Modelling of the hydro-pneumatic suspension system of a rally truck

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Master Thesis

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Looking back at the project a lot of ups and downs have been encountered. The project could never have succeeded without the support and devotion of Ing. Jan-Willem van der Krol from V.S.E. and Dr. Ir. I.J.M. Besseling from the Eindhoven Univeristy of Technology. They helped the project to continue at dead points where critical decisions had to be made.

This project is the end of a Mechanical Engineering education of little over 6 years. During these years I learned the meaning of patience and the power of an overall overview. Sometimes it is better not to put your head in the ground and watch yourself get stuck on a specific topic, but to take one step back and see at what point exactly you are for the development of a thesis.

In these years my motivation for the automotive industry has grown even further. Most of my interest keeps focused on the vehicle dynamics topic. Therefore my special thanks to Prof. Dr. H. Nijmeijer and Ing. R. de Zaaijer, who both supported me in choosing my own thesis assignment.
Summary

The aim of this project is to develop a simulation model containing the hydro-pneumatic suspension system of the competition class GINAF Dakar Rally Truck. The model is validated by measurement data on both component level as well as full vehicle level. The model can be used for further research with respect to active damping control or implementing the system in other groups of vehicles like commercial delivery vans.

The hydro-pneumatic system uses an accumulator to generate spring force (similar to the HydrActive suspension system of Citroën) and a remote valve block to generate damping force. A hydraulic cylinder replaces the damper strut and springs of the vehicle. The cylinder generates oil volume displacement towards the accumulator. The oil is assumed to be incompressible and the volume of the air chamber inside the accumulator is diminished which creates a pressure increase by means of the “ideal gas law”. Higher pressure results in a higher reaction force and so a spring is established. The flows from the piston chambers and rod chambers of the cylinders are led through the tubing system and a remote damper manifold. In the damper manifold the oil flows create a pressure loss over the solenoid valves and other appendages in the tubing system. The valves are placed parallel for each flow and therefore the total flow-through area changes when one of the valves opens or shuts. Due to the pressure losses energy is dissipated and damping is generated.

Using the displacements of the four cylinders all system pressures can be derived with respect to spring and damping behavior. All appendage models are combined in the remote valve block of this suspension system model. The model of the hydro-pneumatic system is validated and appears to be a very good approximation of the system in real life. Some special concerns are built in such as oil fluid division over the valves, overshoot behavior of the pressure relieve valves and the interaction of the valve models.

Finally the full vehicle model is introduced and validated. The full vehicle model uses a multi-body model to relate the pressures within the suspension system with the load forces on the cylinders created by the vehicle mass and road profile. The multi-body model consists of four wheels, two independent suspended axles and a chassis with fixed loading space and cargo compartment. The suspension in this multi-body model consists of a lower triangle, an upper triangle, a cylinder and a tierod for each wheel separately. As a steering mechanism a steering box at the front axle is modeled. The suspension arms in the model create the appropriate installation ratio between wheel displacement and cylinder displacement for any given condition caused by road conditions and weight transfer. The cylinders are connected to the chassis at the top end and to the lower triangle at the bottom end, generating cylinder displacement and velocity.

In the full vehicle model some additional mechanical forces are incorporated as well, such as friction force and weight transfer. Position dependent damping reduces the spiky spring behavior created by the bump stops when the full cylinder stroke is reached. Friction created over the cylinder and the rotation points of the suspension reduces the number of spring movements needed before the vehicle returns to steady state driving. The inertia of the vehicle has a huge influence at the rear axle behavior but implies further tuning of the friction parameters as well to be sure the number of spring movements are kept equal. The height of the centre of gravity of the vehicle and a horizontally translational mass to represent the moving fuel inside the tank do not contribute to the total vertical vehicle driving dynamics much.

At the end of this project a full simulation model is available, validated and suitable for upcoming projects of V.S.E. After validation it appears that the original idea for splitting the damping behavior and spring behavior of the system is allowed to be modeled separately and be added to each other afterwards.
Samenvatting

Het doel van dit project is het ontwikkelen van een simulatie model van het hydropneumatische veersysteem van de GINAF Dakar Rally Truck uit de competitie klasse. Het model is gevalideerd aan de hand van metingen op zowel onderdelen niveau als ook het complete voertuig niveau. Het model is bruikbaar voor verdere studie met betrekking tot actieve demping of voor het onderzoek naar dit veersysteem in andere groepen van voertuigen zoals koeriersbussen.

Het hydropneumatische systeem gebruikt een accumulator om veerkracht te realiseren (hetzelfde als gebruikt in het HydrActive systeem van Citroën) en een gescheiden kleppenblok om demping te genereren. Een hydraulische cilinder vervangt de demperpoot en de mechanische veren van het voertuig. De cilinder creëert een olievolume verplaatsing naar de accumulator toe. De olie wordt onsamendrukbaar verondersteld en het gas in de luchtkamer van de accumulator wordt verminderd waardoor de druk in deze kamer stijgt volgens de “ideale gas wet“. Een hogere druk veroorzaakt een hogere reactie kracht en zo wordt een teer bewerkstelligd. De oliestromen van de zuigerkamers en stangkamers van de cilinders worden door een leidingsysteem en een gescheiden dempinghuis geleid. In het dempinghuis (gescheiden kleppenblok) wordt over de kleppen een drukverlies gecreëerd door de oliestromen als ook over andere appendages in het leidingsysteem. De kleppen in het dempinghuis zijn parallel geschakeld voor iedere stroming waardoor het totale doorstroomoppervlak verandert wanneer één van de kleppen opent of sluit. Door het verlies van druk wordt energie gedissipeerd en komt demping tot stand.

De interne drukken in het systeem met betrekking tot veer en demp gedrag van het voertuig kunnen worden berekend door de verplaatsingen van alle vier de cilinders te gebruiken. Alle aparte appendage modellen zijn gecombineerd in het gescheiden kleppenblok van het model van het dempersysteem. Het model van het hydropneumatische systeem is gevalideerd en blijkt een zeer goede benadering te zijn van het systeem in werkelijkheid. Extra aandacht is besteed aan olieverdeling over de kleppen in het dempinghuis, overshoot gedrag van de drukgevoelige kleppen en de communicatie over en weer tussen de kleppen.

Uiteindelijk wordt het volledige voertuig geïntroduceerd en gevalideerd. Het volledige voertuig model maakt gebruik van multi-body dynamica om de interne drukken in het veersysteem te verbinden met de kracht op de cilinders veroorzaakt door massa en wegprofiel. Het multi-body model bestaat uit vier wielen, twee onafhankelijk opgehangen assen en een chassis met cabine en bagageruim. De ophanging in dit model bestaat uit een onderste draagarm, een bovenste draagarm, een cilinder en een stuurstang voor elk wiel en een stuurhuis is gemodelleerd op de vooras. De draagarmen in het model kopen overeen met de werkelijke afmetingen en veroorzaken dus eenzelfde installatie ratio tussen wiel verplaatsing en cilinder verplaatsing. In het model komen overeen met de werkelijke afmetingen en veroorzaken dus eenzelfde installatie ratio tussen wiel verplaatsing en cilinder verplaatsing in alle gevallen veroorzaakt door gewichtsverplaatsing en wegprofiel. De cilinders zijn met de bovenkant verbonden met het chassis en met de onderkant verbonden aan de onderste draagarm van de ophanging.

In het volledige voertuig model zijn bovendien een aantal mechanische krachten opgenomen zoals wrijving en gewichtsverplaatsing. Positie afhankelijke demping reduceert het stekelige veergedrag veroorzaakt door de bump-stops wanneer de cilinder zijn maximale slaglengte bereikt. Wrijving over de cilinder en scharnierpunt van de ophanging vermindert het aantal veerbewegingen die benodigd zijn voor het voertuig om terug te keren in normale rij staat. De inertia van het voertuig heeft een grote invloed op het gedrag van de achteras maar beheft na installatie in het model een extra tuning van de wrijving om het aantal gesimuleerde veerbewegingen gelijk te houden aan het aantal gemeten veerbewegingen. De hoogte van het zwaartepunt en de horizontale verplaatsing van het zwaartepunt van de brandstof aan boord hebben nauwelijks invloed op het totale verticaal dynamische veergedrag.

Aan het eind van dit project een volledig voertuig simulatiemodel is beschikbaar, gevalideerd en geprepareerd voor het gebruik in verdere projecten van V.S.E. De keus om veergedrad en dempedrad van het voertuig te scheiden van elkaar blijkt na validatie van het systeem toegestaan waardoor ze apart gemodelleerd kunnen worden en aan het eind van de berekening weer bij elkaar opgeteld kunnen worden.
# List of symbols

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<thead>
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<th>Symbol</th>
<th>Description</th>
<th>Unity</th>
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<td>m²</td>
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<tr>
<td>(A)</td>
<td>area</td>
<td>m²</td>
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<td>(C_d)</td>
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<td>(D)</td>
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<td>(\Delta p_{st})</td>
<td>(stagnation) pressure loss</td>
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<td>(L)</td>
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1. Introduction

The GINAF Rally Power Team uses a hydro-pneumatic suspension system on their competition class rally trucks for the Dakar rally. This hydro-pneumatic system is developed by a company that once was part of the development department of GINAF (van GINkel Automobiel Fabriek). This company operates under the name V.S.E. (Vehicle Systems Engineering) and is located in Veenendaal, the Netherlands.

1.1 Project introduction

The system was originally developed for leveling tipper trucks on uneven roads combined with an axle load compensation system. This system is developed for driving over normal uneven road conditions and thus no additional damping system is included. The damping in this leveling system is formed by friction in the suspension joints and in the hydraulic cylinder and pressure loss created over the tubing system.

For the rally trucks this system is further developed by experience in the field and additional damping is added to the system by enhancing a damping manifold. This damping manifold is placed in a remote valve block located outside the damper strut. The development of this system by trial and error and experience takes a lot of effort and time and for new product development assignments this time will not be available.

Additionally to the system installed on the truck is a variable spring and damper system. Not only the ride height and stiffness can be varied but also the damping coefficient can be freely selected by the driver. For now, because of regulations of the Dakar rally, the damping and ride height are set during standstill of the vehicle and are fixed while driving. In the near future it is possible that an active damping system on the truck may be allowed.

In order to analyze the behavior of the vehicle using an active suspension system, it is easier to make a model first and to perform simulations using the control system for the active suspension. Therefore this simulation model should not only represent the overall vehicle behavior accurately but also has to represent the suspension components well.

For future applications time may not be available to such extend as it was for the development of the Dakar truck. One of the reasons is the customer hiring the service and knowledge of V.S.E. expects some bound on the duration of the design period of a new suspension system. Potential clients can be found in high speed delivery services using somewhat heavier vans. Another group of potential clients may be the manufacturers of ambulance vehicles. These vehicles should not only be able to manoeuvre over any obstacle varying from speed bumps to roundabouts; they have to drive over these obstacles fast and comfortably as well. It would be convenient to have a well functioning simulation model of the system to be able to serve upcoming clients. This simulation model can be adapted to the parameters and geometry of different types of vehicles.

1.2 Problem statement

The goal of this project is to develop a simulation model of the hydro-pneumatic suspension system of the GINAF Dakar Rally Truck and to verify the model by measurements. The catch here is that there is no good approximating model of a complete hydro-pneumatic system or applicable literature about this concrete topic available where both spring behavior and damping behavior are caused by oil flows inside the system. Additionally spring and damping behavior are internally influencing each other; different from an ordinary damper system where the coil spring is placed in parallel sequence with the damper strut.
The most convenient way to develop a model for investigating the dynamic behavior of this hydro-pneumatic system is to build a multi-body model of the suspension used in the vehicle which has to be validated as well. The multi-body model generates the installation environment for the hydraulics within the system. For a correct validation the project starts with the validation of the several components of the hydro-pneumatic suspension system.

Secondly the components are combined and validated as a full hydro-pneumatic system. Measurements are used as input signals and the simulated system pressures inside the hydro-pneumatic system are compared to the measured system pressures. Finally the full vehicle behavior is investigated using the multi-body model. Most important is to analyze the influence of various parameters on the vehicle.

1.3 Report outline

The report starts with introducing the working principle of the hydro-pneumatic system and the current stage of knowledge on this topic in chapter 2. Chapter 3 introduces the multi-body model and the most important parameters such as vehicle mass, suspension component masses and dimensions. Next a description of the mechanical part of the model the hydro-pneumatic part is given. The chapter concludes with the visualization of the truck.

The fourth chapter analyses the components used in the vehicle; the valves in the valve block are described as well as other appendages. Chapter 5 uses these components in an isolated suspension model. Here the pressure versus cylinder displacement is validated and it will become clear whether the hydro-pneumatic system is modeled correctly.

Chapter 6 provides a description of additional details in the full vehicle model. Apart from the hydro-pneumatic system described in chapter 5, the total vehicle model also includes additional damping forces in the vehicle as a snubber (position dependent damping) function and friction in the cylinder. For these forces there are no direct measurements available; however it is possible to investigate the influences these forces might have on the vehicle. Finally the conclusions and recommendations are given in chapter 7.
2. Hydro-pneumatic suspension

The majority of the top segment vehicles like the Lexus LS, the Mercedes S-class and the Audi A8 all use air suspension to create a high level of comfort with a low stiffness, adjustable ride height and adjustable damper characteristics. Citroën uses a hydro-pneumatic system well known for its high level of comfort as well and having the same potential to adapt the suspension; ride height and damping characteristics can be adapted to the road conditions the vehicle is driving. The GINAF Dakar rally truck uses a similar system as the Citroën HydrActive system available in their C5 series and C6 series.

In the first section the ordinary damper system and its properties are introduced. In the second section the relationship with the system of Citroën is discussed. In the final sections the hydro-pneumatic system of the GINAF truck is explained.

2.1 Ordinary damper strut

Starting with the principles of an ordinary damping system is the most convenient way to explain how hydro-pneumatic suspension is built-up. Figure 2.1 shows a schematic overview of an ordinary damper used in normal passenger cars [1].

![Figure 2.1: schematic overview of ordinary damper](image)

During compression the piston moves to the left and hydraulic fluid is flowing from the piston chamber towards the rod chamber. The displaced volume of the piston chamber is larger than the displaced volume of the rod chamber. This difference is caused by the stamp volume located inside the rod chamber.

\[ A_{piston} x = A_{rod} x + A_{stamp} x \]  

(2.1)

In this equation \( x \) is the displacement to the left of the piston within the damper, also referred to as the compression of the damper. The areas of the piston, the ring and the stamp are represented by \( A_{piston}, A_{rod} \) and \( A_{stamp} \) respectively.
\[ \Delta V_{\text{piston}} = A_{\text{piston}} x \]  
(2.2)

\[ \Delta V_{\text{rod}} = A_{\text{rod}} x \]  
(2.3)

\[ \Delta V_{\text{stamp}} = A_{\text{stamp}} x \]  
(2.4)

By using (2.2) to (2.4) the displacement of the cylinder can be rewritten to oil volume displacements as well.

\[ \Delta V_{\text{piston}} = \Delta V_{\text{rod}} + \Delta V_{\text{stamp}} \]  
(2.5)

The excess displaced volume has the same volume as the stamp area multiplied by the displacement and is represented by \( \Delta V_{\text{stamp}} \). This excess volume flows to the foot chamber circumcising the piston chamber.

The excess oil thus reduces the volume of the air chamber due to the incompressibility of oil versus the compressibility of the gas inside the air chamber. The pressure in the air chamber increases by the “ideal gas law” given in (2.6). The pressure of the oil chamber is equal to the pressure inside the air chamber.

\[ \frac{pV^n}{T} = C \]  
(2.6)

In (2.6) the temperature \( T \) is assumed to be constant during the pressure increase in the air chamber. \( C \) represents a constant value and \( n \) represents the polytropic coefficient. For adiabatic processes this coefficient is 1.4 while for isothermal processes this coefficient equals 1.0. The relation in (2.6) also applies to an accumulator. Looking closer at figure 2.1 it appears that the oil chamber and air chamber of the damper can be combined to form a similar expansion vessel as an accumulator. The accumulator thus is built inside the strut. When the damper is compressed the adjusted gas chamber pressure is given by:

\[ p_3 = \frac{p_0 V_0^n}{V_3^n} = \frac{p_0 V_0^n}{(V_0 - \Delta V_{\text{stamp}})^n} \]  
(2.7)

In (2.7) \( p_0 \) en \( V_0 \) refer to the pressure and volume as initially can be found in the air chamber of the damper. The pressure \( p_3 \) and volume \( V_3 \) refer to the compressed states inside the gas chamber of the damper strut. The pressure increases by compression and decreases by extension but always returns to the original unloaded situation. In conclusion this principle acts as a spring.

In ordinary dampers this spring principle is too small to support the full vehicle weight. The relation between pressure inside the gas chamber and stamp area is not in the same order of magnitude as the vehicle weight compressing the damper strut. Therefore coil springs are used parallel to the damper strut for support.
In steady state situations when there is no flow, the pressure inside the piston chamber and the pressure inside the rod chamber are equal to the pressure inside the oil chamber; or the pressure inside the piston chamber and rod chamber are equal to the pressure in the air chamber. When no damping is considered the force created by the damper strut is a pure spring force.

\[ F_{\text{spring}} = p_3 A_{\text{stamp}} \]  
(2.8)

The flow through the orifices creates a pressure loss over the orifices; multiplied by the area it is acting on, this loss is responsible for the damper force of the strut. Within the strut the internal pressures and states can be formulated by [1] and [2].

\[ p_1 + \frac{1}{2} \rho v_1^2 = p_2 + \frac{1}{2} \rho v_2^2 + \Delta p_{st} \]  
(2.9)

State 1 refers to the pressure within the piston chamber and state 2 refers to the pressure inside the rod chamber of the damper. Because the piston chamber diameter is much larger than the orifice diameter, the velocity of the fluid in the piston chamber is assumed to be zero. Also there is no tubing between piston chamber and rod chamber, therefore there is no stagnation pressure loss and \(\Delta p_{st}\) can be assumed zero as well.

Using \( v_2 = \frac{Q_{\text{piston}}}{C_d a_{\text{piston}}} \) and \( v_1 = 0 \) and \(\Delta p_{st} = 0\) reduces (2.9) to:

\[ p_1 - p_2 = \frac{Q_{\text{piston}}^2}{C_d^2 a_{\text{piston}}^2} \frac{\rho}{2} \]  
(2.10)

\(Q_{\text{piston}}\) is the flow through the summed area of all orifices in the piston, given by \(a_{\text{piston}}\). The discharge coefficient of the orifice is given by \(C_d\) and is dependent from both the size as well as the form of the orifice perimeter. The same holds for the foot valves between the piston chamber and the oil chamber of the damper strut.

\[ p_1 - p_3 = \frac{Q_{\text{foot}}^2}{C_d^2 a_{\text{foot}}^2} \frac{\rho}{2} \]  
(2.11)

State 3 refers to the pressure within oil chamber of the strut.

Combining equation 2.10 with 2.11 provides the total damper force generated over the orifices within the damper strut. Both the orifices in the piston as well as the foot orifices participate in generating pressure loss.

\[ F_{\text{damper}} = \left( p_1 - p_2 \right) A_{piston} + \left( p_1 - p_3 \right) A_{stamp} \]  
(2.12)

\[ = \frac{Q_{\text{piston}}^2}{C_d^2 a_{\text{piston}}^2} \frac{\rho}{2} A_{piston} + \frac{Q_{\text{foot}}^2}{C_d^2 a_{\text{foot}}^2} \frac{\rho}{2} A_{stamp} \]  
(2.13)
The spring force and damper force normally are combined inside a damper strut and provide a total force created by the strut. This damper force is derived next. In these equations the ambient pressure is left out of the equation because of its small magnitude compared to the pressures inside the damper strut.

\[ F_{\text{strut}} = F_{\text{spring}} + F_{\text{damper}} \]

\[ = p_1 A_{\text{stamped}} + \frac{Q_{\text{piston}}}{C_d} \frac{\rho}{2} A_{\text{piston}} + \frac{Q_{\text{foot}}}{C_d} \frac{\rho}{2} A_{\text{stamped}} \]  

Finally a problem is encountered about the value of the discharge coefficient \( C_d \). From literature it is not clear which value should be used; one source [1] states that in case of a ratio between the length of the pipe (cylinder) and diameter of the orifice exceeds 10 then \( C_d \) can be assumed constant. For smaller ratios when the piston is close to its end points another value for \( C_d \) should be used depending on the roughness of the tubing and the Reynolds number.

As a first approximation for dampers \( C_d = 0.7 \) could be used [2], although this value seems to change most of the times and an iterative process should be used to determine the right value for \( C_d \). The value for \( C_d \) applicable for small orifices used in the valves of the GINAF Dakar suspension system and the orifices in ordinary dampers is estimated to be 0.62 [2]. To compromise between the changing value of \( C_d \) at the end points of the cylinder [1] and the changing value of \( C_d \) by iterative processes between 0.62 and 0.7 [2] a fixed value for \( C_d = 0.65 \) will be used during the remaining part of the project.

### 2.2 Citroën HydrActive system

![Citroën HydrActive System](image)

**Figure 2.2:** overview of the Citroën HydrActive system

- 6 -
Citroën is probably the best known manufacturer using a hydro-pneumatic suspension system. This system forms the first step from ordinary suspension struts to the GINAF Dakar rally truck suspension. Figure 2.2 shows a total overview of the Citroën HydrActive suspension system [3].

In the Citroën system the air chamber and oil chamber are placed outside the main body of the damper. Because of different sizing of the stamp area and the pressure and volume inside the air chamber, this remote accumulator is responsible for the spring behavior of the vehicle and no additional coil springs are needed any further. The main body of the strut consists of the piston chamber and rod chamber, combined with the stamp. This is shown in figure 2.3, a single strut of the Citroën HydrActive suspension system.

The remote accumulator consisting of the air chamber and oil chamber is referred to in figure 2.2 by number 2 for the front axle and by number 5 for the rear axle. The HydrActive suspension system goes further than only the replacement of the damper struts of a conventional vehicle by the struts with remote accumulators; for roll stiffness and bump stiffness different accumulator volumes are used. As becomes clear in figure 2.2, the hydro-pneumatic part of the suspension system of the front axle and the rear axle are exactly the same, only the mechanical geometries of the axles are different. Figure 2.4 provides a schematic overview of a full axle hydro-pneumatic system.

In the centre of figure 2.4 an additional accumulator is located. This is indicated in figure 2.2 by number 3 for the front axle and by number 6 for the rear axle. This centre accumulator also is connected to the vehicle height control system. When driving at high velocity on a flat surfaced road the vehicle is automatically lowered by 13 mm [4].

When driving over a bump with both wheels both struts are compressed and all excess oil of both the left strut as well as the right strut has to be compensated by the air chambers in the accumulators. The pressure will increase accordingly in both the outer accumulators; the flow from the outer accumulators will go on to the centre accumulator. Now all three gas volumes are responsible for the spring behavior of the vehicle. The damping is still similar to the ordinary dampers discussed in the previous section.

When only the left wheel is driving over a bump, the pressure in the left accumulator will increase to such extend that the centre accumulator also will react. Because the road bump takes a short period of time and because
tubing between the accumulators left and right creates too much pressure loss, the pressure in the right accumulator will remain unchanged.

Driving through a left curve compresses the right strut and extends the left strut. Because of the symmetry of the system the excess oil flowing out of the right piston chamber fills the left piston chamber. The oil flow between the cylinders creates a large amount of pressure loss and thus damping. Similar to a vehicle equipped with ordinary coil springs and dampers the roll-in moment is damped strongly; later the vehicle does roll to its equilibrium position caused by the springs. There is practically no excess oil filling the accumulators and thus there is no spring force created against the roll behavior of the vehicle by the hydro-pneumatic system of Citroën. Therefore mechanical roll stabilizers are used in the Citroën in both as well the front axle as the rear axle. These mechanical roll stabilizers support the accumulators to resist the roll behavior of the vehicle.

Figure 2.4: schematic overview of full Citroën HydrActive axle

In conclusion the Citroën HydrActive system is able to have different characteristics for the three possible road conditions: driving over a symmetric road, driving over an asymmetric road and driving through a curve. Additionally the orifice between piston chamber and accumulator can be varied similar to turning the diaphragm of a microscope. This causes a slightly different damping characteristic during driving.

Oil is added to the system when increasing the ride height; gas pressures and gas volumes in the air chambers of the accumulators are not influenced by this process. The relationship between gas pressure and gas volume at the one side of the equation and the mass of the vehicle at the other side is explained by the next equations.

\[ F_{spring} = \frac{m_i g}{2} r_i \]  

(2.16)
The sprung mass for each of the axles is represented by $m_s$ and the final factor $r_i$ refers to the imaginary arm dependent on the geometry of the axle. This installation ratio can be defined by the following principle where $x_{\text{wheel}}$ is the vertical wheel displacement and $x_{\text{damper}}$ is the damper compression:

$$r_i = \frac{x_{\text{wheel}}}{x_{\text{damper}}} \quad (2.17)$$

For non-rolling conditions the pressure in the air chamber of the accumulator ($p_{\text{acc}}$) is equal to the pressure $p_3$ in the oil chamber in the accumulator and to the pressure $p_1$ in the piston chamber of the damper and finally also equal to the pressure $p_2$ in the rod chamber of the damper since there is no flow creating neither pressure loss nor damping; thus (2.16) can be rewritten to:

$$p_{\text{acc}} A_{\text{stump}} = \frac{m_i g}{2} r_i \quad (2.18)$$

In this equation there is no influence from the displaced volume.

### 2.3 GINAF Dakar hydro-pneumatic system

The GINAF Dakar truck suspension system employs the same principles as the Citroën HydrActive system. In the GINAF Dakar system all hydraulic fluid is led outside the cylinder chambers via a tubing system (see figure 2.5). The piston within the damper has no orifices at all and therefore the main body of this strut can no longer be referred to as a damper, but only as a cylinder. Also there is no additional accumulator between the left and the right strut and there is no mechanical roll stabilizer. The roll stiffness of the vehicle is created at the rear axle by using a passive hydraulic roll stabilization system. This will be discussed further on in this section.

Figure 2.5 clarifies the tubing system which leads the hydraulic fluid from the main body of the strut towards the remote damping block. This remote damping block is connected to the accumulator by a tubing system. The damping block contains solenoid valves which act as small orifices but actually are adaptable by changing the electric current controlling these valves. Therefore the orifices creating pressure loss will no longer be referred to as orifices but as valves.

The spring behaviour of the GINAF strut has not changed in comparison to the Citroën strut; for the air chamber (2.6) still holds:

$$\frac{p V^n}{T} = C \quad (2.6)$$

The equilibrium is achieved via the same relation between sprung mass of the vehicle at one hand and gas pressure multiplied by stamp area at the other hand as given in (2.18):

$$p_{\text{acc}} A_{\text{stump}} = \frac{m_i g}{2} r_i \quad (2.18)$$
The installation ratio $r_i$ can even be influenced by the height of the vehicle. When the vehicle height is adjusted, the angle of the cylinder with respect to the vertical axis is altered as well. However this change in angle later seems negligible when looking closer at the vehicle geometry (chapter 3).

![Schematic overview of single GINAF hydro-pneumatic strut](image)

The flow through the valve block creates a pressure loss over the valves similar to the orifices in ordinary shock absorbers. All valves are placed in a parallel sequence and the pressure differences can be calculated [5]:

$$p_1 - p_2 = \frac{\rho Q^2}{2 \left( a_1 C_{d1} + a_2 C_{d2} + ... + a_i C_{di} \right)^2}$$

Figure 2.5: schematic overview of single GINAF hydro-pneumatic strut

Here $p_1$ refers to the pressure in the piston chamber of the cylinder and $p_2$ refers to the pressure in the rod chamber of the cylinder. Areas $a_1$, $a_2$ and $a_i$ refer to the several flow-through areas of valve type $1,2,...,i$ respectively. When all geometries have the same sort of perimeter and are in the same order of size the discharge coefficients for all orifices may be assumed equal to the discharge coefficient of small orifices within an ordinary damper:

$$p_1 - p_2 = \frac{\rho Q^2}{2 C_d^2 \left( a_1 + a_2 + ... + a_i \right)^2}$$

Due to tubing $\Delta p_{st}$ in (2.9) can no longer be assumed to be zero. Therefore (2.20) has to be rewritten as:
The same holds for the pressure difference between the piston chamber of the cylinder and the oil chamber of the accumulator. Here the opened valve areas and the flow will differ from (2.21) to match the flow from the piston chamber towards the accumulator.

\[ p_1 - p_2 = \frac{\rho Q^2}{2 C_d^2 (a_1 + a_2 + \ldots + a_i)^2} + \Delta p_u \]  

(2.21)

The pressure loss over each appendage in the tubing system can be derived using:

\[ \Delta p_u = K \frac{1}{2} \rho v^2 = K \frac{\rho Q^2}{2A^2} \]  

(2.23)

The pressure loss factor \( K \) has different values dependent on the appendage. When looking at \( K \) for small orifices, the same formula for \( \Delta p_u \) appears after rewriting formula 2.5, using \( K = 1/C_d^2 \) [6].

The hydro-pneumatic suspension system at the front axle is a completely independent system for each wheel separately containing only the components as shown in figure 2.5 for each wheel. At the rear axle the hydro-pneumatic system of the left wheel is hydraulically connected to the system of the right wheel (see figure 2.6).

![Diagram of GINAF hydro-pneumatic rear axle](image)

Figure 2.6: schematic overview of GINAF hydro-pneumatic rear axle
When the truck drives a left curve, the left cylinder will extend. Hydraulic fluid is pressed outside the left cylinder rod chamber and will flow to the area where the pressure is lowest which is either the right accumulator or the right cylinder piston chamber. In symmetric road conditions this will be the accumulator and left rod chamber, however in case of a turn the pressure at the piston side of the right cylinder is lower. The oil pressed out of the rod chamber of the left cylinder tries to fill the piston chamber of the right cylinder in order to let the cylinder extend as well. This principle acts against the compression force of the right cylinder during a left curve and the crosslink therefore creates a strong force against roll behavior. There are no electronic or hydraulic actuators used in this system meaning this is a passive hydraulic roll stabilizer.

2.4 The hydro-pneumatic system in detail

In figure 2.7 the hydraulics scheme of the left front wheel is shown. In the appendix D the complete hydraulic scheme is given.

![Hydraulic Scheme](image)

**Figure 2.7: hydraulic scheme single wheel at front axle**

In figure 2.7 the dashed rectangular represents the damper manifold or remote valve block as schematically shown in figure 2.5. The open arrows represent the compression movement of the cylinder and the closed arrows correspond to the extension movement. Figure 2.8 and figure 2.9 show partial enlargements of the damper manifold as represented in figure 2.7.

During compression, the hydraulic oil is flowing from the piston chamber of the cylinder through two similar valves of type 3 which are shown in figure 2.7 and figures 2.8 creating a drop in pressure. These valves are normally closed valves and open when the flow exceeds a certain level while no electric current is applied.
However in the vehicle an electric current is applied to the valves which open the valve to a fixed opening area. No fixed areas are used in the vehicle, because all valves are able to vary their opening area as a function of the electric current. Fixed areas would not be able to create an active damping system.

Figure 2.8: compression side of damping block detail of the hydraulic scheme

When the pressure increases during compression, the fluid flows through a pressure relieve valve referred to in the figures as type 4. This last valve is only opened when the pressure difference between the accumulator pressure and the piston pressure exceeds a certain threshold value. After passing the valve block the oil directly flows to the rod chamber of the cylinder and the excess oil will be led towards the accumulator.

Figure 2.9: extension side of damping block detail of the hydraulic scheme

During extension, the hydraulic oil flows the other way around and is pressed out of the rod chamber of the cylinder. In this mode, the oil is led through only one normally closed valve (valve type 2) and one pressure relieve valve (valve type 5). After passing the valve block, the hydraulic oil directly flows to the piston chamber of the cylinder. The principle of the damper manifold for the flow coming from the rod chamber thus is equal to the principle of the damper manifold for the flow coming from the piston chamber.
The piston chamber of the cylinder can contain a higher volume of oil for the same displacement and additional oil flows from the accumulator towards the piston chamber. Due to the incompressibility of the oil, the volume of oil coming from the accumulator for extension is equal to the volume of the excess oil during compression.

All valves can be controlled during driving; however in the Dakar race it is not allowed to change the settings of the suspension while driving. The rally truck has to stand still before adjusting the settings to have no advantage in time over other teams.
3. The multi-body model

This chapter provides a description of the multi-body model used to simulate the dynamic behavior of the GINAF Dakar truck of 2008. The first section discusses all vehicle parameters applied in the model as well as the location where the data is stacked.

After the data introduction, the mechanical section of the suspension model is described. Here the several modeling layers are discussed and a clear overview in the model is provided.

In the third section the hydro-pneumatic system of the suspension is discussed. This is also modeled in a reduced model isolating the hydro-pneumatic system from the rest of the vehicle for a smarter validation process. This hydro-pneumatic system introduces the steps needed to derive the reaction forces applied on the cylinders of the vehicle. As input signals for this section the cylinder displacements and velocities are used.

Finally some pictures of the Virtual Reality animation are presented, giving an idea about the dimensions of the truck.

As a starting point for the mechanical part of the truck the model of the DAF Dakar Rally Truck of 2007 of M. Pinxteren is used [7]. This model is already equipped with a four wheel independent suspension system. The parameters of the geometry, weight and weight distribution have been updated.

### 3.1 Vehicle data description

In this section the vehicle data is described. First the coordinate system and its origin are discussed. Secondly some basic vehicle parameters will be discussed. Next the data parameter files are discussed which provide all vehicle data for the model.

#### 3.1.1 Axis system

The multi-body model of the GINAF Dakar rally truck is constructed in 3D space. The axis system used is a right handed system in which the directions are:

- Positive x axis: pointing forward
- Positive y axis: pointing to the left
- Positive z axis: pointing upwards

The origin lies at the following point:

- \( x = 0 \): centre of front axle
- \( y = 0 \): longitudinal plane of symmetry
- \( z = 0 \): road level

![Figure 3.1: axis system on GINAF Dakar truck](image-url)
### 3.1.2 Basic vehicle dimensions and masses

The basic vehicle dimensions of the GINAF Dakar rally truck of 2008 are given in table 3.1.

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>wheel base</td>
<td>4.40 m</td>
</tr>
<tr>
<td>Track width</td>
<td>2.10 m</td>
</tr>
<tr>
<td>x axis location of CG ($L_1$)</td>
<td>-1.76 m</td>
</tr>
<tr>
<td>tyre radius</td>
<td>0.64 m</td>
</tr>
</tbody>
</table>

**Table 3.1: vehicle dimensions**

The basic vehicle masses of the GINAF Dakar rally truck of 2008 is 11200 kg which is divided over the front and rear axle by 60 percent and 40 percent respectively.

The summed mass of the axles including rims and tyres is approximately 1500 kg per axle. An overview of the suspension masses is given in the appendix C. The mass of the cabin, loading space, engine, chassis, fuel tanks etc. has to be $11200 - 2 \times 1500 = 8200$ kg. The masses of the suspension parts give an indication about the sprung and unsprung mass of the vehicle. Parts belonging to the unsprung mass are the complete rim and tyre, the complete hub and half of the masses of the rods connected to the hub.

More details on the geometry of the GINAF Dakar Rally Truck of 2008 can be found in appendix A. The coordinates of the various connection points referred to in this appendix can be found in the confidential data appendix (appendix B) and originate from the original CAD design of the Timoney suspension. The rear axle is equal to the front axle rotated by an angle of 180 degrees around the vertical axis and translated by the wheel base over the length axis of the vehicle.

### 3.1.3 Vehicle parameter files

Several parameter files are written to define all parameters used in the vehicle. Most important is the file that starts the other files. The importance of files is shown in figure 3.3; the file on top starts the layer of files below.
In the ‘GINAF_dakar_truckdata.m’ file the overall parameters are given. The parameters can be adjusted here to try out new vehicle setups and concepts. Parameters defined in this file concern the overall vehicle:

- chassis frame parameters as mass and inertia
- chassis sprung and unsprung mass
- chassis weight distribution
- chassis C.O.G. location
- hydraulic fluid properties
- brake distribution
- steering rack properties

This overall script applies the front axle data file and rear axle data file as well.

As a final remark about this file it should be mentioned that the total vehicle mass measured at the RDW test track was 11200 kg. The vehicle is tested carrying 700 kg of fuel. The fuel tank is located at the back of the truck near the rear axle. To compensate the weight distribution as well as to maintain the 60-40 distribution, the frame weight distribution is derived; calculations provide that the longitudinal location of the centre of gravity of the frame including cabin and cargo departments should be 32 percent of the wheelbase removed from the front axle.

The movement of the fuel is prevented by applying a stiff spring without changing the weight distribution of the vehicle. The vehicle may later be investigated with less fuel on board; the model then automatically adapts to this situation and will provide the results for a vehicle carrying less mass and having a different overall weight distribution.
3.1.3.2 **GINAF_dakar_frontaxle and GINAF_dakar_rearaxle**

These files are covered in one section because they practically are exact copies. The values inside might differ from each other but the line-ups of the files are similar to each other. These files provide data concerning:

- track width
- hub mass and inertia
- upper rod and lower rod mass and inertia
- steer rod mass and inertia
- suspension geometry coordinates
- cylinder properties such as initial length and inside areas
- accumulator properties such as initial pressure and volume
- damper manifold properties such as electric currents
- locations of C.O.G. of the suspension links
- orientation of the suspension links w.r.t. the world

At the beginning of these scripts the appendage file is applied to load the data for the various valves in the damper manifold. The axle files also calculate the initial pressure inside the gas chamber using (2.18). The sprung mass for each of the axles is represented by $m_s$ and the final factor $r_i$ refers to the installation ratio.

$$p_{ace} A_{stump} = \frac{m_s g}{2} r_i$$  \hspace{1cm} (3.1)

This derived steady-state accumulator pressure is set by the parameter file as initial accumulator pressure for the model combined with the corresponding accumulator volume.

Introducing the vehicle mass of 11200 kg and a weight distribution of 60-40 over front axle and rear axle respectively, the derived initial pressures are 125 bar for the front axle and 113 bar for the rear axle. The polytropic coefficient $n$ is set to 1 for this process unlike for the driving conditions where $n$ is 1.2. Therefore the steady state calculations are derived in advance and applied.

3.1.3.3 **GINAF_dakar_appendages**

The final file in the overview of figure 3.2 enhances all properties of the appendages used in the vehicle. The properties provided in this file are not given by the supplier but measured afterwards by V.S.E. itself in combination with the supplier. The data concerns:

- pressure loss factors for elbows, fittings and hoses
- properties for check valves
- properties for normally close valves
- properties for pressure relieve valves

The properties for the valves exist of opening area versus electric current and opening area dependency on flow for all valves, crack pressure versus electric current and overshoot behavior function for the PRV valves.

At the end it should be mentioned that for the length of hose tested (1 meter), most of the pressure loss is caused by the two connectors of the hose and not by the hose itself. The hose itself is capable of generating about 2 bar of pressure loss per meter, while the connectors are capable of generating over 30 bar.
3.2 Mechanical section

The mechanical section needs a steering angle, throttle and brake signal in time as input signals. The throttle can be combined with a cruise control to maintain a given velocity. Figure 3.4 shows the overall vehicle model and the signals entering the model to simulate driving the vehicle over the given road conditions.

Figure 3.4: simulation environment

The first layer of the simulation model enhances the chassis, the front axle subframe and the rear axle subframe. This layer is visualized in figure 3.5. Besides the axle systems, a simplified drivetrain and braking system can be found as well. Both these systems generate wheel moments for acceleration and deceleration respectively. The moments are applied in the wheel bearings for each wheel separately. The main focus is the dynamic behavior of the vehicle; the model is simplified concerning drive train and brake installation. Therefore engine parameters are not included in the model. The wheels and tyres are directly connected to the wheel bearings in the Tyre blocks.

In figure 3.5 the hydro-pneumatic suspension system is represented by the “Suspension System” block. This subsystem uses the displacements and velocities of the cylinders from the axle systems to derive internal pressures and forces. These forces are then applied as reaction forces on the cylinders in the axle systems. The “Suspension System” block will be discussed in section 3.3.
3.2.1 Chassis system

The chassis is the main system of this multi-body simulation model. The chassis system enhances the contact to the machine environment (left upper corner of figure 3.6). Here for example the initial vehicle velocity is loaded in the model. The frame body loads the data concerning mass, inertia and location of the vehicle chassis including the cabin and cargo compartment. In the left lower corner fuel is added to the frame and restricted to a pure horizontal movement over the x-axis of the vehicle.

The frame inside the chassis system has a fixed location of gravity independent from the amount of fuel on board. First the overall vehicle weight distribution including fuel has to be derived to derive the horizontal location of gravity for the frame including the cabin and cargo department but without the fuel:
Here \( m_f \) and \( m_r \) are the masses of the vehicle for the front and rear axle respectively. The gas chamber pressures inside the accumulators are represented by \( p_{acc} \). The next equations provide that the horizontal location of the C.O.G. for the frame including cabin and cargo department but without fuel is located at 32 percent of the wheelbase behind the front axle.

\[
m_f = m_{frame,x} + m_{axle} \quad (3.4)
\]

\[
m_r = m_{frame}(1-x) + m_{axle} + m_{fuel} \quad (3.5)
\]
The subframe body is located in the centre of the axle system. This body is connected to the chassis system by a weld at the top of the figure. The location of the axle subframes is provided in the chassis frame body and via the connection the coordinates for the axle subframe are loaded in the model. The wheel hubs are placed at the outside of the figure and form the missing link between the subframe of the axle and the bearings and tyres in the layer above.

![Figure 3.7: front axle model](image)

The connection to the wheel hubs is mirrored for the left and right side. This connection is established by modelling the upper rod, the lower rod and the steering rod. At the front axle the steering rods are connected to a steering rack; at the rear axle the steering rods are directly connected to the subframe by the same universal joints.

The lower rod in reality has two pivot points at the subframe side. For practical reasons these two pivot points are combined to a single revolute joint enhancing the same degrees of freedom. The coordinates for the suspension are loaded in the model in the main bodies of the suspension: the axle subframe body and the wheel hub bodies. The placement of the coordinates can be found in appendix A and appendix B.

The hydraulic cylinders are connected to the subframe by a spherical joint to maintain the three rotational degrees of freedom and a universal joint is used to connect the cylinder to the lower rod. The forces derived in the Suspension System block act on these cylinders as reaction forces to apply the forces on the vehicle behavior. The signals from the cylinders are output signals for the axle subframe systems and sent towards the “Suspension System” block which will be further discussed in section 3.3.
3.2.3 Cylinder system

The “Cylinder” system blocks represent the cylinders in the vehicle where the spring and damping force are applied. The upper section of the cylinder is connected to the chassis by a spherical joint. The lower section is connected to the lower rod by a universal joint. These bodies enhance the one degree of freedom movement by a prismatic joint. The cylinder itself is assumed massless since it has no real contribution to the unsprung mass or to the vehicle dynamic behavior.

![Cylinder model diagram](image)

Figure 3.8: cylinder model

Normally the cylinder would only exist of the left prismatic joint to represent the longitudinal movement of the cylinder. A sensor on this joint measures the displacement and velocity for further use in the “Suspension System”. There the forces created by the hydro-pneumatic system are derived. Spring force and damper force are combined here to generate a total cylinder force.

An additional prismatic joint (the right joint in figure 3.8) is introduced in the cylinder. This joint is responsible for the height control of the vehicle and is comparable to adding oil in the cylinder. The displacement caused by the height control is subtracted from the total displacement in the “Suspension System” block to isolate the driving displacements from the total cylinder displacements. In real the height control is used in static conditions and can be subtracted from the total displacement as well.

3.3 Hydro-pneumatic section

The mechanical section of the suspension is build-up in an equal way for the hydro-pneumatic suspension system as for an ordinary suspension system. The only difference is the replacement of the ordinary linear spring and damper strut by a hydraulic cylinder. Where the oil remains in the strut at an ordinary damper, here all oil is led outside through the damper manifold towards the accumulator. This section introduces the various hydraulic components which are developed for the model.

3.3.1 Suspension System

As mentioned in section 3.2, the displacement and velocity of the cylinder are measured in the “Cylinder” blocks. These signals are sent towards the “Suspension System” block incorporating the accumulators, the damper manifolds, the bump stops and additional damping such as friction. Directly after entering the Suspension System block the signals are diverted to the front axle system and the rear axle system.
Figure 3.9: suspension system model

Figure 3.10: axle suspension box model

Figure 3.11: force balance model

Figure 3.9 shows that the displacement and velocity of all four cylinders enter the "Suspension System" block and are applied to their specific axle system. In the suspension boxes the forces are derived which are applied to the cylinder in the mechanical section of the model.
The forces sent to the cylinders exist of a sum of the cylinder spring and damping force, additional spring force generated by the bump stops and additional damping force generated by friction. This summation is done at the “Force balance” blocks inside the axle suspension boxes (see figures 3.10 and 3.11).

Since the joint sensors in the cylinder measure extension as a positive displacement, the displacements and velocities are adjusted to fit the unwritten law for shock absorbers: compression of the cylinder should be written as the positive direction of displacement. The signal adjustment is located at the -1 gain in figure 3.10 before the signals enter the hydro-pneumatic system.

### 3.3.2 Additional Suspension Forces

The displacement and velocity of the cylinder enter the actual hydro-pneumatic system enhancing the accumulators and the damping manifolds. Parallel the forces created by the bump stops, the position dependent damping and static friction are derived.

![Figure 3.12: bump stops model](image)

The bump stops are modelled as stiff linear springs. The model for the bump stops is given in figure 3.11. First the domains for the limits of the cylinder are defined; then the excess displacement for compression or extension is derived and finally the displacement is multiplied by the stiffness of the bump stop to derive the spring force created by the bump stops.

![Figure 3.13: window function for position dependent damping](image)
Parallel to the bump stops the displacement and velocity signals are used for the position dependent damper model (snubber) and the velocity alone is used for the static friction model. The position dependent damping is created when the cylinder is compressed or extended towards its limits. The position of the cylinder acts as a window function ($W$) in this equation. This position dependent damping reduces the bounce behavior generated by the bump stops.

$$F_{\text{snubber}} = W \cdot \dot{x} \cdot d_{\text{snubber}} \quad (3.6)$$

Here the velocity is represented by $\dot{x}$, the damping coefficient is given by $d_{\text{snubber}}$ and the window function $W$ is defined as shown in figure 3.13.

The window function is given in the “Snubber on/off” block (see figure 3.14) and multiplied by the linear damping coefficient to create the position dependent damping force.

![Figure 3.14: position dependent damping model](image)

The friction is modeled as Coulomb friction and therefore a direct function of the cylinder velocity. The friction is added in the model as shown in figure 3.15.

![Figure 3.15: friction model](image)

### 3.3.3 Hydro-pneumatic system

The actual hydro-pneumatic system incorporates the accumulators, the damping manifolds and the tubing system. The initial idea was to split the spring behavior from the damping behavior of the vehicle and to form two different relations which do not influence each other. Spring behavior relates to the oil volume displacement at the one hand, damping behavior relates to oil volume flow at the other hand. During the project it became clear that it was allowed to separate this spring behavior and damping behavior. The validation discussed in chapter 5 provides great results for the split system where spring force and damper force are added at the end of the line inside the cylinder system to create a single cylinder force. The final hydro-pneumatic system model is shown in figure 3.16.

A separate model has been developed, isolating this hydro-pneumatic system for all four cylinders from the full vehicle model for step-by-step validation. This separate model is used in chapter 5 of this report. Input signals for this isolated suspension-box model are the four cylinder displacements and velocities. Output signals are the flows from the piston chambers, the flows from the rod chambers, the pressures of the piston chambers and the pressures of the rod chamber of the cylinders, the four accumulator pressures and the hydraulic cylinder forces.
The oil volume displacements and velocities are derived next and the equations can be recognised in figure 3.16.

\[
\Delta V_{piston} = A_{piston} \dot{x} \tag{3.8}
\]

\[
\Delta V_{rod} = A_{rod} \dot{x} \tag{3.9}
\]

\[
Q_{piston} = A_{piston} \ddot{x} \tag{3.10}
\]

\[
Q_{rod} = -A_{rod} \ddot{x} \tag{3.11}
\]

The “Accumulator” models the relation between gas pressure and oil volume displacement and the “Damping Manifold and tubing system” models the relation between pressure losses and oil flows from either the piston chamber of the cylinder or the rod chamber of the cylinder. These systems will be discussed in the upcoming sections.

The pressures are converted to forces after passing through the accumulator and damping manifold models. The forces are applied to the hydraulic cylinders. First the forces generated by the bump stops, the position dependent damping and the friction are added to the cylinder forces (inside the “force balance” blocks, section 3.3.1) before sending them to the mechanical section of the full vehicle multi-body model.

\[
F_{piston} = A_{piston} P_{piston} \tag{3.12}
\]

\[
F_{rod} = A_{rod} P_{rod} \tag{3.13}
\]

\[
F_{cylinder} = F_{piston} - F_{rod} \tag{3.14}
\]

\[
= A_{piston} P_{piston} - A_{rod} P_{rod} \tag{3.15}
\]

At the rear axle a hydraulic roll compensator is modelled. By connecting the rod volume displacement and flow of the left cylinder with the right piston volume displacement and flow and the right accumulator with damping manifold, this crosslink is established. Figure 3.17 shows the hydro-pneumatic system for the rear axle.
3.3.5 Accumulator

The accumulator is modelled on basis of the “Ideal Gas Law”. First this excess oil displacement is derived, and then the gas states within the air chamber of the accumulator are calculated:

\[ A_{\text{stamp}} = A_{\text{piston}} - A_{\text{rod}} \]  \hfill (3.16)

\[ \Delta V_{\text{stamp}} = \Delta V_{\text{piston}} - \Delta V_{\text{rod}} \]  \hfill (3.17)

\[ = A_{\text{piston}} x - A_{\text{rod}} x \]  \hfill (3.18)

\[ = A_{\text{stamp}} x \]  \hfill (3.19)

The displaced oil volume diminishes the volume for the gas inside the accumulator causing a pressure increase.

\[ V_{\text{acc}} = V_0 - \Delta V_{\text{stamp}} \]  \hfill (3.20)

\[ p_{\text{acc}} = \frac{p_0 V_0^n}{V_{\text{acc}}^n} = \frac{p_0 V_0^n}{\left(V_0 - \Delta V_{\text{stamp}}\right)^n} \]  \hfill (3.21)
In (3.20) and (3.21) state 0 refers to the unloaded gas states of the accumulator; pressure and volume as initially in steady state can be found in the air chamber of the accumulator. The pressure increases by compression and decreases by extension but always returns to the original unloaded situation; in conclusion the accumulator acts as a spring. Figure 3.18 shows the accumulator system as developed in the simulation model.

Figure 3.18: accumulator model

The initial compression due to ride height control is subtracted from the total displaced volume. In reality the ride height control is independent from driving conditions as well. This way only the oil volume displacements caused by driving are used by the accumulator function. The initial compression is sent towards the mechanical section to establish a pre-set ride height as installed by the user. The rest of the accumulator system is needed to derive (3.21) in real time simulations.

3.3.6 Damping Manifold and tubing system

The main damper system represents the damper manifold with all its valves. Each valve is the main part of a separated valve model; these valve models will be further explained in chapter 4. To understand the model of the damper manifold a schematic overview is given in figure 3.19 for a flow from either a piston chamber or a rod chamber.

Figure 3.19: schematic overview damper manifold for one flow

In figure 3.19 $Q$ refers to the flow from one of the cylinder chambers. In the model for the hydro-pneumatic suspension two flows are considered: one flow from the piston chamber of the cylinder to the rod chamber of the cylinder and the accumulator; the other flow from the rod chamber of the cylinder to the piston chamber of
the cylinder. The total flow-through area \( a \) of the damper manifold is dependent from the pressure loss and flow which creates an algebraic loop in the model. This makes the model slower but not less functional.

The damper manifold is modelled only for its function: to model pressure loss generated by the flows from the cylinder chambers, using equation:

\[
\Delta p = \frac{Q^2}{C_d^2 a^2} \frac{\rho}{2}
\]  

\( Q \) is the flow through the total area of all the valves in the piston flow, given by \( a \). The discharge coefficient of the orifice is given by \( C_d \) and is dependent from both size as form of the orifice perimeter. Here a fixed value for \( C_d \) is chosen as explained in chapter 2.1 of the report.

The total flow-through area per flow \( a \) is generated by several valves in a parallel sequence. These valves are influences not only by the pressure loss or oil flow, but also are influenced by each other. The amount of oil flowing through the valves for example is dependent on the total flow-through area formed by all valves together and the opening area of the valve individually. The block in figure 3.19 where the total damper manifold flow-through area is derived therefore should be further worked by adding a flow division over the separate valves inside this oil flow. The flow distribution is visualized in figure 3.20.

Figure 3.20: flow division through the several parallel placed valves
Additionally there is a stagnation pressure loss added in the “Damping Manifold and tubing system”. This pressure loss is generated over the tubing system between the cylinder and the damper manifold and between the damper manifold and the accumulator. The equation combining the pressure loss generated over the damper manifold and the stagnation pressure for each flow separately is:

\[ \Delta p = \frac{\rho Q^2}{2C_d(a_i + a_s + ... + a_t)} + \Delta p_{st} \]  

(3.23)

The stagnation pressure loss \( \Delta p_{st} \) can be divided into a stagnation pressure loss before the damper manifold and in a stagnation pressure loss after the damper manifold. After the damper manifold there only is one flow from the damper manifold towards the accumulator. Due to the signal routing in the hydro-pneumatic model, the actual single flow is considered for the stagnation pressure created between damper manifold and accumulator.

The stagnation pressure loss is modelled by:

\[ \Delta p_{st} = K \frac{1}{2} \rho v^2 = K \frac{\rho Q^2}{2A^2} \]  

(3.24)

The pressure loss factor \( K \) is a summation of the independent appendage pressure loss factors.

![Figure 3.21: stagnation pressure loss model](image)

### 3.4 Multi-body model visualization

This section shows the multi-body model visualization created using the Virtual Reality toolbox in Simulink. In figure 3.22 the complete multi-body model is shown, consisting of a chassis, a cabin, a loading space, a front axle, a rear axle and four wheels. Figure 3.23 shows a close up of the front axle and rear axle in virtual reality; the parts can be seen in table 3.1. For the figures an equal lay-out is used as in the report of M. Pinxteren [7].

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>upper cylinder rod</td>
<td>4</td>
</tr>
<tr>
<td>2</td>
<td>lower cylinder rod</td>
<td>5</td>
</tr>
<tr>
<td>3</td>
<td>upper rod</td>
<td>6</td>
</tr>
</tbody>
</table>

Table 3.1: virtual reality axle parts
(a) perspective view

(b) bottom view

Figure 3.22: multi-body model in virtual reality

(a) front axle

(b) rear axle

Figure 3.23: multi-body models of the axles in virtual reality
4. Hydraulic components validation

The damping of the suspension system is achieved in the hydraulics of the hydro-pneumatic system. Valves in the remote valve block cause a pressure loss created by the flow through these valves. The valves in the block consist of five different. One of these valves is a check valve, two are ‘normally closed’ (NC) valves and the final two valves are ‘pressure relieve’ (PRV) valves. This chapter discusses the modeling of these valves. V.S.E. has put a lot of time, effort and money in the experiments needed for this validation step and has chosen to keep this data classified to authorized personnel only. Thus for confidentiality reasons the flows and pressures in this chapter are scaled to the maximum measured flow (over all appendages) and maximum measured pressure (also over all appendages) respectively.

All data is scaled to one certain factor for either the flow or the pressure loss. These scaling factors can be found in appendix E of the confidential data appendix. The opening areas of the valves also are written in rate of opening without providing the actual size of the valve. The original figures can be found in appendix E of the confidential data appendix as well. The values in the experiments and in the simulation results are scaled equally to be able to compare the results and conclude whether the validation is done properly.

4.1 Check valve

The check valve (referred to as valve type 1 in section 2.4) opens when the pressure difference over the valve exceeds 0.5 bar. Calculating the corresponding flow gives a flow of approximately 0.1 L/min in the direction from accumulator to cylinder. The simulation model needs the rewriting to a flow-dependency in order not to get stuck with high stiffness equations. When looking at figure 4.3 this assumption still is a good approximation and during simulations the 0.5 bar appears to be negligible. The valve seems to be fully opened at 3.5 % of the maximal measured flow by experimental validation. This can be modeled by the following equation where the flow is given by $Q$:

$$ r_{CV} = \frac{1}{4} \sqrt{Q} \quad (4.1) $$

![Figure 4.1: opening rate for type 1 valve](image)
In (4.1) $r_{CV}$ is the opening rate for the check valve as a function of the flow. This equation provides a problem during simulation; the valve will have an infinite fast opening rate at the start of opening. Therefore a smoothing function is applied in the model. The formula for the rate of opening for the check valve is shown in figure 4.1.

In the model of the valve, the formula for the opening rate of the valve is applied in the “area” lookup table and multiplied by the maximum opening area the valve is able to create. Afterwards the opening area is limited to this same maximum opening area. The model is shown in figure 4.2.

**Figure 4.2: type 1 valve model**

The flow through the valve creates a pressure loss given by equation 4.2:

$$\Delta p = \frac{Q^2}{C_d} \frac{\rho}{2}$$

(4.2)

By several experiments combined in one graph figure 4.3(a) has been obtained; by simulations figure 4.3(b) has been established. When setting up the experiment in equal conditions these same results can be measured and the simulation would still fit perfectly.

**Figure 4.3: type 1 valve characteristics**
4.2 Normally closed valves

Two different normally closed valves are used in the suspension system of the GINAF Dakar rally truck. The valves are already introduced in section 2.4 as valve types 2 and 3. First the valve of type 2 is discussed here, which is placed in the flow from the rod chamber of the cylinder to the piston chamber of the cylinder. The valve characteristics are dependent on the electric current and the flow through the valve block.

The opening area of the valve is given as a function of the current applied to the valve. The data given by the supplier implies there is a proportional relationship between the actual flow-through area and the electric current. The measurement data for the normally closed valves is shown in figure 4.5a and figure 4.6a. The orifice area is derived from these results. For confidentiality reasons the actual opening area is not shown here, but the opening rate of the valves is plotted instead. The opening rate is equal to the opening area of the valve scaled with the maximum opening area for that particular valve. Figure 4.4a shows the opening rate for the type 2 valve and figure 4.4b shows the opening rate for the type 3 valve.

\[
\frac{Q}{A} = \frac{Q_{\text{nom}}}{A_{\text{nom}}} \cdot \frac{A}{A_{\text{nom}}}
\]

(a) opening rate for type 2 valve 
(b) opening rate for type 3 valve

Figure 4.4: opening rate for type 2 and type 3 valves

The valve opening area is derived by applying the next equation to the lowest constant flow measurements.

\[
a = \frac{Q^2 \rho}{2C_d \Delta p}
\]

The data of the supplier suggests that when the electric current for the valve is set, the valve will remain at that given position. But experiments show that the pressure loss at higher flows is not as high as in using a pre-set area, thus the area of the valves increases with higher flow.
At 11 percent of the maximum applied flow through the type 2 valve the valve is normally opened, but at 22 percent of the maximum applied flow the type 2 valve is maximally opened. This phenomenon is encountered when the opening areas for the higher constant flow measurements are derived by applying (4.3).

Similarly to the valve of type 2, the type 3 valve is validated. The area again is derived from the measurements using equation 4.3. The results of the simulations are compared to the experiments in figure 4.6. It should be mentioned for figure 4.5 and figure 4.6 that several experiments obtaining the same results have been performed, but only one experiment is visualized for clarity.

Figure 4.5: type 2 valve characteristics

Figure 4.6: type 3 valve characteristics
The type 3 valve is larger than the type 2 valve and is able to cope with a higher flow. This valve is fully opened at 44 percent of the maximum measured flow. Again the area derived at 11 percent of the maximum measured flow is taken as the reference where the flow has no influence on the opening area. For all constant flow measurements the opening area is derived at each fixed electric current. The opening area increased with a higher flow. The difference between installed area opening (the lowest graphs in figures 4.5 and 4.6) and maximum opening area (the highest graphs in figures 4.5 and 4.6) relates to the flow. For each flow a percentage of this opening difference which is opened by the power of the flow is given in table 4.1.

<table>
<thead>
<tr>
<th>Flow [L/min]</th>
<th>50</th>
<th>100</th>
<th>150</th>
<th>200</th>
</tr>
</thead>
<tbody>
<tr>
<td>Additional rate of opening [%] for type 2 valve</td>
<td>0</td>
<td>100</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>Additional rate of opening [%] for type 3 valve</td>
<td>0</td>
<td>33</td>
<td>67</td>
<td>100</td>
</tr>
</tbody>
</table>

Table 4.1: additional rate of opening for type 2 and type 3 valves

The type 2 and type 3 valves are modeled in a similar fashion, only differing in the data they load. The electric current versus opening area is loaded in the “area” lookup table (see figure 4.7). In this lookup table the data of figure 4.4 is loaded for the specific valve. The data of table 4.1 is applied in the “flow to opening” lookup table.

![Figure 4.7: type 2 and type 3 valve model](image)

As shown in figure 4.7, the additional rate of opening is multiplied by the valve area which is not opened by the electric current yet. This additional opening area is added to the opening area dependent on the electric current. At the end of the model, the final opening area is limited to the maximum opening area of the valve.

For all valves additional tests are performed at a fixed electric current. Varying the flow at a fast rate could result in a different flow-through area than using a semi-constant flow. Due to the fast change in flow, the flow might exert a larger power to the valve piston inside the valve (see figure 4.8). The leak flow passage of the valve causes the pressure behind the valve piston to lag behind the pressure in front of the valve piston.

The direction of the flow does not influence the pressure lag inside the valve; the pressure lag is equal when the oil is flowing from port 1 to port 2 as the pressure lag when the oil is flowing from port 2 to port 1. A larger power due to a fast change in flow might open the valve further and faster than in case a semi-constant flow is applied.
The type 2 valve provides disputable results over the two tests. The disputable results are given in table 4.2 and are derived by applying (4.3) on the semi-constant flow measurement data and on the fast variance in flow measurement data to derive the opening areas and then to convert them into an opening rate.

<table>
<thead>
<tr>
<th>Electric current</th>
<th>400 mA</th>
<th>600 mA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rate of opening in slow variation of flow</td>
<td>0.92</td>
<td>1</td>
</tr>
<tr>
<td>Rate of opening in fast variation of flow</td>
<td>0.059</td>
<td>0.12</td>
</tr>
</tbody>
</table>

Table 4.2: opening rates for type 2 valve in two different flow measurements

One suggestion for this behavior is that the valve is already influenced by the power of the flow at 11 percent of the maximum measured flow and the reference orifice area should be derived at a lower flow of approximately 5 percent of the maximum flow. During the fast flow measurements with a constant electric current, the maximum pressure loss rate of figure 4.5 is reached at 5 percent of the maximum measured flow. The reference opening area should thus be smaller than implied by the measurements of figure 4.5. The smaller orifice areas corresponding to the fast variation in flow measurements would provide the most accurate valve characteristics.

4.3 Pressure relieve valves

The pressure relieve valves (valves type 4 and type 5 of section 2.4) are more complicated than the normally closed valves. Where the normally closed valves only use electric current and flow as input signals, the pressure relieve valves also are dependent on the pressure difference over the valve itself. The crack pressure (when the valve starts opening) for the valve versus the electric current is given by the manufacturer.

It should be noticed that the crack pressure also is scaled to the maximum measured pressure during the experiments. For a clear comparison between measured results and simulated results the scaling factors are all applied to both the experiments as well as to the simulations. This provides a clear validation without publishing sensitive information.

The valve is supposed to be proportional with respect to the flow, similar to the normally closed valves. This does not imply a linear relation between electric current and opening area rate. Measurements indicate a quadratic pre-stage followed by a linear stage in the midsized electric currents as electric current versus opening rate relation.
At the higher electric currents the flow-through area of the valve reaches a plateau, which is shown in figure 4.10. The opening areas of the valves are scaled to the maximum area for that particular valve. When the opening rate is 0, the valve is fully closed; when the opening rate is 1, the valve is fully opened.

The flow influences the opening areas of the pressure relieve valves similar as it does to the normally closed valves. In this valve model the additional opening area is not dependent of the remaining area of the valve which is still closed, but a fixed factor has been chosen between the flow and additional opening. This factor is shown in table 4.3.
For the opening areas corresponding to the higher electric currents the flow-through area is close to its maximum. When the factor for additional opening area is applied, the flow-through area would exceed the maximum valve opening area. Therefore the opening area for the valve is limited in the model at the end of the calculations.

<table>
<thead>
<tr>
<th>Flow [L/min]</th>
<th>50</th>
<th>100</th>
<th>200</th>
</tr>
</thead>
<tbody>
<tr>
<td>Additional rate of opening [%] for type 4 valve</td>
<td>0</td>
<td>90</td>
<td>240</td>
</tr>
<tr>
<td>Additional rate of opening [%] for type 5 valve</td>
<td>0</td>
<td>95</td>
<td>280</td>
</tr>
</tbody>
</table>

Table 4.3: additional rate of opening for type 4 and type 5 valves

Figure 4.12 shows the model developed for this valve. The “area” lookup table again loads the nominal opening area dependent on the installed electric current. This area is not directly set by the valve but dependent on the pressure as well. When the pressure difference over entrance and exit port of the valve exceeds the crack pressure, the valve starts opening following the curve given in figure 4.11. The curve is given in a rate of opening development in the model, obtained by (4.4) and applied to the model by the “opening area development” lookup table.

\[
\rho_{\text{PRV}} = \frac{1}{3} \sqrt{\Delta p - p_{\text{crack}}} \tag{4.4}
\]

![Figure 4.11: opening rate development for type 4 and type 5 valves](image)

In the model the opening rate development curve is first multiplied by the area dependent on the electric current. This provides the actual opening area of the valve. The opening area is limited to the nominal opening area; this provides the same results as limiting the opening rate development to 1.
The overshoot behavior in the lower section of figure 4.12 will be discussed later in this section. First the results obtained with this model are discussed. The results are shown in figure 4.13b for the type 4 valve and 4.14b for the type 5 valve. The results are scaled to the maximum pressure measured during the experiments. Similar to the normally closed valves several experiments have been performed per flow and per valve. Eventually only the best compromise of the measurements is provided in this report for comparison. Scaling of the simulation results is equal to the scaling of the experimental results for a fast comparison.

Figure 4.13: type 4 valve characteristics
When testing the valves with a fast variance in flow an overshoot behavior during the opening of the valve occurs. These valves have a very similar lay-out as the valve shown in figure 4.6. The overshoot behavior can be explained by the geometry of the valve. The flow enters the valve at port 1 and due to the high flow rate a force is created which is higher than the force given by the electric current. For this reason the valve initially opens further in a dynamic flow than in a constant flow and overshoot behavior occurs.

Behind the valve piston (where the electric spool is located) the pressure lags behind and catches up due to the leak flow passage of the valve. When the pressure is equalized over the front and rear area of the valve the overshoot behavior is gone and the valve returns to its installed position. This overshoot behavior is modeled by super-positioning a transfer function on top of the actual pressure loss created by the valve. The overshoot behavior is modeled in the lower section of figure 4.12. The results of the overshoot behavior measurements for the experiments and the simulations are shown in figure 4.15.
4.4 Other appendages

Besides the valves in the valve block other appendages have been validated as well. For elbows, hoses and fittings a stagnation pressure loss factor $K$ as defined in (2.23) is derived from the measurements and inserted in the model. The indices in the legend of the graphs refer to the size which is used during the measurements; the index type A appendages have smaller flow-through areas than the type B appendages, placed in the vehicle.

Figure 4.16: elbow piece characteristics

Figure 4.17: fitting piece characteristics
In figures 4.16 to 4.18 the pressure loss and the flow are scaled to the maximum values measured during the experiments. The pressure loss over a hose piece is still comparable to the pressure loss over an elbow piece, however true data is not shown here. In the appendix E.4 of the confidential data appendix the actual characteristics can be found as well as the pressure loss factors.

![Graphs of pressure loss over hoses](image)

(a) experiment

(b) simulation

**Figure 4.18: hose piece (length is 1 meter) characteristics**

It should be noticed that for the hose measurements a piece of 1 meter length is used. The maximum pressure loss over this short length of hose is approximately 30 to 50 times lower than the pressure loss over 2 connectors. The connectors which are inserted in the hose have smaller inner diameter which is responsible for the high pressure loss compared to elbow and fitting appendages. For this reason it is more important to look at the number of hoses and connectors that are used than to take the length of the hose into account. In the vehicle most of the connections are established by hoses, therefore pipes are not included in the measurements.
5. Isolated suspension-box model validation

In chapter 3 the simulation model of the suspension system is discussed. An important part of the full vehicle model is the hydro-pneumatic suspension system. This system can be tested separately as well and an isolated system model is developed for this validation process. This model uses the displacements and velocities of the cylinders as input signal. The displaced oil volume can be derived by multiplying the displacement by the corresponding cylinder areas. In the same manner the flow can be derived by using the velocity of the cylinders.

As a road condition a ramp is placed on a flat road area for retrieving experimental results. These tests are performed on the RDW test track in Lelystad (NL). The ramp is 6 meters long and progressively increases to a height of 25 cm. The velocity of the vehicle during measurements was 77 km/h and during simulations this same velocity is used.

For confidentiality reasons the data is concealed as previously done in chapter 4. For both the experiments as well as the simulation results, the pressures, forces and flows are scaled with equal factors and these scaling factors are given in appendix F of the confidential data appendix. The displacements are scaled to a displacement rate with a total range of 1 between the bump-stops for compression and extension. The range is divided for 40 percent of the stroke in extension and 60 percent in compression, referring to the neutral position. The total stroke thus ranges from minus 0.4 for extension to plus 0.6 for compression between the bump-stops.

5.1 Spring behavior

The cylinder displacements which are measured during the experiments are used for the simulation model as well for validating the system. These measured displacements are scaled and shown in figure 5.2.

![Figure 5.1: simulated road area](image)

Figure 5.1: simulated road area

For confidentiality reasons the data is concealed as previously done in chapter 4. For both the experiments as well as the simulation results, the pressures, forces and flows are scaled with equal factors and these scaling factors are given in appendix F of the confidential data appendix. The displacements are scaled to a displacement rate with a total range of 1 between the bump-stops for compression and extension. The range is divided for 40 percent of the stroke in extension and 60 percent in compression, referring to the neutral position. The total stroke thus ranges from minus 0.4 for extension to plus 0.6 for compression between the bump-stops.

![Figure 5.2: measured cylinder displacements](image)

Figure 5.2: measured cylinder displacements
Most interesting is the rear axle displacement. First the vehicle hits the road bump and the cylinder is partly compressed, then the vehicle is in flight until the rear axle hits the ground. During this ‘flight’ the cylinders are extended to a negative displacement rate of 0.2 at the rear axle. Back on the ground the movement of the rear axle immediately disappears while the front axle needs one more extension and compression movement. When an incompressible fluid is assumed, the displacement is directly responsible for the accumulator pressure:

\[
p_{\text{acc}} = \frac{p_0 V_0^n}{(V_0 - \Delta V_{\text{stamp}})^n}
\]

(3.21)

Figure 5.3: accumulator pressures

Figure 5.4: accumulator pressure characteristics
A plot of the accumulator pressure versus the cylinder displacement should result is a similar graph as spring force versus spring displacement. The accumulator is an exponential spring and additionally influenced by the gas polytropic coefficient.

For fast cylinder movements the polytropic coefficient will tend to 1.4; for slower movements with more heat exchange during the compression the coefficient will tend to 1. During experiments there is not a single value for the polytropic coefficient causing a chaos for these plots as shown in figure 5.4.

The tangle of lines at the rear axle is even worse than at the front axle. This is due to the crosslink of the passive hydraulic roll stabilizer. To obtain the characteristics on a symmetric road both the displacements should be averaged as well as the accumulator pressures. This provides a much more distinct graph of the characteristics of the system (see figure 5.5a). In all characteristics plots the simulation model uses a fixed polytropic coefficient of 1.2. Figure 5.5b shows the same averaged rear axle characteristics including different values for the polytropic coefficient. Here it seems that the value of $n=1.2$ is especially in the extension region of displacement the best solution and in the compression region of displacement a very nice approximation.

![Averaged accumulator pressure characteristics rear axle](image)

(a) averaged rear axle characteristics $n=1.2$  
(b) averaged rear axle characteristics varying $n$

Figure 5.5: averaged accumulator pressure characteristics rear axle

To provide an idea about the spring stiffness a tangent can be drawn at the static position of the vehicle. The spring stiffness of the front axle and the spring stiffness of the rear axle can both be found in appendix F.

<table>
<thead>
<tr>
<th>steady state initial displacement rate [-]</th>
<th>front</th>
<th>rear</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.2</td>
<td>0.35</td>
</tr>
</tbody>
</table>

Table 5.1: steady state displacement rate and spring stiffness

The spring stiffness of the front axle and rear axle will be used later in section 5.2.
5.2 Damping behavior

The input velocities from the cylinders cause a hydraulic fluid flow through the remote valve block of the suspension system. In the valve block and through the several appendages the hydraulic fluid creates a pressure loss. The pressure loss created by the flow from the piston chamber of the cylinder is referred to as piston pressure loss. The pressure loss created by the oil flowing from the rod chamber is referred to as rod pressure loss.

![Piston pressure loss front axle](image1)

![Piston pressure loss rear axle](image2)

(a) front axle  (b) rear axle

**Figure 5.6: pressure loss caused by flow from piston chamber**

In the flow from the rod chamber to the piston chamber and accumulator the disputable valves of the previous chapter are used for creating pressure loss. The flow-through area derived from the data of the fast variation in flow valve-measurements is used as suggested to be the most accurate valve data as well.

![Rod pressure loss front axle](image3)

![Rod pressure loss rear axle](image4)

(a) front axle  (b) rear axle

**Figure 5.7: pressure loss caused by flow from rod chamber**
Figure 5.7 shows that using the type 2 valve opening areas derived from the fast variation in flow experiments approximate the isolated suspension system measurements very well including the short peaks in pressure loss.

The overshoot behavior modeled in the PRV valves is responsible for the spiky behavior of the front axle around a simulation time of 27 seconds. When investigating the damping characteristics, the overshoot behavior should be taken out of the equation. The overshoot behavior at fast variations in pressure makes it possible that at a certain amount of flow different pressure losses can be created by the valves. A clear overview of the characteristics can only be provided when there is a single solution for pressure loss corresponding to flow. The overshoot behavior causes more than one solution for pressure loss at a specific amount of flow. Only for this analysis the overshoot behavior is therefore turned off in the model to create a clear perspective of the properties of the system. In all other simulation conditions this overshoot behavior should be turned on.

Figure 5.8: pressure loss versus flow characteristics

The rod pressure loss versus flow characteristics equals the expectations raised in chapter 4. In the first stage only the normally closed valves are responsible for the pressure loss; the second stage is influenced by the opening behavior of the pressure relieve valves.

Finally both valves are open and both valves contribute to the pressure loss generated by the oil flow. At the front axle the additional opening area of the NC valves caused by the high flows and the opening behavior of the PRV valve influence each other, because they use the same domain of flow. The intermittence between these valves is responsible for a smooth transition between the various stages of the pressure loss characteristics. The rear axle shows a more clear contrast between the various stages in opening behavior of the valves.

The graphs of figure 5.8 can be converted to ordinary damping force versus velocity figures to compare the characteristics of the GINAF Dakar truck with other vehicles. Here it should be mentioned that a positive flow was regarded when the flow was coming from the cylinder and flowing towards the accumulator. For the flow from the piston chamber this would imply that extension would be suggested to be positive. Therefore the sign of the graph changes when rewriting the flow of the oil towards a cylinder velocity. The damping force versus cylinder velocity graphs is shown in figure 5.9.
Figure 5.9: damper force versus cylinder velocity characteristics

Figures 5.9a and 5.9b can be combined in a total damping force characteristics graph for better comparison with other vehicles. This has been done in figure 5.10. In figures 5.9 and 5.10 the damping force is scaled as well as the cylinder velocity. The original figures and the scaling factors can be found in appendix F of the confidential data appendix.

Figure 5.10: total damping characteristics
To derive the damping factor for this system the next equation should be applied with the index \( i \) either front or rear:

\[
\zeta_i = \frac{d_i}{2\sqrt{k_i \frac{m_{s,i}}{2}}}
\]  

(5.1)

In (5.1) the spring stiffness is represented by \( k \) and the damping coefficient is represented by \( d \). The problem here is that in linear mass-spring-damper systems these parameters are constant. In this system however both stiffness and damping depend on cylinder displacement and cylinder velocity. In the previous section the spring stiffness of front axle and rear axle have been derived. These values will be used here as well. The sprung mass per axle \( m_{s,i} \) is used and should be divided by 2 to represent the mass per cylinder.

The variation in damping coefficient is large and therefore the scaled damping factors for the vehicle are plotted against the scaled velocity in figure 5.11a.

![Damping Coefficient and Factor Plots](image)

(a) damping coefficient versus cylinder velocity  
(b) damping factor versus cylinder velocity

**Figure 5.11: damping characteristics versus cylinder velocity**

Figure 5.11b shows the damping factor derived by applying equation 5.1 damping coefficient. Figure 5.11b is a non-scaled figure for the damping ratio, but does not provide any sensitive information when the other parameters are still unknown. The damping ratios clearly show that the vehicle without additional friction would be heavily under-critically damped.

When the damping ratio or damping factor \( \zeta \) is equal to 1, the system is critically damped. Normal vehicles have a damping ratio between 0.2 and 0.4. The GINAF Dakar truck tends more towards 0.05.
Some final thoughts regarding the isolated suspension system:

1. The valves used in the isolated hydro-pneumatic suspension model are all stand-alone valves and can directly be replaced by other valve models. The characteristics of the valve models used in this suspension-box model are discussed in chapter 4.

2. For the type 2 valve the smaller derived areas from the fast variance in flow valve-measurements have been applied in the model, since it gives the most accurate results.

3. The pressure relieve valves are subjected to an overshoot behavior. When the pressure increases at a very high rate, the pressure relieve valve will open fast and pass the area set by the electric current.

4. The flow per valve is derived directly from the flow-through area of the specific valve compared to the total flow-through area of the flow in the valve block. This way each valve in the model receives the flow which actually passes the valve in the vehicle as well.

5. A last thing worth mentioning is the sensitivity of the model with respect to the algebraic loop. The problem occurs when the flow-through area of a valve changes infinitely fast as in a sudden opening due to overshoot behavior. For this reason the pressure loss created by the transfer function in the pressure relieve valves is directly added to the total pressure loss created by the total flow from the cylinder chambers. The total pressure loss is not only generated over the valves but over the various appendages as well.

The isolated suspension system model approximates the measurements fairly well. However one should keep in mind that the valve opening areas used in the model are derived from measurements as well. This implies that for new valves similar measurements are necessary for an accurate valve simulation model.
6. Parameter study at the full vehicle model

In this chapter the full vehicle model is discussed focusing more on the overall vehicle motions while driving over the road bump defined in chapter 5 and visualized in figure 5.1. The vehicle velocity in this chapter is kept equal to the velocity in the previous chapter: 77 km/h. Now the complete vehicle model is considered, additional influences are considered here as static friction, position dependent damping, height of centre of gravity and a horizontally translational mass simulating the fuel inside the fuel tank. In this chapter the displacements of the cylinders, the pressures inside the system and the forces generated by the system all are scaled to preserve the confidentiality of the report. The scaling factors used for both the experiments as well as the simulation results and the original figures for this chapter can be found in appendix G of the confidential data appendix.

6.1 Baseline simulation results

First the suspension displacement of the baseline vehicle containing only the hydro-pneumatic system as described in chapter 5 is simulated. This vehicle is modeled as discussed in chapter 3 but without the position dependent damping, without the friction, with the fixed location of C.O.G. for the fuel, with a tyre stiffness of 600,000 N/m for the front axle and rear axle and having a normal frame inertia including the cabin and cargo department.

The simulation with this baseline model provides a clear perspective in what sense the hydro-pneumatic suspension contributes to the complete vehicle and what is caused by other factors. The simulation model thus has no position dependent damping, no static friction, the original height of centre of gravity and no horizontally translational mass. The results are shown in figure 6.1 and compared to the measurements. The first compression stroke (positive displacement) is caused by driving up the ramp. The fast extension is due to the flight of the vehicle when no load is acting on the wheels.

![Cylinder displacement front axle](image1.png)
(a) front axle

![Cylinder displacement rear axle](image2.png)
(b) rear axle

Figure 6.1: measured cylinder displacement and simulated cylinder displacement for baseline vehicle model

Figure 6.1 shows that the measurements and the simulations do not correspond. The simulations show a far more under-critically damped system than the vehicle during the experiments. In addition the rear axle displacement of the measurements show a behavior which is not even closely met by the simulations.
6.2 Position dependent damping

As starting adjustment a position dependent damping (snubbing) is added to the vehicle. When the cylinder is compressed or extended towards its limits additional damping force is created. The position of the cylinder acts as a window function \( W \) in this equation.

\[
F_{\text{snubber}} = W \cdot \dot{x} \cdot d_{\text{snubber}}
\]  

(6.1)

Here \( \dot{x} \) is the cylinder velocity, \( d_{\text{snubber}} \) the damping coefficient and the function \( W \) is defined as in figure 6.2.

![Window function for snubber](image1.png)

Figure 6.2: window function for position dependent damping

![Simulated cylinder displacement](image2.png)

Figure 6.3: simulated cylinder displacement for baseline model with position dependent damping
The model for the position dependent damping is discussed in section 3.3.2. This position dependent damping reduces the bouncing behavior of the unsprung mass at the bump stops; as a result the spikes at the maxima of the front axle and rear axle displacement are reduced when hitting the bump-stops (see figure 6.3).

6.3 Increase in height of Centre of Gravity

The second adjustment considered in this chapter is the location of the centre of gravity (C.O.G.). The horizontal location is known by measurements of the various wheel loads; the vertical position is not very well known. As for a start the height of the centre of gravity is estimated to be 1.25 m. To investigate the influence of this parameter, the centre of gravity is relocated to a 150 percent of this originally estimated value.

The influence of this parameter on the given road area does not really result in a different response of the vehicle as expected in advance. The pattern and number of cylinder movements remains unchanged; the only difference can be found in the increase of amplitude of the simulated displacements by less than 5 % (see figure 6.4). The graph for the higher C.O.G. also includes the model for position dependent damping.

Thus the influence of this parameter is negligible to the pitch behavior of the vehicle. The influence of the height of the C.O.G. to the roll behavior of the vehicle is not further investigated here. Additionally a vertical location of the centre of gravity higher than 1.60 m seems unlikely. Therefore this parameter is reset to its original value of approximately 1.25 m.

![Figure 6.4: simulated cylinder displacement for vehicle model with increase of height C.O.G.](image)

6.4 Static friction

Static friction might be the most influencing and important adjustment to the total vehicle behavior. The static friction is added in the model and discussed in section 3.3.2. Especially the front axle is under-critically damped by the hydro-pneumatic system alone. The measurements in figure 6.1 show that the front axle needs 3 compression strokes and the rear axle directly damps out while entering the second compression stroke; the compression stroke when the wheels touch the ground.
The simulated front axle behavior approximates the experiments accurately; the rear axle still shows a complete second compression stroke. The magnitude of the static friction in the GINAF Dakar Rally Truck is unknown and is tuned during several simulations to approach the measured displacements as close as possible. The friction force for the front axle is tuned to a scaled value of 0.0225 and the rear axle friction force is tuned to a scaled value of 0.01. The scaling factors can be found in appendix G of the confidential data appendix to derive the correct values for friction in Newtons.

The static friction in the front axle seems to be about twice as high as in the rear axle. This might be caused by the dimensions and properties of the two different systems used for front and rear axle; such as different accumulator pressure and volume, different cylinder dimensions and different sprung masses. The front axle cylinder and rear axle cylinder have the same outside dimensions; they do not share identical dimensions internally. Friction can also be found at the rotation points of the suspension. This friction is also considered here under the same topic as cylinder friction. The results in figure 6.5 compare the baseline vehicle model with position dependent damping and friction with the experimental results from the measurements at the RDW test track in Lelystad.

The simulated results for the full vehicle model with these two adjustments for the front axle are satisfactory and above all expectations; the simulated results for the rear axle are not sufficiently accurate compared to the measured data when the same road area is applied.

### 6.5 Increase in vehicle inertia

The inertia of the vehicle model is modified in order to find the cause of the single measured displacement of the rear axle. The inertia of the frame originally is estimated and applied to the model. In this investigation process, the full vehicle mass, height, width and length are used.
The inertia is changed by adjusting the mass, the width, the height and the length of the frame.

\[
J = \begin{bmatrix}
\frac{m}{12} (b^2 + h^2) & 0 & 0 \\
0 & \frac{m}{12} (L^2 + h^2) & 0 \\
0 & 0 & \frac{m}{12} (L^2 + b^2)
\end{bmatrix}, \quad m = 11200, \quad b = 2.2, \quad h = 4.2, \quad L = 6.7
\] (6.2)

The figure shows an improvement in the simulated rear axle compression movement after the wheels touch the ground. Still the results of this model with position dependent damping, static friction and increased inertia are not an exact copy of the measurements, but still a better approximation has been reached. The number of movements however is increased as well as the amplitude of the cylinder movements. As a solution for these extra movements at the front axle, additional damping by friction can be considered.

Comparing figures 6.6 and 6.7 show that indeed additional friction is needed; the rear axle needs far less cylinder movements as without additional friction. As a result the front axle and rear axle now form a compromise between compression curve for the rear axle and amplitude of displacement at the front axle. The static friction force is increased to a scaled value of 0.045 for the front axle and to a scaled value of 0.02 for the rear axle.

The results in figure 6.7 seem to be the best compromise available but it also shows that some further investigation in this topic may be needed for more accuracy.

Figure 6.6: measured cylinder displacement and simulated cylinder displacement for vehicle model with increased inertia

(a) front axle

(b) rear axle

The inertia is changed by adjusting the mass, the width, the height and the length of the frame.
6.6 Horizontally translational mass

As a possible different cause for the vehicle behavior at the rear axle the moving fuel on board is considered as well. The fuel is not stacked between the front and rear axle as in normal street trucks; two tanks on board are located near the rear axle. One tank is located in front of the rear axle at the left side of the chassis, the other behind the rear axle between the chassis beams. The total volume for these tanks is about 1000 L corresponding to a summed mass of 850 kg.

![Figure 6.8: location of modeled fuel tank](image)

The fuel mass is connected to the chassis by a spring without damping. When there is no movement of the fuel considered such as in the baseline vehicle model, the spring joint modeled in figure 3.6 is replaced by a weld joint to guarantee that the location of the C.O.G. is fixed with respect to the frame. In this case the horizontal movement of the fuel is the focus of the investigation and a spring joint is applied with a spring stiffness of 1000 N/m. The simulation results for the location of the C.O.G. of the fuel can be seen in figure 6.9.
The fuel is modeled as a single mass located a little behind the rear axle (see figure 6.8). This would approximate the actual displacement of the fuel inside the tank. The location of the centre of gravity of the fuel might move up horizontally to 20 cm to the front or 20 cm to the rear starting from its neutral position. The model is described in section 3.2.1 where the chassis system of the multi-body model is explained.

In figure 6.9 the spring with a stiffness of 1000 N/m is used as spring stiffness and no damping is applied. This causes a fuel centre of gravity displacement of at least 20 cm. To isolate this influence, the inertia is set back to the normal frame inertia and the vehicle model parameters are returned to the parameters corresponding to section 6.4 where only friction and position dependent damping are applied.

Figure 6.9: simulated fuel displacement with soft spring stiffness

Figure 6.10: simulated cylinder displacement for vehicle model with horizontal moving fuel and cylinder displacement for vehicle with friction and position dependent damping
The graphs of the vehicle model with moving fuel are similar to the graphs of the vehicle model without this feature and no changes can be observed by implementing horizontally translational mass to the model. A possible explanation for the lack of influence might be the location of the fuel. All fuel is stacked in tanks close to the rear axle. There is no real weight transfer of the fuel mass from the rear axle towards the front axle; the weight transfer remains concentrated at the rear axle of the vehicle. Due to the lack of influence on vehicle spring and damping behavior, the translational mass is restricted in displacement by a weld to the chassis.

When the fuel is a fixed part of the frame, cabin and cargo department the vehicle mass still equals 11200 kg and the weight distribution of 60/40 over the axles will be kept intact. However, when changing the amount of fuel on board, the weight distribution over the frame should be recalculated as well as the total mass of the vehicle. By modeling the fuel as a separate body, the influence of driving with a full or empty tank to the dynamic behavior of the vehicle can more easily be investigated. Now only the fuel mass has to be adjusted in the parameter files without having to calculate the new vehicle mass etc.

So the low influence of fuel displacement on the dynamic behavior would not imply that driving a full or empty tank might not influence the behavior of the vehicle; a complete full fuel tank creates a different weight distribution over the axles of the truck than a truck driving a fuel tank nearly empty.

### 6.7 Tyre stiffness

Finally the vertical tyre stiffness is considered. Lowering the tyre stiffness directly caused additional cylinder movements and larger amplitudes. The simulations of the dynamic behavior of the vehicle deteriorated compared to the experimental results of the tests at the RDW test track. Therefore the vertical tyre stiffness is increased from its original 600,000 N/m as used during the whole project to 1,000,000 N/m. The value of 600,000 N/m was used in the DAF Dakar project of M. Pinxteren [7] and was assumed to be a similar value for the GINAF Dakar truck.

However GINAF inflates their tyres to a higher pressure for flat road areas such as during the measurements in Lelystad. The tyre on the GINAF truck itself also is thicker and stiffer than the tyre on the DAF truck. Therefore the tyre stiffness is increased to correspond to the tyre stiffness of normal long-distance traveling trucks. For more certainty and a correct validation, it may be useful to measure the tyre stiffness of the vehicle for better understanding of the real vehicle behavior.

The rear axle behavior improved without worth-mentioning changes to the front axle. The results are shown in figure 6.11 and are compared to the cylinder displacements measured at the experiments in Lelystad. A tyre stiffness increase even reduces the second compression movement at the rear axle and thus approximates reality even further. Without additional validation of the tyre stiffness, it can not be concluded that the results shown here are absolutely correct. There is a kind of uncertainty about the stiffness of the rally truck tyres reaching the same stiffness as applied on long-distance hauling commercial trucks.

In the next section this higher vertical tyre stiffness will be applied to investigate the dynamic behavior of the vehicle. Additionally the position dependent damping, adjusted vehicle inertia and static friction are still applied to the model for their improvements on the vertical driving dynamics.
Finally the simulation model as defined in section 6.7 is used for other types of roads as well. Most important to investigate while driving over different road conditions is the dynamic behavior of the vehicle in vertical direction. As an additional test the vehicle model is supposed to drive over a road consisting of blocks or small bumps driving at 77 km/h. The vehicle velocity is kept equal to the velocity used at the road bump to be able to look closer to the order of magnitude of the pressures inside the system.

Figure 6.12: measured cylinder displacement and simulated cylinder displacement for vehicle model with increased tyre stiffness, increased inertia, tuned friction and position dependent damping

6.8 Full vehicle model driving over blocked road profile

Finally the simulation model as defined in section 6.7 is used for other types of roads as well. Most important to investigate while driving over different road conditions is the dynamic behavior of the vehicle in vertical direction. As an additional test the vehicle model is supposed to drive over a road consisting of blocks or small bumps driving at 77 km/h. The vehicle velocity is kept equal to the velocity used at the road bump to be able to look closer to the order of magnitude of the pressures inside the system.

Figure 6.12: simulated blocked road profile
The impact on the suspension is highest when the bump is rectangular shaped. The road profile is a-symmetric; the bumps for the left wheels and right wheels follow-up after each other for 200 meters of road profile. For licensing reasons, the tyre is not able to envelop over the road block, but is modelled as spring at a single road contact point.

The road profile is defined in figure 6.12. The bumps are 1 meter long and 10 cm high. For each wheel the bumps are separated by 10 meters.

The cylinder displacements are smaller than the wheel displacements by the installation ratio. The cylinder displacement is measured as output data of model and scaled to 60 percent of the stroke needed for compression and 40 percent needed for extension before the bump-stops are hit. The results are shown in figure 6.13.

Figure 6.13 shows that the front axle cylinder displacement is exactly as expected with one small additional spring movement after the direct hit of the block on the road. The rear axle is about half a period behind the front axle and has a smaller displacement during the direct hit. If the vehicle would have started with a higher ride height, the displacements of the rear axle would not exceed the limit where the bump stops support the hydro-pneumatic system at a displacement rate of 0.6 for compression. The opposite is possible as well; the vehicle ride height could also be lowered towards a displacement rate of -0.4. In this situation the bump stops for extension will be hit by the suspension system.

![cylinder displacement front axle](image1)

(a) front axle

![cylinder displacement rear axle](image2)

(b) rear axle

Figure 6.13: simulated cylinder displacement over blocked road profile

The displacement of the cylinder is directly related to the accumulator pressure as well as the flow is directly related to the pressure losses generated in the damping manifold. The pressures in the accumulators are in the same order of magnitude as in the road bump test. When a symmetric road would be considered, the pressures at the rear axle accumulators would be lower than the pressures at the front axle accumulators. However the pressure of the rear axle accumulators exceeds the pressures at the front axle due to the hydraulic roll stabilizer and the a-symmetric road area.
The higher accumulator pressures at the rear axle are caused by the displaced oil volume of the piston chamber during compression that is not able to flow to the rod chamber of the other cylinder since that cylinder is not influenced by the road at that moment. All oil pressed out of the piston chamber is directly led towards the accumulator increasing the pressure. The fast pressure increase relates to a higher reaction spring force in the cylinder and this explains the smaller cylinder movements of the rear axle given in figure 6.13.

The pressure loss created by the rod flow is larger than the pressure loss created by the piston flow. If only the magnitude of the flow would be considered, the expectation would be otherwise. But the smaller opening areas of the valves inside the damper manifold for the rod flow create a pressure loss exceeding the pressure loss created by the piston flow.
Additionally the flow from the rod chamber is led through only one normally closed valve, while the flow from the piston chamber is led through two larger normally closed valves. Finally with these high amounts of pressure loss not only the normally closed valves are opened but also the pressure relieve valves will be engaged by the internal system pressures for all flows.

Figure 6.16: simulated pressure loss due to rod flow over blocked road profile
7. Conclusion and Recommendations

To design a simulation model for the competition truck of the GINAF Rally Power Team several hurdles had to be taken. First of all, in contradiction to ordinary damper struts, all oil flows are led outside the cylinder chambers via a hydraulic tubing system. Secondly no coil springs are used in the vehicle but the spring behavior is realized by remote accumulators. Maybe the most interesting aspect is the damping system consisting of various valves and the passive hydraulic roll compensator.

All of these hurdles have been investigated during this project and as a final result a validated simulation model now is available representing the vehicle’s vertical dynamic behavior. The model can be used in further projects concerning parameter optimization, active damping control or application to other types of vehicles such as commercial vans.

7.1 Conclusions

The model is validated by measurements at componential level in laboratories and by measurements of the full vehicle. By the validation process some conclusions can be drawn:

1. The cylinder with remote damping manifold and remote accumulator can be modeled as an ordinary damper strut which includes these features inside the main housing. Additional pressure losses over the bypass system have to be taken into account, especially for the rear axle where a lot of tubing is used to create a hydraulic roll compensator.

2. The accumulator as spring mechanism is only indirectly related to oil displacement. In the same manner the damping force is only indirectly dependent on the oil flow. Therefore the force derivations can be modeled separately and later be combined in a total cylinder force.

3. During the validation of the valves it appeared that the supplier data is not applicable in the model and independent testing is needed. At the end the valves are measured by V.S.E. and these data are used inside the valve models.

4. Most accurate in the suspension of the vehicle are the geometry and the hydro-pneumatic system. With promising results the hydro-pneumatic system is validated separately from the full vehicle.

5. A parameter study at the level of the full vehicle simulation model has come up with surprising results. The most important changes to the baseline vehicle enhance position dependent damping at the end points of the cylinders, static friction, increase of inertia to match the full vehicle inertia and vertical tyre stiffness. Together these adjustments create a similar displacement curve as been found during measurements. The height of the centre of gravity and the influence of the fuel moving in the tank are both negligible on the suspension deflection.

6. Step-by-step validation seems to be the key word for the success of this thesis. First all parameters of the vehicle are checked and adjusted. Secondly all components of the hydro-pneumatic system are validated. Thirdly the hydro-pneumatic system is isolated from the full vehicle and validated as an independent system. As a final step the full vehicle model is considered.

7. The rear axle spring and damping behavior needs further investigation. A compromise is taken at this point between static friction and vehicle inertia. Maybe a follow-up project might bring some additional clarity on this specific topic.
7.2 Recommendations

Some recommendations with respect to the development of the simulation model of the GINAF Dakar Rally Truck are given below:

1. Measurements of the valves used in the vehicle are necessary; the data given by the supplier seems to be unreliable and cannot be applied. This difference might be caused by the context the valve is designed for. Normally these valves are used in a static flow. The valves inside the damping manifold of the truck have to keep up with flows varying not only in direction but also in magnitude. Before modeling a valve for the damping manifold, the valve should be tested separately to investigate its characteristics.

2. The rear axle displacement during the measurements at the road bump test is damped out in only one extension and compression movement of the cylinder. This makes the rear axle look supercritically damped. This level of damping is not caused by the hydro-pneumatic system of the suspension as shown in the validation process of the isolated suspension system. The displacement of the rear axle seems to be strongly influenced by the inertia of the vehicle. This phenomenon is not further clarified yet and some additional investigation might be justified.

3. The spring and damping behavior of the vehicle are mainly considered during this project causing a lack of attention to the overall vehicle dynamics. It would be wise to investigate the roll-behavior of the vehicle as well to investigate the principles of the passive hydraulic roll compensator at the rear axle. The fishhook test implies that even the bump stops are used to prevent the vehicle from rolling over.

4. As additional recommendation it should be noticed that this model is ready for further use. It is very well possible to develop a control loop for the electric currents through the valves and so to create active damping. The intention of the model is to work on improving the vehicle dynamics using virtual testing. This saves time and money for not having to rebuild the system for each adjustment. The model could also be applied to other vehicles as well when adjusting the geometric parameters, the mass parameters and the scaling parameters of the system.

5. The hydro-pneumatic isolated model can directly be applied in vehicles having different suspension geometries. The cylinder displacements and velocities are then used to derive spring force and damping force of the cylinder. The mechanical section of the model can be adjusted by means of the SimMechanics Toolbox for MATLAB Simulink. Different suspension geometries as rigid axles, Mc Pherson suspension and double-wishbone suspension can all be modeled in SimMechanics and can all be combined with the hydro-pneumatic system. For the GINAF Dakar truck these different geometries might not be as interesting due to the high forces involved but for other types of vehicles it might be interesting to investigate which system provides the best results in combination with the hydro-pneumatic system.

6. The total accuracy of the model should be investigated further. Now only the vertical dynamic behavior of the vehicle is considered. Up to the isolated hydro-pneumatic suspension system, the vehicle is validated to be a very accurate approximation of reality for these dynamics. Further research in handling, a-symmetric driving, cornering etc has to be done.
References


Appendices
Appendix A: geometrical node-placement suspension

The geometry of the suspension is given in figure A.1. The joints inside the suspension system are given by coordinates c01 until c09 for the left side of the vehicle and by coordinates c11 until c19 for the right side of the vehicle. The right side of the vehicle is equal to the left side of the vehicle but mirrored over the longitudinal axis of the vehicle. The locations of these coordinates are given in the confidential data appendix A.

Figure A.1: node placement front axle

The rear axle coordinates are equal to the front axle coordinates but mirrored in the lateral axis of the vehicle. Where the c08 joint is pointing forward at the front axle, the same joint is pointing to the rear at the rear axle. Further all joints are translated horizontally over the longitudinal axis of the vehicle by the wheelbase.