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Top-Down Validation Framework for Efficient and Low Noise Electric-Driven Vehicles with Multi-Speed Gearbox

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Abstract: The shift towards e-mobility is resulting in new technological challenges. Thus, new, more efficient product development methods and a better product understanding are required. During product development, validation is essential both to achieve significantly increased knowledge of the system in question and to ensure that customers' expectations of characteristics are met. Based on existing top-down validation approaches, this article discusses an innovative both-ends-against-the-middle-approach (BEATM) developed by the author. A validation framework which combines physical and virtual elements is presented. By way of example, a development approach for the toothing validation layer of an electric vehicle powertrain with a multi-speed gearbox is introduced.

Keywords: electric drive; noise; gear; powertrain; simulation

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1. Introduction

Facing todays challenges in electrical powertrain development, multi-speed gearboxes are increasingly used, as they improve the efficiency of utilisation [1,2].

Various methods may be employed in the search for the appropriate motor-gearbox combination. Any of these methods can help achieve an optimised powertrain topology both in terms of efficiency [2–4] and with regard to the customers' requirements [1,5]. Multi-speed gearboxes also make use of lighter electric motors with higher possible maximum rotational speeds, resulting in a decrease in costs [1]. However, both high rotational speeds and the absence of acoustic masking in combustion engines lead to an increase in actual gear noises as well as their perception by the driver. In particular, the tooth mesh and its behaviour are a source for noises and must be considered throughout the entire development process.

To achieve improved acoustic characteristics, both the macro- and the micro-geometry of the teeth can be optimised [6]. Furthermore, specific modifications of the teeth's surface will also be discussed [7]. Established analytical models are used to describe the influence of profile shifts on the mesh stiffness of spur gears [8]. Additionally, the vibration and acoustic radiation of gearbox housings based on 3D simulations will be explored in detail, as will the influence of different gearbox macro-designs on efficiency and vibration at high speeds [9,10]. While methods of generating acoustically-optimised gears and efficiency-optimised gearbox designs are well established, there is a distinct lack when it comes to a holistic view of the interaction between efficiency and noise behaviour in the early stages of product development.

Within the product development process, validation is the main source of knowledge and is essential in ascertaining the fulfilment of customer expectations with regard to product characteristics [11]. Using a mixed physical–virtual validation environment allows for early studies of product behaviour [12]. The x-in-the-loop framework (XiL) supports validation activities and the development of validation environments, leading to faster and more efficient product development [13–15]. Examples of the established methods to create a physical validation environment, in particular for the acoustic analysis, will be discussed [16,17].

XiL is used to validate product properties over a wide range of development stages without the need for a whole-system prototype. Considering validation as a multi-layer activity, the X stands for the system in question on the particular layer. Independently of the layer at which a system is validated, the user and its environment are an essential part of the XiL framework. This can be shown particularly clearly in the validation of vehicles, since, here, the driver closely interacts with the system as well as the environment, which includes factors such as routes chosen, weather conditions, etc.; see Figure 1. When setting up the validation layer, the first step is the definition of the unit under test as well as the specification of the rest-system properties required for the particular purpose of the test. The system is then divided into subsystems, which are either physical or virtual—without losing the closed control loop shown in Figure 1.

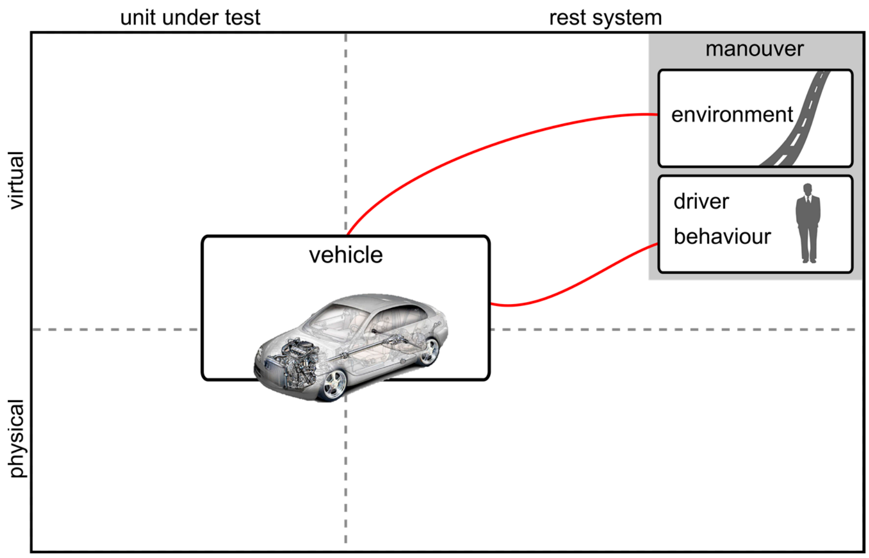


Figure 1. System layout, vehicle-in-the-loop.

By combining the approaches and frameworks shown above with meta models such as the V-Model [18], the integrated Product engineering Model (iPeM) [19], or the Munich Procedural Model (MPM) [20], the product development process can be streamlined and made more efficient. Considering XiL as the basis of a top-down planned validation, the established processes of Simultaneous Engineering and Concurrent Engineering can be combined to form a new validation approach. This optimised top-down validation approach can lead to the simultaneous development of physical as well as virtual models in a both-ends-against-the-middle approach [14,21].

Above all, the challenge is to develop virtual models which then have to be validated by physical tests. These virtual models are intended to evaluate both acoustic quantities and the efficiency in the early stages of the product development process.

2. Top-Down Framework and Model Implementation

The XiL framework introduced above, together with the top-down validation approach, will determine the validation set-up. Planning the validation from the top-level, in this particular setting, leads to a four-layer environment. The main focus when working on solutions for optimising powertrain units is still the vehicle as a whole, with its real behaviour in a real environment with a real driver. In particular, the drivers' experience of vibration and sound must be considered in all validation layers. Therefore, the top-level vehicle tests are planned as set out in Figure 1.

Stripping the chassis, the wheels, the side shafts, and the battery from the real vehicle—and transferring these to the virtual rest system—results in the second validation layer (layer 2, see Figure 2). Within that layer, the motor-gearbox-unit is the physical unit under test, thus forming a powertrain test rig. The environment and the driver behaviour will also be part of the virtual rest system; see the system layout in Figure 2. The red lines indicate the interaction between the components. Where they represent a cross-over between the physical and virtual domains, suitable sensors and actuators are needed, as well as, ideally, a real-time-capable simulation software.

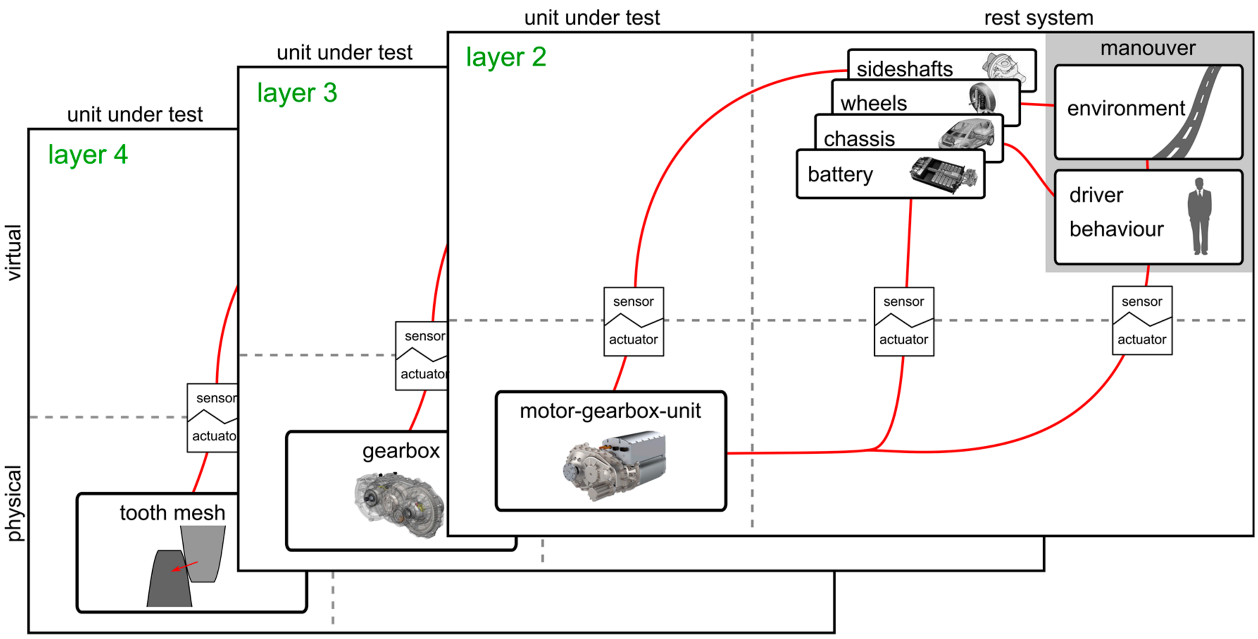


Figure 2. XiL system layout.

Reducing this system by the traction motor leads to a gearbox test environment on the third validation layer, which is not further discussed in this article. However, it is worth mentioning that the interaction between all validation layers is always taken into account. The fourth validation layer examines the tooth mesh as the physical unit under test; here. all the other vehicle components are part of the virtual rest system.

Planning the validation in a top-down order clearly defines all the validation layers as well as the system interfaces. The implementation of the tests (both physical and virtual) is usually performed in a bottom-up order; please refer to the discussion of the extended validation approach in Section 5.

The model implementation as described below (Sections 3 and 4) is based on the system layout as stated above. Starting the implementation activities on the fourth validation layer leads to a gear mesh study with the objective of gathering results for gears optimised in terms of both sound emission and efficiency. The virtual domain of that layer is described in Section 3, for the physical domain see Section 4.

In addition to the framework introduced above, a model-based validation environment is established as shown in Figure 3. It shows the content of both the virtual and physical domains as well as the interfaces between them. The objective is to achieve validated models of the gear mesh. Therefore, the detailed toothing data are calculated by an industry project partner using gear calculation software. These parameters are used as input for the virtual domain as well as manufacturing parameters in the physical domain.

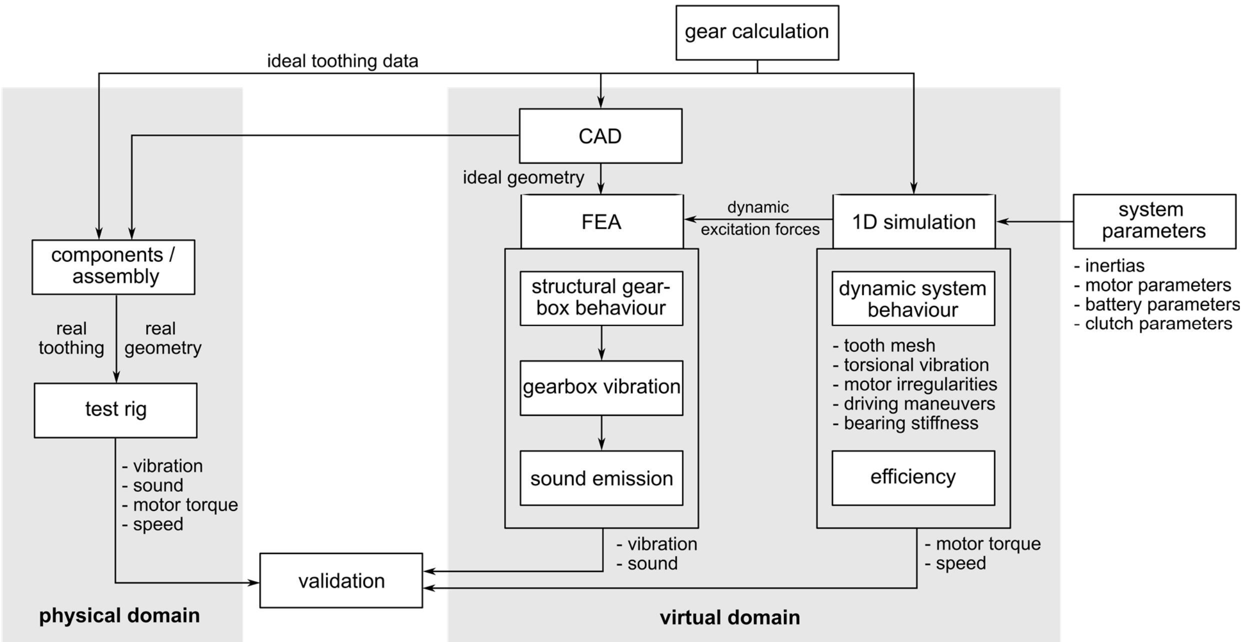


Figure 3. Model implementation.

3. Virtual Model Implementation

To calculate the dynamic system behaviour and the efficiency of the gear pair, a 1D simulation software is used. Subsequently, the dynamic excitation forces at the housing are passed on to a Finite Element Analysis (FEA) where, firstly, the structural behaviour of the ideal gearbox housing geometry is considered. By combining the structural behaviour and the excitation forces from the 1D simulation, the gearbox vibration and the sound emission (both air-borne and structure-borne) can be estimated.

The 1D simulation model contains the shafts (including their masses), helical gears (see Table 1, which includes their masses, inertias, and gearing parameters), and the bearings (including their stiffness and damping coefficients). Figure 4 shows an overview of the implemented system. The dynamic behaviour is predominantly influenced by the gear element. Therein, properties of the tooth contact are defined, which result in irregularities of the output rotational speed. In this model, particularly, the effect of the mesh stiffness on the overall system is examined.

The dynamic equations of the gear system shown in Figure 4 can be written as:

$$[\mathrm{M}]\{\ddot{\mathrm{q}}\} + [\mathrm{D}]\{\dot{\mathrm{q}}\} + [\mathrm{C}]\{\mathrm{q}\} = \{\mathrm{F}(\mathrm{t})\}$$

where M is the matrix consisting of masses and inertias, D is the damping matrix, and C is the stiffness matrix; q is the displacement vector and F the vector including the forces. Periodically varying mesh stiffness leads to the systems' dynamic behaviour. A spring element between two meshing teeth is used to describe the mesh stiffness. The values of the specific tooth stiffness are specified with respect to the normalised meshing length, see Figure 5b.

Table 1. Gear pair data.

Parameter	Symbol	Value
No. of teeth (-)	z_1, z_2	38, 57
Normal module (mm)	m	1.75
Face width (mm)	b	20
Helix angle (°)	β	15
Normal pressure angle (°)	α	20

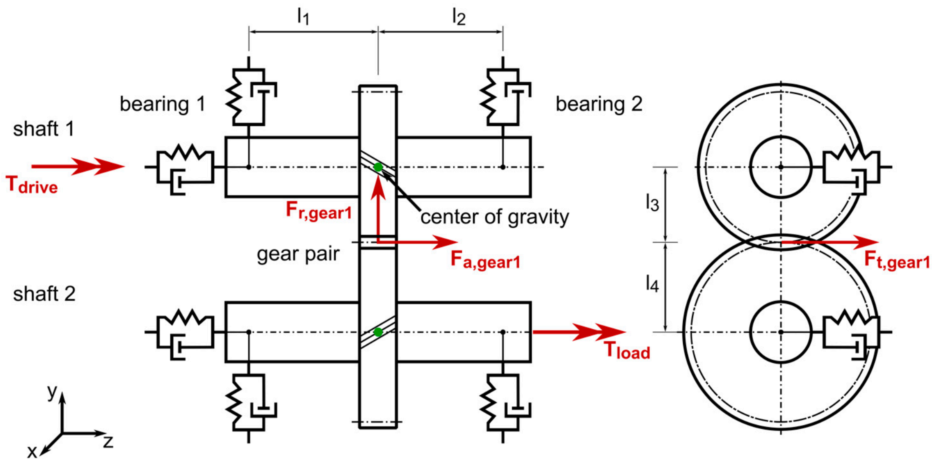


Figure 4. Gear system.

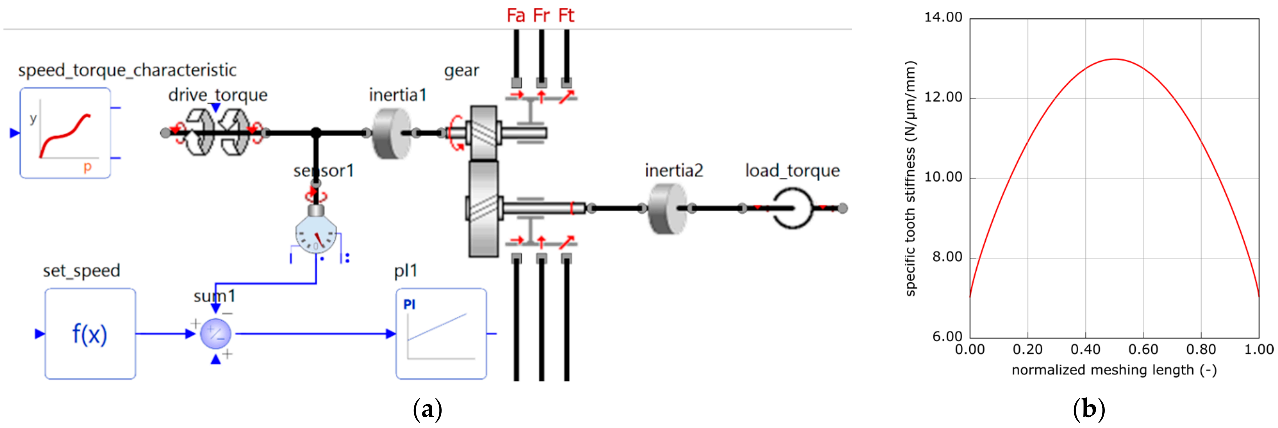


Figure 5. (**a**) Rotational system; (**b**) specific tooth stiffness for the mesh stiffness calculation.

The simulation model is divided into a rotational and a translational domain. In the rotational model an electric motor sets a drive torque. A real electric motor's speed–torque-characteristic is used. With the help of a simple control loop, different load torques can be defined, depending on the speed desired. The gear pair model calculates the gears' dynamic behaviour, including the parameters from the gear calculation. Figure 5a shows the details of the rotational system in the 1D simulation software.

The translational system is used for the calculation of the bearing forces. For this, the tooth forces are provided by the gear pair model. Each shaft is represented by two translational models which contain both the moment and the force equilibrium. To calculate the bearing forces at the gearbox housing in the z- and y-direction, the axial and radial tooth forces are used, see the coordinate system in Figure 4. The second translational model calculates the bearing reaction force in the x-direction using the tangential tooth forces. To consider the bearing stiffness and damping, spring-damper-elements are used. A simulation model detail for gear shaft 1 is shown in Figure 6. The inner forces of the spring-damper models represent the resulting forces in three directions and can be used for the FE analyses described below.

To perform dynamic 3D FE analyses, the project schematic, as shown in Figure 7, is set up. The geometry block contains the solid domain (the gearbox housing) for structural analyses as well as the fluid domain (the acoustic region and air enclosing the gearbox housing) for the acoustic analysis. Based on a modal analysis for the determination of the eigenfrequencies and their shapes, a harmonic (frequency) response analysis with modal superposition is performed. The dynamic bearing loads derived from the 1D simulation are used as input for the harmonic response analyses. The calculated structural behaviour is used as input for the acoustic analyses which result in the acoustic parameters.

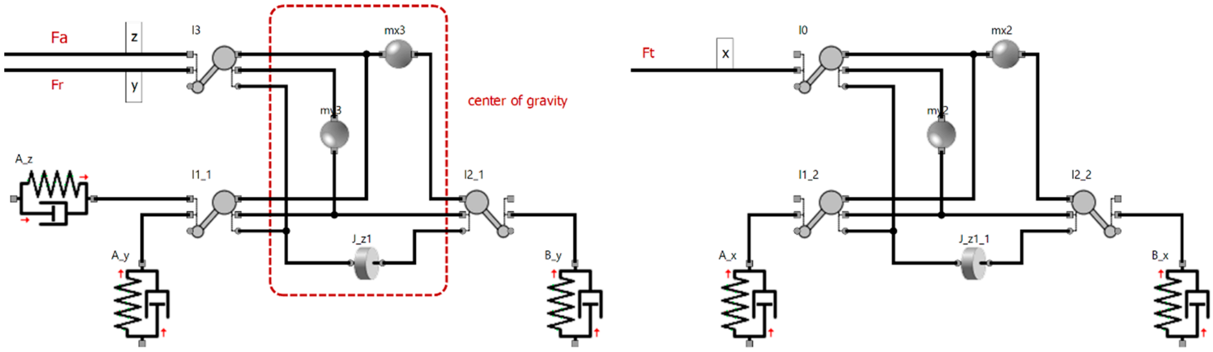


Figure 6. Translational systems for calculating bearing forces in z- and y-direction (**left**), and x-direction (**right**).

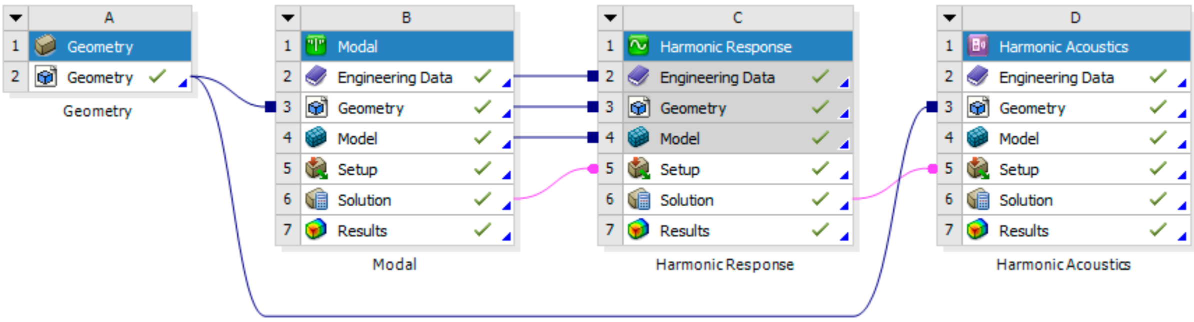
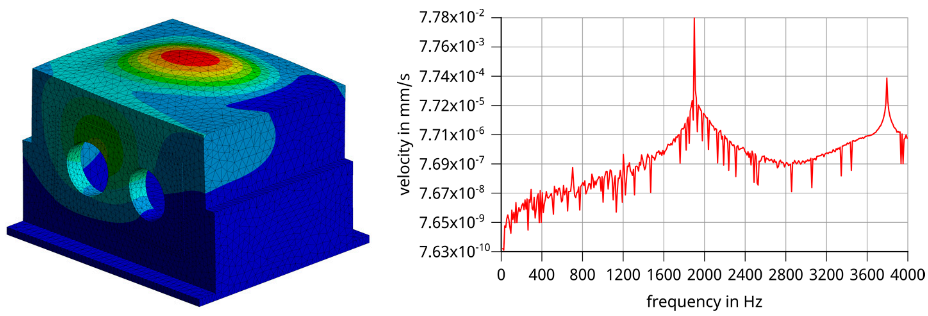


Figure 7. FEA project schematic.

Figure 8 shows a sample result of the harmonic response analysis. In the velocity amplitude spectrum (right) for a specific gearbox housing surface, the eigenfrequencies are indicated by higher amplitudes which are fed by the bearings' dynamic forces. Figure 8a shows an example of the deformation of the gearbox housing at a specific eigenfrequency. To predict the acoustic behaviour of the system (in particular the air-borne sound emission), the acoustic analysis uses the results of the harmonic response analysis.



(**a**) (**b**)

Figure 8. (**a**) FEA deformation result of the harmonic response analysis; (**b**) velocity amplitude spectrum.

The acoustic analysis uses the fluid domain. The load is applied by combining the surface velocities calculated using the harmonic response analyses with the interface surface of the fluid (air). The sound parameters in the fluid region are calculated using the acoustic coefficients of radiation. The sound pressure level L_p is used to rate the simulated gear noise. It is calculated as:

$$L_p = 10\,\lg\left(\frac{p}{p_0}\right)^2 = 20\lg\left(\frac{p}{p_0}\right)$$

where p is the root mean square sound pressure and p_0 is the reference sound pressure with a value of 20×10^{-6} N/m^2. This approach has been established and is described in former publications [9,22,23].

By way of example, the sound pressure distribution within the fluid domain at a specific frequency is shown in Figure 9a. The frequency spectrum (Figure 9b) shows the sound pressure level (SPL) at a specific point. These results can be used as a reference for physical tests, such as structure-borne or air-borne sound measurements for validating the implemented virtual models (both 1D and 3D). Additionally, they can be used for further optimisation loops of the gears and the housing.

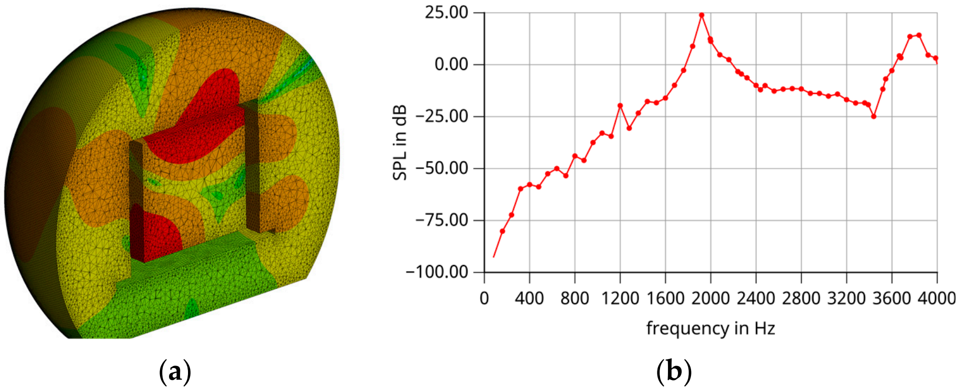


Figure 9. (**a**) FEA sound pressure result; (**b**) sound pressure spectrum.

4. Physical Validation Environment and Methods

To validate the results of the virtual models described above, the physical domain is developed according to Figure 3. The objective is to obtain validated virtual models of the gear mesh as well as information about the quality of the acoustic simulation. Since the parameter changes in the gears are in a micron range, a geometrically precise test rig is needed. Due to this, the high rotational speeds (up to 20,000 rpm) and high system frequencies (in particular from the gear mesh), exceptionally high accuracy sensors for torque, speed, and sound pressure (structure-borne and air-borne) are needed. Figure 10 shows a rendering of the test rig. Similarly to the simulation model described in the previous section, the test rig consists of a gear pair which is driven by two electric motors. Both are controlled by a real-time-capable automation software. This now makes the simulation of the rest system according to Section 2 feasible.

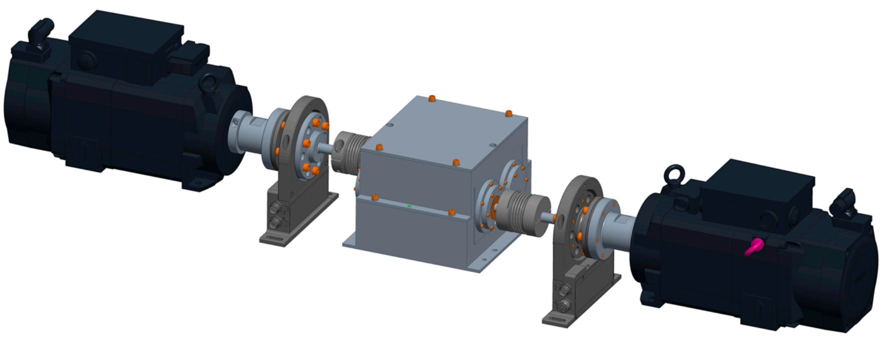


Figure 10. Physical validation environment.

The design process took into account both the bearing rollover frequencies and the torsional eigenfrequencies of the entire powertrain. The gearbox housing has a specific structural behaviour as described above. Thus, critical operating points and ideal positions for the placing of accelerometers can be determined. Due to the high influence of the concentricity on the vibration excitation, the gears are connected to the shaft by an optimised cone clamping sleeve. This ensures that the quality of gear alignment following any replacement will remain consistent.

The 3D simulation results suggest that the air-borne sound pressure level is comparatively low. Therefore, the usability of the results must be checked. Appropriate software systems and data acquisition modules for data recording and processing are already available. It is also possible to determine the efficiency of the gear pair enabled by the high precision speed and torque measurement data. This allows for a combined analysis of efficiency and acoustics as well as the interaction between them.

To validate the simulation models which were implemented during the development of the test rig, physical testing must be used. For the validation of the calculated eigenfrequencies and their shapes (gearbox housing), an experimental modal analysis will be applied. Real experimental studies on the rollover frequencies of bearings and natural frequencies of the torsional vibrations will also be performed. In addition, the frequency response of the gearbox housing is measured at certain operating points (e.g., certain speed). In addition, the acoustic simulation will be validated with real measurements. For this purpose, the same points as in the simulation are selected and the structure-borne sound is measured at this location. The hardware required for the physical analyses is available. This includes acceleration sensors for the modal analysis of the gearbox housing as well as high-resolution microphones and structure-borne sound sensors for the acoustic analysis. A dedicated analysis software is also available.

5. Extended Validation Approach

There is an ongoing discussion and expansion of the available validation approaches and frameworks; the both-ends-against-the-middle-approach introduced by the author contributes a new perspective. The focus here is to suggest that validation needs to combine both top-down and bottom-up activities.

The concept of the BEATM-approach is visualised in Figure 11. Starting the product development process with planning the validation activities from the top to the bottom (from vehicle to gear pair, as described in Section 2) allows for the appropriate consideration of the rest system behaviour at each validation layer (compare the top-down planned activities shown in Figure 11). Following this top-down planning of activities, the simultaneous development of virtual validation models as well as the physical validation environment is performed according to Figure 3 in Section 2. The integration of the model itself is mostly a stand-alone activity without a validation layer interface. Hence, a bottom-up approach as shown in Figure 11 also results. However, depending on the time of model implementation, not all the information required for this may be available. Nevertheless, the idea of result transferability from layer to layer should be kept in mind. To achieve this, the validation of all (sub)models implemented is essential. Consequently, it is possible to arrive at validated whole-system models by combining validated sub-models, although the model complexity increases from layer to layer and over the time of development.

Using the method described above, a real frontloading of validation activities within the product development process can be achieved. Working with these validated sub-models guarantees that there is awareness of the customers' requirements over the entire development process, regardless of which validation layer is being worked on. Upon completion of a project, all the implemented and physically validated models can be used for the development of further products with similar requirements. They can be transferred within a product development and reference process framework.

By combining the bottom-up and top-down activities as described above, existing validation approaches may be expanded. Following Figure 11, the content of both the virtual and the physical domain meets at the end of the development and validation time line. At this point, the product is fully developed and validated, and both domains together describe it in its entirety. The approach described herein is new and offers a method to gain a better understanding of the inter-relations between a product's development and its validation.

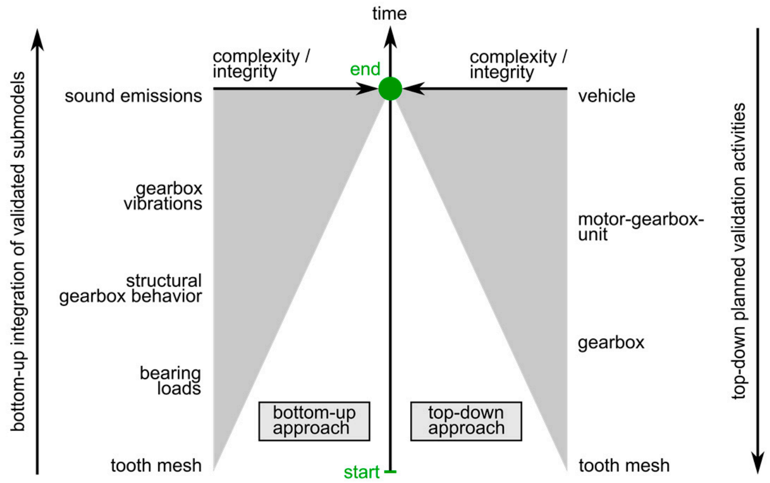


Figure 11. Both-ends-against-the-middle-approach (BEATM).

6. Conclusions

Based on the approaches and frameworks currently available, a physical–virtual coupled validation environment for noise and efficiency studies has been introduced. Existing approaches are extended by using a combination of bottom-up and top-down validation activities.

The virtual domain for a gear pair test is represented by 1D simulation models, which enable the estimation of the dynamic system behaviour of such a gear pair as well as the respective shafts, including the bearing behaviour by stiffness and damping coefficients. Thus, the reaction forces at the housing can be calculated. Subsequently, a 3D FE analysis is performed in which, first, the modal behaviour is computed and, second, the frequency response is estimated using the reaction forces from the 1D simulation as input. Based on the results of the frequency response analysis, respectively, the surface velocities at the gearbox housing, an estimation of the emitted air-borne as well as structure-borne sound can be made through an acoustic analysis.

Controlled by a real-time-capable automation software, a test rig including two electric motors represents the physical domain. The main objective of the test rig operation is the validation of the results from the 1D and 3D simulation models. The test rig behaviour itself needs to be validated, in the future, prior to the start of testing. There are also plans for the extension of the virtual models with regard to efficiency. Finally, the both-ends-against-the-middle-approach remains to be transferred into a reference process model.

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