Generation and validation of a real time multibody model for off-highway vehicles, aiming at the design of suspension systems

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Abstract.

This study presents the logical process followed to develop a multibody model of an off-highway vehicle with independent suspension that can run in real-time regardless of the imposed maneuver. This multibody model has been developed to support in the next future the study, design, and optimization process of different hydro-pneumatic suspension systems for the vehicle.
The starting point was the development of the complete 3D multibody model of the vehicle of interest in Simcenter Amesim®, with particular focus on the suspension mechanism. This model was used as the reference for the real-time one. The final real-time model was developed step by step in the same simulation environment and validated by comparing the complete 3D multibody model dynamic response referring to standard maneuvers (ISO 5008, moose test, lateral bumps test) after the introduction of inertial contributions, tires model, and road profiles.
Lastly, it was possible to adjust some parameters to make the model able to run with a fixed step solver to support model-in-the-loop simulations. The main result obtained is a significant reduction of simulation time from several minutes to a few seconds. This is a clear benefit that can be further emphasized in the optimization processes of hydro-pneumatic suspension systems.

Keywords. Independent suspension, off-highway, real-time model, multibody model.

NOMENCLATURE

m_{sp}	vehicle sprung mass
$m_{wheel_{f/r}}$	front/rear wheel mass
$m_{strut_{f/r}}$	front/rear wheel structure mass
m_{axle}	front axle mass

$m_{wishbone_{low/up}}$	lower/upper wishbone mass
$m_{steer_{link}}$	steering link mass
$m_{steer_{cyl}}$	steering cylinder mass
w	vehicle wheelbase
w_f	distance between vehicle CoG and front axle
$F_{z_{susp_{f/r}}}$	vertical load on front/rear axle
$F_{z_{tire_{f/r}}}$	vertical load on front/rear tire
$R_{w_{f/r}}$	free radius of front/rear wheel
$R_{w_{load_{f/r}}}$	loaded radius of front/rear wheel
$k_{z_{tire_{f/r}}}$	vertical stiffness of front/rear tire
$perc_{sprung}$	sprung percentage for intermediate masses
$x_{axle_{CoG}}$	x coordinate of axle CoG
g	gravitational acceleration

1. INTRODUCTION

To study the dynamic behavior of a vehicle and the interaction with the vehicle subsystems, primarily suspension, steering or braking systems, the multibody representation of the vehicle is one of the most effective approaches, and it is adopted always more often by researchers and engineers. Nevertheless, the multibody modeling approach presents some issues, for example the modeling and the parametrization of the tire-soil contact or the general parametrization of some unknown information, as the friction contributions, and the computational time needed for the simulation. In this work, the multibody modeling approach is applied to an agricultural tractor, a very complex vehicle per se, with the aim to create a virtual platform where it will be possible to integrate very detailed models of several architectures of hydro-pneumatic suspensions system, in order to compare and optimize different design.

Modern agricultural tractors are sophisticated vehicles, and are always more often, even the smaller ones, equipped with a hydro-pneumatic suspension at the front axle. Hydro-pneumatic suspension is undoubtedly the best solution for this vehicle [1], where integration of the suspension system in the already present hydraulic circuit, which controls several functions of the tractor, seems the more efficient solution. With the freedom given by the chance to adopt different architectures, damping, and stiffness characteristics for the suspension system, comes the responsibility to choose the best solution (a compromise between comfort and safety) and tuning. That is a real challenge and often involves long and exhausting experimental tests on ISO tracks and fields, to test several dynamic conditions [2]. It is recognized in the literature that, though extensive knowledge exists on suspension systems different layouts, and tuning, there are topics where improved understanding is needed. In particular, the comparison between alternative concepts on the architectures of the suspension and guidelines on the process of the suspension "tuning", because generic tuning strategies have not been widely developed [3]. In this context, a virtual tool, able to simulate the system and perform the requested tests would be very helpful, not only to compare different architectures or to carry out the tuning, but also to reach a better insight of the suspension and vehicle behavior. In fact, considering that the achievement of an optimal behavior is a very challenging task, a full understanding of underlying system dynamics would be very useful. To do this, the virtual model must give a detailed description of both the suspension system and the vehicle. The integration of the hydraulic model of the suspension with the kinematic multibody model of the vehicle, though numerically challenging, seems the right way to approach the problem. An example of a detailed multibody model of a tractor may be found in [4] and [5]; great attention has been focused here on the description of the tire-terrain contact characteristics and the correlation with experimental data. Biral et al. in [6] and [7] analyzed a semi active hydro-pneumatic suspension system for a tractor with the focus of optimizing the control logic of the system. Here the experimental and numerical analysis is developed by referring to a quarter car model of the vehicle. In this work, the focus was the design of opportune control logic for the suspension and the analysis of the main limits of the different semi-active control strategies, that were found to be due to the damper dynamics and the non-linear behavior of the hydraulic suspension, in particular due to the friction forces acting on the piston-cylinder assembly. However, recognizing the importance of considering the pitching motion of the tractor in the suspension analysis to develop an optimized control strategy, the authors proposed in [8] a half car model of the vehicle coupled also with a more detailed thermo-hydraulic model of the hydraulic suspension system.

It is doubtless difficult to develop a detailed model of the hydro-pneumatic suspension interacting with a full car model of an agricultural tractor but, at the same time, only with this approach all the dynamic motions of the tractor are considered. Another challenging aspect is that once the virtual tool is ready, simulation time may be discouraging in utilizing it, especially in the industry daily work, and particularly if the idea is to apply an optimization process to guide the design and choice of the suspension characteristics. For this reason, this work focuses on developing a simplified off-highway vehicle model able to reduce simulation time, while keeping the same steady-state and dynamic behavior of a more complex and detailed model.

A tractor multibody model was developed in Simcenter Amesim® starting from an MSC Adams® multibody reference one, which was previously validated with experimental tests. We chose Simcenter Amesim® because it allows both a multibody modeling and a more simplified one leveraging the multi-environmental libraries within the software, which is suitable for the creation of a Real Time (RT) model. Once the multibody model was aligned with the reference one, an embedded Amesim® tool has been exploited to create several look-up tables that replicate the kinematic behavior of the suspended axle. Some intermediate models were then used to consider the inertial effects even in the simplified model. We subsequently employed another software tool to analyze the system stability and modify some parameters to match the frequency range of interest of a fixed step solver. This is a fundamental step to make the system stable with this kind of integration and so with RT simulation.

Lastly, some standard maneuvers have been simulated with both the multibody model and the Real Time one. In this way, it was possible to compare the results and validate the entire methodology presented.

2. OFF-HIGHWAY VEHICLE MODEL IN SIMCENTER AMESIM®

The off-highway vehicle taken as reference for the present study is shown in Figure 2.1. It is a 116 kW tractor with independent suspension on the front axle, while the rear one is not suspended.

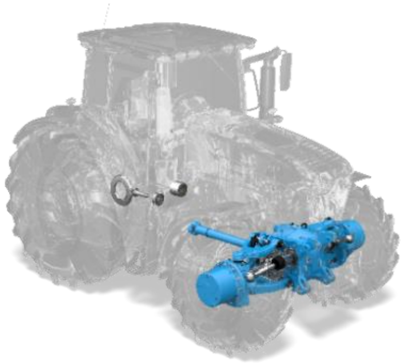

Figure 2.1. Tractor with a suspended axle used as reference

These vehicles, or the tractors in general, are specifically designed to deliver a high tractive effort (or torque) at slow speeds, to haul a trailer or machinery such as that used in agriculture, mining, and/or construction. Usually, a ballast is used so that the load is not too unbalanced on one axle concerning the other when machinery is coupled with the vehicle. Vehicles with an independent suspension on the front axle have both a more comfortable and safer driving on-road and performance optimization on the field.

The presented activity focuses on the realization of the RT model having as a starting point the detailed lumped parameter multibody model of the examined vehicle. The model has been developed with Simcenter Amesim®. The multibody model was used not only as a starting point for the development of the RT simplified one, but also as a benchmark for the RT model validation. Several main components of the vehicle have been identified to facilitate both the multibody model development and comprehension.

These components are indeed easily distinguishable and have been interfaced to obtain the detailed vehicle model. These main components are:

- the vehicle sprung mass;
- the steering cylinder;
- the wheel upright and hub (unsprung mass);
- the detailed model of the front axle with the examined independent suspension;
- the Pacejka tire and road model;
- the capability of considering the connection with the hydraulic circuit of a hydro-pneumatic suspension.

These elements are highlighted in Figure 2.2, where the block diagram of the multibody model is reported.

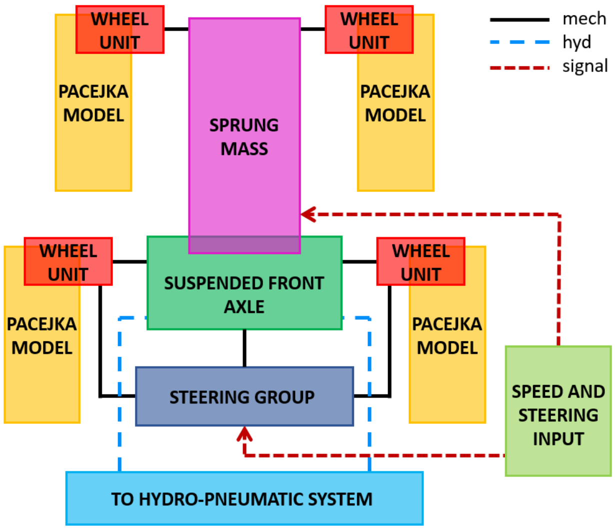

Figure 2.2. Scheme of the Simcenter Amesim® multibody model

As anticipated, the model has been developed in a modular way so that it is easy to modify or replace one or more components. The modification of the suspension geometry or even the replacement of the whole suspension type could be very interesting to compare the different suspension performances.

The study reported in this paper presents the case of a double-wishbone suspension. Nevertheless, it would be sufficient to replace the 3D model of the suspension kinematics to evaluate the performance of another type of suspension on the same vehicle.

Moreover, the parametric characteristic of the model allows to vary all the vehicle parameters as masses, inertias, and/or geometries in a very easy way.

In this way, the numerical model is adjustable to both the substitution of the examined suspension and the modification of the whole vehicle size.

This lumped parameter multibody model developed in Simcenter Amesim® was then compared with another multibody model, previously developed in MSC Adams®. This reference model was created and experimentally validated thanks to on-road and on-field tests by the off-highway engineering department of Dana Italia. The impossibility of simplifying the MSC Adams® model to make it able to run in real time and the lack of capability of including the detailed model of the hydraulic circuit led to the need for developing the present Simcenter Amesim® model.

The results comparison was carried out considering several simulations to validate both the kinematic and dynamic aspects of the vehicle model.

While the suspension kinematics turned out to be perfectly aligned between the two models, some differences have been noticed in the vehicle dynamic simulations, which were the ISO 5008 test [9] and the asymmetric bump test.

In particular, the latter highlighted different dynamic responses of the wheels mainly due to the usage of different wheel-soil contact models in the two software. In fact, while MSC

Adams® uses a contact surface, the Simcenter Amesim® limits the choice between a single contact point and a model called “tandem egg”, where two contact points are considered.

Despite the possible difficulty of aligning two different contact models, a further investigation on this particular topic has been started with Siemens (i.e., the developer of Simcenter Amesim®) support. This activity is still ongoing and may be the subject of future work.

Anyway, the presented methodology in this work to develop a RT vehicle model is valuable, regardless of the differences in the contact model.

3. DEVELOPMENT OF THE SIMCENTER AMESIM® REAL TIME MODEL

3.1. Kinematic Tables generation and validation

The first logical step of the proposed methodology is the creation of several kinematics lookup tables of the suspension and their subsequent numerical validation. These lookup tables are necessary to replace the 3D suspension kinematics model, while ensuring the correct response to pure bounce, pure roll, or pure steer input.
A Simcenter Amesim® tool named K&C (Kinematics and Compliance) Data Generator was used for these lookup tables creation. This tool works as a virtual test bench for the examined suspended axle. The inputs for the K&C Data Generator are thus the suspension type and the hardpoints coordinates. This tool already provides a hardpoints template for McPherson and Double Wishbone suspensions (Figure 3.1), as the examined one, but it is also possible to create a template for other types.

The hardpoints coordinates must refer to the left wheel and must be expressed in the chassis reference frame. With particular reference to Figure 3.1, the hardpoints to be defined for a Double Wishbone suspension are the following:

- A: wheel center
- B: center of tire contact patch
- C1: upper wishbone link with chassis (external)
- C2: upper wishbone link with chassis (internal)
- D: spindle link with lower wishbone
- E1: lower wishbone link with chassis (external)
- E2: lower wishbone link with chassis (internal)
- M: spindle link with tie rod of steering system
- N: tie rod link with steering rack body or cylinder
- K: spindle link with upper wishbone
- R1: damper lower link with spindle
- R2: dumper upper link with chassis
- W: external point defining the wheel rotation axis

Once the information regarding the suspension geometry is entered, it is necessary to set the variation range for vertical wheel displacement and steering rack displacement and the intermediate steps to analyze the suspension behavior.

This step defines the total number of combinations of vertical displacement and steering displacement to consider, so that is the total number of batch runs for the virtual bench. The total number of runs is thus $n \cdot m$ where m is the dimension of the vertical displacement vector and n is the dimension of the steering rack displacement vector.

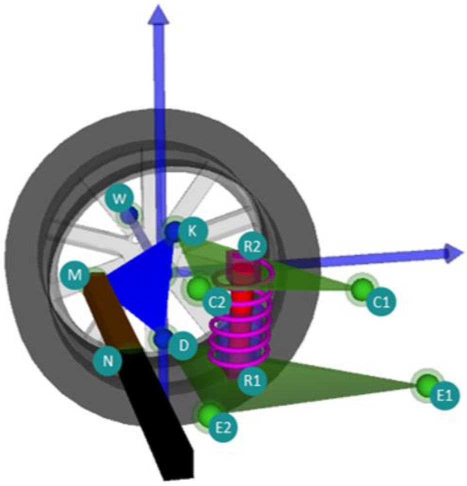

Figure 3.1. K&C template for double-wishbone suspension

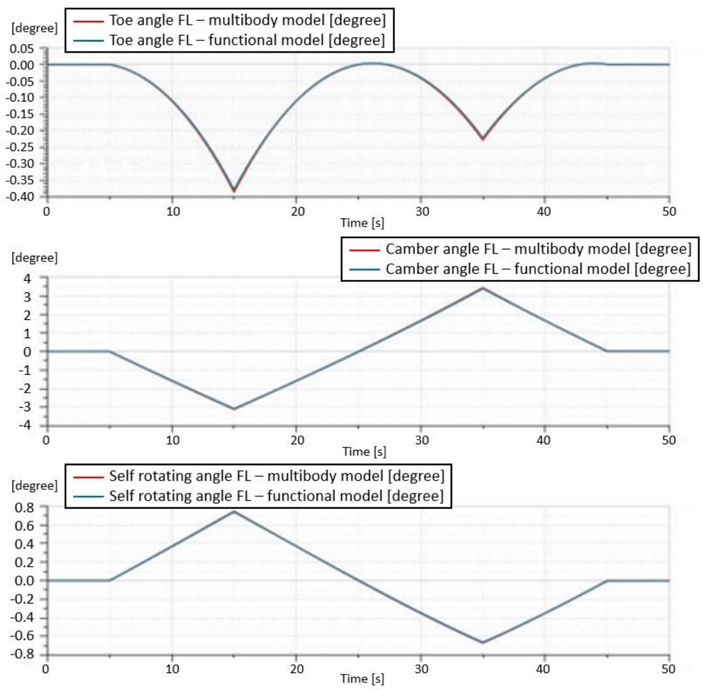

Figure 3.2. Kinematics comparison between multibody (red) and functional model (blue)

The outputs of this tool are the 3D lookup table for the suspension kinematics parameters as a function of the vertical wheel displacement and the steering rack displacement.
These kinematics parameters are the toe angle, the camber angle, the dive angle, X and Y displacement, the damper length, and the damper length variation.
There are 14 lookup tables, 7 for the left wheel and 7 for the right one. As anticipated, the subsequent step is the numerical validation of the newly created kinematic lookup tables.

Both the starting multibody model of the vehicle and the simplified functional model where the kinematics lookup tables are entered were considered in the same Simcenter Amesim® sketch to facilitate the comparison.

This functional model is based on a specific sub-model (VDCAR15DOF1 [10]) available in the Vehicle Dynamics library of the software. The input elements and sensors of the K&C demo are replicated for both models. This allows considering the same combination of pure bounce, pure roll, and pure steer for both of them. Furthermore, both the models are clamped.

The comparison of some suspension kinematics parameters of the two models with a pure bounce input is depicted in Figure 3.2, where toe, camber, and caster angles are reported for the left wheel. The results for the right wheel are obviously similar, so they were not reported in figure.
It is easy to note that the graphs are almost perfectly aligned. Similar results were obtained with a pure roll input, a pure steer, and with a combined input.

The kinematics lookup tables are thus validated, and the kinematics behavior of the functional model is aligned with the multibody model one.

3.2. Vehicle Dynamics integration

The following steps regard the introduction of the dynamic aspects in the functional model, like gravity, masses, inertias, and the tire model.

Firstly, all the information related to masses and inertias was introduced into the Global Parameter. This passage was repeated for the following main system components:

- sprung mass m_{sp}
- tire and wheel structure m_{wheel_f}, m_{wheel_r}, m_{strut_f}, m_{strut_r}
- front axle m_{axle}
- suspension wishbones $m_{wishbone_{low}}$, $m_{wishbone_{up}}$
- steering cylinder and links $m_{steer_{cyl}}$, $m_{steer_{link}}$

The first model considered for the vehicle dynamics validation was a Comfort Bench Model, which was used only for the analysis of the vertical dynamics. Both the multibody model and the functional one are subjected to a frequency sweep displacement of the front wheels to evaluate the frequency response.

In addition to the entering of masses and inertias into the VDCAR15DOF1 element of the functional model, it was necessary to also include the tire vertical stiffness, the tire lateral and longitudinal stiffness, the road vertical excitation and its connection to the vehicle. The tire parameters were entered in the model from the datasheet or the correspondent **.tir* file.

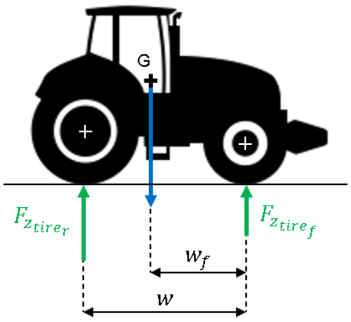

Figure 3.3. Calculation of vertical tire forces

Moreover, the equilibrium for the functional model was calculated. The vehicle front and rear vertical load were evaluated using the vehicle wheelbase (Figure 3.3).

$$F_{z_{susp_f}} = (w - w_f)/w \cdot m_{sp} \cdot g \tag{3.1}$$

$$F_{z_{susp_r}} = w_f/w \cdot m_{sp} \cdot g \tag{3.2}$$

From these, the vertical tire forces were obtained by adding the weight of the tire and wheel assembly. These forces and the tire vertical stiffness were thus used to evaluate an estimate of tire loaded radius.
Note that at first not all masses were considered in this calculation for simplicity.

$$F_{z_{tire_f}} = \frac{F_{z_{susp_f}}}{2} + \left(m_{wheel_f} + m_{strut_f}\right) \cdot g \tag{3.3}$$

$$F_{z_{tire_r}} = \frac{F_{z_{susp_r}}}{2} + \left(m_{wheel_r} + m_{strut_r}\right) \cdot g \tag{3.4}$$

$$R_{wload_f} = R_{w_f} - \frac{F_{z_{tire_f}}}{k_{z_{tire_f}}} \tag{3.5}$$

$$R_{wload_r} = R_{w_r} - \frac{F_{z_{tire_r}}}{k_{z_{tire_r}}} \tag{3.6}$$

Some of the results obtained from the Comfort Bench model are depicted in Figure 3.4, where there is a comparison between the functional model (in blue) and the multibody one (in red). The alignment is acceptable but not perfect, probably because of the simplification in the calculation above.

The following step provides for the substitution of the longitudinal and lateral stiffness values with the Pacejka 2002 model for both the front and rear axle and the subsequent driving on road. The vertical excitation was replaced with the 3D road model and a properly calibrated velocity controller (PID) was integrated. The Pacejka model parameters were taken from the **.tir* tire files.

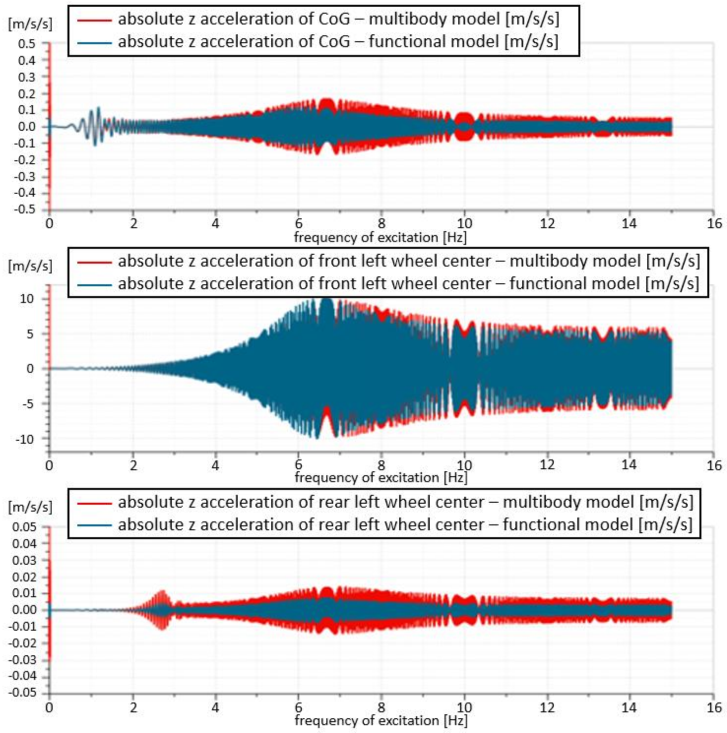

Figure 3.4. Comfort bench comparison between multibody (red) and functional model (blue)

The standard maneuver considered for this further model comparison was the asymmetric bump. This because this test is short but at the same time quite demanding for the model.

A refinement of equilibrium calculation for the functional model with the inclusion of the other masses was considered necessary before continuing. Firstly, the new position of the system center of gravity was evaluated considering, as said, not only the vehicle sprung mass, but also the front axle, the suspension wishbones, and the steering links. All the added masses were considered as centered in the center of gravity of the front axle for simplicity. The system was thus simplified to a system of two particles (Figure 3.5) where the center of gravity is defined by (3.7).

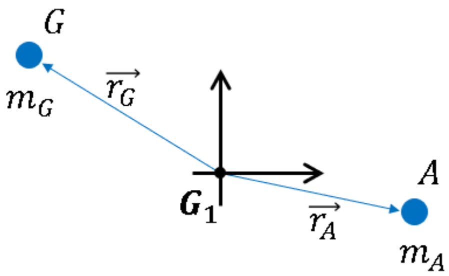

Figure 3.5. Two particles system

$$\begin{pmatrix} x_{G1} \\ y_{G1} \\ z_{G1} \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ 0 \end{pmatrix} = \frac{\sum m_i \vec{r_i}}{\sum m_i} = \frac{m_G \vec{r_G} + m_A \vec{r_A}}{m_G + m_A} \tag{3.7}$$

The terms m_G and m_A are defined in (3.8)-(3.9).

$$m_G = m_{sp} \tag{3.8}$$

$$m_A = m_{axle} + perc_{sprung} \cdot (m_{wishbone_{up}} + m_{wishbone_{low}} + m_{steer_{link}}) \tag{3.9}$$

The term $perc_{sprung}$ was introduced to consider only the sprung percentage of the wishbone and link masses. In fact, these components are not completely sprung or unsprung due to their intermediate position, but the percentage value is not known. However, this percentage was set to 50% after a brief sensitivity analysis. Equations (3.10)-(3.11) show the modification of the front and rear axle loads.

$$F_{z_{susp_f}} = \left((w - w_f) \cdot m_{sp} + \left(w - x_{axle_{CoG}}\right) \cdot \left(m_{axle} + 2 \cdot perc_{sprung} \cdot \left(m_{wishbone_{up}} + m_{wishbone_{low}} + m_{steer_{link}} \right) \right) \right) \cdot \frac{g}{w} \tag{3.10}$$

$$F_{z_{susp_r}} = \left(w_f \cdot m_{sp} + x_{axle_{CoG}} \cdot \left(m_{axle} + 2 \cdot perc_{sprung} \cdot \left(m_{wishbone_{up}} + m_{wishbone_{low}} + m_{steer_{link}} \right) \right) \right) \cdot \frac{g}{w} \tag{3.11}$$

Similarly, (3.12) shows the modification of the vertical tire force.

$$F_{z_{tire_f}} = \frac{F_{z_{susp_f}}}{2} + \left(m_{wheel_f} + m_{strut_f} + \left(1 - perc_{sprung}\right) \cdot \left(m_{wishbone_{up}} + m_{wishbone_{low}} + m_{steer_{link}} \right) \right) \cdot g \tag{3.12}$$

Finally, all the inertias were modified using the Huygens–Steiner theorem by referring to the new center-of-mass frame.

After entering all these modifications into the Global Parameter and into the VDCAR15DOF1 element, it was possible to analyze the simulation results of the asymmetric bump test.
The comparison between the functional model behavior (in red) and the multibody model one (in blue) regarding the most interesting results is depicted in Figure 3.6. These results are the front wheel center positions and the vehicle Euler angles (i.e. roll, pitch and yaw).
The results qualitative comparison turned out to be satisfying. Note that this was only a qualitative comparison, while a quantitative one will be presented for the final RT model.

The Amesim® functional model was thus considered validated concerning the Amesim® multibody one.

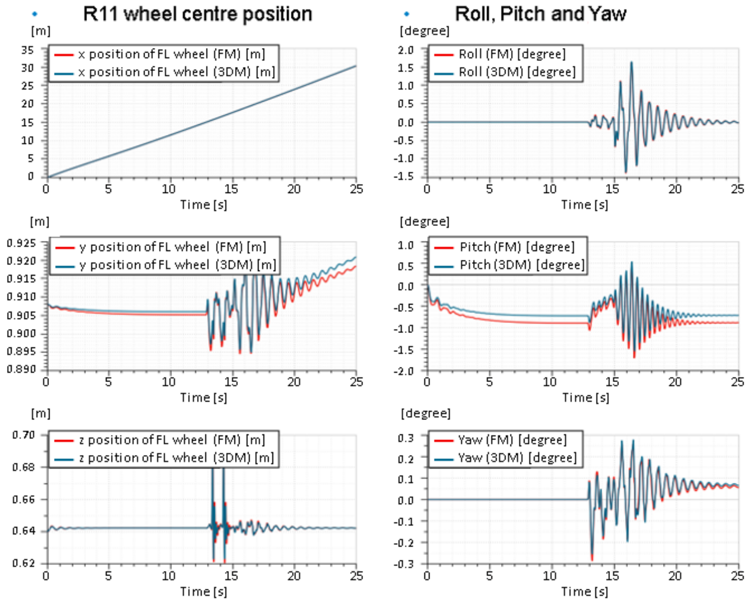

Figure 3.6. Drive on road comparison between multibody (blue) and functional model (red)

3.3. Model modifications for fixed step simulations

As said, the last step regards the modification necessary to make the functional model able to run in real time. In fact, some quantities must be adapted to match the frequency range of interest of a fixed step solver. The Performance Analyzer tool of Simcenter Amesim® was used to do so. The tool template is reported in Figure 3.7.

Switching from a variable time step solver to a fixed time step solver is a pre-requisite for using a model for Real-Time study. To ensure numerical stability and real-time target limitation, the value of the fixed time step must be in a dedicated range that depends on the model.

This tool allows visualizing all the information regarding the numerical stability region of the system and the critical modes for the selected integration method. Firstly, it is necessary to set an appropriate number of linearizations during the run (in orange in Figure 3.7). After selecting a guestimate of the fixed step solver (in blue), the tool shows the stability region for the eigenmodes (in pink). After the simulation has run, the user has access to the modes for each linearization time, to the maximum time step for the given solver method (in light blue), and to the time and contributors causing the maximum problem (in green and red) [10].

This information is necessary to modify these contributors of the modes causing troubles and then to choose the maximum fixed time step. Once the parameters have been appropriately modified, it is possible to test the Real Time model.

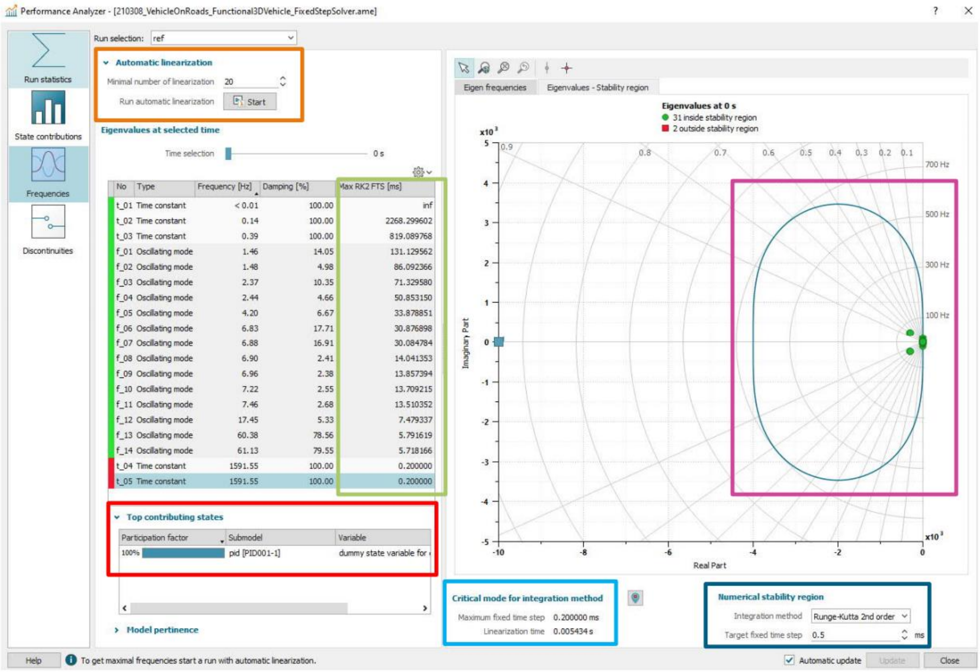

Figure 3.7. Simcenter Amesim® Performance Analyzer

4. REAL TIME VEHICLE MODEL

The vehicle Real Time model created following the previous steps was used to simulate three standard maneuvers, which are:

- Asymmetric bump test
- ISO5008
- Moose Test (Double line change)

The RT model results were then compared with the starting multibody model ones.

The figures below depict the comparison between these two models' general behavior with the asymmetric bump test. Figure 4.1 shows the positions of the front left wheel center in the RT model compared to the 3D model ones. It is easy to note that the results are quite well aligned. The small differences in the y position graph can be considered negligible, since the lateral shift is in the order of millimeters after tens of meters of longitudinal drive.

Figure 4.1 shows also the comparison of the three Euler angles in the two considered models. As before, the results are generally well aligned. The small offset in the pitch graph can be explained by a remaining small inaccuracy in the system mass identification or balancing. Nevertheless, the results remain satisfactory.

To confirm that, Fig. 4.2 depicts a histogram of the percentage errors of some of the most interesting results of the RT model concerning the multibody one.

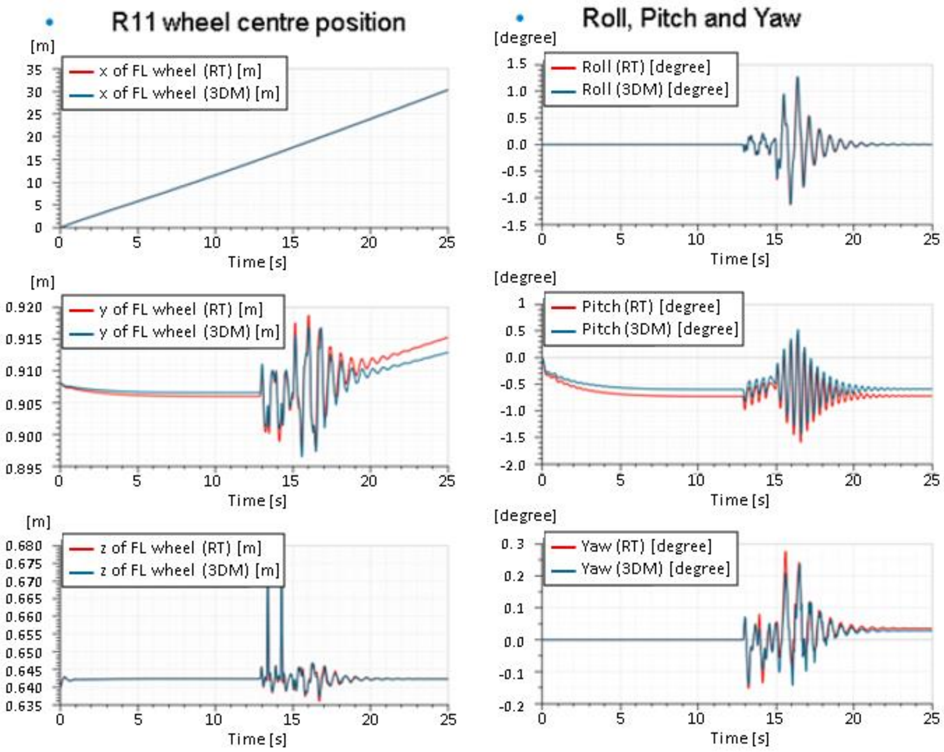

Figure 4.1. Comparison between multibody (blue) and RT model (red) for asymmetric bump test – position of front left wheel center and vehicle Euler angles

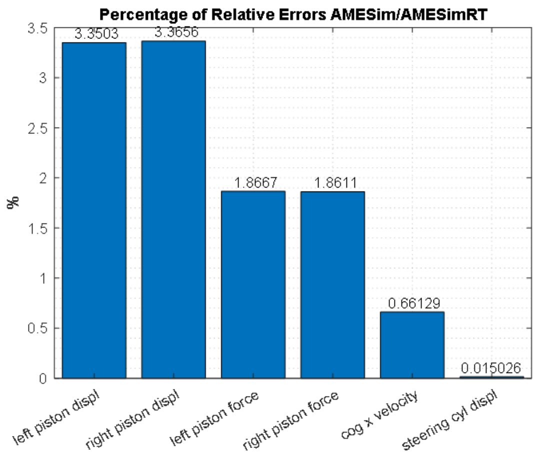

Figure 4.2. Percentage errors for asymmetric bump test

Similar results alignments were obtained also for the ISO 5008 and the moose test. The histograms of the main percentage errors regarding the ISO 5008 and the moose test are reported in Figure 4.3 and 4.4.

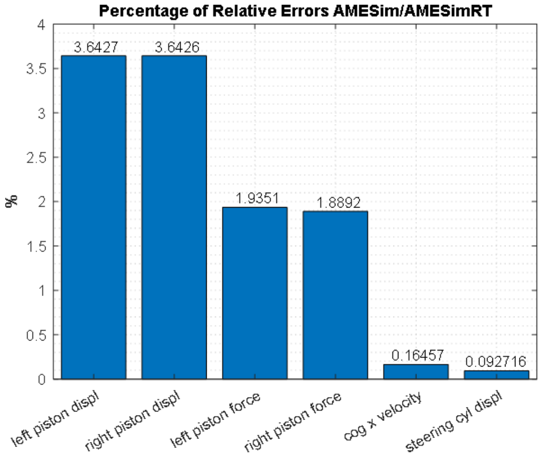

Figure 4.3. Percentage errors for ISO 5008

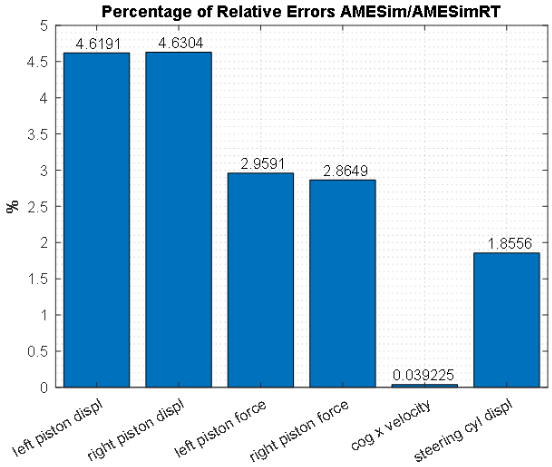

Figure 4.4. Percentage errors for moose test

In summary, the results are quite well aligned for all three considered tests. This aspect is highlighted by the percentage error values, which are all below the 5% threshold. The RT model could thus be considered validated with respect to its multibody counterpart.

Another aspect to highlight is the remarkable improvement in CPU time to run these simulations. In fact, the CPU time is reduced to few seconds for all three considered tests, while the multibody model needs minutes or tens of minutes depending on the simulation.

5. Conclusions

5.1. *Achievements*

The development of the presented methodology in Simcenter Amesim® enabled the creation of a tractor simplified model with independent suspension which can run in Real Time. The followed steps allowed us to consider both the kinematics and dynamics aspects of the starting multibody model. Thanks to this, the results of the RT model turned out to be well aligned with the multibody reference ones. The key aspect of this work is the improvements in terms of computational time to run the simulations. In fact, while both the Amesim® multibody model and the Adams® reference one has computational times in the range of minutes (depending on the simulated maneuver), the RT model can simulate all the tested maneuvers in just a few seconds.

5.2. *Real Time future usage*

The possibility to have a vehicle model that can run in Real Time and provide reliable results is already a benefit. However, this aspect is highlighted if this model can be used within an optimization workflow for the whole independent suspension system where it is necessary to evaluate the performances of a high number of consecutive designs. Passing from minutes to seconds of computational time for a single simulation, it is easy to imagine that the total time-saving during the entire optimization can easily be tens of hours. If we instead consider the same amount of time, this model ensures to evaluate more designs and so a more robust and accurate optimization.

5.3. *Future work*

As anticipated, the alignment between the Amesim® multibody model and the reference Adams® one is not perfect due to the different wheel-soil contact models used. This open point is already under investigation, and it is the first we would like to clarify to validate the RT model completely.

The next activity will concern the analysis and modeling of the hydro-pneumatic system of the suspension, which was not considered in this work, that had the aim to generate and validate the dynamic model of the vehicle. Despite it is not easy to make a hydraulics system model capable of running in Real Time due to its high-frequency features, the usage of a neural network could solve this problem. The creation and the appropriate training of a neural network could thus be the subject of future work.

6. References

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Biographies

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