

Power Train Model Refinement Linked with Parameter Updating Through Nonlinear Optimization

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ABSTRACT

In the virtual development process validated simulation models are requested to accurately predict power train vibration and comfort phenomena. Conclusions from refined parameter studies enable to avoid costly tests on rigs and on the road. Thereby, an appropriate modeling approach for specific phenomena has to be chosen to ensure high quality results. But then, parameters for characterizing the dynamic properties of components are often insufficient and have to be roughly estimated in this development stage. This results in a imprecise prediction of power train resonances and in a less conclusive understanding of the considered phenomena. Conclusions for improvements remain uncertain.

This paper deals with the two different aspects of model refinement and parameter updating. First an existing power train model (predecessor power train) is analyzed whether the underlying modeling approach can reproduce the physical behavior of the power train dynamics adequately. Thereby especially rotational irregularities of the power train causing the low frequency boom noise are considered. Based on the example "tire model", different model improvements are investigated by means of sensitivity analysis. Efficient measures are chosen to validate the quality of the results in comparison to test results.

A manual approach by a stepwise analysis of each parameter is time consuming and often does not lead to accurate results due to nonlinear model behavior. Furthermore, the interdependence of an increasing number of parameters can hardly be managed manually. Therefore, the second part of

the paper focuses on parameter identification methods through nonlinear optimization based on full vehicle measurements. The whole optimization process - from the formulation of the objective function to the analysis of the results - is discussed.

The model improvements as well as the parameter optimization are based on vehicle tests of an all-wheel driven passenger car. The combination of model refinement and updating with non linear optimization methods demonstrates an effective approach to fine-tune simulation models for an optimum support in the product development process.

INTRODUCTION

Multi Body Simulation models (MBS) of the power train are used for various applications in the development process of passenger cars. The goal is to save costs and time replacing expensive prototype tests by virtual model analyses on computers. For example the analysis of power train vibrations in the design stage is essential.

One important aspect in the development of power trains is the optimization of drivability. Drivability describes the complex and subjective feelings experienced by a driver while interacting with a vehicle [8]. The calibration task concerning drivability is closely linked with low frequency power train vibration and comfort phenomena, like engine idle, full load, Tip in (Shuffle), etc [6], and it is furthermore closely related to the ECU (Engine Control Unit) and TCU (Transmission Control Unit) calibration. The various controllers increasingly influence the behavior of a car (X by wire). More and more development capacities are spent in the

calibration and data adjustment of implemented control algorithms. Hereby, it is important, that the driver is not aware of all these complex processes in the background. For the development process [12] linking of hardware with virtual models is therefore another important aspect. For these SiL/HiL (Software in the Loop/Hardware in the Loop) applications real time simulation models of power train and the entire vehicle are needed.

The power of modern gasoline and diesel engines is still permanently increasing. The development of new charging devices and ongoing improvements in injection technique cause high fluctuations of engines torques and therewith high vibration excitation of the whole power train. To further reduce fuel consumption, especially low speed driving conditions become important. Relevant simulation analyses require precisely adjusted models in order to provide high precise results.

The following article focuses on the rotational vibrations in the whole power train excited by a so-called “wide open throttle” acceleration maneuver. High torque fluctuations at low speeds can be amplified in the power train by the dynamic behavior of the different subsystems. These vibrations excite global modes of the chassis at the mounting points between the suspension or aggregates and the vehicle body and can result in the uncomfortable boom noise [3], [4].

SIMULATION ENVIRONMENT AND MODEL SETUP

The multi body simulation model is defined in an object oriented simulation environment with good compatibility to Matlab Simulink (the model can be easily imported as an S-Function). Matlab Simulink is a wide spread simulation environment for real time simulations. It is based on a signal flow based concept and has thereby some disadvantages compared to the object oriented approach. This is valid especially when developing complex and large models. The advantages of the object oriented approach are [2]:

- Based on equations and not on data flow (adaptable to more than one data flow context);
- Better reuse of classes because the underlying equations do not represent a data flow;
- Object oriented language with class concept or templates in C++;
- Reuse of components and evolution of models available;
- An architectural description language for complex physical systems;

The simulation model represents a Volkswagen Golf V, 4MOTION with a turbo charged direct injection Diesel engine and a six gear manual transmission. The engine has a

displacement of 2l and a maximal torque of 320 Nm. Figure 1 gives an overview of the different subsystems of the power train. For the rotating parts and their elastic connections (joints) only the torsional degrees of freedom are taken into account, except for the prop shaft model. The prop shaft models consist of two flexible bodies which are connected through a constant velocity joint with a linear stiffness and damping.

(See Figure 1 after last section of paper)

The engine model consists of a rotation angle dependent inertia Θ representing the rotating and oscillating masses of the crank train [7]. The crank angle dependent inertia is excited by a crank angle dependent torque M . This torque represents the sum of the tangential pressures p_{TG} multiplied with piston area A_K and the radius r_K for all 4 cylinders (Equation 1).

$$\begin{aligned} \Theta(\varphi) \cdot \ddot{\varphi} + \frac{1}{2} \Theta'(\varphi) \cdot \dot{\varphi}^2 &= M(\varphi) \\ M(\varphi, n) &= \sum p_{TG}(\varphi, n) \cdot A_K \cdot r_K \end{aligned} \quad (1)$$

The highly nonlinear model consists of the crank angle dependent inertia and the speed and crank angle dependent moment. Thus, it has two degrees of freedom ($\dot{\varphi}, \varphi$). It can represent the torque and speed fluctuations of the engine and was developed for high calculation efficiency.

The dual mass flywheel model consists of two inertias and speed dependent stiffness and damping parameters.

Several inertias connected with 2 linear stiffness and damping elements model the transmission. The two gear stages are modeled as kinematic gear joints.

The front axle differential and Power Take Off (PTO) consist of simple inertias and corresponding kinematic gear joints and the differential gear.

$$M_{in} = 0.5M_{out,1} + 0.5M_{out,2} \quad (2)$$

For the prop shaft different models are available, depending whether they are used for offline calculations or for real time implementations in HiL environments. In our case a flex body prop shaft model is used to achieve high result quality. To achieve an efficient parameter and optimization result only a limited number of modes is implemented (first 4 dynamic modes). In former publications it could be shown, that even with a low number of modes, results of sufficient quality in the regarded low frequency range can be achieved [5].

In the rear axle drive the VW typical Haldex clutch is implemented. For full load accelerations the Haldex clutch is more or less closed. This effect was analyzed in [3], [4] where it could be shown that small Haldex slip has nearly no effect on prop shaft torsional vibrations. This condition is assumed here, too.

The side shafts are modeled as torsional 2 mass oscillators and for the tire model the Pacejka approach (steady state slip model) is implemented.

MODEL REFINEMENT

For the optimization process it is important that the used simulation model can describe and represent the expected model vibration behavior. For example if the prop shaft model is discretized too roughly dynamic behavior of rotational vibrations cannot be simulated adequately [3],[4] and [5].

Simulations of the basic model showed that the amplitude curves of rotational irregularities at the wheels (front and rear) showed a nearly 10 times too small amplitude level for the low frequency (not affected by noise disturbance) range compared to the measurements (Figure 2). Numerous parameter variations and sensitivity checks concerning the tire model and side shafts were carried out and didn't give any reasons for the low amplitudes. The parameter studies only effected the characteristics of the amplitudes but they don't increase the general level.

As a next logic step the steady state tire slip model (Pacejka Formula) was refined. In literature [10] a Single Contact Point Transient Tire Model (SCP) was introduced for relatively low frequency and large wavelengths transient and oscillatory vehicle motions. The idea behind the model is that the wheel slip point S, which is attached to the rim at a level near the road surface, is connected with a linear spring with the contact point S'. The introduced spring represents the compliance of the tire carcass (Figure 3).

(See Figure 2 after last section of paper)

(See Figure 3 after last section of paper)

The different velocities of the two points S and S' (located in same plane through wheel axis normal to road) cause a deflection u of the spring. The derivative for the longitudinal (x-direction) deflection becomes:

$$\frac{du}{dt} = -(v_{sx} - v'_{sx}) \quad (3)$$

For the lateral direction (y-direction) the same approach is used but not quoted here. The relaxation length σ_κ describes the ratio between slip C_{FK} and tire stiffness C_{Fx} .

$$\sigma_\kappa = \frac{C_{FK}}{C_{Fx}} \quad (4)$$

So the differential equation for the force and the lateral deflection can be written as follow

$$\frac{du}{dt} + \frac{1}{\sigma_\kappa} |v_x| u = |v_x| \kappa = -v_{sx} \quad (5)$$

with κ as the longitudinal wheel slip ratio.

$$\kappa \approx \frac{v_{sx}}{|v_x|} \quad (6)$$

The longitudinal force can be obtained by multiplying the deflection u with the longitudinal tire stiffness C_{Fx} .

For relaxation length following approach is used (relaxation length is a function of the vertical load):

$$\sigma_\kappa = F_z \cdot (p_{Tx1} + p_{Tx2} \cdot df_z) \cdot e^{p_{Tx3} \cdot df_z} \cdot \frac{R_0}{F_{z0}} \cdot \lambda_{\sigma\kappa}$$

$$df_z = F_z \cdot \frac{1 - \lambda_{Fz0}}{\lambda_{Fz0}} \quad (7)$$

With:

df_z change in vertical load
 F_z nominal load

With tire parameters:

p_{Tx1} 0.8
 p_{Tx2} -0.005
 p_{Tx3} -0.3
 $\lambda_{\sigma\kappa}$ 1.157 scaling factor
 R_0 0.314 [m] (undeformed wheel radius)
 F_{z0} 7100 [N] (nominal load)
 λ_{Fz0} 0.906 (nominal load scaling factor)

The simulations with the refined Single contact point transient tire model show a correct behavior of the wheel rotational irregularities. The amplification in amplitude can be found at 1400 rpm (Figure 2).

RIG TESTS AND MODEL VALIDATION

For model validation and the subsequent optimization process experimental investigations on a chassis dynamometer for 4WD vehicles were performed. Thereby the rotational irregularities of the power train were measured and analyzed. The rotational irregularities were measured with rotating laser vibrometer (wheels and shafts) and with inductive sensors (dual mass flywheel and transmission) (figure 4). The rotational irregularities at the wheels are compared to the other measurement positions the smallest ones and over 2000 rpm the electronical and optical noise is higher than the occurring movements (rotational movements) of the wheels. In this operating range over 2000 rpm the measurement results for the wheels are not valid and no conclusions can be drawn from them. The highest amplitudes for the wheels can be found at about 1400 rpm and so they are not affected by the noise disturbance. More details on the measurements of rotational irregularities and the noise phenomenon can be found in [3]. The measurement results are used for parameter identification of the power train model. The approach is described in the subsequent chapter.

(See Figure 4 after last section of paper)

PARAMETER UPDATING WITH NONLINEAR OPTIMIZATION

Fitting a model with two or three roughly estimated parameters can be done manually. Each parameter is manipulated step by step until the calculation fits best to the underlying measurements. But if the number of parameters increases, model calibration becomes difficult, because the complexity and number of possible variations increase enormous. In this case a systematic model refinement with the help of nonlinear optimization can save a lot of time and work.

The aim is to identify certain parameters of the simulation model, so that the deviation compared to the measurements is a minimum.

For the definition of the objective function a mathematical approach is needed to calculate the difference between the vibration amplitudes of the measurements and the simulation

Test data were measured in time domain and analyzed via FFT-analysis in frequency domain. The different amplitude curves (A_1 to A_m) represent the considered measurement

points. They are given as discrete data points as a function of frequency f (equations 8).

$$\begin{aligned} A_1 &= (A_1(f_{1,1}), A_1(f_{1,2}), \dots, A_1(f_{1,n})) \\ A_2 &= (A_2(f_{2,1}), A_2(f_{2,2}), \dots, A_2(f_{2,n})) \\ &\vdots \\ A_m &= (A_m(f_{m,1}), A_m(f_{m,2}), \dots, A_m(f_{m,n})) \end{aligned} \quad (8)$$

with:

1 ... m measurement points
1 ... n frequency index

Also the simulation results in time domain can be analyzed via FFT in frequency domain. It is important that the FFT settings for the measurement and simulation results are identical and that the discrete amplitude ($\bar{A}$) values are given for the same frequencies as the measurements (equations 9). This can be done easily by interpolation.

$$\begin{aligned} \bar{A}_1 &= (\bar{A}_1(f_{1,1}), \bar{A}_1(f_{1,2}), \dots, \bar{A}_1(f_{1,n})) \\ \bar{A}_2 &= (\bar{A}_2(f_{2,1}), \bar{A}_2(f_{2,2}), \dots, \bar{A}_2(f_{2,n})) \\ &\vdots \\ \bar{A}_m &= (\bar{A}_m(f_{m,1}), \bar{A}_m(f_{m,2}), \dots, \bar{A}_m(f_{m,n})) \end{aligned} \quad (9)$$

The sum of the squared difference of all amplitude values between measurement and calculation gives the objective functions J_i for the measured amplitudes.

$$\begin{aligned} J_1 &= \sum_{i=1}^n (\Delta A_1(f_{1,i}))^2 \\ J_2 &= \sum_{i=1}^n (\Delta A_2(f_{2,i}))^2 \\ &\vdots \\ J_m &= \sum_{i=1}^n (\Delta A_m(f_{m,i}))^2 \end{aligned} \quad (10)$$

In general, all amplitudes show a different level and different minimum and maximum values. For easier handling the amplitudes are normalized. Thereby all values are between 0 and 1. The normalized squared amplitude differences are defined as