Vehicle dynamic simulation and powertrain simulation of a heavy hybrid vehicle with interconnected suspensions

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Master Thesis in Vehicle Engineering

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Abstract

This thesis presents two simulations of a heavy hybrid vehicle, the first part of the thesis is focused on the specifications of the vehicle designed in accordance with the requirements based on the literature study of the soils where the vehicle will travel. The second part presents the study of the vehicle through two simulations. The first simulation is oriented on the dynamical behavior of the vehicle. The second simulation focuses on the energy management of the vehicle. The presented thesis is a multi-disciplinary study, combining knowledge on vehicle dynamics, hydraulic suspensions and hybrid systems.

The dynamical simulation of the vehicle has been performed with Matlab/Simulink and the third party program Delft-Tire for the tire modelling. Specials features of Matlab have been used; SimMechanics for the modelling of the parts, links and joints of the vehicle, and SimHydraulics for the modelling of the hydraulic suspensions. The principal tests performed on the vehicle by the dynamical simulation are the tests defined by the NATO - STANAG standards as AVTP 03-170. The tests are a crossing obstacle test and different sine wave roads. The obstacle of the obstacle crossing test is an APG-10 obstacle, an 10 inch high step with vertical edges. The objective of this simulation is to verify the design of the suspension and to observe the forces created in each link of the suspension system in order to design the chassis and the suspension system. The sine wave driving tests are performed to highlight the influence of the different hydraulic connections. Finally the slalom test presents the influence of the hydraulic anti-roll bar.

The results show that the vehicle suspension verifies the STANAG standard. The results show also that the forces applied at the wheel by the obstacle crossing defined in the AVTP 03-170 are directly related to the diameter and the stiffness of the tire. The maximum forces encountered at the wheel corresponds to 2.5 G vertically and 1.5 G longitudinally. The sine wave driving and the slalom test are showing the benefits and the need for advanced hydraulic suspensions.

The second simulation is the modelling of the hybrid power management of the vehicle. The simulation has been performed with the objectives to create a tool for sizing series hybrid powertrain. This simulation has also been performed with Matlab/Simulink and the Simscape Library.

The tool created show that when, the vehicle is equipped with 150 kW of power generation and 300 kW of battery would be able to drive at a constant speed of 10 km/h with the terrain inputs evaluated from the literature study, but to create sufficient result the input parameters of the tools need to have a better definition.

This thesis has taken place at CNIM – La Seyne sur Mer in south of France.

Keywords: hybrid vehicle, off-highway, low pressure tires, obstacle crossing
Preface

This thesis is the first part of a study of a heavy hybrid vehicle for extremes terrains. The hybrid vehicles are a new field of study for CNIM initiator of this project. CNIM has already a good experience with specials purpose vehicles and want to extend their capabilities with a new project of heavy hybrid vehicles with extreme off-road capabilities. CNIM aims to create a vehicle able to drive in extremes areas, with high capabilities for crossing punctual difficulties such as big rocks or bumps. The CNIM capabilities in vehicles for special purpose are proven with the SPRAT capabilities [2]. The SPRAT is a military support vehicle, capable to carry and launch automatically a bridge of 26 meters long or 2 bridges of 14.3 meter long where tank up to 80 tones can cross. All maneuvers to recover the bridge or to drop the bridge are fully automatically done, no physical intervention is needed else than the operator supervision. The vehicle has the capability to climb hills up to 60%, to cross steps with vertical edges up to 80cm and to drive over a gap of 3m. The high scale of abilities of the SPRAT places CNIM as a privileged actor for future off-highway vehicles with high capabilities. The studied vehicle is in the continuity of the off-highway capacities of the SPRAT.

The project started after a market study performed by David Myard in 2011 of exploration in hostiles areas and continued by Sheila Felix in 2013 [3]. The market study highlighted the future increase of interest for exploration of hostile’s areas by many companies. The market study was followed by a study of the state of the art performed from October 2013 to December 2013. The objective of this study was to observe the difficulties encountered in low accessibility areas, the standard vehicles used in these areas and the future tendency of use. A summary of the vehicles used in low accessibilities areas is performed in the introduction of this master thesis. The important study of the extreme environments also performed in the state of the art study, has allowed identifying the keys parameters for vehicles in hostiles areas which are summarized in the first chapter. These parameters have led to design of an innovative suspension system described in chapter II of this master thesis.
This thesis has the objective to present the development of a dynamic model and a powertrain model of a heavy hybrid vehicle with a high scale of off road capacities on the following aspects:

- Presentation of soft soils problems.
- Presentation of an innovative suspension design.
- Dynamical simulation of the studied vehicle.
- Development of series hybrid simulation.
Acknowledgements

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Chapter I.

1 Introduction

The following part aims at presenting the origins leading to the project and the different objectives of the project.

1.1. Background

Some areas are still isolated from the rest of the world. The environment is a large barrier isolating large places where it is extremely difficult for the human to develop. The temperatures are one of the major elements limiting the development. The accessibility of these areas is most of the time only available by air. The rare roads are only negotiable during summer and highly dangerous due to the roughness of the environment. The mud and the melting snow can lead to a serious sinkage of the vehicles. The absence of communication means lead to a strong increase of the isolation factor. The recent years have shown a regain of interest in the exploration of hostile’s areas by ground vehicles [3]. This regain of interest is explained by the strong increase of prices of the aerial means of transportation in regard to the mass or number of persons transported.

1.2. Objectives of the research project

Following the tendencies presented in the background, CNIM’s objective is to produce a vehicle for extremes areas. The objectives of the part of the project presented in this report are as a first step to create a dynamical model of the vehicle in order to validate that the design of the suspension respects the requirements of the STANAG AVTP-03-170 [1] and to find the maximum stress applied to the suspension to design its different parts. In the second phase the objective is to create a tool modelling the powertrain of the vehicle to test the influence of the different components.

1.3. Presentation of the master thesis

The master thesis project is divided into six major parts. The objectives of these parts are listed below:

Chapter II- Study of extremes areas, historical review of the vehicles used in these areas, key factors of these areas, and requirements for a vehicle suspension

Chapter III-presentation of the design of the vehicle suspensions in accordance with the requirements expressed in the first part. Presentation of the different elements of the vehicle suspension: tires, hydraulic suspension, powertrain.

Chapter IV-Study of the rolling resistance in soft soil, in order to have realistic values for the powertrain simulation.

Chapter V-Dynamic simulation of the vehicle, validation of the vehicle suspension regarding the STANAG standards, observation of the maximal forces applied to the suspension.

Chapter VI-Powertrain simulation, presentation of the tool created to size the powertrain.

Chapter VII-Conclusion, and further work.
Chapter II.

2 Literature study of the key parameters to design a suspension for extreme areas

2.1. Introduction

The exploration of extreme areas has evolved quite spectacularly directly after the Second World War, the improvement on vehicle mobility conducted by the important research performed during the war and the numerous military vehicles left by the conflict, has allowed explorers to use modified military vehicles to explore areas with a low accessibility and an hostile environment. During the period of 1970-80’s the low capacity of aerial transports which can land on unprepared terrains and the lower availability of military vehicles demobilized after the war, has conducted engineers to design new vehicles. Most of these new vehicles are renovations or adaptations of standard military vehicles as the Vityaz [4], a tracked vehicle based on the architecture of the Russian T64 tank, but some companies like the society Foremost or NOV have during this period designed from scratch new vehicles as the tracked Husky 8 [5] vehicle, or the Rolligons [6] vehicles equipped with low pressure tires. During this period (1970-80’s) these vehicles have known an important success, and numerous of Vityaz, Husky 8 and Rolligons are still in use in extreme areas.

![Figure 1: Pictures of Rolligon [6], Vityaz [4] and Husky 8 [5]](image)

The control of extreme environment areas has also been a common objective for the United States and the URSS during the cold war. The two countries have developed different concepts and prototypes to explore low accessibility areas, the challenge for the two countries was the control of the Antarctica continent where each country had deployed a basis, close to the magnetic pole [7] for the URSS and near the south pole for the American [8].
But not all produced prototypes have succeeded in their different missions. The most famous prototypes are the American Snow Cruiser who only drove a few meters before staying stuck in the Antarctica [9], and the URSS Kharkovchanka concept [10], which was able to drive only on short distance due to the extremely high consumption of the vehicle.

Figure 2: Picture of the Snow Cruiser [9] (above) and the Kharkovchanka (below) [11]

After the late 1980’s, the increase of the transport capacity of aerial transporters with helicopters like the Mil Mi-26 [12] or planes capable of transporting heavy loads over long distance has considerably conquered the market of extreme area vehicles. The supplying of bases is still today performed most of the time by aerial bridges.

Figure 3: Mil Mi-26 [12] and Twin Otter [13]

But nowadays the increase of price of oil, combined with the relative low safety for the crew in case of crash and the low transportation capacity of aerial means compared to a ground vehicle, has conduct companies to reconsider the needs for off-highway ground vehicles. The technical advances in low pressure tires for soft soils combined with the advances in power and traction capacities of off-highway vehicles have changed the market. The needs for access for these areas have also highly increased during the last years [3].
The difficulties experienced by the numerous vehicles lost by staying blocked in these areas due to the soil difficulties added to the difficulties to create a rescue mission, highlight the needs for a special design vehicle for extreme terrains.

2.2. Soft soils

The study of soft soil which is the major difficulty encountered by off road vehicles in isolated areas can help to highlight the principal parameters of an extreme terrain vehicle suspension. This part aims at studying the soft soils in order to explain the key parameters of these areas which have led to the design of the suspension presented in chapter III.

Soft soil can lead to the sinkage of the vehicle and most of the vehicles which stayed blocked in hostile areas have been after an important sinkage. Moreover it is highly difficult to extract a vehicle from a sinkage position without exterior help. The absence of traffic also of help is an additional factor which is responsible for the dangers of hostile’s areas.

For off highway transportation and logistics, typically in mines or worksites, the sinkage of the vehicles is also an important problem not in terms of danger of staying blocked but in the productivity lost each time a vehicle stay blocked for a few minutes or hours [14].

The basic principle of the sinkage phenomenon is due to the important ground pressure between the wheel and the soil which creates an important compaction of the soil before it can carry the weight of the vehicle [15]. But the problem is more complex, typically soils are constituted of different layers, which have different resistances and responses to a certain load.

Different types of soils are studied in this part, the first part aims at studying soils composed of different layers each, with a different resistance to compaction. The second part focuses on the study of the soil as a homogenate soil but with a vegetation and moisture layer on the top of the soil.

2.2.1. Soils with different layers

Soils like snow can be subject to a repetitive melt-freezing [15], which creates a rigid ice layer on the top of the ground, which can be covered by another layer of snow as shown in figure 4.

![Figure 4: Snow soil description [15]](image)

The ice layer is a fragile material, which resist loading with almost no deformation, but breaks in fragile mode under a too high load. Oppositely the two different snow layers can be seen like highly plastic materials.
By applying a pressure on the upper snow layer it is possible to create a relation between the pressure applied and the sinkage. This test is performed with a bevameter. The result of this test performed on the snow presented in figure 4 is shown in figure 5.

![Figure 5: Pressure/sinkage relation](image)

The results show that the curve of pressure/sinkage, is a discontinue curve. The first part of the curve represents the compaction of the top snow layer and the second part the lower snow layer, the discontinuity of the curve represents the break of the ice layer.

### 2.2.2. Soils with vegetation layer

Soils with vegetation experience during the passage of the vehicle a compaction which destructs the vegetation. The vegetation of the soil keeps the soil strength high and highly increases the admissible shear stress of the soil.

The vegetables of the soils can be seen as fibers, and the earth a matrix, vegetation and earth create together a low fiber composite on the topsoil. The destruction of the vegetation through the important compaction or due to the shear stress created by the vehicle passage destructs the fibers (vegetation) and leaves the soil exposed to erosion. The soil can also be separated as two different layers, one layer (topsoil) highly resistive until the break of the fiber, and under a homogenous layer (subsoil) without vegetation as shown in the figure 6.

![Figure 6: Vegetation soil description](image)

The same test performed on the snow soil is performed on a soil with vegetation. The results reproduced in figure 7 show the pressure/sinkage relation curve. The muskeg area (bog or swamp area) resists to the applied pressure until the fibers of the topsoil break lowering the necessary pressure to penetrate the soil. This specific behavior is highlighted by the decrease of the necessary pressure to penetrate the soil after a certain sinkage.
Moreover, a soil with a destructed vegetation becomes exposed and through the attacks of the environment (rain), the resistance of the soil is lowered due to an increase of the water content in the soil which is responsible for the sinkage of the vehicles.

### 2.2.3. Conclusion

The two different types of soils presented have a similar behavior under a vertical pressure load. The difference between the two types of soil is the change in the pressure/sinkage relation, with a highly fragile topsoil like for a snow terrain, the pressure/sinkage relation has a discontinuity oppositely with a vegetation topsoil the relation is continuous but both soils show an easy penetration if the top soil is destructed. The results show the necessity for the vehicle to not exceed a certain pressure above which the sinkage of the vehicle becomes extremely important.

The observations permit to draw the first necessity of a vehicle which is to have a low ground pressure in order to not break the topsoil. But other parameters have an important influence on the soil response, the shear stress applied, the numbers of passes of the vehicles, and the power traction mode – tracked or wheeled.

### 2.2.4. Shear stress

The shear stress applied to the soil by the wheels or tracks is an important parameter of the disturbance applied to the soil. Soils with a fragile topsoil are not very sensitive to shear stress, homogenate soils or vegetation soils are oppositely highly disturbed by the shear stress applied.

The shear stress applied to the soil is directly related to the slip, the acceleration and the direction of motion of the vehicle. The slips of wheels highly disturb the soil and is directly responsible for creating big ruts in soft soils. Changes in the vehicle speed with large accelerations and braking are directly responsible for a considerable wheel slip since the power traction in a soft soil is a function of the slip coefficient [15]. The high influence of the shear stress is due to the absence of tensile strength in soils, also the shear stress applied by the traction of the vehicle highly disturb the soil, digging big ruts [16] [17]. The changes of directions are also important parameters since during cornering the shear stress created by the vehicle is much more important than during straight driving. The shear stress is highly dependent to the traction model: wheels or tracks.
2.2.5. Multi-passes

A road is by definition an area where numerous vehicles drive. The soil receives an important hardening since the vehicle wheels or tracks are passing at the exactly same location at each passage. The multiple loads of the terrain have cumulated influence which leads to serious damages of the soil. On a vegetation soil, the vegetation is strongly affected and suffers from the vehicle passages even if the topsoil is not deeply disturbed. The several loads on the ground, hardening the soil lead to an important soil compaction, and an even more soil exposure. This problem of multi pass driving destructing the vegetation is a major problem in military terrain, which sometimes need years before healing from the passage of one or more vehicles. The problem of soil exposure after several passes of vehicles is also experimented in mines where sometime a stabilization of the soil is necessary in order to avoid a too large and sometimes dangerous erosion of the soil. On homogenous soil the influence of multi-passes is not important, and this is also the case for soils with fragile topsoil like snow if the ice layer is not destructed [15] [18] [19].

J.Y. Wong [15] has defined different criteria to select the maximal admissible pressure of a vehicle to permit the passage as function of the soil type and the number of passes (table 1).

<table>
<thead>
<tr>
<th>Terrain</th>
<th>Ideal (multipass operation or good gradability)</th>
<th>Satisfactory</th>
<th>Maximum acceptable (mostly trafficable at single-pass level)</th>
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<tbody>
<tr>
<td>Wet, fine-grained</td>
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<tr>
<td>• Temperate</td>
<td>150</td>
<td>200</td>
<td>300</td>
</tr>
<tr>
<td>• Tropical</td>
<td>90</td>
<td>140</td>
<td>240</td>
</tr>
<tr>
<td>Muskeg floating mat</td>
<td>30</td>
<td>50</td>
<td>60</td>
</tr>
<tr>
<td>European bogs</td>
<td>5</td>
<td>10</td>
<td>15</td>
</tr>
<tr>
<td>Snow</td>
<td>10</td>
<td>25-30</td>
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2.3. Wheeled

Off-highway wheeled vehicles are almost all designed on the same basis, the dump truck architecture. A dump truck is built from two chassis with a flexible link between the two chassis. The steering is made by a hydraulic control between the two chassis. The interest of articulated chassis is the capacity of trucks to drive out from deep ruts by rotating one chassis relative to the other. Several others architectures have been tested, but it remains that an all steerable and powered wheels vehicle is the most attractive architecture [15].

The agricultural industry has also performed important research on soft soil, not in terms of vehicle mobility but in terms of soil productivity. The destruction of the vegetation by the soil compaction or disturbance applied by the wheels, leads to an important decrease of the soil productivity. Tire companies as Michelin or Goodyear have developed low pressurized tires in order to minimize the ground pressure applied under the wheel. Agricultural tires try to reduce the disturbance on the ground by the passage of the wheel by having a large footprint reducing the influence of the wheels passage [20].
2.4. Tracked vehicles:
The tracked vehicles compared to wheeled have a larger contact surface between the tracks and the ground which added to the important sculptures of their tracks offer them a higher traction power and a lower ground pressure. The combination of these two parameters makes the tracked vehicles less sensitive to sinkage and in case of an important sinkage gives them a better capacity to extract themselves from a difficult position. The tracked vehicles have also a better mobility capacity in rough terrains, but the disturbances created by the tracks, especially the shear stress applied; have a much more large effect on the ground than wheeled vehicles. During cornering the tracks are literally ploughing the soil and destructing the vegetation [21].

The road driven by tracked vehicles are more sensitive to the number of passes of the vehicles. In order to increase the capabilities of roads in resistance to repetitive load of tracked vehicles it is important to limit the turning radius of the curves.

2.5. Requirements for extremes areas vehicles
The softs soils have been studied for years by armies for increasing the mobility of the vehicles, by agricultural and mining companies in order to increase the productivity and other companies. The observations made by this soil study based on the type of power traction, the differences of response coming from the soils as function of the type of load applied and the soil composition have allowed to highlight the key parameters to design a vehicle for soft soils.

The principal parameter is the soil pressure which has to be lower than a critical ground pressure defined by the break of the topsoil layer. Moreover the minimization of the shear stress applied is a key parameter in order to increase the resistance of the roads to repetitive passes of vehicles. All these observations have conducted to design a vehicle with an innovative suspension.
Chapter III.

This chapter aims at presenting the suspension of the vehicle and the different components of the full suspension: the low pressure tires, the hydraulic struts and the powertrain selected in accordance with the requirements of the previous section.

The different elements of the suspension are studied in the vehicle dynamical simulation and in the powertrain simulation.

3 The suspension design

The important requirements presented in the conclusion of the soft soil study have conducted to design a wheeled vehicle in order to limit the destruction of the soil by applying a low shear stress, with a suspension offering simultaneously a low ground pressure and all wheels powered and steerable. The suspension is presented in figure 8.

![Suspension design](image)

The suspension is constituted of a rigid diabolo creating a connection between the two wheels. Inside the diabolo are the wheel hub motors. The diabolo can rotate freely relative to the lower part of the suspension, which can rotate relative to the upper part of the suspension.

The wheel travel is performed by the revolute joint between the upper part of the suspension and the lower part, and the hydraulic strut as shown in figure 9. The vehicle has a wheel travel of 500 mm, in order to fulfil the requirements of high off road capabilities.
Figure 9: Wheel travel description

The revolute joint between the diabolo and the lower part of the suspension is used to better adapt the load of each tire to the ground as shown in figure 10.

Figure 10: Rotation of the diabolo

The steering of the suspension is performed by a rotation of the full suspension around the Z-axis as shown in figure 11. The vehicle is an all-axles steering vehicle. The turning radius of the vehicle is presented in figure 60.

Figure 11: Steering of one axle
The steering system, the upper part, the lower part, the diabolo, the wheels, and the hydraulic strut create an axle. The vehicle is equipped with 8 axles, four on each side of the vehicle. Two axles create a pair of axles (figure 12).

![Figure 12: Full suspension of the vehicle](image)

The vehicle has the following specifications (table 2):

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of the vehicle (fully loaded)</td>
<td>50 tons</td>
</tr>
<tr>
<td>Load per axle</td>
<td>6.25 tons</td>
</tr>
<tr>
<td>Load per wheel</td>
<td>3.125 tons</td>
</tr>
<tr>
<td>Height of the center of gravity</td>
<td>2.2 m</td>
</tr>
<tr>
<td>Longitudinal position of the center of gravity in reference to the front pair of axles</td>
<td>6 m</td>
</tr>
<tr>
<td>Length</td>
<td>12 m</td>
</tr>
<tr>
<td>Width</td>
<td>6 m</td>
</tr>
</tbody>
</table>
3.1. Low pressure tires

The parameter of a low ground pressure is directly related to the size of the footprint of the vehicle tires. In order to minimize the ground pressure, the vehicle has to have a tire footprint as large as possible. To fulfill this requirement, the vehicle is equipped with an innovative suspension which permits to double the number of tires. The number of tires is not the only parameter permitting to increase the size of the global footprint. In order to increase the footprint, the studied vehicle is equipped with agricultural low pressure tires. Presented in chapter II, low pressure tires reduce the soil compaction by their large footprint. Moreover the rotation of the diabolo presented in figure 10 allows keeping a large contact patch between the tire and the soil.

Michelin has selected a tire in accordance to the specifications of the studied vehicle presented in table 3:

Table 3: Michelin Data of the selected tire [20]

<table>
<thead>
<tr>
<th>Dimensions profil gonflé - Inflated dimensions</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Largeur Jante de mesure - Measuring rim width</td>
<td>21.00</td>
<td>21.00</td>
</tr>
<tr>
<td>Pression - Pressure</td>
<td>1.60 b</td>
<td>23 ps</td>
</tr>
<tr>
<td>Largeur Boudin - Section Width</td>
<td>675 mm</td>
<td>26.6 in</td>
</tr>
<tr>
<td>max en service - Max Grown</td>
<td>669 mm</td>
<td>26.1 in</td>
</tr>
<tr>
<td>-- NOMINAL ETRTO</td>
<td>876 mm</td>
<td>34.5 in</td>
</tr>
<tr>
<td>-- MAXIMUM ETRTO</td>
<td>740 mm</td>
<td>29.0 in</td>
</tr>
<tr>
<td>Diamètre Extérieur - Overall Diameter</td>
<td>1 865 mm</td>
<td>73.4 in</td>
</tr>
<tr>
<td>max en service - Max Grown</td>
<td>1 873 mm</td>
<td>73.7 in</td>
</tr>
<tr>
<td>-- NOMINAL ETRTO</td>
<td>1 833 mm</td>
<td>72.2 in</td>
</tr>
<tr>
<td>-- MAXIMUM ETRTO</td>
<td>1 873 mm</td>
<td>73.7 in</td>
</tr>
</tbody>
</table>

In a low pressure tire, the inflation pressure changes the vertical stiffness of the tires. When the pressure changes the stiffness of the selected tires can vary from 240 N/mm to 430 N/mm. Even when the tire is highly pressurized the vertical stiffness is extremely low compared to the vertical stiffness of a regular truck tire which is minimum around 800 N/mm. Michelin documentation provide information about the evolution of the tire stiffness as function of the inflation pressure as shown in figure 13.

The load at one wheel is around 3000 kg, Michelin propose three different pressure/vertical stiffness configurations. On higher inflations, the tire has not been tested by Michelin with a load of 3000 kg because the tire becomes over inflated leading to an increase of the wear in the middle of the tire.
The tire has also speed limitations as function of the tire pressure. Table 4 shows that the authorized speed at a certain load increase with an increase of the pressure. This increase of the authorized speed as function of the inflation pressure is due to the augmentation of the deformation of the tire with the diminution of the inflation pressure, which can lead to a fast wear of the tire and in extremes cases to the destruction of the tire.

The second reason for these speed limitations is the risk issues with low pressurized tires at high speed. Low pressurized tires can leave the rim during cornering, the low pressure applied on the rim by the low pressurized tire is not large enough for keeping the tire on the rim.
Table 4: Possible driving speed as function of the load and the inflation pressure of the tire [20]

<table>
<thead>
<tr>
<th>Press. psi</th>
<th>50 km/h Kg</th>
<th>40 km/h Kg</th>
<th>30 km/h Kg</th>
<th>10 km/h Kg</th>
<th>40 km/h Dual Kg</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30 mph Lb</td>
<td>25 mph Lb</td>
<td>20 mph Lb</td>
<td>6 mph Lb</td>
<td>25 mph Dual Lb</td>
</tr>
<tr>
<td>0.40</td>
<td>6</td>
<td>2,510</td>
<td>5,530</td>
<td>3,230</td>
<td>7,130</td>
</tr>
<tr>
<td>0.50</td>
<td>7</td>
<td>2,995</td>
<td>6,600</td>
<td>3,830</td>
<td>8,440</td>
</tr>
<tr>
<td>0.60</td>
<td>9</td>
<td>3,050</td>
<td>6,720</td>
<td>3,250</td>
<td>7,170</td>
</tr>
<tr>
<td>0.70</td>
<td>10</td>
<td>3,210</td>
<td>7,080</td>
<td>3,425</td>
<td>7,550</td>
</tr>
<tr>
<td>0.80</td>
<td>12</td>
<td>3,365</td>
<td>7,420</td>
<td>3,600</td>
<td>7,940</td>
</tr>
<tr>
<td>0.90</td>
<td>13</td>
<td>3,525</td>
<td>7,770</td>
<td>3,775</td>
<td>8,320</td>
</tr>
<tr>
<td>1.00</td>
<td>15</td>
<td>3,680</td>
<td>8,110</td>
<td>3,950</td>
<td>8,710</td>
</tr>
<tr>
<td>1.10</td>
<td>16</td>
<td>3,840</td>
<td>8,470</td>
<td>4,125</td>
<td>9,090</td>
</tr>
<tr>
<td>1.20</td>
<td>17</td>
<td>4,000</td>
<td>8,820</td>
<td>4,300</td>
<td>9,486</td>
</tr>
<tr>
<td>1.30</td>
<td>19</td>
<td>4,155</td>
<td>9,160</td>
<td>4,475</td>
<td>9,870</td>
</tr>
<tr>
<td>1.40</td>
<td>20</td>
<td>4,315</td>
<td>9,510</td>
<td>4,650</td>
<td>10,250</td>
</tr>
<tr>
<td>1.50</td>
<td>22</td>
<td>4,470</td>
<td>9,850</td>
<td>4,825</td>
<td>10,640</td>
</tr>
<tr>
<td>1.60</td>
<td>23</td>
<td>4,630</td>
<td>10,210</td>
<td>5,000</td>
<td>11,020</td>
</tr>
<tr>
<td>1.70</td>
<td>25</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.80</td>
<td>26</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.90</td>
<td>28</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

In the dynamical simulation of the vehicle presented in chapter V some tests are performed with a tire inflated at a pressure under the minimum requirements in order to push the vehicle to its limits. In order to avoid the problem of the tire leaving the rim, each wheel will receive a beadlock fixing the tire on the rim.
3.2. Hydraulic suspension
This part aims at presenting the hydraulic suspension and the different features offered by hydraulics suspensions.

Hydraulic suspensions have been invented by the French vehicle car manufacturer Citroen. The first hydraulic suspension adapted on a series produced vehicle was adapted on the Citroen DS in 1955 [22]. During years hydraulic suspensions have been the exclusive property of Citroen and used under license by a few other car manufacturers. Nowadays there are a few civilian vehicles with hydraulics suspension; the price and the complexity make the use of hydraulic suspensions extremely rare in passenger cars. Moreover, the advantages produced by a hydraulic suspension are multiplied with the number of wheels, this increase of capacities as function of the number of wheels make the hydraulic suspensions extremely interesting for military vehicle, and special purpose vehicles with many wheels, but less interesting for classic 4 wheel passenger cars.

3.2.1. Principle of a basic hydraulic suspension
In a basic hydraulic strut, the spring and the damper are replaced by a single acting actuator and a spring or gas charged accumulator. When the wheel hits a bump the oil in the cylinder chamber flows through a hydraulic restriction and compresses the gas in the gas charged accumulator – figure 14. The gas charged accumulator acts like a spring, and the hydraulic restriction acts like a damper which regulates the flow of oil going to the accumulator. The spring function of the suspension is also created by a mechanical spring included in each actuator and the spring function offered by the gas or spring compression in the accumulator. A gas charged accumulator is more interesting than a spring loaded accumulator because the gas offers a non-linearity of the spring rate in the suspension which is highly demanded.

![Figure 14: Scheme of a simple hydraulic suspension](image)

With hydraulic suspension, the level of the suspension is controllable, by filling in the hydraulic circuit with an external pump, the increase of fluid increases the pressure of the oil in the circuit, lifting the vehicle by changing the equilibrium position of the hydraulic actuator. Each wheel is equipped with its own hydraulic circuit. Each hydraulic circuit can be filled
separately in order to compensate the vehicle static roll if the vehicle is more loaded on one side. Example old Citroen DS cars have different chassis positions [22]. The upper position is used to drive on rough roads, and the lower position more stable to drive on highways. The major problem of lifting up a chassis with a standard hydraulic suspension is that the stiffness of the suspension is changed by the change of level of the suspension. The increase of oil in the suspension increases the load on the single acting actuator by compressing the spring; also the stiffness of the suspension is affected.

The need for an external pump for charging each circuit is mandatory; a hydraulic circuit has always a little oil leakage, after a long time the vehicle suspension will be in the lowest position if there is not an external filling pump.

The restriction plays the role of the damper in the suspension, the oil flow is restricted and slowed down in the restriction, by changing the restriction size, the damping can be changed. With a tight restriction the oil has more difficulties to flow through the restriction and to charge the gas or spring charged accumulator, the suspension has a high damping. Oppositely, with a large restriction, the damping of the suspension is small.

The basic features of a hydraulic suspension have been now explained, but hydraulic suspensions exist also in more advanced configurations.

### 3.2.2. Advanced hydraulic suspension

Advanced hydraulic suspensions, are based on the same principle of a simple hydraulic suspension: actuator, restriction and gas charged accumulator. But the major change is the replacement of the single acting actuator by a double acting actuator. The two chambers of the actuator are connected together and to the same restriction and gas charged accumulator. Of course in this configuration, the rod of the cylinder has to be connected to the chassis and the piston chamber to the wheel in order to create a positive lift force. Since the oil pressure is the same in the two chambers, the force created by the actuator is directly related to the surface ratio between the head chamber surface and the piston chamber surface. Figure 15 illustrates the hydraulic scheme of an advanced hydraulic suspension.

![Figure 15: Scheme of an advanced hydraulic suspension](image)
The spring function of the suspension is here only performed by the gas charged accumulator compared to a simple hydraulic suspension where there is still a mechanical spring.

The lift of the vehicle can be done exactly as in a simple suspension by filling in the hydraulic circuit of the suspension. Compared to a basic hydraulic suspension, the stiffness of the suspension is not changed by the lift of the vehicle.

By comparing an advanced hydraulic suspension to a mechanical suspension it is possible to highlight the advantages of a single hydraulic suspension compared to a mechanical one.

The first advantage of a hydraulic suspension, is the strong non linearity of the spring rate as function of the wheel travel. The suspension is still stiff even when the wheel is completely deployed. The wheel will increase the response with down travel as shown in figure 16. Moreover the spring force highly increases when the wheel lift up. This strong increase of the spring rate, makes vehicles equipped with hydraulics suspensions less sensitive to wheels touching upper hard stops.

![Figure 16: Evolution of the spring force as function of the suspension displacement][23]

The second comparative element is the natural frequency of the two suspensions (figure 17). A hydraulic suspension has a natural frequency which increases with the load, oppositely the natural frequency of a mechanical suspension has a natural frequency which decreases with the load.
This change of natural frequency enables to have a vehicle with a more rapid response, allowing a better filtration of the road bumps. Moreover it is possible to observe in figure 17 that the evolution of the natural frequency is less important on a hydraulic suspension than on a mechanical one.

3.2.3. Advanced hydraulic control of suspension

Some car manufacturers have invented semi-active suspension (called regularly active suspension) where the restriction area is controlled by the ECU to regulate the damping of the suspension as shown in figure 18 [24].

With a semi active suspension the damping, related to the restriction area, varies to adapt the damping of the suspension to the road roughness. For energy consumption and reliability reasons semi active suspensions will not be adapted to the studied vehicle.
3.2.4. Advantages of hydraulics suspensions

The hydraulic suspensions capacities are multiplied with the number of wheels as announced in the introduction. The special capacities are coming from the capacity of hydraulic suspensions to be connected together. By setting up the right connection between the different hydraulic struts, it is possible to create a hydraulic anti-roll bar and to equilibrate the load applied on different wheels. This chapter aims first on the capacity of hydraulic suspensions to equilibrate the load on each wheel; the second part focuses on the capacity to create a hydraulic anti roll bar.

3.2.4.1. Load equilibrium under different wheels

One of the capacities produced by hydraulics suspensions is the capacity to equilibrate the load under different wheels placed on the same side of the vehicle. To create this feature, the two front cylinders on each side are connected together. Each head chamber is connected with the other head chamber on the same side of the vehicle. The same connection exists for the two cylinder chamber as described by figure 19.

![Figure 19: Hydraulic scheme of one side of the vehicle (lateral point of view)](image)

When the vehicle hits a bump, there is an increase of load on the front tire which is climbing the bump, the oil flows out from the piston chamber, but instead of charging the accumulator, the oil flow goes in the other piston chamber, the oil has the opposite flow orientation between the two head chambers, the hydraulic connection equalize the oil pressure in the two cylinders also equalizing the load under each wheel as shown figure 20.
Off road vehicles have a high tendency of letting a wheel slip in rough terrain, because the load on the tire is not sufficient to transmit the torque to the ground letting the wheel spinning and slipping. In order to avoid the slip of one wheel, off road vehicles are most of the time equipped with lockable differentials. With a well selected hydraulic suspension, the load on each wheel is equilibrated, also the slip of one wheel is reduced since the load on each tire is equalized.

When different wheels are connected on the same hydraulic circuit, it is possible to see the different wheels like a single wheel driving on the average surface of the different wheels. Figure 21 illustrates this simplification.

The feature of equilibrium force between each wheel connected on the same hydraulic circuit is highly recommended for off road driving, not only for the behavior of the vehicle on the bumps but also to avoid the sinkage or slipping of one wheel.
3.2.4.2. Anti-roll bar functionality

The second capacity of hydraulic suspensions is to create a hydraulic anti-roll bar between the two sides of the vehicle. To provide this feature, the hydraulic struts placed on the two sides of the vehicle are connected through specific hydraulic connections. Each head chamber of one side is connected with the piston chamber of the other side as described in figure 22.

Figure 22: Hydraulic scheme of the cross stabilization connection (upper point of view)

Supposing that a vehicle is taking a right curve without a hydraulic anti-roll bar, the inertial force applied on the body pushes the vehicle outside the curve. The load on the outer suspension of the curve increases and the load on the inner suspension of the curve decreases. The increase of the load on the outer suspension creates an overpressure in the piston chamber of the outer strut and respectively decreases the pressure in the piston chamber of the inner suspension of the curve. This difference of created pressure makes the vehicle roll as described in figure 23.

Figure 23: Forces during cornering

By adding the proper hydraulic connections between the different hydraulic struts as shown in figure 23 it is possible to compensate the roll.

When the vehicle is cornering, the pressure in the left cylinder chamber increases, this creates a flow of oil to the head chamber on the other side of the vehicle. This flow of oil compensates the roll over force. On the figure 24 the red arrow shows the flows of oil and the forces created.
3.2.4.3. **Overpressure problems**

Off road vehicles can hit an obstacle at high speed; this can make a wheel lift up extremely rapidly. Such lift can create an important pressure in the piston chamber. A big flow of oil tries to flow through the restriction to the accumulator. If the restriction is too tight the oil pressure will increase up to extreme values in the piston chamber. This overpressure can damage the hydraulic circuit, and perforate a pipe or the cylinder. In this case it is necessary to add a security valve between the accumulator and the cylinder, which opens in case of a high pressure difference between the accumulator pressure and the cylinder pressure (figure 25).

![Figure 24: Force correction with the cross stabilization](image)

In case of an obstacle hit at high speed the oil bypass the restriction through the check valve and charge the accumulator without creating any overpressure in the cylinder.

The same security valve can be added oppositely to allow the accumulator to discharge oil in the cylinder rapidly in order to allow the wheel to get down faster and the suspension to adapt extremely rapidly to the ground (figure 26).
3.2.5. Conclusion

The list of advantages provided by the hydraulic suspensions meet precisely the list of requirements of an off road vehicle. The equilibrium of load on the different wheels, reducing the wheel slipping and sinkage, combined with the anti-roll bar feature permit the vehicle to have better behavior than regular off road vehicles equipped with mechanical suspensions.

3.3. Presentation of the hydraulic suspensions of the vehicle

It has been decided that the vehicle will be equipped with hydraulic suspension since hydraulic suspensions have capacities which meet precisely the requirements of the vehicle.

The vehicle is equipped with 8 hydraulic struts connected on two different hydraulic circuits as described in figure 27.
The two front wheels on each side of the vehicle have also the same load on each axle and the two rear wheels on each side of the vehicle as well. Different other hydraulic connections have been tested in the vehicle dynamical simulation chapter V.

3.3.1. Dimensioning the hydraulic components

The dimensioning of the hydraulic components of the suspension is an important step to select the proper stiffness and damping of the suspension. The dimensioning of the hydraulic components has been made by a hydraulic suspension supplier. Here is presented a simple modelling and dimensioning of hydraulic components with a method from the hydraulic suspension supplier participating to the project and a literature study. Hydropneumatic suspensions systems 2011 [23] and AMESIMIUT [25] give important knowledge for dimensioning and understanding a hydraulic suspension.

3.3.1.1. Assumptions:

During working the gas of the charged accumulator can be estimated as working in an adiabatic mode and quasi static transfer mode, the speed is sufficiently high to limit the heat exchange with the rest of the world (equation 1). Also it is possible to write:

Figure 27: Full hydraulic scheme of the vehicle suspension
\[ PV^\gamma = \text{constant} \]  \hspace{1cm} (1)

Where \( P \) is the pressure, \( V \) the volume of the chamber, and \( \gamma \) is the perfect gas constant.

The mechanical suspension stiffness can be written as: the force variation over the displacement (equation 2):

\[ K_{\text{mech}} = \frac{\Delta F}{\Delta X} \]  \hspace{1cm} (2)

Where \( K_{\text{mech}} \) is the mechanical stiffness of the suspension, \( \Delta F \) is the force variation and \( \Delta X \) is the displacement variation.

For a hydraulic suspension the stiffness \( K_{\text{hydraulic}} \) can be written as the variation of the pressure over the variation of volume of the gas charged accumulator (equation 3).

\[ K_{\text{hydraulic}} = \frac{\Delta P}{\Delta V} \]  \hspace{1cm} (3)

Where \( K_{\text{hydraulic}} \) is the stiffness of the hydraulic suspension, \( \Delta P \) the differences of pressure in the two chambers, and \( \Delta V \) the variation of volume of the strut.

With these different assumptions it is possible to calculate the hydraulic stiffness of the suspension.

### 3.3.1.2. Calculation of the hydraulic stiffness

To calculate the stiffness of the hydraulic suspension, it is important to consider that the accumulator works in an adiabatic mode and quasi-static transfer mode, so by differentiation it is possible to write (equation 4):

\[ \Delta(PV^\gamma) = V^\gamma \Delta P + P \gamma V^{\gamma-1} \Delta V = 0 \]  \hspace{1cm} (4)

The variation of stiffness of the hydraulic suspension becomes (equation 5):

\[ \Delta K_{\text{hydraulic}} = -\gamma \frac{P}{V} \]  \hspace{1cm} (5)

It is important to remark that even if there is a minus sign, the variation of the stiffness is still positive, as for a compressed spring, where an augmentation of the length induces a reduction of the effort, an augmentation of the volume leads to a reduction the pressure in the hydraulic suspension, also the stiffness is positive.

The selected suspension stiffness is 430 N/mm. With the previous equations it is possible to relate the hydraulic stiffness to the mechanical stiffness (equation 6).

\[ P = \frac{F}{S}, \text{and } V = SX \]  \hspace{1cm} (6)

Where \( S = S_1 - S_2 \) surface of the cylinder, \( V \) volume of displaced oil, \( X \) the displacement of the cylinder, and \( F \) the force.

The following relation can be established, relating the mechanical stiffness \( K_{\text{mechanical}} \) to the hydraulic stiffness \( K_{\text{hydraulic}} \) (equation 7):

\[ K_{\text{hydraulic}} = \frac{\Delta P}{\Delta V} = \frac{\Delta \left( \frac{F}{S} \right)}{\Delta (S \ast X)} = \frac{1}{S^2} \frac{\Delta F}{\Delta X} = \frac{1}{S^2} K_{\text{mech}} \]  \hspace{1cm} (7)
The accumulator needs to be precharged at a pressure $P_0$ and the data of the initial volume $V_0$ is also needed. When the accumulator is plugged on the hydraulic circuit it works at a pressure and volume of working, respectively $P_i$ and $V_i$. Initially the system is at an equilibrium position, the law of perfect gas can be written (equation 8):

$$P_iV_i = P_0V_0$$  \hspace{1cm} (8)

By knowing the initial pressure given by the system, it is possible to get the initial volume of working. As explained above during the working of the suspension it is possible to consider that the gas works in an adiabatic and quasi static phase, also the following equation can be applied during the working phase (equation 9):

$$PV^\gamma = P_iV_i^\gamma$$  \hspace{1cm} (9)

Finally, the stiffness of the accumulator is given by the stiffness around the static load point by (equation 10):

$$K_{\text{hydraulic}} = \gamma \frac{P_i}{V_i}$$  \hspace{1cm} (10)

### 3.3.1.3. Dimensioning of the accumulator

The selected actuator has a displacement of 500 mm, a rod of 100 mm diameter and a piston of 140 mm diameter. The effective area of the cylinder is also (equation 11):

$$S = S1 - S2 = \frac{\pi}{4} (140^2 - 100^2) = 7540 \ mm^2$$  \hspace{1cm} (11)

The mechanical stiffness and the strut section are yet selected. But the pressure and volume of working, $P_i$ and $V_i$ are still unknown, neither $P_0$ and $V_0$. The load of the vehicle enables to determine $P_i$ (equation 12):

$$P_i = \frac{F}{s} = \frac{M_{\text{car}} \cdot g}{N \cdot s} = \frac{(50 \ * \ 10^3 \ * \ 9,81)}{8 \ * \ 7540} = 81 \ \text{Bar}$$  \hspace{1cm} (12)

Where $M_{\text{car}}$ is the mass of the car, $N$ the number of struts, $g$ the gravity constant, and $S$ the difference of areas in the cylinder presented in equation 11.

In a hydraulic suspension, the volume of the gas accumulator has to be at least as large as the maximal volume swept by the cylinder during working. With $X_{\text{equilibrium}}$ as the equilibrium position of the cylinder, and $S$ the difference of areas in the cylinder during working it is possible to calculate the minimum volume of the gas accumulator (equation 13):

$$V_{\text{min}} = 2 \ * \ X_{\text{displacement}} \ * \ S = 3770 \ \text{cm}^3$$  \hspace{1cm} (13)

If the minimum value of volume is selected for the gas charged accumulator, when the cylinder has his maximum volume swept, the volume left for the gas in the accumulator is equal to 0 cm$^3$. Due to this observation it is necessary to increase the volume of the gas charged accumulator in order to let the suspension having a maximum travel distance. The hydraulic supplier proposes by experience as first test value for this vehicle project to increase the value of the gas charged accumulator to (equation 14):

$$V_0 = 4200 \ \text{cm}^3$$  \hspace{1cm} (14)

From this point it is possible to calculate the rest of the system (equation 15):
\[
K_{\text{mech}} = S_{\text{cylinder}}^2 \ast K_{\text{hyd}} = S_{\text{cylinder}}^2 \ast \gamma \ast \frac{P_i}{V_i} = S_{\text{cylinder}}^2 \ast \gamma \ast \frac{P_i}{P_0V_0} \div P_i
\]

\[
= \gamma S_{\text{cylinder}}^2 \ast \frac{P_i^2}{P_0V_0} = 430 \text{ N/mm}
\]

Also the precharge of the gas charged accumulator is (equation 16):

\[
P_0 = \gamma \ast S_{\text{cylinder}}^2 \ast \frac{P_i^2}{K_{\text{mech}} \ast V_0} = 24 \text{ bar}
\]

With Matlab/Simulink it is possible to simulate only the cylinder and accumulator, with this simulation it is possible to verify that the suspension has the selected stiffness, and to see how the stiffness of the suspension evolves with the strut displacement.

\[
\text{Force (N)}
\]

\[
\text{Displacement (m)}
\]

Figure 28: Stiffness of the suspension as function of the displacement

The results of the simulation show that the stiffness of the suspension corresponds to the selected stiffness around the equilibrium point which is when the strut is in the middle position. When the strut is compressed it is possible to see an important increase of the stiffness of the suspension due to the gas compaction in the gas charged accumulator. The results show also a high similitude with the presentation of the stiffness of a hydraulic suspension as function of the displacement presented in Hydropneumatic suspensions systems, 2011 [23], and reproduced in figure 16.

3.3.2. Conclusion on the suspension design

The vehicle has been presented with a focus on the different specificities of the suspension: low pressure tires, hydraulic strut and the powertrain. All design choices have been made in order to fulfil the different requirements of the soft soils and high off road capabilities presented in the second chapter of this report.
3.4. Powertrain system

The powertrain of the vehicle is a series hybrid powertrain. The selection of a hybrid system has been made by considering the numerous constraints given by the vehicle requirements and design.

The specific design of the vehicle suspension has conducted to design a vehicle with wheels hub motors. The choice of implementing a mechanical power transfer from motors placed above the suspensions to the wheels would have conduct to design an extremely complex and heavy transmission. The increase of weight would have compromised the ground pressure of the vehicle directly related to the weight of the vehicle. Moreover, wheel hub motors let the possibility to be controlled separately increasing the mobility of the vehicle in off-road terrains.

3.4.1. Basic Principle of series hybrid powertrain

Series hybrid powertrains already exists in numerous civilian vehicles. The Chevrolet Volt is the first series produced vehicle with a series hybrid powertrain. But series hybrid powertrains are not new they exist since years on sub marines and since 1986 on train [26].

The working principle of a series hybrid powertrain is explained in figure 29:

![Figure 29: Series hybrid powertrain scheme [27]](image)

The thermal motor (prime mover) provides the power of the vehicle through the generator. The battery plays the role of the power buffer and storage. The maximal output of a regular vehicle without hybrid powertrain is equal to the maximal power of the thermal engine. In a hybrid series powertrain the maximal output power is equal to the sum of the thermal engine power and the maximal output of the power storage device. This sum of power permits to downsize the thermal engine. Example a Chevrolet Volt has only a 60 kW thermal engine for a power at the wheels of 111 kW [28].

The objective of the implemented series hybrid powertrain is not only to avoid a complex transmission between the wheels and the motor which is the case with a regular transmission, but it is also to optimize the efficiency of the thermal engine by making the thermal engine turning at a fixed speed and load optimizing the efficiency, in order to optimize the range and the consumption of the vehicle.

As function of the speed and the load, the efficiency of a thermal engine can vary considerably as shown in figure 30. By fixing the load and the speed of the engine it is possible to highly increase the efficiency of the engine.
3.4.2. Implementation of series hybrid powertrain management

For classic vehicles the NEDC cycle of driving is the reference cycle of driving [30]. The calculations performed for downsizing the thermal engine of a series hybrid vehicle are calculated on this reference cycle. For an off road vehicle, standards cycles of driving don’t exist since the encountered terrains and missions can vary significantly.

In order to find a reasonable value of the size of the thermal engine and the power storage buffer, a simulation of the powertrain management is presented in chapter VI. Moreover a special management of the output power of the thermal engine is set up in the simulation in order to maximize the downsizing of the thermal engine. When the charge of the battery drops the thermal engine starts to turn at maximal efficiency point with the objective to recharge the battery (green point of load in figure 31). But if the battery continues to discharge due to a high demand of power coming from the driver, the thermal engine can be pushed to its maximal power with the effect of decreasing the efficiency but increasing the output power, in order to let the driver continue to drive at his wanted speed (red point of load figure 31).

With this special powertrain management it is possible to simultaneously get a vehicle with an extreme high power at the wheels corresponding to the sum of the power of the storage
buffer and the power of the thermal engine, a transmission capable to transmit the power to the wheels without a complex mechanical transmission and vehicle with a high range by the optimization of the thermal engine efficiency.

3.4.3. Conclusion on the powertrain

The powertrain of the vehicle offers the capacity to reach far away positions, by adapting the efficiency of the engine to the load. The series hybrid powertrain offers in the same time a large downsizing of the thermal engines.
Chapter IV.

4 Simple modelling of softs soils

This chapter aims at calculating a reasonable value of the rolling resistance of the vehicle in order to have realistic values for the powertrain sizing tool simulation present in chapter VI.

The rolling resistance is the first parameter responsible of the consumption of a vehicle; also it is necessary to have reasonable values of the soil rolling resistance. The study focus on a flexible tire model created with the Bekker equations and presented in Terramechanics and Off-Road Vehicle Engineering [15].

The modelling of different types of soils is a complex problem, which needs to take in account the different layers of the soil; the modelization of the soil as different layers is hard to approach with a simple modelling and need a FEM study [31]. The method presented in this part focus on considering the soil as a homogenous material. This approximation can be done if the fragile layer is not broken by the passage of the vehicle which happens when the pressure applied by the vehicle is less important than the critical pressure needed to break the fragile layer of the soil [15].

CREEL has especially studied the rolling resistance of vehicle on ice and snow areas as function of the density of the snow and the vehicle parameters. CREEL has developed a tool, the SNOW MOBILITY MODEL to approach the rolling resistance in snow. Sadly the first and the second version of the tool present a high level of error (up to 300 % of error [32]). The high level of error between the tool developed by the CREEL and the reality comes from the highly different types of snow encountered which can’t be approached with the simple variable of the density of the snow. The snow is made of crystal, the size and the crystallization level of the crystals of the snow creates cohesion in the snow. This cohesion is directly responsible for the load capacity of the snow. As consequence the method presented here focus on unfrozen soil as vegetation soil or homogenous soils as clay [16] [32].

In the method of a flexible tire, the rolling resistance is calculated from two different factors, the first factor is the energy used to compact the soil, and the second factor is the energy used in the tire deflection.

4.1. Flexible tire method

The vehicle is equipped with low pressure tires in order to minimize the vertical pressure and the disturbances applied to soil and increase the resistance capacity of the road to repetitive loadings as explain in chapter II.

The calculations are based on the calculation on a ground pressure applied by the tire lower than the ground critical pressure, if the tire ground pressure is above this certain ground critical pressure; the tire breaks the fragile top soil layer leading to an important sinkage of the vehicle. In this case the tire can be assumed as a rigid rim. Oppositely if the tire ground pressure is lower than the ground critical pressure, the tire has an important deflection, and cannot be assumed as a rigid rim, in this case a part of the rolling resistance comes from the tire deflection (figure 32).
The objective of the method is in a first step to calculate the sinkage of the vehicle in order to calculate the energy used to compact the soil, in a second step to estimate the value of the tire deflection in order to calculate the energy consumed by the tire deflection and finally to calculate an estimation of the rolling resistance of the vehicle in different types of soils.

The critical ground pressure $P_{gcr}$ described in the introduction as the pressure above which the fragile layer of the soil breaks can be approached with the Bekker equation (equation 17), where $W$ is the masse of the vehicle, $k_c, k_\phi$ and $n$ are soils constants, $b, b_{ti}$ and $D$ are tires parameters with $b$ the width or length of the contact patch, $b_{ti}$ the width of the contact patch and $D$ the diameter.

$$P_{gcr} = \left(\frac{k_c}{b} + k_\phi\right)^{\frac{1}{2n+1}} \cdot \left(\frac{3W}{(3-n) \cdot b_{ti} \sqrt{D}}\right)^{\frac{2n}{2n+1}}$$  \hspace{1cm} (17)

If $P_{gcr} > P_g$ where $P_g$ is the tire ground pressure, the vehicle ground pressure is less large than the critical ground pressure, the sinkage of the vehicle is low, and the tire has a large deflection.

The ground pressure under the tire can be approached as the pressure of the tire for low pressure tire in a first approximation (equation 18):

$$P_g = \text{tire pressure}$$  \hspace{1cm} (18)

The $b$ term in calculation of the critical pressure equation is the smallest parameter between the length and the width of the contact patch, by calculating and comparing the length of the contact patch to the width of the tire it is possible to obtain the $b$ term.

The length of the contact patch $l_t$ can be approached as:

$$l_t = \frac{W}{b_{ti} \cdot P_g} = 0.98m \quad > \quad b_{ti} = 0.675m$$  \hspace{1cm} (19)

As said above the $b$ term in the critical ground pressure calculation is the smallest parameter between the length and the width of the contact patch. Since the length $l_t$ of the contact patch is larger than the width the $b_{ti}$ term in the equation 17 is equal to the width of the contact patch.
Terramechanics of off road vehicles [15] presents different Bekker constant for different types of homogenous soils or with a small moisture layer. The calculation of the ground critical pressure has been performed for each of these soils (table 5):

<table>
<thead>
<tr>
<th>Terrain type</th>
<th>Bekker constants</th>
<th>Critical ground pressure kPa</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$n$</td>
<td>$k_c (kN/m^{n+1})$</td>
</tr>
<tr>
<td><strong>LETE sand</strong></td>
<td>0.705</td>
<td>6.94</td>
</tr>
<tr>
<td></td>
<td>0.611</td>
<td>1.16</td>
</tr>
<tr>
<td></td>
<td>0.804</td>
<td>3.93</td>
</tr>
<tr>
<td></td>
<td>0.728</td>
<td>NaN</td>
</tr>
<tr>
<td></td>
<td>0.578</td>
<td>9.08</td>
</tr>
<tr>
<td></td>
<td>0.781</td>
<td>47.8</td>
</tr>
<tr>
<td></td>
<td>0.806</td>
<td>155.9</td>
</tr>
<tr>
<td><strong>Upland sandy loam</strong></td>
<td>1.1</td>
<td>74.6</td>
</tr>
<tr>
<td></td>
<td>0.97</td>
<td>65.5</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>5.7</td>
</tr>
<tr>
<td></td>
<td>0.74</td>
<td>26.8</td>
</tr>
<tr>
<td></td>
<td>1.74</td>
<td>259</td>
</tr>
<tr>
<td></td>
<td>0.85</td>
<td>3.3</td>
</tr>
<tr>
<td></td>
<td>0.72</td>
<td>59.1</td>
</tr>
<tr>
<td></td>
<td>0.77</td>
<td>58.4</td>
</tr>
<tr>
<td></td>
<td>1.09</td>
<td>24.9</td>
</tr>
<tr>
<td></td>
<td>0.7</td>
<td>70.6</td>
</tr>
<tr>
<td></td>
<td>0.75</td>
<td>55.7</td>
</tr>
<tr>
<td><strong>Rubicon Sandy loam</strong></td>
<td>0.66</td>
<td>6.9</td>
</tr>
<tr>
<td></td>
<td>0.65</td>
<td>10.5</td>
</tr>
<tr>
<td><strong>North ower Clayley loam</strong></td>
<td>0.73</td>
<td>41.6</td>
</tr>
<tr>
<td></td>
<td>0.85</td>
<td>6.8</td>
</tr>
<tr>
<td><strong>Grenville loam</strong></td>
<td>1.01</td>
<td>0.06</td>
</tr>
<tr>
<td></td>
<td>1.02</td>
<td>66</td>
</tr>
</tbody>
</table>

It appears that the ground critical pressure of all the soils tested is inferior to the ground pressure applied by the tire equal in a first approximation to the inflation pressure of the tire. The ground critical pressure is between 1.1-2 bar, and the tire inflation pressure is between 0.4 and 1.2 bar.

Since the tire pressure is lower than the critical pressure it is possible to evaluate the rolling resistance by the flexible tire method.

The rolling resistance is composed of two terms calculated separately, the first term is the rolling resistance due to the soil compaction $R_c$, the second term is the rolling resistance due to the deflection of the tire $R_f$. The sum of the two terms of rolling resistance defines the total rolling resistance of the vehicle.
Since the tire pressure is under the critical soil pressure $P_{g_{cr}}$, the sinkage $z_e$ of the vehicle can be approached as (equation 20):

$$z_e = \left( \frac{P_g}{k_c + \frac{k_c}{b} + k_f} \right)^{\frac{1}{n}}$$  \tag{20}$$

With the sinkage of the vehicle it is possible to calculate the rolling resistance due to the compaction of the soil $R_c$ defined as (equation 21):

$$R_c = b_{ti} \times \left( \frac{k_c}{b} + k_f \right) \times \frac{z_e^{n+1}}{n+1}$$  \tag{21}$$

The results of the rolling resistance of the soil compaction $R_c$ are presented in table 6.

The results of the calculation of the rolling resistance from the tire deflection are more complex. The calculation is based on a semi empirical equation combining the definition of the angle of pressure $\alpha$ of the tire sinkage (figure 33), and an empirical factor $\varepsilon$.

![Figure 33: Flexible tire definition [15]](image)

To calculate $\alpha$ it is necessary first to evaluate the tire deflection $\delta_t$ (equation 21, equation 23, and figure 33).

$$\delta_t = D - \frac{D\delta_t}{2} \left( \frac{D}{2} - \left( \frac{l_t}{2} \right)^2 \right)$$  \tag{21}$$

The contact angle $\alpha$ can be estimated as (equation 22, figure 33):

$$\alpha = \cos^{-1} \left( \frac{D - 2\delta_t}{D} \right)$$  \tag{22}$$

The empirical factor $\varepsilon$ (equation 23) is combining the type of tire (radial or bias), and the height of the tire.

$$\varepsilon = 1 - e^{-k_e \frac{\delta_t}{n}}$$  \tag{23}$$

Where $\delta_t$ is the tire deflection (equation 21), $k_e$ a special variable ($k_e = 7$ for radial tires and $15$ for bias) of the tire and $h$ the tire section height.
With these different terms it is possible to calculate the rolling resistance coming from the tire deflection \( R_f \) (equation 24):

\[
R_f = \frac{3.581 \times b_{ti} \times D^2 \times p_g \times \varepsilon \times (0.0349 \times \alpha - \sin(2\alpha))}{\alpha (D - 2\delta_i)}
\]  

(24)

The results are presented on table 6. The total rolling resistance can be calculation by the sum of the two rolling resistance calculated (equation 25):

\[
R_{\text{total}} = R_f + R_c
\]

(25)

| Terrain type            | Bekker constant |  |  |  |  |  |  |  |
|-------------------------|-----------------|---|---|---|---|---|---|
|                         | \( k_c \) \( (kN\) n) \( /m^{n+1} \) | \( k_\phi \) \( (kN\) n) \( /m^{n+2} \) | critical ground pressure | sankage | \( R_c \) \% | \( R_f \) \% | \( R_{\text{total}} \) \% |
| Upland sandy loam       | 0.705 6.94 505.8 | 103 | 0.022 | 1.27 | 5.20 | 6.5 |
|                         | 0.611 1.16 475 | 108 | 0.014 | 0.84 | 5.20 | 6.0 |
|                         | 0.804 3.93 599.5 | 104 | 0.028 | 1.58 | 5.20 | 6.8 |
|                         | 0.728 NaN 1348 | 151 | 0.006 | 0.38 | 5.20 | 5.6 |
|                         | 0.578 9.08 2166 | 224 | 0.001 | 0.05 | 5.20 | 5.2 |
|                         | 0.781 47.8 6076 | 260 | 0.001 | 0.07 | 5.20 | 5.3 |
|                         | 0.806 155.9 4526 | 227 | 0.002 | 0.12 | 5.20 | 5.3 |
|                         | 1.1 74.6 2080 | 139 | 0.023 | 1.11 | 5.20 | 6.3 |
|                         | 0.97 65.5 1418 | 131 | 0.020 | 1.05 | 5.20 | 6.2 |
|                         | 1 5.7 2293 | 149 | 0.015 | 0.75 | 5.20 | 5.9 |
|                         | 0.74 26.8 1522 | 158 | 0.006 | 0.34 | 5.20 | 5.5 |
|                         | 1.74 259 1643 | 131 | 0.099 | 3.63 | 5.20 | 8.8 |
|                         | 0.85 3.3 2529 | 172 | 0.006 | 0.35 | 5.20 | 5.5 |
|                         | 0.72 59.1 1856 | 176 | 0.004 | 0.22 | 5.20 | 5.4 |
|                         | 0.77 58.4 2761 | 194 | 0.003 | 0.19 | 5.20 | 5.4 |
|                         | 1.09 24.9 3573 | 164 | 0.014 | 0.68 | 5.20 | 5.9 |
|                         | 0.7 70.6 1426 | 162 | 0.005 | 0.27 | 5.20 | 5.5 |
|                         | 0.75 55.7 2464 | 190 | 0.003 | 0.19 | 5.20 | 5.4 |
| Rubicon Sandy loam      | 0.66 6.9 752 | 127 | 0.009 | 0.56 | 5.20 | 5.8 |
|                         | 0.65 10.5 880 | 137 | 0.007 | 0.41 | 5.20 | 5.6 |
| North ower Clayley loam | 0.73 41.6 2471 | 194 | 0.003 | 0.16 | 5.20 | 5.4 |
|                         | 0.85 6.8 1134 | 128 | 0.016 | 0.88 | 5.20 | 6.1 |
| Grenville loam          | 1.01 0.06 5880 | 203 | 0.006 | 0.31 | 5.20 | 5.5 |
|                         | 1.02 66 4486 | 185 | 0.008 | 0.41 | 5.20 | 5.6 |

The results show that on homogenous soil, the rolling resistance of the vehicle equipped with flexible tires can be estimated as an average of 6%. For the vehicle powertrain simulation presented in chapter VI, it is necessary to input a realistic rolling resistance value. Since the simulation is made to test the powertrain in its limits, the value of the rolling resistance will be increased to 7% in order to maximize the power consumption of the vehicle.
Chapter V.

5  Vehicle dynamical simulation

A vehicle dynamic simulation is used to learn about the future vehicle behavior. The simulation phase during the conception of the vehicle allows to simulated numerous of tests manoeuvers like double lane change, slalom or others tests to observe the vehicle response and optimize it, by changing, for example the parameters of the suspensions, or the mass distribution on each axle. The objective of the simulations is also to avoid future unwanted phenomenon like the rollover of the vehicle during cornering. Examples for race cars, simulations of the vehicle are performed to improve the speed response and the stability of the vehicle. For off road vehicles, simulations of obstacle crossing allow analysing the forces applied in the suspension parts, and the accelerations experienced by the driver during the manoeuvers, and finally validate the design of the suspension.

5.1. Objectives of the simulations

The principal objective of the simulation for CNIM is to determine the forces experienced by each of the links and joints of the suspension when the vehicle is crossing a standard obstacle. This information is extremely important, and will help to design each part of the axles. The perquisite objective is to ensure by simulation that the accelerations taken by the driver will be in accordance with the maximum acceptable accelerations defined in the test manoeuver standards.

5.2. Description of the tests

For military vehicles, the tests applied on the vehicles are determined by the STANAG standards. The STANAG are the standards for NATO vehicles [33], these standards define obstacles and manoeuvers that vehicles have to be able to cross at a certain speed. The STANAG are highly exigent standards.

Even if the new CNIM vehicle is not a military vehicle, the tests performed in simulation on the vehicle are the STANAG tests. The STANAG tests are roughs tests, the ability for the vehicle to succeed in these tests will guarantee the capacity of the vehicle to pass in extreme terrains without breaking.

The tests of obstacle crossing and the test of sine wave driving are performed as presented in the STANAG AVTP 03-170, the 3rd test is only performed to highlight the need of an anti-roll bar on the vehicle.

5.3. Material for the tests

The dynamic simulation of the vehicle is performed with Simulink/Matlab and the third party program Delft-Tire from TNO-Automotive for the tire modelling which is not a feature included in Simulink/Matlab.

Delft-tire is an add-on on program for Simulink and Adams permitting to modelize tires. Delft-tire includes 5 different levels of tire modelling, from the simple Magic formula models, to rigid rings models [34], in this study, only simple models of tires are used since Michelin has not performed a full analysis of their tire yet, the tires parameters for advanced tire simulation are not available.
The model of the vehicle is divided in subsystems. Each subsystem represents a mechanical, a hydraulic, a control, or a sensor system, part of the full vehicle. The subsystems are connected one to each other, in order to build the full vehicle model (figure 34 and 35, table 7).
Figure 35: Full model - Simulink
### Table 7: Full model connections description

<table>
<thead>
<tr>
<th>Legend</th>
<th>Subsystem</th>
<th>Function</th>
<th>Connected to</th>
<th>Type of connection</th>
<th>Special inputs</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>World</td>
<td>Reference frame to the vehicle model</td>
<td>Body 2</td>
<td>6 – dof</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Cabin sensors 10</td>
<td>6 - dof</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Body</td>
<td>Inertia of the body</td>
<td>Steering actuator 7</td>
<td>Rigid</td>
<td>Cruise speed</td>
</tr>
<tr>
<td></td>
<td></td>
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<td>sensors 10</td>
<td>6 - dof</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Mechanical suspension</td>
<td>Link, parts and joins of the mechanical suspension</td>
<td>Wheels 6</td>
<td>Revolute</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Mechanical strut 4</td>
<td>Spherical</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Steering actuator 8</td>
<td>Rigid</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Speed calculator 9</td>
<td>Value (torque)</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Strut Mechanical</td>
<td>Link, parts and joins of the mechanical strut</td>
<td>Mechanical suspension 3</td>
<td>Spherical</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Hydraulic strut 5</td>
<td>Potential variables (speed and force)</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Hydraulic strut</td>
<td>Simulate the strut and the gas accumulator</td>
<td>Hydraulic connectors 11</td>
<td>Hydraulic connections</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Mechanical Strut 4</td>
<td>Potential variables (speed and force)</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Wheels</td>
<td>Simulate one wheel</td>
<td>Mechanical suspension 3</td>
<td>Revolute</td>
<td>Tire file and road file</td>
</tr>
<tr>
<td>7</td>
<td>Steering calculator</td>
<td>Calculate the angle of each axle</td>
<td>Steering actuator 8</td>
<td>Value (angle)</td>
<td>Turning radius</td>
</tr>
<tr>
<td>8</td>
<td>Steering actuator</td>
<td>Actuate the angle of each wheel</td>
<td>Steering calculator 7</td>
<td>Value (angle)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Mechanical suspension 3</td>
<td>Rigid</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Body 2</td>
<td>Rigid</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Speed calculator</td>
<td>Calculate the torque to the wheel to keep cruise speed</td>
<td>Mechanical suspension 3</td>
<td>Value (torque)</td>
<td>Cruise Speed</td>
</tr>
<tr>
<td>10</td>
<td>Cabin sensors</td>
<td>Sense the accelerations and positions of the cabin</td>
<td>Body 2</td>
<td>Rigid</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>World 1</td>
<td>6 - dof</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Hydraulic connectors</td>
<td>Connect all hydraulic struts</td>
<td>Hydraulic strut</td>
<td>Hydraulic connections</td>
<td></td>
</tr>
</tbody>
</table>
For a full description of each subsystem, please report to Appendix A – description of the vehicle model.

5.3.1. Hypothesis and simplifications of the model

On the presented model of the vehicle several hypothesis have been made, some hypothesis have been made due to Simulink limitations, other hypothesis come directly from the modelling which is never the perfect representation of the reality.

Hypothesis:
- The parts of the model are considered as fully rigid
- The joints of the system are considered perfect. All joints have no stiffness or damping. The joints are not bushing joints as in Adams - Car.
- The stiffness and damping of the suspension come only from the hydraulic components: the restrictions and the check valves of the system.
- Some joints have been modified in order to have an iso-static model
  - The mechanical connection between the mechanical strut and the mechanical suspension are normally revolute - bushing joints, they have been modified to spherical joints
  - The mechanical connection between the suspension and the body, normally allow a certain rotation, it has been replaced by a perfect revolute joint.
- The electric motors are not modelled. The torque applied to the wheel is a computed value.
- The hydraulic connections between the hydraulic cylinder and the gas charged accumulator has been adapted to increase the simulation speed as explained in the paragraph beneath.

The hydraulic connection between the hydraulic cylinder and the gas charged accumulator is normally constituted of two security valves and a restriction. The restriction limit and slow down the oil going to the gas charged accumulator making the damping, and the two check valves limit the phenomenon of over pressure if the vehicle hit an obstacle at high speed leading to a rapid lifting of one axle as explain in chapter III.

When the full hydraulic connection is modelled as on the scheme in figure 36, the solver has difficulties to make the model converge, leading to extra small steps of calculations. This is due to difficulties to equilibrate the oil flow between each check valve and the restriction.

![Figure 36: Extract of one of the hydraulic scheme](image)

To avoid extra small steps of calculation, it is possible to simplify the hydraulic modelling without modifying the level of the modelling. Check valves have a leakage parameter, which can be modified. The leakage parameter represents a fixed area which is always open letting the oil
flow through the check valves, even when the valve is closed. In the scheme in figure 37, it is possible to see how check valve operates.

![Diagram of check valve operation](image)

**Figure 37: Opening area of the check valve as function of the pressure [35]**

By setting up the leakage parameter of each check valve to the half of the value of the restriction, the effect of the two checks valves is the same as two perfect check valves without leakage and a restriction. This modification makes the hydraulic model of the suspension much easier to solve, leading to much more rapid simulation (figure 38).
5.3.2. Inputs of the simulation

The simulation needs two specific inputs files, the road file describing the road, and the tire file, describing all variables of the tire.

The road file describes the vertical elevation of the road as function of the traveled distance. Pre-created roads are available (sine road, plank, flat road), or a 3D discretize road can be created. In the presented dynamical simulations only pre created roads are used, for the obstacle crossing simulation the PlankRoad file is used, the file describes a road with a parameterized step. The sine road file is used for the others tests of the STANAG AVTP 03-170 [1], and the flat road for the slalom test [36].

The tire file is composed of all data of the tires. In this study, the most important data is the vertical stiffness of the tire, which varies simultaneously with the pressure of the tire. Since the parameters of the lateral stiffness are not available yet, the slalom test performed to study the stability of the vehicle is not representative of the reality and also only used to highlight the influence of the different hydraulic cross stabilizations configurations.

For a more advanced description of the roads or tires files please report respectively to appendix C and D.

5.3.3. Position of the sensors

The results of all simulations are extracted from different sensors placed on the vehicle. The sensors can record accelerations, speeds, and positions of a specific point or the forces and momentum applied on a joint.
The acceleration sensor and the pitch & roll sensor used on the obstacle crossing, sine wave driving and slalom test are placed in the cabin of the vehicle as shown in figure 39.

![Figure 39: Picture of sensor placement](image)

The forces sensors are placed in the wheels hubs, the results shown by the sensors cumulate the forces acting on the two wheels of one axle. The force sensors are placed on the first front axle pair, because the first front axle pair appears to be the most stressed of the vehicle.
5.4. Obstacle crossing test

5.4.1. Description of the obstacle crossing test

The obstacle crossing test is defined in the STANAG AVTP 03-170 [1], NATO standard test. The obstacle is constituted of an 10 inch high step with vertical edges.

The TNO_PlankRoad file is used with the following parameters (figure 40):

![Diagram of obstacle crossing test parameters](image)

Figure 40: Picture of the parameters of the plank road [33]

The parameter \(L_{Bevel\, edge}\) is set to 0, in order to create a step with vertical edges, the \(H\) parameter is set to 254 millimeters corresponding to the conversion of 10 inch in mm. And the length parameter of the step: \(L\) is set to 1 m.

The vehicle encountered the obstacle after 50 meters of driving \(x_{start}=50\,000\) mm. During this distance of travel, the vehicle equilibrates itself to reach an equilibrium state when it encountered the obstacle. The modelling in Simulink doesn’t allow to start the vehicle directly from an equilibrium point, and it is impossible in such complex system to find the prefect equilibrium start point of the vehicle. Also the vehicle has small oscillations during the first 50 m before the obstacle.
The pictures on figure 41 show the vehicle crossing the obstacle during the simulation:

![Figure 41: Picture of the vehicle crossing the obstacle](image)

Before establishing the criteria on the forces created at the wheel, by the passing of the obstacle, it is important to set up the values of the hydraulic restriction to select a good damping of the suspension, and to observe the influence of the parameters chosen during the design phase (wheel’s diameter, stiffness etc).

### 5.4.2. Influence of the restriction size

The restriction producing the damping in the suspension is the first studied parameter. By modifying the size of the restriction it is possible to modify the damping of the suspension. But the problem is more complex, because the damping of the suspension is nonlinear, during the obstacle crossing, the check valves open leading to nonlinear damping.

Different sizes of restrictions have been tested (by changing the leakage area parameter of the check valves). The variations of the size of the restriction directly impact the behavior. With a large restriction the oil flows rapidly in the accumulator and the chassis has a tendency to bounce, because the damping of the suspension is too low. Oppositely with a small restriction the suspension becomes highly damped but still bounce due to the opening of the check valves.

The tests have been performed with the lower tire pressure of 0.4 bar (equivalent tire stiffness of 240 N/mm) in order to create a maximum bouncing effect. Even if Michelin recommend driving at less than 30 km/h with tires inflated at less than 0.6 bar, the tests will be still performed with tires at low pressure, in order to maximize the bouncing effects.
The results in figure 42 show that with a too large restriction the vehicle continue to bounce after the obstacle and the bounce are almost not damped. With a too tight restriction the vehicle bounces are highly damped but the vehicle still bounce, this effect is due to the non-linearity of the damping of the suspension created by the opening of the check valves.

The results show also, that the cabin drops before being lifted up by the crossing of the bump. This behavior is due to the position of the sensor in the cabin above the first pair of axles and the high center of gravity of the vehicle. When the vehicle hit the bump, the vehicle is slowed down by a longitudinal force at the bottom of the wheel, leading to let the inertia force applied in the center of gravity of the vehicle pushing the body forward. The two forces create a momentum conducting the vehicle to pitch and also the sensors placed in the front of the vehicle to fall down at the start of the obstacle crossing (figure 42).

![Restriction opening influence on vertical motion of the cabin](image)

**Figure 42: Restriction opening influence on vertical motion of the cabin**

To select a good suspension damping which permits the suspension to absorb the bumps and simultaneously to avoid the bouncing of the chassis, a criteria selection is that the chassis makes only one bound and one rebound after a bump (figure 43).
Typically on the curve above, the 10 mm² restriction allows a 2nd bound and rebound before a stabilization leading to two full bounce, this behavior can make the driver feel dizzy and decrease the stability of the vehicle.

The results show that with a restriction of 18 mm² the vehicle makes only one significant bound and rebound after crossing the obstacle. When the restriction is fixed at 18 mm² the vehicle has also the optimal behavior. All furthers tests will be performed with the restriction fixed at 18 mm².

### 5.4.3. Influence of the tire pressure

Michelin has selected for the project a tire in accordance to the specifications of the vehicles (load per tire, available place, speed requirements, low ground pressure, etc), the parameters of the tire changes with the inflation pressure, with the simulation it is possible to see the influence of this tire parameter.

It is possible to see that normally, tires inflated at pressures under 0.6 bar should not be driven at more than 30 km/h [20]. Also the test of obstacle crossing defined in the AVTP 03-170 should not be performed with tires inflated as less than 0.6 bar. But in order to test the vehicle at its limits the tests are performed with under pressurized tires.

By observing the forces created at the wheel, it is possible to see the influence of the tire pressure on the applied torque by the crossing of the obstacle (figure 44).
The results show that the forces created at the wheel are up to 140 000 N with a highly pressurized tire and only 110 000 N for a low pressurized tire. The vertical stiffness of the tire is also a 1st order parameter in this obstacle crossing test. To establish the criteria of the maximum forces at the wheel, it is important to perform the maneuver with tires inflated at the highest pressure in order to maximize the force criteria of the suspension design.

5.4.4. Influence of the tire diameter

The tire diameter has also a role in the forces created in the tire. With a larger diameter the vertical forces created are less high, because the wheel hits the obstacle at a lower point decreasing the vertical forces. The tests of crossing are performed with a tire inflated at 0.6 bar and a vertical stiffness of 350 N/mm (figure 45).
The results show, that the maximization of the tire diameter is a factor to reduce the forces created at the wheel, but the results show also that the difference of created forces is relatively low. The influence of the tire diameter shows its limits when the tire diameter becomes so large that the size of the obstacle becomes negligible in front of the tire diameter, which is the case in this test.

The results show that the option to increase the diameter of the tire is not the best solution in order to reduce the forces created at the wheel since the influence of the tire diameter for a big wheel is small. Moreover for dimensional reasons (available space under the vehicle chassis), it is not possible to increase the diameter. The unloaded diameter of the wheels is also fixed to 1880 mm corresponding to the data of Michelin.

### 5.4.5. Accelerations passing the obstacle

The AVTP 03-170 standard define the maximum acceptable acceleration and pitch motion in the cabin of the driver, during the crossing of the obstacle the acceleration in the cabin should not be superior to 2.5 G vertically and the pitch angle 6° peak to peak. With a higher acceleration or pitch motion, the driver health can be affected.

The test of accelerations is performed with the tires at the highest pressure (1.2 bar, 430 N/mm) in order to maximize the accelerations in the cabin, and to be sure that the results show the maximum acceleration experienced by the driver. Moreover since all parts of the system are perfectly rigid, the acceleration experienced by the driver will be probably less high on the real vehicle, because several parts of the suspension should deform under the large load.

![Figure 46: Vertical acceleration in the cabin](image)
The results in figure 47 show the capacity of the vehicle to stay stable during the manoeuvre. The angles experienced by the driver peak to peak is maximum 2° and also don’t exceed the maximum acceptable limit of 6° defined in the AVTP 03-170 [1].

5.4.6. Forces at the wheel

The perquisite objective has allowed verifying that the vehicle equipped with the presented suspension fulfils the STANAG standards.

The first objective of the crossing obstacle test is to have an estimation of the maximum forces encountered by the suspensions of the vehicle in order to design the chassis and all the parts of the suspension.

All previous tests have enabled to observe and quantify the influence of the different parameters of the tire, they have also allowed finding the setting up of the tires parameters leading to a maximum solicitation of the suspension during driving.

By setting up the stiffness of the tire and the inflation pressure to the maximal values of 430 N/mm, and 1.2 bar, it is possible to maximize the forces created at the wheel (figure 48).
The results show that the maximal forces created at the wheel correspond to 2.33 times the vertical load vertically, and 1.5 times the vertical load longitudinally. The maximal forces can be expressed in G, the maximum forces are also 2.33 G vertically and 1.5 G longitudinally. These results allow selecting a criteria on the maximum forces to later design all the suspension parts. The criteria is defined by adding a security coefficient on the maximum encountered forces. The criteria is defined as 2.5 G vertically and 2 G longitudinally.

5.4.7. Conclusion of the obstacle crossing

The results of the simulation show that the vehicle should meet the requirements of the AVTP 03-170 obstacle crossing test, in terms of vertical acceleration, and in terms of angles of the cabin.

This validation of the behavior of the vehicle on the obstacle crossing test, has led to select a criteria on the forces applied on the wheel to design the suspension. The results show that the maximum forces on the wheel will be inferior to the selected criteria of 2.5 G vertically and 2 G longitudinally chosen with a security coefficient.
5.5. Sine wave driving

5.5.1. Description of the test

One of the other tests performed on the studied vehicle to be conform to the AVTP 03-170 [1] standards, is the sine wave driving. The AVTP 03-170 [1] defines different sine wave roads, with different vertical amplitudes and periods, which have to be taken at a cruise speed of 50 km/h. The NATO standards fix the maximal vertical accelerations in the cabin at a maximum of 2.5 G and the angle 10°.

The AVTP 03-170 [1] test defined the following sine wave roads (figure 49):
- Sine wave, period L=7 m, amplitude peak to peak H=20 cm
- Sine wave, period L=4 m amplitude peak to peak H=30 cm
- Sine wave, period L=1.8 m amplitude peak to peak H=15 cm

![Figure 49: Picture of the parameters of the sine wave road [33]](image)

Before validating the vertical acceleration in the cabin, it is possible on a sine wave road to observe the change of behavior of the vehicle as function of the different hydraulic connections between the hydraulic struts. Different configurations of hydraulic circuits can be imagined. In chapter III, the hydraulic configuration of the circuit is two circuits of two pairs of axles, which is the configuration equipping the vehicle during the slalom test and the obstacle crossing test. But it is also possible to imagine different hydraulic connections. Five different models are tested on sine waves roads, in order to compare the influence of the different hydraulic connections, and to compare the studied vehicle to a vehicle with regular mechanical suspensions.

5.5.2. Description of the different models tested

Five different models are tested in the sine wave driving test, four models are equipped with different hydraulic suspensions, and the last model is equipped with a regular spring and damper suspensions.

The first model is constituted of four independent hydraulic circuits; each pair of axles has its own. The model has the following hydraulic scheme (figure 50):
This model is called 1/1/1/1. This model simulates a basic hydraulic suspension; the load on each wheel is not equilibrated between the axles.

The second model is the model studied on the obstacle crossing test and slalom test. The vehicle has two hydraulic circuits, the two pair of front axles are connected together and the two rear pair of axles are connected together. This model is called 2/2. This model equilibrates the weight between the two front axles and the two rear axles.

The third model is called 3/1, the three front pair of axles are connected on the same hydraulic circuit, and the rear pair of axle is on a separated hydraulic circuit, as on the following scheme (figure 51).
The model called 1/3, has the front pair of axle on a single hydraulic circuit and the third rear pair of axles connected on another hydraulic circuit (figure 52).
In the last model, the hydraulic struts are replaced by mechanical springs and dampers struts.

5.5.3. Results

When the vehicle encountered the first bump, there is an important transitional behavior before the vehicle is stabilized. The results presented will focus on the stabilized behavior of the vehicle and not take in account the transitional behavior of the vehicle during the manoeuver.

To maximize the response of the vehicle to the road all sine wave driving tests are performed with highly pressurized tires at 1.2 bar with a vertical stiffness of 430 N/mm.

All models are first tested on a sine wave road with an extremely low period of 1.8 m to highlight the influence of the different hydraulic connections and at low speed of 20 km/h. A low period of sine wave is necessary to highlight the influence of the hydraulic connection, if the period is too long (more than the distance between two consecutive wheels), the difference of elevation between two consecutive axle is small, and do not permit to observe the equilibrium performed by the hydraulic suspension.

Figure 53 shows the results of driving on sine wave road with a period of 1.8 m and amplitude of 20 cm, the sine wave roads is taken at a cruise speed of 20 km/h.
Figure 53: Comparison of models on a sine wave road

The results show that the optimal configuration is 2/2 configuration. The 3/1 and 1/3 configurations have a similar behavior, but 1/3 configuration appears to be slightly more interesting than the 3/1 configuration. This is due to two different factors, the first is that the most solicited axle is the first front axle, also it is normal that the configuration 3/1 has a slightly better behavior, but the reason why the difference is more than remarkable is that the sensor is placed directly above the first axle, also in the configuration 1/3, the sensor senses more directly the roughness of the road. The mechanical suspension configuration appears to have the lowest filtering capacity due to the linearity of the spring.

It is possible to observe the vertical forces created by each strut on the 2/2 configuration (figure 54):

Figure 54: Forces on the different axles for the 2/2 configuration

The results show that the connected hydraulic struts are creating an almost equal force. The simulation shows that the equilibrium provided by the 2/2 connection is excellent since the
forces on the front axles are almost the same and the forces on the two rear axle (3 and 4) are also almost equal at any time.

Due to the important time consuming of the simulations at high speed, the different sine wave roads described in the AVTP 03-170 [1] will be tested only by the 2/2 model.

Results for the 7 m period sine wave road are shown in figure 55 and 56:

![Figure 55: Vertical acceleration 7 m – 20 cm](image1)

![Figure 56: Angle of the cabin sine wave road 7 m – 20 cm](image2)

For the sine wave of period 7 m and amplitude 20 cm the equilibrium of load made by the hydraulic circuits is less visible since the period of the sine waves is much longer than the distance between two pairs of axles. Also there is only a small distance to compensate between the two pairs of axles, since the two pairs of axles are almost at the same height.

The results show a relatively stable cabin, the accelerations and angles experienced by the driver are acceptable.
Results for the 4 m period sine wave road are shown in figure 57 and 58:

The results highlight that the 4 m period sine wave is the roughest road for the vehicle.

Results for the 1.8 m sine wave road are shown in figure 59 and 60.
The results show an almost unperturbed vehicle, this specific behavior is due to the extreme size of tires, which literally jump from one bump to the next one.

The accelerations and cabin angles experienced by the pilot of the vehicle are conform to the STANAG [1] standard which define the maximum acceptable acceleration to 2.5 G and the maximum angle experienced in the cabin to 10 ° peak to peak.

5.5.4. Conclusion on the sine wave roads tests
The tests have highlight the superiority of the 2/2 hydraulic configuration, and tests have enable to verify that the vehicle is conform to the STANAG AVTP 03-170 [1] standards in terms of sine wave driving.
5.6. Influence of crosswise stabilization

The hydraulic cross stabilization is a specific hydraulic connection between the different hydraulic struts of the vehicle creating a hydraulic anti-roll bar.

By changing the hydraulic connections between the hydraulic struts subsystems, it is possible to highlight the influence of the cross stabilization made by the hydraulic cross connections between each sides of the vehicle.

5.6.1. Description of the test

To highlight the influence of the hydraulic cross stabilization, a slalom manoeuver is inputted on the steering calculator subsystem of the vehicle. Even if Michelin has explain that the tire model will not simulate the real tires forces since the model is too simple to model the real lateral forces of a low pressure tire, the objective of this simulation is not to conclude on the lateral stability [36] of the vehicle but only to highlight the influence of hydraulic cross stabilization connection.

The input manoeuver is a slalom, a sine wave of 10 ° angle and period 1 rad/sec (figure 61) is inputted on the steering calculator subsystem, the vehicle will make successive curves with a minimum turning radius of 22 m which corresponds to the peak steering angles taken by the front and rear axles. The steering calculator computes the angle taken by each wheel as shown in figure 62. The center of rotation of the vehicle has been selected on the middle of the vehicle.

![Figure 61: Sine wave input](image-url)
5.6.2. Description of the three different models

Three different models are tested on the slalom manoeuver. The slalom manoeuver has for principal objective to highlight the need for an anti-roll bar, but also to see if a restriction on the cross stabilization is needed in case if the vehicle has a roll bouncing effect in the curves.

The first model is the regular model, with cross stabilization as explained in the chapter III.

The second model has the same characteristic of the first model, but the cross stabilization has been removed (figure 63).

In the 3rd model, the hydraulic suspension has been replaced by a standard mechanical suspension. This model has no anti roll bar.

5.6.3. Slalom manoeuver

On the manoeuver the sensor placed in the cabin will measure the roll of the cabin. Since the center of gravity of the vehicle is high, the vehicle has naturally, an important tendency to roll (figure 64).
The tests are performed with the parameters selected on the first part, the restriction is fixed at 18 mm², the inflation pressure of the tire is 0.6 bar (350 N/mm), and the vehicle is locked at a speed of 50 km/h by the speed calculator (figure 65).

As expected, the results show that the model equipped with the hydraulic cross stabilization has the best response. The second model has also a slightly better response than the 3rd model equipped with mechanical suspensions. This difference is due to the non-linearity of the spring rate of the hydraulic suspension. When the vehicle roll, the stiffness increases leading to a smaller roll than on the vehicle equipped with regular spring and damper suspension. The results show also that without an antiroll bar the vehicles cabin can roll up to 11°, which is extremely uncomfortable.

The results show that there is no need for another restriction between the two sides of the crosswise stabilization. If the results would have shown a roll bouncing of the vehicle, a restriction on the pipe of the crosswise stabilization will be needed. This lateral bouncing effect comes when the oil flows too easily from one side to the other. The modification of the following hydraulic scheme can be imagined to reduce this effect if it would appear (figure 66).
The restriction added limits the flow of oil switching between each side, creating a roll damping.

### 5.6.4. Conclusion on the slalom manoeuvre

The slalom manoeuvre has enabled to highlight that an anti-roll bar is a necessary feature on the future vehicle due to the high position of the center of gravity of the vehicle. Moreover, even if the test has not permit to conclude on the lateral stability of the vehicle, it has permit to show that there is no need for a restriction on the hydraulic cross stabilization since there is no roll bouncing effect of the vehicle during cornering.

### 5.7. Conclusion on the vehicle dynamical simulations

The different simulations presented in this chapter, has permit to find the optimal restriction opening in the hydraulic suspension, but also to find the best configuration of the hydraulic connections between the struts. Finally the different tests have permit to identify the configuration where the suspension is under its most important load, leading to validate the vehicle as conform to the AVTP 03-170 [1] (perquisite objective) and finally to select a criteria on the maximum forces applied on the suspension in order to later design the suspension.
Chapter VI.

6 Powertrain simulation

The selection of the size of the powertrain is an important step in the vehicle design process. Especially in a hybrid architecture where there are much more components to size than on classical powertrain architectures.

This part aims at presenting the powertrain tool created in order to later test and size the powertrain of the studied vehicle. The simulation of the power generation management is studied using Simulink with the Simscape library. The model created simulates the powertrain with the thermal engine, the alternator, the battery system, the electric motors and the vehicle in order to observe in a future study the influence of the different elements of the powertrain.

6.1 Objectives of the simulation:

The objective for CNIM is to create a simple model of a series hybrid motorization in order to test different size of components and study their influence in order to select the optimal size of the different components. Since the components of the vehicles have not been selected yet, this model created is only a tool created for a future study of the hybrid management.

6.2 Material of the test

6.2.1 Description of the model

Exactly like for the model created for the dynamical simulation, the powertrain model is divided in subsystems, each represents a specific function of the powertrain.

The model is based on two thermal engines, providing current to a battery system through an alternator, in order to power up the electric motors of the vehicle (figure 66 and 67).

For a more advanced description of the model please report to appendix D.

![Figure 67: Principle block diagram](image)
6.3. Hypothesis and simplifications of the model

On the model different hypothesis and simplifications have been made. Most of the hypothesis comes from simplifications done to increase the speed of the simulation and to have a model simple to calculate.

Hypothesis:

- The rolling resistance is a fixed value.
- The road is discretized; the road can be assimilated to different ramps.
- There are no losses between the electric motors and the wheels.
- The battery of the system is assimilated to a voltage source and a resistor in series. The battery is extremely simplified since the modelling of a battery is a complex area.
- The electric motors have no power limitations.

The battery is assimilated to a constant voltage source and a resistance placed in series. An ammeter is placed in series with the resistance. The ammeter counts the number of amperes going in or out from the battery as shown in figure 69. This information is used to calculate the level of charge of the battery.
Figure 69: Battery subsystem

The electric motor subsystems are assimilated to a transformation of the electric power to a mechanical power with a fixed efficiency. The electric motor subsystems take as much as needed power to move the vehicle at the selected speed from the battery block and the alternator block. Since there is not any saturation block on the demands of power from the electric motors, the electric motors have no power limitation.

6.4. Inputs of the simulation and parameter set up

The simulation needs different inputs files: the road file describing the road, the different levels of charge of battery triggering the start of the two thermal engines to their maximal efficiency or maximal power load point and the wanted speed of the vehicle.

The road file shown in figure 70, describes the height of the road as function of the travelled distance. The road file calculates the slope of the road between each given point. The road is discretizing in different slopes with a fixed gradient.

Figure 70: Road description

The test road describes an imaginary road in mountains; where the downhill have been replaced by uphill in order maximize the power demand of the vehicle. The road starts at an altitude of 0 m and finish above 9000 m altitude on an 80 km distance. This road in used as test in order to describe the most severe road that the vehicle can encounter.
The soft soil study performed in the first part of this report, has permit to find reasonable parameters of rolling resistance which is in a heavy vehicle the first parameter responsible for the vehicle consumption. From the data of table 6 it is possible to select an average rolling resistance encountered in extremes terrains. The rolling resistance has been set up to 7 %.

The working points of the two thermal engines are given points, the level of charge of the battery trigger the start of the two thermal engines. As explains in the hybrid power description chapter III the powertrain has two workings points, one point where the thermal engines works at maximum efficiency, in order to limit the fuel consumption and one point where they work at maximum power in order to compensate the high demand of power of the electric motors.

The parameters of triggering have been selected as under 80 % of level of charge of the battery, the engines start to recharge the battery by turning at its maximum efficiency level, and with a charge under 20 % the engines start to turning at full load with a decrease of efficiency. At the start of the simulation the battery is charged at 90 %.

For this test case the power of the two thermal engines is 150 kW for each engine and 300 kW for the battery system. These parameters have allowed the car to travel at a constant speed of 10 km/h in the defined test road (figure 70). The selection of this value is only to test the model.

6.5. Results

The study is focused on the charge and discharge of the battery regarding the output power of the electric motors.

Figure 70 describes the evolution of the charge of the battery. The curves of the power demand of the electric motors and the power provided by the two thermal engines are presented in figure 71.

![Figure 71: Output power of the thermal engines (blue) and the electric motors (red)](image-url)
The results show that when the battery hit a discharge of 80% at 4 km, the thermal engines start. But the result highlight that the thermal engines are not capable to recharge the battery system since between 4 km and 35 km the battery system continues to be discharged.

After 35 km the battery hit the level of 20% of charge triggering the launch of the thermal engines to their maximum output power, it is possible to see that even if the road goes seriously uphill (23% of gradability), the battery system is recharged. When the battery system are finished to be charged (level of charge above 90%), the thermal engines stop, and the vehicle is continuing to drive with the battery system only.

It is possible to see that the vehicle has the capacity to travel during important slopes to a fixed speed of 10 km/h which need a large power of propulsion, larger than the combined power of the two thermal engines.

On this specific test case, the powers of the different components have led to verify that the vehicle could be able to drive on the test road with the selected components. But since all components have not been selected, the efficiency selected on the different components (electric motors, thermal engines) are not representative of the real powertrain.

6.5.1. Conclusion on the powertrain simulation
The presented model is an efficient tool to pre size the powertrain of a series hybrid architecture of a special purpose vehicle which is not designed to be driven on roads. The special powertrain management of the vehicle permits a maximal downsizing of the thermal engine without decrease of the maximal output power of the vehicle.
Chapter VII.

7 Contributions, further work and conclusion

7.1. Contributions of the project

The first part of this project presented a summary of the literature study on off road vehicles, the study focus on the influence on the soil of the vehicle passages with the objective to find a vehicle which permits a maximization of the numbers of passages on a road.

The second part presented the suspension of the vehicle which completes the requirements presented in the literature study and the different specifications of the vehicle suspension: the low pressure tires, the hydraulic strut and the powertrain. This master thesis permits particularly to understand the interconnected suspensions.

Chapter IV aims at calculating the rolling resistance of the vehicle with a method from the literature permitting to taking in account that the vehicle is equipped with low pressure tires. The objective is to create reasonable values of the rolling resistance for the powertrain simulation in chapter V. Therefore this master thesis permits to understand and estimate the rolling resistance in soft soils of low pressurized tires.

In chapter V the focus is on the dynamical study of the vehicle equipped with interconnected suspensions. This master thesis highlight clearly by different simulations the advantages of interconnected suspensions presented in chapter III. Moreover the results of the simulations can be used for designing the suspensions.

Chapter IV presents the powertrain management sizing tool. This master thesis permits to better understand the specific power management presented in the chapter III, and permits simultaneously to have a start of sizing of the powertrain components.

Finally the report has presented the use of a new tool for studying vehicle dynamics – Matlab Simulink/Simscape is for the moment not a widespread simulation tool for studying vehicle dynamics since the version of Simulink/Simscape permitting the 3D simulation presented in this report is only available since September 2013.

7.2. Further work

This report provides the first results of a dynamical simulation of a heavy hybrid vehicle, focused on the dynamical behavior and the powertrain. The simulations have proven their benefits by permitting to pre-size the components of the powertrain and the different parts of the suspension. Some further work can still be done:

- The dynamic simulation has to be improved with the real data of the tires in order to study more precisely the lateral stability. Moreover the vehicle dynamical simulation has to be verified by a prototype to ensure the quality of the results.
- The forces applied at the suspensions have to be defined in terms of lateral force applied. During cornering or when the vehicle is blocked in ruts, the forces needed to turn the wheels can be high and need to be taken in account when the suspension is designed.
- The powertrain management simulation has to be improved with the real datas of all the components.
• The capacities of thermal engines to turn to their maximum load capacities have to be proven.
• The powertraction of the vehicle has to be estimated in order to validate the gradability capacities.

7.3. Conclusion

The study of the soft soil has enabled to highlight the key parameter of off road vehicles with a focus on the multi-passes capacities of the roads. The literature study has permit to highlight problems like the different layers of soils and the difficulties of soils to heal after the passage of different vehicles. The literature study has moreover permit to design the suspension presented in chapter III.

The dynamical simulation has enabled to highlight that hydraulic suspensions are a major advantage for off-road vehicles. The combination of the multiple specific features offered by hydraulic suspensions fulfil the requirements of the studied vehicle. The specific design of the suspension leading to a high center of gravity of the chassis is compensated by the anti-roll feature offered by hydraulic suspensions. More over the ability of hydraulic suspension to compensate the load applied on each tire is extremely useful for vehicles driving on soft soils where the wheels of the vehicle can slip or sink easily.

The dynamical simulation has also allowed having knowledge on the maximal forces which can be applied to the suspension longitudinally and vertically. This information is important in order to later design the parts of the suspension. It appears also that the large size of the tires diameter is considerably lowering the forces applied to the suspension, this effect of lowering the forces at the suspension is reinforced by the low stiffness of the tire which filtrate the forces applied.

The powertrain simulation has enabled to highlight the benefits of the specific power management of the series hybrid powertrain. This specific powertrain management allows minimizing the power output of the thermal engines without reducing the maximal output power of the vehicle. The combination of the power of the power storage device in combination with to the power of the thermal engine enables to keep a high power output for low durations.

Simulink has proven to be an extreme useful tool for dynamic simulations, the large capacities of the program to simulate different problems has enabled to simultaneously simulate the powertrain of the vehicle and the dynamical simulation. Moreover it is possible to connect the two simulations in order to simulate accurately the powertrain of the vehicle.
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8 Appendix A

This appendix presents the different subsystems of the full model of the vehicle.

8.1. World

The world block, fix the referential frame of the vehicle model. The model is connected to the world by a 6 dof joint.

8.2. Body

The body block has 9 communicators. All communicators are mechanical communicators, 8 communicators represent the links to each suspension. One communicator is a 6 degree of freedom link connected to the world frame.

8.3. Mechanical Suspension

The suspension block is composed of 5 mechanical communicators and 1 input variable. The link to the chassis, the links to each wheel, and the upper and lower link to the cylinder of the suspension are the mechanical communicators of the system. The subsystem input variables is the torque applied to each wheel by the electrics motors (figure 73).

Figure 73: Mechanical suspension block
8.4. Strut - mechanical
The mechanical cylinder subsystem corresponds to the two revolute joints on each side of the cylinder and to the prismatic joint of the mechanical cylinder subsystem. The mechanical part of the cylinder has two mechanical communicators, one input variable, and one output variable. Each mechanical communicator, correspond to one side of the strut where there are spherical joints. The input is the force produced by the hydraulic of the cylinder on the prismatic joint representing the strut, and the output is the displacement speed of the prismatic joint (figure 74).

Figure 74: Mechanical strut block

8.5. Strut- hydraulic
The hydraulic part of the strut subsystem has two hydraulic communicators, one input and one output variable. The communicators are hydraulic communicators, one for each chamber of the cylinder. Each chamber is connected to another chamber of another strut to create the anti-roll bar feature and to equalize the forces on each wheel. The input is the speed displacement of the prismatic joint of the mechanical part of the cylinder. The output is the force created by the hydraulic cylinder (figure 75).
As shown in the figure 75 the hydraulic connection between the cylinder and the hydraulic gas accumulator is composed of two check valves and no restriction. The hydraulic restriction let the oil flow only in case of a high differential pressure between the cylinder and the gas accumulator (figure 76).
In the model the check valves have a leakage variable, by setting the leakage variable of each check valve to half of the value of the restriction; the two check valves have the same behavior as two check valves and one restriction.

8.6. Wheel
Each wheel subsystem has one mechanical communicator. The subsystems communicator is the mechanical link to the suspension subsystem. The subsystem simulate the behavior of the tire, the subsystem needs the tire file and the road file (figure 77).

![Figure 77: Wheel block](image)

8.7. Steering calculator
The steering calculator subsystem computes the angle of each axle; the system has one input and 8 outputs. The input is the turning radius and the outputs are the angles of each axle.

8.8. Steering actuator
The steering actuator subsystem is composed of 2 mechanical connections and 1 input. The input is the turning angle of the axle. The subsystem turns the axle from the given angle calculated by the steering calculator. One mechanical connection is connected to the body, the other one to the mechanical suspension (figure 78).

![Figure 78: Steering block](image)
8.9. Speed calculator
The speed calculator computes the torque given at the wheels to keep the vehicle at cruise speed.

8.10. Cabin sensors
The cabins sensors are different sensors placed in the cabin. The sensors record accelerations, positions and angles taken by the cabin during the tests manoeuvres.

8.11. Hydraulics connections
Others blocks are added on the model to simulate the hydraulics pipe between each hydraulic cylinder and sensors are added to see the forces acting on each joint and the accelerations of the body and the drivers seats.
9 Appendix B
This appendix defines the different road files uses in the different tests manoeuvres.

9.1. Flat road – used for slalom tests

!
! road data file for TNO_flat_road
!
$-----------------------------------------------units
[UNITS]
LENGTH = 'meter'
FORCE = 'newton'
ANGLE = 'radians'
MASS = 'kg'
TIME = 'sec'
$-----------------------------------------------definition
[MODEL]
METHOD = '2D'
ROAD_TYPE = 'flat'
$-----------------------------------------------parameters
[PARAMETERS]
MU = 1.0 $ peak friction scaling coefficient
OFFSET = 0 $ vertical offset of the ground wrt inertial frame

9.2. Sine wave road

!
! road data file for TNO_sine_road
!
! single sine (wavelength: 2 m, height: 0.1 m)
! 0 deg. angle with respect to the driving direction
!
$-----------------------------------------------UNITS
[UNITS]
LENGTH = 'meter'
FORCE = 'newton'
ANGLE = 'degree'
MASS = 'kg'
TIME = 'sec'
$-----------------------------------------------MODEL
[MODEL]
METHOD = '2D'
ROAD_TYPE = 'sine'
$-----------------------------------------------PARAMETERS
[PARAMETERS]
MU = 1 $ peak friction scaling coefficient
OFFSET = 0 $ vertical offset of the ground wrt inertial frame
ROTATION_ANGLE_XY_PLANE = 0 $ definition of the positive X-axis of the road wrt inertial frame
HEIGHT = 0.15 $ height of the sinewave
START = 50 $ distance along the X-axis of the road to the start of the sine wave
LENGTH = 2 $ wavelength of the sine wave along X-axis of the road
DIRECTION = 0 $ rotation of the bump about the Z-axis wrt to the X-axis of the road
N_BUMPS = 60 $ number of consecutive sine bumps
9.3. Step road

! road data file for TNO_plank_road
!
! single cleat (length: 50 mm, height: 10 mm)
! 45 deg. angle with respect to the driving direction
!
$---------------------------------------------------------------------UNITS
[UNITS]
LENGTH = 'mm'
FORCE = 'newton'
ANGLE = 'degree'
MASS = 'kg'
TIME = 'sec'
$---------------------------------------------------------------------MODEL
[MODEL]
ROAD_TYPE = 'plank'
$---------------------------------------------------------------------PARAMETERS
[PARAMETERS]
MU = 1.0 $ peak friction scaling coefficient
OFFSET = 0 $ vertical offset of the ground wrt inertial frame
ROTATION_ANGLE_XY_PLANE = 0 $ definition of the positive X-axis of the road wrt inertial frame
HEIGHT = 254 $ height of the cleat
START = 50000 $ distance along the X-axis of the road to the start of the cleat
LENGTH = 1000 $ length of the cleat (including bevel) along X-axis of the road
BEVEL_EDGE_LENGTH = 0 $ length of the 45 deg. bevel edge of the cleat
DIRECTION = 0 $ rotation of the cleat about the Z-axis wrt to the X-axis of the road
DRUM_RADIUS = 1250 $ drum radius
10 Appendix C – tire file

This appendix defines the tire file used for the simulations.

```plaintext
[MDI_HEADER]
FILE_TYPE   = 'tir'
FILE_VERSION = 3.0
FILE_FORMAT  = 'ASCII'

FILE_NAME: TNO_car205_60R15_reduced_input.tir
TIRE_VERSION: MF-Tyre/MF-Swift 6.2
MF-TOOL TEMPLATE: TNO_mfswift62

COMMENT: Tire 205/60 R15
COMMENT: Manufacturer  Delft-Tyre
COMMENT: Nom. section width  (m)  0.205
COMMENT: Nom. aspect ratio  (-)  0.6
COMMENT: Rim radius  (m)  0.1905
COMMENT: Measurement ID
COMMENT: Test speed  (m/s)  16.7
COMMENT: Road surface  Asphalt
COMMENT: Road condition  Dry
COMMENT: Road curvature  0
DATESTAMP: 24 jul 2013, 07:11:10
USER: MF-Tool 6.2

Generated by: TNO
Copyright TNO 2013

$---------------------------------------------------------------- 
units
[UNITS]
LENGTH = 'meter'
FORCE = 'newton'
ANGLE = 'radians'
MASS = 'kg'
TIME = 'second'

$---------------------------------------------------------------- 
model
[MODEL]
FITTYP = 62  $Magic Formula Version number
TYRESIDE = 'Left'
LONGVL = 16.7  $Nominal speed
VXLOW = 1  $Lower boundary of slip calculation
ROAD_INCREMENT = 0.002  $Increment in road sampling
ROAD_DIRECTION = 1  $Direction of travelled distance

The next lines are only used by ADAMS and ignored by other MBS codes

USE_MODE specifies the type of calculation performed:
0: Fz only, no Magic Formula evaluation
1: Fx,My only
2: Fy,Mx,Mz only
3: Fx,Fy,Mx,My,Mz uncombined force/moment calculation
4: Fx,Fy,Mx,My,Mz combined force/moment calculation
```
| 5: Fx,Fy,Mx,My,Mz combined force/moment calculation + turnslip |
| +0: steady state behaviour |
| +10: including relaxation behaviour |
| +20: including relaxation behaviour (nonlinear) |
| +30: including rigid ring dynamics |
| +100: smooth road contact |
| +200: smooth road contact (circular cross section, motorcycles) |
| +400: road contact for 2D roads (using travelled distance) |
| +500: road contact for 3D roads |

Example: USE_MODE = 434 implies:
- combined slip
- rigid ring dynamics
- road contact for 2D roads

PROPERTY_FILE_FORMAT  = 'USER'
FUNCTION_NAME  = 'TNO_DelftTyre_Adams_interface::TYR815'
N_TIRE_STATES  = 5
USE_MODE  = 114 $Tyre use mode switch (ADAMS only)
HMAX_LOCAL  = 2.5E-4 $Local integration time step (ADAMS only)
SWITCH_INTEG  = 0.1 $Overrule local integrator when set to 0 (ADAMS only)

[DIMENSION]
UNLOADED_RADIUS  = 0.940 $Free tyre radius
WIDTH  = 0.883 $Nominal section width of the tyre
RIM_RADIUS  = 0.3883 $Nominal rim radius

[OPERATING_CONDITIONS]
INFLPRES  = 120000 $Tyre inflation pressure

[INERTIA]
FNOMIN  = 30000 $Nominal wheel load
VERTICAL_STIFFNESS  = 430000 $Tyre vertical stiffness

[STRUCTURAL]

[CONTACT_PATCH]
ELLIPS_MAX_STEP  = 0.001 $Maximum height of road step
ELLIPS_NWIDTH  = 1 $Number of parallel ellipsoids
ELLIPS_NLENGTH  = 20 $Number of ellipsoids at sides of contact patch

[INFLATION_PRESSURE_RANGE]
PRESMIN  = 20000 $Minimum valid tyre inflation pressure
PRESMAX  = 1000000 $Minimum valid tyre inflation pressure

[VERTICAL]
### VERTICAL_FORCE_RANGE

- **FZMIN** = 1000 \( \text{Minimum allowed wheel load} \)
- **FZMAX** = 200000 \( \text{Maximum allowed wheel load} \)

### LONG_SLIP_RANGE

- **KPUMIN** = -1.5 \( \text{Minimum valid wheel slip} \)
- **KPUMAX** = 1.5 \( \text{Maximum valid wheel slip} \)

### SLIP_ANGLE_RANGE

- **ALPMIN** = -1.5 \( \text{Minimum valid slip angle} \)
- **ALPMAX** = 1.5 \( \text{Maximum valid slip angle} \)

### INCLINATION_ANGLE_RANGE

- **CAMMIN** = -0.175 \( \text{Minimum valid camber angle} \)
- **CAMMAX** = 0.175 \( \text{Maximum valid camber angle} \)

### SCALING_COEFFICIENTS

- **LFZO** = 1 \( \text{Scale factor of nominal (rated) load} \)
- **LCX** = 1 \( \text{Scale factor of Fx shape factor} \)
- **LMUX** = 1 \( \text{Scale factor of Fx peak friction coefficient} \)
- **LEX** = 1 \( \text{Scale factor of Fx curvature factor} \)
- **LKKX** = 1 \( \text{Scale factor of Fx slip stiffness} \)
- **LHX** = 1 \( \text{Scale factor of Fx horizontal shift} \)
- **LVX** = 1 \( \text{Scale factor of Fx vertical shift} \)
- **LCY** = 1 \( \text{Scale factor of Fy shape factor} \)
- **LMUY** = 1 \( \text{Scale factor of Fy peak friction coefficient} \)
- **LEY** = 1 \( \text{Scale factor of Fy curvature factor} \)
- **LKY** = 1 \( \text{Scale factor of Fy cornering stiffness} \)
- **LKZC** = 1 \( \text{Scale factor of Mz camber stiffness} \)
- **LHY** = 1 \( \text{Scale factor of Fy horizontal shift} \)
- **LNY** = 1 \( \text{Scale factor of Fy vertical shift} \)
- **LTR** = 1 \( \text{Scale factor of Peak of pneumatic trail} \)
- **LRES** = 1 \( \text{Scale factor for offset of Mz residual torque} \)
- **LXAL** = 1 \( \text{Scale factor of alpha influence on Fx} \)
- **LYKA** = 1 \( \text{Scale factor of kappa influence on Fy} \)
- **LNYK** = 1 \( \text{Scale factor of kappa induced Fy} \)
- **LS** = 1 \( \text{Scale factor of Moment arm of Fx} \)
- **LMX** = 1 \( \text{Scale factor of Mx overturning moment} \)
- **LVMX** = 1 \( \text{Scale factor of Mx vertical shift} \)
- **LMY** = 1 \( \text{Scale factor of rolling resistance torque} \)
- **LMP** = 1 \( \text{Scale factor of Mz parking torque} \)

### LONGITUDINAL_COEFFICIENTS

- **PDX1** = 0.7411 \( \text{Longitudinal friction Mux at Fznom} \)
PKX1 = 6.504 $Longitudinal slip stiffness Kfx/Fz at Fznom

[OVERTURNING_COEFFICIENTS]

PDY1 = 0.8784 $Lateral friction Muy
PKY1 = -12.56 $Maximum value of stiffness Kfy/Fznom
PKY2 = 3.378 $Load at which Kfy reaches maximum value

[ROLLING_COEFFICIENTS]

[ALIGNING_COEFFICIENTS]

[TURNSLIP_COEFFICIENTS]
11 Appendix D – energy management model

11.1. Thermal engine
The thermal engine subsystem represents one thermal engine of the vehicle. The subsystem is composed of one input and one output. The input is the required speed [rpm] of the thermal engine, and the output is the mechanical rotational port of the thermal engine.

The thermal engine is directly represented by the Simulink/Simscape thermal engine block.

11.2. Load controller
The load control subsystem is the control block of the speed of the thermal engine, following the 3 points strategy presented in chapter III to achieve a maximal ratio of efficiency/power of the motor.

The subsystem takes as input the three different working points of the engine (speed and load torque), and the levels of charge of the battery to make a change of load point of the engine.

11.3. Alternator
The alternator has one rotational port, one electric port, and one input variable. The rotational port is the mechanical input of the alternator. On the rotational port is placed a resistive torque which correspond to the load inputted from the load controller permitting to calculate the output power of the alternator. The output power is then multiplied by the efficiency of the alternator, and finally converted in an electric wire connected to the battery and electric motor.

11.4. Battery pack
The battery pack is modeled from a constant voltage source and a resistance. The calculation of the level of charge of the battery is made by counting the amperes flowing though the constant voltage source. The battery pack has one electric port, corresponding to the electric wire connected to the alternator and the electric engine, and one output corresponding to the level of charge of the battery.

11.5. Electric motor
Each electric motor subsystem has one electric port and one rotational output. The electric port corresponds to the connection with the alternator and the battery, and the mechanical rotational port is connected to the vehicle.

11.6. Vehicle
The vehicle subsystem, has only one rotational input, the block simulate the vehicle as a point mass vehicle.