value to be designed; x absorber at x coupling point of the isolation subframe
- **Isolation subframe**: Reduced FE-model based on CAD-Data (7); three couplings points to the engine mounts and four to the subframe mounts
- **Subframe mounts**: Three-dimensional stiffness and damping values, to be designed; four subframe mounts placed between the isolation subframe and the car body
- **Car body**: Coupled three-dimensional transfer functions based on measured data; four coupling points to the subframe mounts.
- **Transfer path to passenger’s ear**: Uncoupled transfer functions, based on measured data; twelve transfer paths from the car body to the passenger’s ear.

A holistic numerical simulation model is built up using the software Matlab/Simulink. The corresponding block diagram is depicted in Figure 8. The light blue boxes represent admittance submodels (engine, subframe and car body), the grey boxes represent impedance models (engine and subframe mounts). The transfer paths to the passenger’s ear are modeled as finite impulse response (FIR) filter.

![Figure 8 Block diagram of the simulation model](image)

**5. OPTIMIZATION AND DESIGN**

The set up system model is used to design the optimal vibration absorber. It is easily possible to analyze the sensitivity of arbitrary parameters. Two different parameter studies are performed and in this chapter the results are discussed.

In Figure 9, the influence of the absorber mass on the system up to 100 Hz is shown. The absorber mass is varied between 0 kg and 10 kg and the impact on the forces introduced into the car body is investigated (left figure). The results are plotted in a normalized view in dB (re 1 N), where blue colors represent lower and yellow/red colors higher transmissibility of structure-borne noise with respect to...
the initial design without an absorber. In the lower frequency range, the natural frequencies of the system stay basically the same. With a higher absorber mass, the magnitudes of these frequencies rise while the neutralization effect increases. The neutralization frequency of the absorber is set to 41.67 Hz - the excitation frequency of the one-cylinder engine’s 0.5th order at operational speed of 5000 rpm. Due to the additional absorber mass, a new natural frequency occurs between 42 Hz and 55 Hz. This resonant frequency grows with a higher absorber mass, while the magnitude of the peak decreases.

![Figure 9 Variation of the absorber mass](image)

On the right hand side in Figure 9, the influence on the displacement of the engine’s center of gravity is depicted. The deflection of the engine should not exceed certain limits in order to comply with durability demands. High absorber masses (> 8 kg) may lead to critical deflections in the frequency range below 10 Hz. Therefore, absorber designs with high masses are not considered further.

Similar investigations with variable absorber stiffness are shown in Figure 10. The neutralization frequency is also set to 41.67 Hz and the stiffness is varied between 30 N/mm and 500 N/mm.

![Figure 10 Variation of the absorber stiffness](image)

The position of the absorber on the subframe is slightly moved towards the engine’s center of gravity. The overall effects are comparable to the absorber mass variation. For a small stiffness, an anti-resonance frequency in the displacement of the engine’s center of gravity appears. Absorbers with high stiffness tend to be critical, due to low frequencies with high displacement magnitudes.

In Figure 9 and Figure 10 the impact of the absorber parameters on the NVH behavior and the resulting forces introduced into the car body are shown. With a higher absorber mass or stiffness the neutralization effect gets stronger with an increased width between neutralization and resonance frequencies. Especially for systems with a variation of excitation frequency, this design is beneficial.
On the other hand, a lower mass or stiffness keeps the system light and the engine does not tend to vibrate with large magnitudes.

A good compromise is e.g. an absorber with $m_r = 3 \text{ kg}$, $k_r = 205.65 \text{ kN/m}$ and $f_r = 41.67 \text{ Hz}$. The frequency response function (FRF) of the overall system with (orange) and without (green) an absorber is shown in Figure 11. The function expresses the response forces introduced into the car body at one subframe mount to an applied excitation force at the engine’s center of gravity. During operation the frequencies between 12.5 Hz and 42 Hz (grey marked area) are excited.

![Figure 11 FRF: response forces at a subframe mount to excitation forces of the engine](image)

Due to the added absorber mass, which increases the overall mass of the system, the transmitted response forces get smaller between 17 Hz and 48 Hz. Furthermore an additional resonance frequency occurs at 52 Hz. At operational speed (5000 rpm, 41.67 Hz) the absorber causes an anti-resonance frequency with a substantially smaller response magnitude.

An engine run-up from 1500 rpm to 5000 rpm is simulated for both the original system as well as the system with absorber. In Figure 12 the impact of the absorber is depicted in the time domain. Compared to the original system, a slightly higher force magnitude occurs between $t = 6-7$ s. At all other times and especially at operational speed ($t > 15$ s), the resulting forces introduced into the car body are considerably reduced. A high absorber mass could cause even more reduction at operational speed with conceivable drawbacks during the run-up.

![Figure 12 Resulting forces at a subframe mount (left) and sound pressure in the interior of the car (right)](image)

The resulting sound pressure in the interior of the car is shown on the right in Figure 12. Here, too, the design with absorber leads to improvements on the NVH behavior of the system. The absolute reduction of the sound pressure is smaller, compared to the reduction of the resulting forces at the subframe mount. This is due to a multitude of different transfer paths into the interior of the car, which affects the summed-up sound pressure. At operational speed the sound pressure is reduced from
1500 μPa to 500 μPa. This corresponds to a reduction of approximately 8 dB (re 20 μPa) sound pressure level.

6. CONCLUSIONS

In this paper a systematic approach to model, simulate and assess multi physical domain systems is presented, in order to support the development process of vibration optimization measures. Furthermore, a concept for a novel low emission generator-electric vehicle with a steady-state operational speed of the generator is proposed, thus working at a high efficiency level. The expected impact of the operational control strategy on the NVH comfort is discussed and identified to be significant.

On the basis of this generator-electric vehicle the model-based development of a vibration absorber, integrated in a mounting system, is presented. Numerical simulations of stationary and transient operational conditions of the generator are performed. The numerical results have been presented and the performance of vibration reduction measures has been evaluated. By these results a feasible vibration absorber can be proposed.

The presented approach can be used to estimate noise, vibration and harshness performance in all phases of the design process. Thus, it helps to identify negative effects of the designed system in an early stage and to benchmark suitable active and passive concepts for the reduction of vibration in preliminary studies.

ACKNOWLEDGEMENTS

This work was partly conducted within the framework of the LOEWE center AdRIA. The support by the state of Hesse is gratefully acknowledged. Furthermore, the support by our colleague Hendrik Buff, who performed and analyzed the experimental determinations, is kindly acknowledged.

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