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VALIDATION OF A HUMAN-AND-HARDWARE-IN-THE-LOOP CONTROL ALGORITHM

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This study proposes the development and validation of a human-and-hardware-in-the-loop (H2iL) simulator based on an electric vehicle (simulator) with fully customizable parameters, on which any type of vehicle characteristics can be loaded.

During the first stage of the development, the built model will be customized with the parameters of a real vehicle and subjected to a number of basic traffic maneuvers (launch, braking, accelerating, gear change). The model's behavior and response will be compared to recorded data to check the reliability of the results.

Keywords: electric vehicle, powertrain, comfort, simulation, H2iL.

1. Introduction

Vehicle design, simulation and control are complex domains which evolved and expanded in tight connection with the increase in computing power. Currently there is a large number of simulation software available, all of them being very flexible, allowing accurate modeling and tackling various fields of interest. However, all these advantages come with a setback: the simulation software cannot offer real feedback on the vehicle comfort and driving pleasure when used in "offline" simulation. The now becoming classical HiL (hardware-in-the-loop) simulations offer a degree of feedback from individual components of the real system [1] and the modern H2iL (human-and-hardware-in-the-loop) systems give a limited feedback of the human operator [2], [3]. Despite becoming more complex and expensive, the H2iL systems offer data on simulated vehicle comfort and performances under simulated driving conditions. Current H2iL systems can be used on studying a large variety of parameters: the dynamic and the energetic performances, the passive and the active safety features, the in-car entertainment systems etc.

This project proposes the development of a powertrain H2iL simulator that can be used for the study of comfort for different maneuvers (launch, gearshift, tip-in). In order to choose the solution, an analysis of some of the simulators and methods used for vehicle dynamics is done.

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The “inDrive Simulator” project from Ingenieurgesellschaft Auto und Verkehr (IAV) was developed as a testing platform for future cars and can be used before the first prototype is even built [4]. This is a simulator capable of being drive in real-life road traffic which can be used by virtually any individual after appropriate training. Using the accelerator pedal, the brake pedal and the gear selector lever, the driver of the simulated vehicle will not control the actual drive of the base vehicle, but a virtual drive in a virtual vehicle. These virtual components are simulated as mathematical models on the computer and calculated in real time on the basis of the virtual vehicle’s operating state and driver instructions. The results will then provide the input values for controlling the vehicle’s longitudinal dynamics and for computing further target variables. The main advantage of this project is that it closes the gap between the early design and prototype testing because there is no need to finalize the target hardware at this stage. All that is required are mathematical models and a base vehicle, not necessarily the target vehicle, but one as similar as possible.

In this paper, the model used in the simulation is an evolution of the one developed in a previous research [5]. It has been proven that the first three gear changes can be simulated using a converted electric vehicle by following an imposed cycle or acceleration pattern within acceptable tolerances. The current model will be enhanced by considering elastic drive shafts and study their influence on the vehicle’s dynamic behavior during simple maneuvers. Matching the model’s response with the experimental recording will determine the parameters for the elastic drive shafts: stiffness and damper rating.

The final part of this paper will approach the study of more complex traffic maneuvers to further prove the model’s coherence. The traffic maneuvers used as input data for the simulation are based on real recordings made on test vehicles in real traffic conditions. The recordings have been made using an FA3403 Series triaxial accelerometer from FGP Sensors & Instrumentation [6]. The triaxial accelerometer was mounted on the interior side of the driver’s seat rail in order to ensure a good correlation with the acceleration applied on the driver.

2. Model presentation

The electric vehicle is based on a standard, front wheel drive, C segment car (figure 1). The drive force supplied by a 20kW electric motor is amplified by a reduction gear and then distributed to the front wheels through the elastic drive shafts. In this case, the reduction gear has a fixed ratio of 5 and the elastic drive shafts have been simulated using a rotary spring damper block with a preset stiffness of 228 Nm/degree and a damper rating of 0.5 Nm/(rev/min) [5].

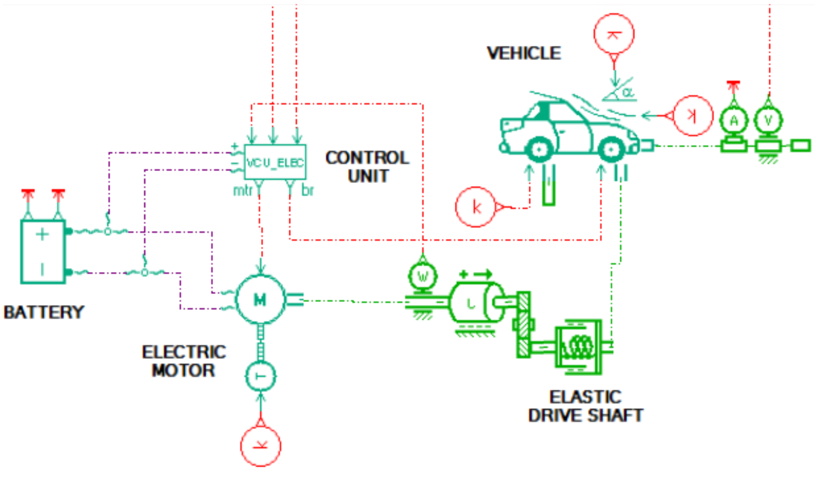

Fig. 1. Vehicle model

The control of the vehicle acceleration (figure 2) is based on a PID block that has a cycle input with data measured and stored under the form of time dependent acceleration. The signal from the source is filtered using a general transfer function block and then integrated to obtain speed. It is linked to the vehicle's electronic control unit with braking and acceleration controls. Using positive (0 to +1) and negative (-1 to 0) saturation blocks for separating the acceleration and braking signals, it adjusts the vehicle's speed by decreasing the error between the model's actual speed and the input speed from the cycle, in meters per second. The vehicle's velocity is obtained by integrating its longitudinal acceleration measured using high sampling frequency. The measured signal is first filtered using a 3rd order Butterworth low-pass filter with a cutoff frequency of 10Hz [8].

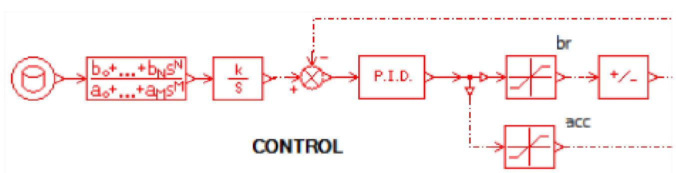

Fig. 2. Control using a PID block.

This study will investigate the influence of the stiffness and damper rating of the elastic drive shafts on the vehicle's capability of following the imposed cycle during various maneuvers.

Due to the influence of the differential on the drive shafts, an average, equivalent, value of the stiffness had to be used. It was calculated using left and right stiffness values from a basic, C segment vehicle. Due to the importance of the drive shaft stiffness in the simulation, the procedure for determining the equivalent value is presented below.

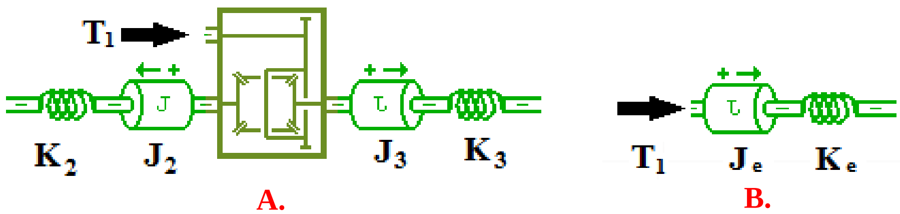

Fig. 3. A. – Differential with modeled elastic drive shafts;
B. – Equivalent model of two drive shafts.

Using the classic differential layout in AMESim, the equivalent motion equations are obtained:

$$T_1 = 2 \cdot T_2 = 2 \cdot T_3 \tag{1}$$

$$\omega_1 = \frac{1}{2}(\omega_2 + \omega_3) \tag{2}$$

where T_1 is the input torque, T_2 and T_3 are the left and right wheel torque, ω_1 is the input rotary velocity, ω_2 and ω_3 are the left and right drive shafts velocities.

The next step is to include the drive shafts, by adding inertias and rotary springs (figure 3 – A.), where $K_{2,3}$ are the left and right drive shafts stiffness' and $J_{2,3}$ are the left and right drive shafts moments of inertia. By applying dynamic principles for rotational motion for both left and right drive shafts:

$$T_2 - K_2 \cdot \theta_2 = J_2 \ddot{\theta}_2 \tag{3}$$

$$T_3 - K_3 \cdot \theta_3 = J_3 \ddot{\theta}_3 \tag{4}$$

where $\theta_{2,3}$ are the angular positions of the left and right drive shafts. For the equivalent model with two combined drive shafts (figure 3 – B.), the following equation is obtained:

$$T_1 - K_e \cdot \theta_e = J_e \ddot{\theta}_e \tag{5}$$

where K_e is the equivalent drive shaft stiffness, J_e is the equivalent drive shaft moment of inertia and θ_e corresponds to the angular position of the equivalent drive shaft and is equal to the angular position of the differential's sun gear θ_1.

By using (1) and (2) the following equation is obtained:

$$K_2 \cdot (2 \cdot \theta_1 - \theta_3) + J_2 (2 \cdot \ddot{\theta}_1 - \ddot{\theta}_3) = K_3 \cdot \theta_3 + J_3 \ddot{\theta}_3 \qquad (6)$$

Because the study focuses on the entire vehicle, the inertias of both elastic drive shafts are considered negligible [7] (figure 1); equations (4), (5) and (6) become:

$$T_3 = K_3 \cdot \theta_3 \qquad (7)$$

$$T_1 = K_e \cdot \theta_e \qquad (8)$$

$$K_2 \cdot (2 \cdot \theta_1 - \theta_3) = K_3 \cdot \theta_3 \qquad (9)$$

By applying (1), the following equivalence is obtained:

$$T_1 = 2 \cdot T_3 = 2 \cdot K_3 \cdot \theta_3 = K_e \cdot \theta_1 \qquad (10)$$

Finally, by using (9) and (10), the stiffness of the equivalent drive shaft is obtained:

$$K_e = \frac{4 \cdot K_2 \cdot K_3}{K_2 + K_3} \qquad (11)$$

3. Results

The simulation consists of subjecting the model to three complex maneuvers: a moderate acceleration with two gearshifts, a fast launch and a tip-in / tip-out maneuver. The first test of the simulation has been made for a simple cycle, with a moderate acceleration and two gearshifts, with default values for the elastic drive shafts’ stiffness and damper ratings. The acceleration has been measured with a 100 Hz sampling frequency on a front wheel drive, C segment car. The control with the PID block ensures a good correlation between the imposed vehicle acceleration and the simulated vehicle acceleration. As in [8], the acceleration profile is retarded with 0.2 s (figure 4), but it is following the imposed acceleration very closely (figure 5) during the first gearshift. This proves that the first two and, possibly, three gear changes can be simulated by the theoretical platform by following an imposed cycle or acceleration profile.

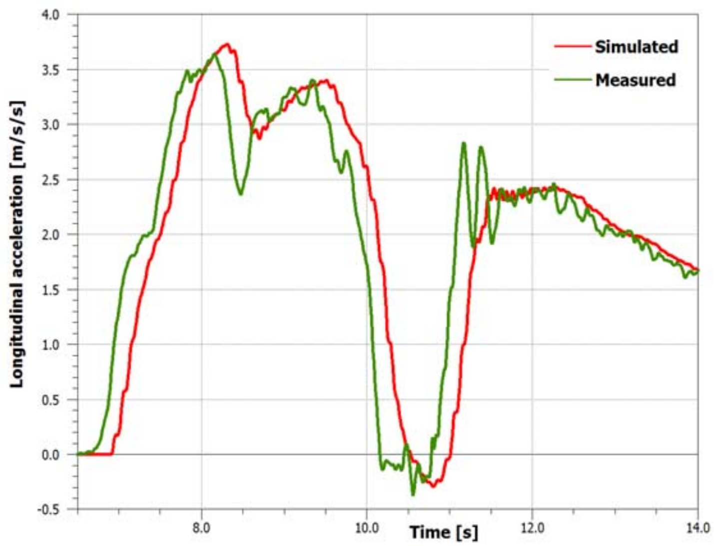

Fig. 4. Imposed and simulated vehicle acceleration

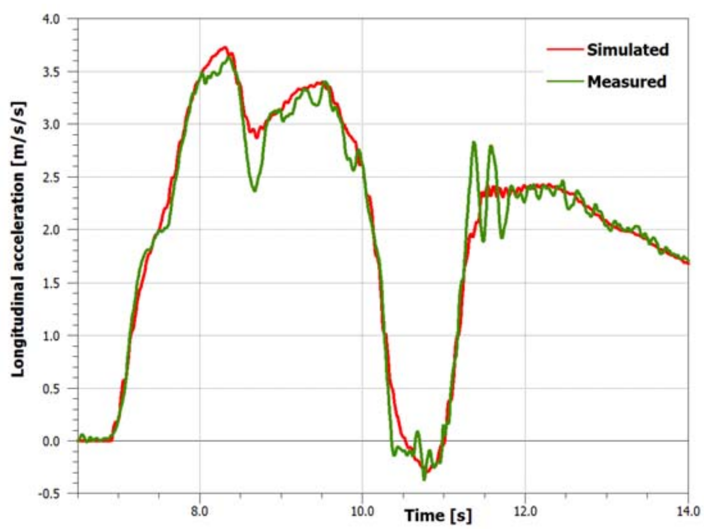

Fig. 5. Imposed vehicle acceleration (retarded with 0.2s) and simulated vehicle acceleration

The next step is to investigate the influence of the elastic drive shafts on the vehicle's dynamic behavior. This is done by running simulations with a discrete variation of the stiffness and damper rating values, considering a reference value and a ± 10% variation for high and low values. The reference value for the elastic drive shaft stiffness was considered 228 Nm/degree, with 23 Nm/degree steps above and below this number. An average value of 0.5 Nm/(rev/min) was selected for the damper rating, with a 0.49 Nm/(rev/min) positive and negative variation. The simulation spans over 13.5 seconds and covers the first two gear shifts. During this time, the vehicle reaches a top speed of 72 km/h and a maximum acceleration of 3.8 m/s^2.

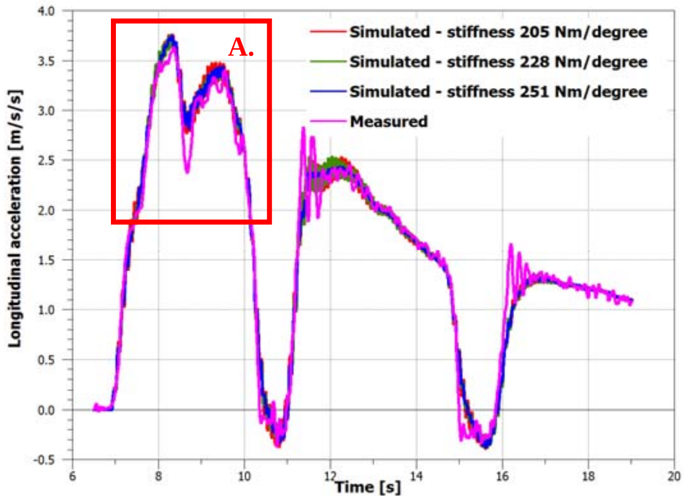

Fig. 6. Influence of the stiffness variation on longitudinal acceleration

From the three considered cases, the best results are obtained with the highest value for the stiffness. It is also noticeable that the influence of the stiffness on the longitudinal acceleration is gear dependent. The impact of the stiffness on the acceleration increases in the higher gears (figure 7).

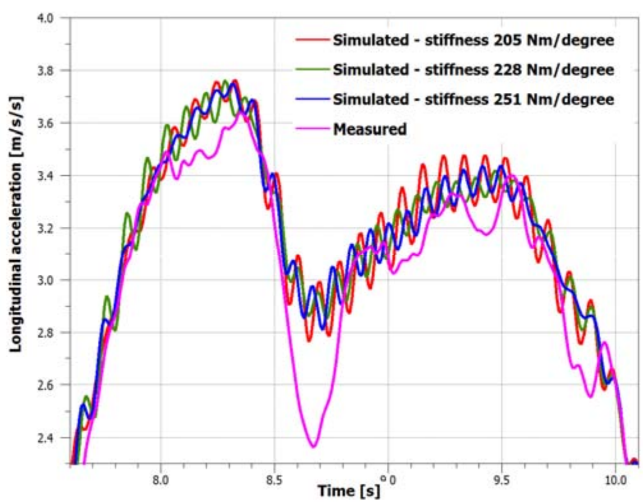

Fig. 7. Influence of the stiffness variation on longitudinal acceleration – detail A.

Using the stiffness of 251 Nm/degree, a study of the damper rating influence on the longitudinal acceleration has been made (figure 8). At low damping rates the controlled process input is unstable (i.e. its output diverge) as can be seen in figure 8 after the second 11. Stabilization of response and a good correlation with the measured data is obtained with a damper rating of above 0.5 Nm/(rev/min).

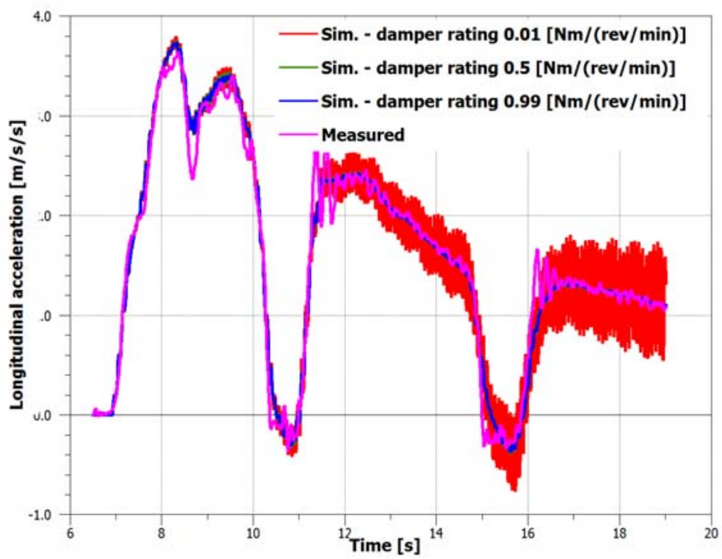

Fig. 8. Influence of the damper rating variation on longitudinal acceleration

The second maneuver, the fast launch, is a rapid start with the acceleration pedal depressed 50%. The whole recording takes 4 seconds. Subjecting the model to this maneuver proved that the electric motor of the theoretical vehicle can reach a maximum acceleration of 4.1 m/s^2 and cannot cope with the high demanded acceleration of the imposed cycle, which is 5.8 m/s^2 (figure 9 – Case 1).

In order to reach a higher acceleration, the gear ratio of the reduction gear was increased from 5 to 7. Thus, a higher acceleration was obtained, but the system began to oscillate during the second half of the simulation (figure 9 – Case 2). In order to stabilize the system, a modification of the elastic drive shaft's parameters was necessary. The stiffness was increased from 251 to 300 Nm/degree and the damper rating from 0.5 to 1.5 Nm/(rev/min) (figure 9 – Case 3).

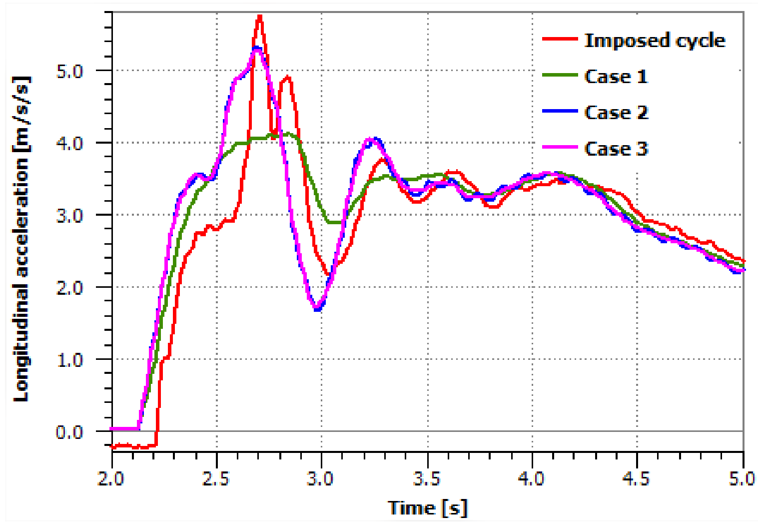

Fig. 9. Fast launch with 50% acceleration pedal

The model was also subjected to a tip-in/tip-out type maneuver. It consists of a deceleration followed by a 2 second period of high acceleration (3 to 4m/s^2) (figure 10). During this period, the vehicle's speed increases from 14km/h to 34km/h. The model reacts very well to the regime change, following very closely the imposed acceleration. The difference in the peak speed occurs because of the way the speed is calculated (by being integrated from acceleration values).

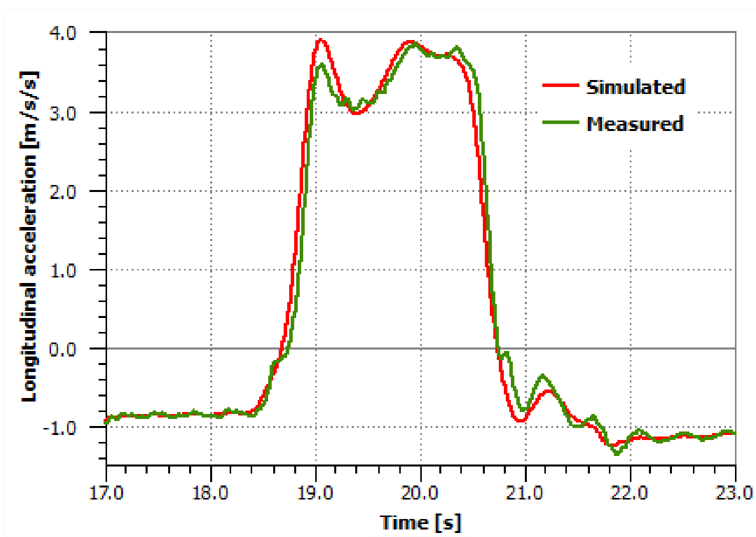

Fig. 10. Tip-in / tip-out maneuver

4. Conclusions

Using the designed model, it has been shown that an electric vehicle, as described in the introduction, is feasible and can be used for the study of dynamic maneuvers (launch, gear change). In order to have a good correlation between the simulation and the measured data, a higher stiffness value has to be used, up to 300 Nm/degree during the fast launch. The investigation of using a torsional damper on the drive shaft leads to the conclusion that a rating higher than 0.5 Nm/(rev/min) will ensure higher accordance of the simulation results with the measured data. A value of 1.5 Nm/(rev/min) was needed to help stabilizing the model during the fast launch. This leads to the conclusion that, if the real test vehicle will be built, an easy way of adjusting its stiffness and damper rating will have to be developed.

In a future work, the model will be further developed by introducing front and rear axle suspension and mounting the vehicle's electric motor on elastic mounts.

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R E F E R E N C E S

[1]. *Schuette, H., Waeltermann*, P., Hardware-in-the-Loop Testing of Vehicle Dynamics Controllers – A Technical Survey, SAE paper no. 2005-01-1660, SAE 2005 World Congress & Exhibition, Detroit, USA, April 2005.
[2]. *Morse, P.,* Right on Cue, Automotive Testing Technology International, September 2012, Stuttgart, Germany, pp 28.
[3]. *Schreiber, U., Todtermuschke, K., Back, O.,* Experiencing and definition of the real shifting confort at the virtual synchroniser in the virtual powertrain, 8th CTI Symposium, Innnovative Automotive Transmissions, Berlin, 2009.
[4]. *Jaensch, D.,* Driving a Car That Does Not Even Exist Yet, Automotion, Edition 02, April 2008, Berlin, Germany, pp 4.
[5]. *Stoica, I., Bataus, M.,* Advantages of developing an Electric Vehicle for Advanced Real Life Vehicle Simulations, 2nd International Conference, Motor Vehicles and Transportation, Timisoara, 2012.
[6]. ****,* FA3403 Series Triaxial Accelerometer Datasheet, FGP Sensors & Instrumentation, 2005.
[7]. *Eller, B.,* Contribution a l'Etude Amont de l'Agrement de Conduite de Systemes de Propulsion de Nouvelle Generation, These de doctorat, Ecole Centrale de Nantes, 2011.
[8]. *Stoica, I., Bataus, M., Croitorescu V.,* Human-and-Hardware-in-the-Loop Simulator for Studying Vehicle Longitudinal Dynamics, SIMEX, Brussels, 2013