STEP BY STEP MODEL BASED SYSTEM TESTING APPROACH FOR DRIVELINE TORSIONAL VIBRATION STUDY WITH APPLICATION FOR BOOMING AND TIP IN ATTRIBUTES

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ABSTRACT

Over the past decades many drivers have made the automotive world change. Going towards lower fuel consumption, reaching higher safety level for driver and passenger, even significant steps towards autonomous vehicles have been taken in the recent years. But whatever the innovations, step forwards, mind changes in our mobility habits, the industry considers the driving experience, subjective feeling, comfort as being at the center of the design choices.

Torsional vibration issues generated by the torque irregularities of a combustion engine or transient phenomena is a typical concern for auto OEMs or driveline manufacturers. This paper describes a Model Based System Testing approach for optimization of the driveline. A three-step approach is presented starting with system simulation model creation. Model parameterization is done either from specifications or retrieved from component reverse engineering. The model is then validated using the Functional Mockup Interface (FMI) 2.0 standard for direct test/simulation comparison. Finally, the validated model is used for NVH target optimization in view of next vehicle design. Two concrete example cases are used to illustrate the concept: booming noise optimization in a benchmark vehicle with CPVA and tip in vibration in a known vehicle using different ECU control parameters.

INTRODUCTION

In recent years the combined use of test and simulation has become a cornerstone of the development cycle [1]. When it comes to vehicle driveline torsional vibration driven by torque irregularities from combustion engine, or from transient event like sudden throttle variation, the 1D system simulation approach is an ideal trade-off between ease of parameterization, processing time and prediction accuracy.

In this paper 1D simulation models of the driveline are created using

Simcenter Amesim and are integrated as FMI in Simcenter Testlab Process Designer to ease the model validation and comparison with test data. Once the models are validated, they are used to check the effect of component parameterization as well as control implementation towards torsional vibration behaviour and global vehicle NVH perception. Two cases are studied:

- Low frequency booming noise and vibration. The study focusses on the frequency range up to 100Hz and the effect of a Centrifugal Pendulum Vibration Absorber in the driveline.
- Tip in shock. Tip in is the sudden reacceleration after a coast down. The typical unwanted behaviour is a strong and undamped shock typically below ~20Hz in the longitudinal direction. The effect of different ECU control parameters is evaluated.

MODEL CREATION

The main objective of the simulation model is to understand the complete dynamic of the driveline and more precisely to quantify 1) the effect of CPVA towards seat rail vibration and 2) the influence of control parameters towards vehicle longitudinal acceleration.

Low frequency booming

For the low frequency booming case the vehicle under test has a 4-cylinder petrol engine with a 7-gear automated gearbox. The transmission is done at the rear wheels through a propeller shaft, differential and driveshafts.

In order to ensure a good accuracy up to ~100Hz, the gearbox model is built using distributed inertias. The suspensions are modelled with 2D components and dynamic tire properties. The powertrain is built with a 3D inertia block and the mount stiffness properties are measured. The transfer to the interior noise and vibration is done using measured FRFs in trimmed body configuration. The vehicle body is represented by its inertia and geometrical parameters. The model is driven using dynamic torque input at the flywheel. The model overview is shown in Figure 1 below:

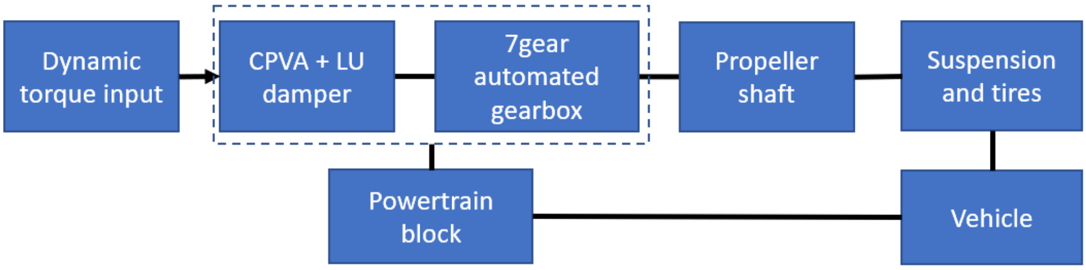

Figure 1 - System simulation model for booming

The key element of the booming study is the Centrifugal Pendulum

Vibration Absorber. The CPVA is a torsional damper designed to damp a torsional order. It's damping properties are mainly linked to the inertia, friction parameters and the tracks of the surrounding masses. The CPVA is built-in together with a torque converter and lock up damper at the input of the gearbox. It is combined with a set of linear springs and arc springs as shown in Figure 2

Figure 2 - CPVA assembled in the torque converter

All geometrical parameters, inertia and spring stiffness are measured, and a CAD model is built to have an accurate description of the tracks. These tracks are then extracted and implemented in the system simulation model using a 2D library. Figure 3 shows the 3D model and the corresponding system simulation layout:

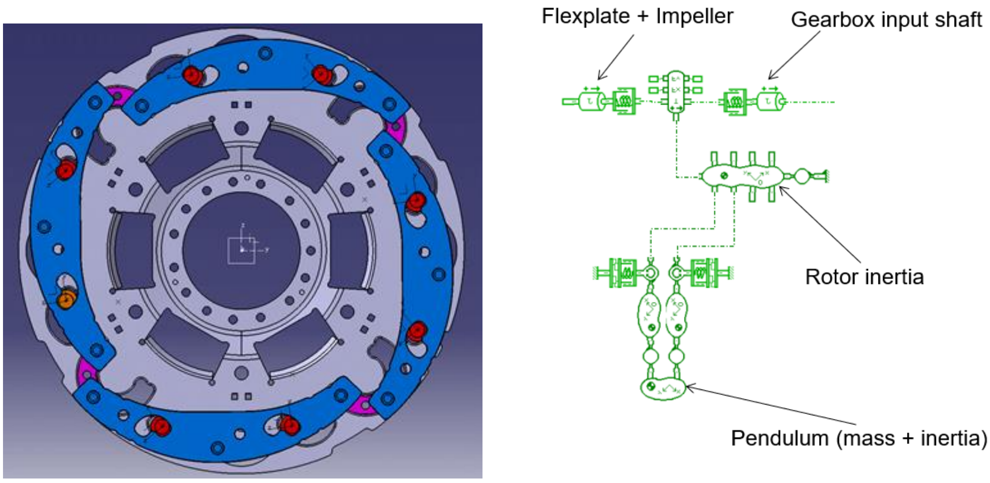

Figure 3 - 3D CPVA assembly and corresponding 1D model

Tip in

For the tip in case the vehicle under study is known and all design parameters are used for the model creation. The frequency range of interest is lower therefore the gearbox is built using simplified ratio with equivalent inertia and clearances. Tires, suspension and vehicle body are modelled using 2D components with known inertia and stiffness. The powertrain block is modelled as a 3D component. The overview is shown in Figure 4.

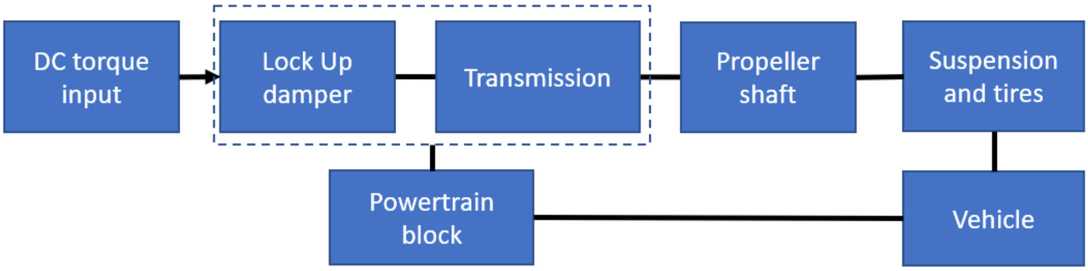

Figure 4 - System simulation model for tip in

A controller is added on top of the plant model, using a co-simulation between Simcenter Amesim subsystem with solver and a MatLab Simulink subsystem with solver as shown in Figure 5. The control loop will ensure that the model is driven with the correct torque profile depending on the vehicle speed, engine speed, throttle input…etc.

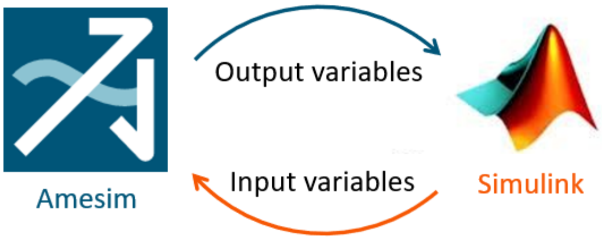

Figure 5 – Co-simulation Amesim / Simulink for embedded controls

MODEL VALIDATION

The validation step consists in making a comparison between the result of the simulation model and the result of operational tests. From the operational measurement a set of data is used to drive the simulation model and the selected target channels are used for direct comparison of the simulation result. In order to avoid a tedious import/export/comparison work for correlation the Functional Mock-up Interface (FMI) 2.0 for Co-Simulation (open standard to conveniently exchange models prepared in different simulation software environments) is used. A model using the FMI interface is called a Functional Mock-up Unit (FMU). An FMU can be imported into the complete process as a standard method, as shown in Figure 6 in Simcenter Testlab Process Designer:

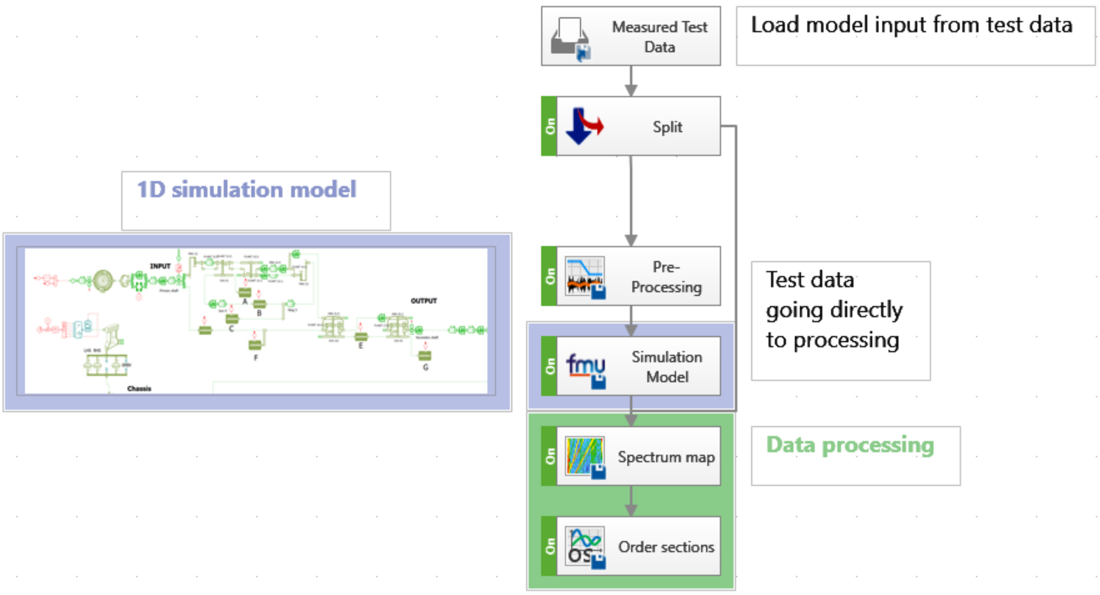

Figure 6 - Functional mock-up interface

The validation is done both at component level and full vehicle level as is described in the next chapter.

CPVA model and full vehicle booming model validation

A dedicated component test bench (Figure 7) is designed to extract the dynamic parameters of the CPVA. The bench can reproduce the dynamic from the combustion engine and is being driven in multiple rpm / torque conditions. Hysteresis effect and friction/damping parameters are estimated.

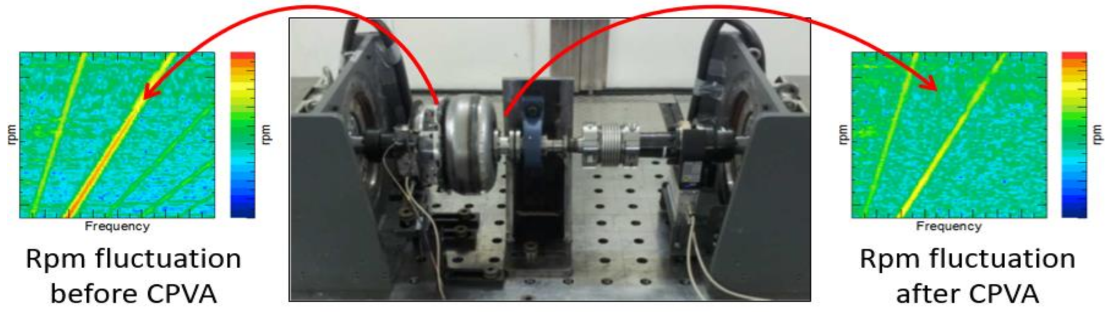

Figure 7 - Lock up damper with CPVA on the test bench - filtering effect overview

The CPVA component created based on the reverse engineering is exported as an FMU and driven using the measured test bench data. The measured bench torque is applied and the torque at the input and output of the CPVA is compared between test and simulation. Friction and damping parameters were updated using automatic optimization tool. It did not fundamentally change the dynamic response but was required to stay within the targeted accuracy level. Final correlation is shown in Figure 8:

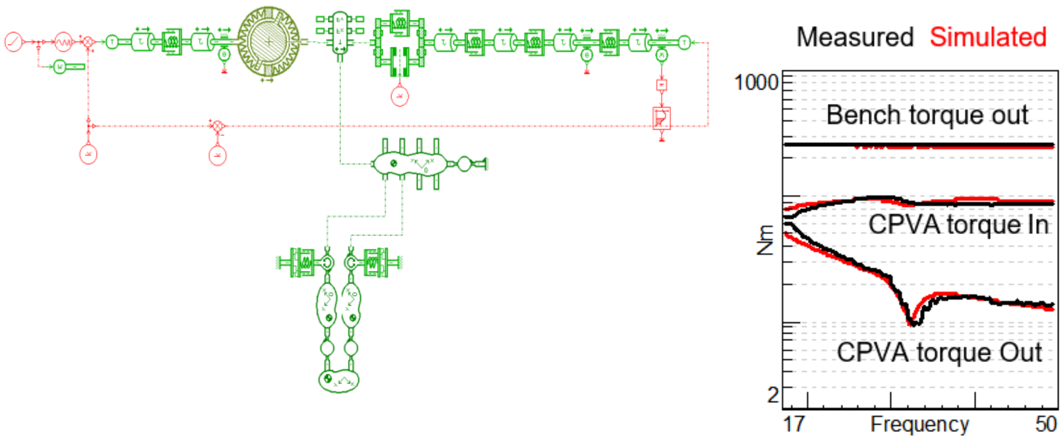

Figure 8 - CPVA component model and Order 2 correlation from bench testing

A series of booming measurement are done with full vehicle instrumentation. The validated CPVA model is included in the full vehicle model and the assembly is exported as an FMU. A new process is created using the dynamic engine torque as an input to the model and the validation is done by comparing the propeller and driveshaft torque order 2 fluctuation as well as the order 2 seat rail acceleration as shown in Figure 9:

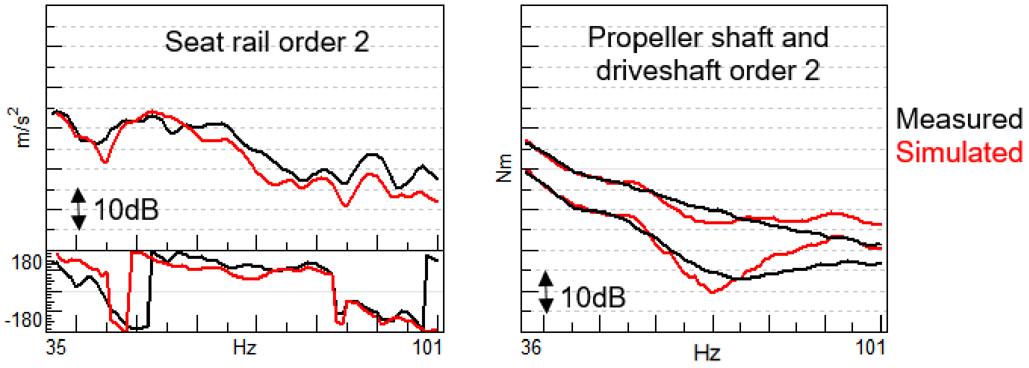

Figure 9 - Full vehicle model validation at seat rail acceleration, propeller shaft and driveshaft torque order 2

Although the correlation level achieved is very high (~3dB for the highest amplitude peak in the low frequency range) the accuracy starts deviating at higher rpm/frequency. This is mainly due to the physical limitations of a 1D model. At those frequencies, 3D effects become more and more important and we are hitting the boundary of what 1D can achieve.

Tip in model validation

As for the booming case, the full vehicle tip in model is validated by comparison with full vehicle operational test data. The measured throttle input, selected gear, engine speed and vehicle speed are used as an input to the simulation model. The controller integrated in the FMU is making use of these different inputs to generate the torque request at the driveline input. The driveshaft torque and seat rail acceleration are then compared between test and simulation in Figure 10:

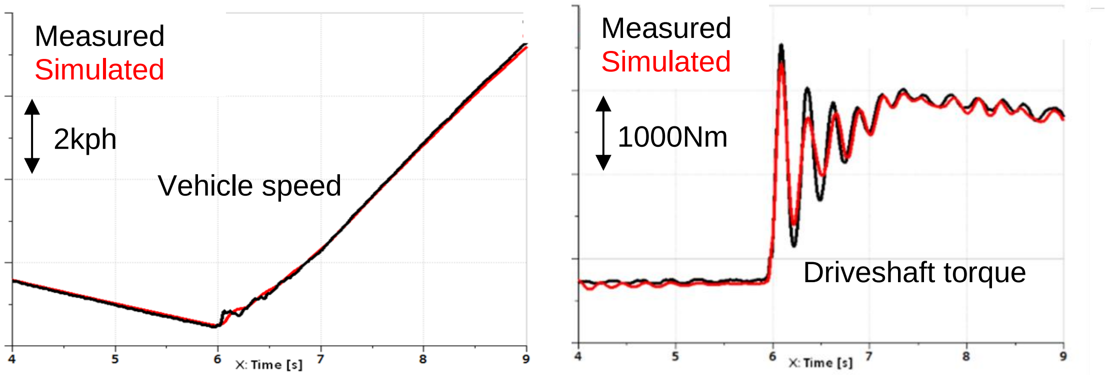

Figure 10 - Vehicle speed and driveshaft torque correlation

The measured and simulated vehicle speed and driveshaft torque are matching well both timing wise and in amplitude.

MODEL ANALYSIS AND UPDATING

Now that the simulation models are validated, we can study the actual effect of the CPVA in the driveline towards the seat rail vibration and answer to the question of the added value of this expensive component. An extra study is also done on parameter tuning for -booming- performance improvement. The CPVA length, radius and masses are modified to try and improve the performance. A difference of ~5dB can be seen at the max acceleration level at the seat rail in Figure 11:

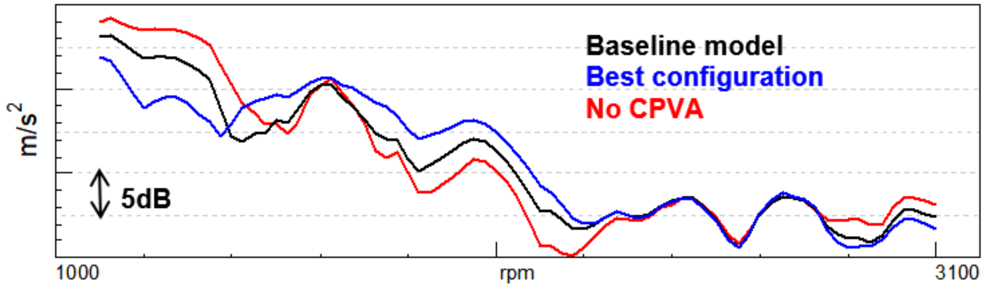

Figure 11 - Seat rail order 2 vibration sensitivity towards CPVA parameters

For tip in performance different control algorithms are tested in order to see the impact towards the shock at the seat rail acceleration (Figure 12). The original measured case is compared with two different settings: strong limitation on the reaction to avoid shock (case 1), and compromise response (case 2):

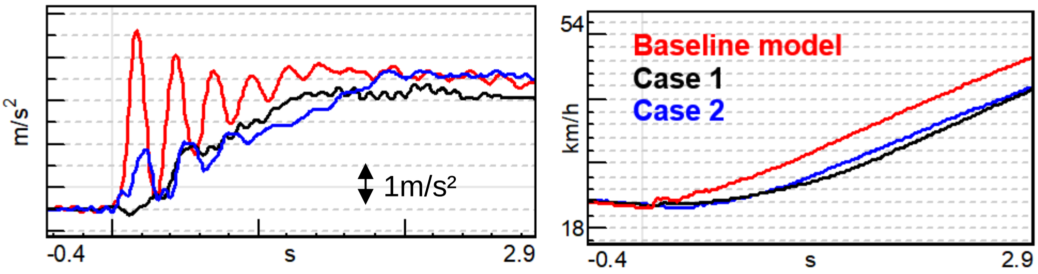

Figure 12 - Seat rail acceleration and vehicle speed for the different controls

CONCLUSION

This paper has shown the application of the Model Based System Testing approach to optimally balance the use of test and simulation for product engineering. The use of the FMI standard in a Test environment enabled the efficient combination of test data and simulation models for model validation and parameter studies. The parallel processing of both data sources ensured a perfect one to one comparison, while the use of one single environment saved the usual time-consuming data import/export process between different tools.

The predictive models developed in this work can be further used in a Multi Attribute Balancing scope to optimize the vehicle performance early in the development cycle [2].

REFERENCES

[1] Fabio Luis Marques dos Santos, Roland Pastorino, Bart Peeters, Cassio Faria, Wim Desmet, Luiz Carlos Sandoval G´oes, and Herman Van Der Auweraer. *Model based system testing: Bringing testing and simulation close together*. In Structural Health Monitoring, Damage Detection & Mechatronics, Volume 7, pages 91–97. Springer, 2016.

[2] Steven Dom, Jan Deleener, Tom Van Houcke, Tristan Enault, Nicolas Sabatier, Masanori Kawagoe, Tomohiro Yamaguchi - *Multi Attribute Balancing of NVH, Vehicle Energy Management and Drivability at Early Design Stage Using 1D System Simulation Model* – technical paper SAE international - 2019-26-0178