Suspension System Dimensioning and Modelling for Co-Simulation with Simulink and Adams

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Master Thesis in Vehicle Engineering

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Dedication

This thesis is dedicated to my beloved family, particularly my parents. I can not find words to express my gratitude to you. Thank you for your love, support and belief in me. You have always been there cheering me up and standing by my side through the good times and bad. Without you, I would not be the person I am today.

"He who practices patience will never be deprived of success, even though it may take long time to reach him.”

Ali ibn Abi Talib (as)
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My master thesis would not have been possible without your help.

To my dear friends, you know who you are, thank you for your support and care. I greatly value your friendship and I deeply appreciate your belief in me.

October 2014, Örnsköldsvik

Hassan Muhieddine
Sammanfattning


Abstract

This master thesis addresses the dimensioning of a hydraulic suspension system for a hybrid forest truck (HFT) under development. Furthermore, the suspension system has been modelled in MATLAB/Simulink and an environment has been set-up for a co-simulation with the vehicle model in MSC Adams. The interface between the programs has been determined and the required preprocess is described, step by step. Numerical problems and challenges that have arisen have been discussed, together with recommendations for how they could be solved.

The purpose of the co-simulations have been to analyse the vehicle vibration levels and tune component parameters, such that, the international standard for whole-body vibrations, ISO 2631-1 (1997) and AFS 2005:15 (2005) can be met. The results are presented and clearly shows that the vibration levels in the vehicle cab are acceptable according to the standards. Moreover, recommendations are given for future work regarding how co-simulation problems could be solved, but also how to develop this passive suspension system to be of active type.
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Glossary

Adams  MSC ADAMS (multibody dynamics software).
AFS  Arbetsmiljöverkets Författningsamling.
BAE Systems  BAE Systems Hägglunds.
CAD  Computer Aided Design.
CES  Continuously controlled Electronic Suspension.
HFT  Hybrid Forest Truck.
ISO  International Organization for Standardization.
OEM  Original Equipment Manufacturer.
Simulink  MATLAB/Simulink.
Nomenclature

$\rho$  Hydraulic fluid density [kg/m$^3$]
$\tau$  Sampling time [s]
$a$    Acceleration [m/s$^2$]
$A_p$  Piston area [m$^2$]
$A_{O.f}$ Fixed orifice area [m$^2$]
$C_q$  Flow coefficient [-]
$D_p$  Piston diameter [m$^2$]
$F_D$  Damping force [N]
$F_S$  Spring force [N]
$F_{SD}$ Spring and damping force [N]
$l_{cyl}$ Cylinder length [mm]
$n$    Polytropic exponent [-]
$P_0$  Accumulator precharge pressure [bar]
$P_1$  Accumulator minimum working pressure [bar]
$P_2$  Accumulator maximum working pressure [bar]
$P_{acc}$ Accumulator pressure [bar]
$q$    Flow rate [liter/min]
$s_p$  Piston stroke [m]
$t$    Time [s]
$V_1$  Nitrogen volume at $P_1$ [liter]
$V_2$  Nitrogen volume at $P_2$ [liter]
$v_p$  Piston velocity [m/s]
$x_p$  Piston position [m]
1 Introduction

Ever since horses and sleighs were replaced by tractors, forwarders on wheels have been used to collect and transport logs from the forest to roads where they could be loaded on trucks for further transportation. These forwarders have been developed over decades, but the progress has lately stagnated on the three main challenges of forest terrain transport: 1. Cost/productivity 2. Accessibility 3. Damages to ground vegetation and soil on the forest floor.

A group of Nordic forest-based companies and BAE Systems investigated for two years if the tracked technology used for military all-terrain vehicles could be utilized to overcome the aforementioned challenges. The results showed that BAE Systems tracked technology in combination with hybrid electric drive would significantly improve the forwarder productivity, increasing raw-material accessibility and reduce the ground floor damages. Based on this, a decision was made proceed with the development of these forwarders. The main objectives of developing the HFT could be divided in four categories:

- Development and demonstration of novel technology enabling increasing productivity.
- Optimisation of raw-material flow.
- Improved wood mobilisation through minimized soil disturbance.
- European industry leadership.

The role of BAE Systems within the commitment is to support the forestry machine OEM developing the system specifications for tracked electric forwarders. Their main responsibility is the design and development of a track assembly, suspension system and associated components [1].

One of the goals at BAE Systems system has been to reduce the emerging accelerations in the vehicle cab. The simulations done so far to study the vibration levels has been performed in Adams [2] where a very simple model have been used to represent the suspension system. Still, it has been possible to show that the vibrations are below the fatigue limit for eight hours exposure time according to ISO 2631-1 (1985) and AFS 2005:15 (2005).

The objective of this thesis is to dimension and develop a more realistic model of the hydraulic suspension system. The primary goal is to still be able to meet the standards for whole-body vibration exposure with the new suspension system. A secondary goal is to also reduce the accelerations measured in the vehicle cab in comparison to the values obtained from earlier simulations.

The project can be divided into five phases:

1. Pre-study to gain more knowledge about HFT, what the specifications are, constraints, what have been done so far and what can be used here.

2. Analyse the CAD-model to get information about geometries, dimensions, degrees-of-freedom and to ensure that different parts can move freely during suspension without colliding with each other.

3. Dimensioning the components of the suspension system.
4. Model the suspension system in Simulink [3], establish an environment for co-simulation with the vehicle model in Adams. Run the co-simulation and tune parameters to reduce the vibration levels in the vehicle cab.

5. Develop a control system for controlling the active valves in the suspension system.

6. Summarize and present the results.

The computer used for modelling and co-simulation in this project has a 64-bit operating system and the processor is an Intel(R) Xeon(R) CPU E5-1650 0 @3.20 GHz.

Some of the customer specified requirements for the vehicle are presented in Table 1.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total life expectancy of vehicle</td>
<td>15000</td>
<td>hours</td>
</tr>
<tr>
<td>Maximum vehicle total weight</td>
<td>28</td>
<td>ton</td>
</tr>
<tr>
<td>Vehicle load capacity</td>
<td>15</td>
<td>ton</td>
</tr>
<tr>
<td>Average speed in terrain</td>
<td>7</td>
<td>km/h</td>
</tr>
<tr>
<td>Maximum speed in terrain</td>
<td>15</td>
<td>km/h</td>
</tr>
<tr>
<td>Average speed on road</td>
<td>20</td>
<td>km/h</td>
</tr>
<tr>
<td>Maximum speed on road</td>
<td>40</td>
<td>km/h</td>
</tr>
<tr>
<td>Ambient temperature range</td>
<td>-32 to 35</td>
<td>°C</td>
</tr>
<tr>
<td>Operation in slope up/down</td>
<td>27</td>
<td>degrees</td>
</tr>
<tr>
<td>Operation in side slope</td>
<td>10</td>
<td>degrees</td>
</tr>
<tr>
<td>Maximum vehicle width</td>
<td>2.89</td>
<td>m</td>
</tr>
<tr>
<td>Maximum transportation width</td>
<td>3.1</td>
<td>m</td>
</tr>
<tr>
<td>Maximum transportation height</td>
<td>3.8</td>
<td>m</td>
</tr>
<tr>
<td>Minimum ground clearance</td>
<td>0.6</td>
<td>m</td>
</tr>
<tr>
<td>Maximum vertical travel during suspension</td>
<td>± 0.2</td>
<td>m</td>
</tr>
</tbody>
</table>
2 Standards for Whole-Body Vibration

Whole-body vibration measure procedures and analysis methods have been standardised since the 1970’s. Today, the most used standard is the ISO 2631-1 (1997) which is the only international standard for evaluating whole-body vibrations. Among many other things it include the analytical equations for calculating representative values and technical details and requirements for the measurement equipments and installation of the sensors. It defines the evaluation procedures for health, comfort, perception and motion sickness. The focus is put on a seated posture and the effects of the lumbar spine [5].

The effects from vibration exposure is usually analysed by a method called the weighted root mean square (r.m.s). The r.m.s method is a statistical measure of the magnitude of a varying quantity. It is especially suitable for calculating mean values of signals that are both positive and negative as it is here for the acceleration signals. R.m.s is calculated as

$$ a_w(T) = \left( \frac{1}{T} \int_0^T a_w^2(t)dt \right)^{\frac{1}{2}} \quad (1) $$

where $a_w(t)$ is frequency weighted acceleration. The purpose of weighting the acceleration data is to model the frequency response of the human body. Weighting curves for the three coordinate axis x, y and z are presented in Figure 1. The coordinate system of the vehicle in this project is defined in accordance to the ISO standards, as described in ISO 8855.

![Figure 1: Frequency weighting factors for whole-body vibrations in the different vibration directions (x, y and z).](image)

However, applying frequency weighting will more likely have a small effect on discomfort predictions as the vibration frequency range presented in Figure 1 is heavily skewed at low frequencies [5]. Moreover, the highest value of the weighting factor is around 1. Therefore, frequency weighting is neglected for post-processing in this project, instead a vector sum equation is used where each value is multiplied with a multiplying factor depending on vibration direction.

$$ a = \sqrt{1.4a_x^2 + 1.4a_y^2 + a_z^2} \quad (2) $$

This implies that if the values obtained with (2) satisfies the standards for whole-body vibration, then they even do it with margin at lower frequencies. Table 2 presents the discomfort response for a human at different vibration values. The limit value for whole-body vibrations according to the Swedish work environment authority, Arbetsmiljöverket (AFS 2005:15), is 0.5 m/s² [6].
Table 2: Likely reaction to various vibration values (ISO 2631-1, 1997)

<table>
<thead>
<tr>
<th>Magnitude of overall vibration total value</th>
<th>Discomfort response</th>
</tr>
</thead>
<tbody>
<tr>
<td>Less than 0.315 m/s²</td>
<td>Not uncomfortable</td>
</tr>
<tr>
<td>0.315 m/s² to 0.63 m/s²</td>
<td>A little uncomfortable</td>
</tr>
<tr>
<td>0.5 m/s² to 1.0 m/s²</td>
<td>Fairly uncomfortable</td>
</tr>
<tr>
<td>0.8 m/s² to 1.6 m/s²</td>
<td>Uncomfortable</td>
</tr>
<tr>
<td>1.25 m/s² to 2.5 m/s²</td>
<td>Very uncomfortable</td>
</tr>
<tr>
<td>Greater than 2.0 m/s²</td>
<td>Extremely uncomfortable</td>
</tr>
</tbody>
</table>

The standard ISO 2631-1 explains the causes of vibrations exposure as:

“Whole-body vibration may cause sensations (e.g. discomfort or annoyance), influence human performance capability or present a health and safety risk (e.g. pathological damage or physiological change).”

This shows that discomfort and problems relating to vibration exposure are important, acknowledged and must be considered by the engineers designing and developing new products.
3 CAD-Model

A CAD-model of the vehicle has been frequently updated during the period 2012-2014. Several parts has been designed in more detail based on results from structural analysis for instance. However, some times, the focus has been put on a certain specific part or component without considering the whole model due to lack of time or relevance at that moment. Thus, an overall control is required in order to ensure that no parts are colliding, control arms can move freely according to the requirements and so on. This has been done here, but since it is not the main objective of this thesis, only the most obvious deficiencies in the model are fixed. Furthermore, it has been verified that the suspension cylinders can move freely based on specifications that is presented later.

A decision was made by BAE Systems in the beginning of the development phase for this vehicle to use the track assembly from the BVS10 [7] as a starting point for designing the track assembly aimed to be used on the HFT. On this basis, together with identified load cases and various requirements, the track frame was designed in detail in a preceding work [8]. The shape and dimensions was changed relative to the original one, as well as the position for some of its features. The distance between the attachment points for the control arms was increased but no further adjustment was made at that moment in the CAD-model to compensate for that. This was here fixed by extending the chassis beam so the control arms could be moved to their intended position and assembled correctly to the track frame.

![Figure 2: Control arm attached to the track frame](image)

It was also found that the designed flanges mounted on the chassis beam, at where the rear control arms are attached, was to small. The control arms were already colliding and it would not have been possible for them to move freely according to the requirements. The specified degrees of freedom for the rear control arms are

- $\pm 19.9^\circ$ along the x-axis
- $\pm 12.3^\circ$ along the y-axis
- $\pm 1.2^\circ$ along the z-axis

Moreover, according to the specification [4], the vehicle should have a vertical suspension travel of $\pm 200$ mm. New flanges was designed based on these requirements, see drawings in the Appendix B. It should here be mentioned that only geometrical requirements was considered and a stress analysis is required before any production decision can be made.
Figure 3 presents the designed suspension cylinder, the dimensioning procedure is described in Subsection 4.1.

Figure 3: Suspension cylinder
4 Suspension System Components

The simplest possible hydropneumatic suspension system consists of an accumulator which is directly mounted on the cylinder, and of course the hydraulic fluid. In case of space requirements for instance, the accumulator and cylinder can be separated by oil lines and fittings necessary for the connection. The gas-charged accumulator work as a spring, while its characteristics depends among other things on the accumulator type, its precharge pressure and its volume. The cylinders are the components who transfer the forces between the input side and isolated side and its characteristic are mainly determined by the piston diameter and stroke. Desired damping characteristics of the system can be achieved by adding orifices and/or valves to the suspension system.

4.1 Suspension Cylinder

As already mentioned the cylinders are the load-carrying components in the suspension system. Furthermore, they enable the travel of the suspension which reduces the accelerations on the isolated side. This results in an energy exchange between the mechanical setup (chassis, control arms etc.) and the hydraulic suspension system. In addition to the cylinder dimensions, some other important specifications are presented below [9]:

- Permissible operating pressure
- Temperature ranges
- Compression and rebound stop: Should the cylinder provide this function or are there external end stops
- End-of-stroke damping if applicable
- Maximum piston velocity
- Qualification testing in particular in terms of fatigue endurance
- General operating and environmental conditions.

One of the possibilities to categorize cylinders is by their functional principle which can be distinguished between the single-acting cylinders and the double acting cylinders. A suspension system with only low static load variations or with external mechanical preload, as it is in this case, can be run with single-acting cylinders. An option would be to use a simple plunge cylinder whose advantages are low production cost, high safety factor for buckling and robust connection of the rodside support element. However, if built from solid material, there is a disadvantage of increased unsprung mass and a solution would be to make the rod hollow. Another possibility is to use a more complex single-acting cylinder with piston and rod but connecting the rod chamber to the oil reservoir or simply attach an air filter at the connection port. The cylinder design will here be delimited to determining the required piston diameter and stroke, and based on the results find a proper suspension cylinder on the market.

The design case for dimensioning the piston diameter $D_p$ is the scenario where the pistons are fully retracted and the vehicle needs to be raised to normal drive position. Hydraulic oil of high pressure must then be pumped into the cylinder, so that, the created piston force is high enough to overcome the acting loads and extract the vehicle to normal drive position. The static loads acting over the suspension cylinders was determined by a static simulation in
Adams. Fixed joints were placed in the cylinders to prohibit any possible piston displacement and the vehicle velocity was set to zero. The results are presented in Figure 4.

Figure 4: Static forces acting over the suspension cylinders

The different suspension cylinders are marked and can be identified in Figure 5 where the Adams model of the vehicle is presented.

Figure 5: Adams HFT model

However, in reality, the suspension cylinders on each side of the vehicle, both in the front and the rear frame, are connected to each other by a common accumulator, see Figure 6. This implies that the results in Figure 4 must be interpreted with care and the difference between the model and reality must be considered.

Figure 6: Illustration of how each two cylinders are connected to the same accumulator

In reality, this type of connection implies that the acting loads will be distributed over each couple of interconnected cylinders. For instance, an estimation of the real value for the static load $F_{\text{rear, static}}$ acting over the rear cylinders can be determined by the taking the mean value of the forces Rear first and Rear second from Figure 4.
\[ F_{\text{rear.static}} = \frac{F_{\text{rear.first}} + F_{\text{rear.second}}}{2} = 82634 \text{N} \]  

(3)

The same can be applied on the cylinders in the front which gives

\[ F_{\text{front.static}} = \frac{F_{\text{front.first}} + F_{\text{front.second}}}{2} = 41119 \text{N} \]  

(4)

Assuming that the hydraulic pump can deliver oil with a pressure \( P_{\text{system}} \) of 200 bar, then the required piston diameter can be calculated as

\[ D_p = \sqrt{\frac{4F_{\text{rear.static}}}{\pi P_{\text{system}} \cdot 10^5}} = 0.0725 \text{m} \]  

(5)

Based on the standard piston dimensions the manufacturers can provide, it would here be appropriate to choose a piston with a \( D_p \) of 0.08 m.

The required piston stroke depends on the suspension requirements for the vehicle. The track frame should according to the specifications [4] have a vertical suspension travel of ±200 mm. Figure 7 shows how the suspension cylinder is mounted between the chassis beam and the control arm that is attached to the track frame. During suspension, the extraction and retraction of the piston will among other things depend on the control arm motion. The length of the control arm \( L_{ca} \) is 1140 mm which in turn will have the same displacement as the track frame at its attachment point. Thus, for a vertical displacement of ±200 mm, the control arm will have an angular displacement \( \theta_{ca.disp} \) of

\[ \theta_{ca.disp} = \pm \sin^{-1} \left( \frac{200}{1140} \right) = \pm 10.1^\circ \]  

(6)

\[ \text{Figure 7: Suspension cylinder mounted between the chassis beam and the control arm} \]

The suspension cylinder will have the same angular displacement as the control arm which is used to determine \( s_p \) from the CAD-model. The cylinder length in normal drive position is 642.8 mm and as can be seen in Figure 8, this length will vary between 552.8 mm and 732.8 mm which gives that \( s_p = 0.18 \text{ m} \). Furthermore, the rod should be extended to stick out with 82 mm when the piston is fully retracted.
Based on these dimensions, a cylinder that could be used for this suspension system is CDL2MP5/80/50/180D1/B11CFUMWLY=82mm from Bosch Rexroth [10]. However, this cylinder does not have any end-cushions or piston position sensor. Since this is desired here, the company should be contacted and asked if these features could possibly be added and in that case what the final cylinder dimensions would be. An other possible company to contact is Stacke Hydraulik [11] who manufactures plunge cylinders of different dimensions. At the end, a compromise has to be made between functionality and price, where in this case the functionality should have higher priority.

4.2 End-of-Stroke Damping

A passive suspension system that under no circumstances reaches its end positions, is not dimensioned to be soft enough. A softer suspension system can better reduce arising accelerations by allowing a larger piston travelling distance. However, the stroke is limited, and a required piston displacement that exceeds this stroke results in a bottoming suspension system. The consequence of this is among other things, reduced comfort and high loads on the cylinder cap. A possibility is to stiffen the suspension system to avoid this problem, on the other hand, the comfort will be reduced in all conditions. Thus, a compromise has to be done where the stiffness can be controlled as function of the piston position.

The solution is to use end-of-stroke damping which can be adapted in different ways. An option is to mount a damping-cap on the piston that goes in to the cylinder top at the end position, see Figure 9 for illustration. The play between the cylinder top and damping-cap is very small which reduces the oil flow and a pressure increment counteracts the motion. The end-of-stroke damping should only be active during the motion towards the end stop and let the piston move freely on the way back to the design position. Sizing the constant flow resistor is always a compromise: the deceleration might not be sufficient if the opening area is chosen to be too large. On the other hand, the deceleration will be too high with a too small opening area and in both cases high level acceleration will arise and reduce the comfort.

For a preliminary design of the end-of-stroke damping, it has here been considered sufficient to model the end-cushion with a variable area hydraulic orifice whose opening area is controlled as function of $x_p$ and $v_p$. This is explained in more detail in Subsection 6.4.
Hassan Muhieddine  
Suspension System Components

![Figure 9: Typical design of an end-cushion [12]](image)

4.3 Accumulator

The gas-filled accumulators are the elements in the suspension system which provides the elastic medium for the spring function. The gas (usually nitrogen gas, \(N_2\)) is compressible and the pressure increases with increased compression. It should be mentioned that accumulators are subjected to certain regulations due to the pressurized gas which represents a potential hazard. The directives determine for instance how to dimension an accumulator for the respective pressure ranges, what needs to be considered for their production and so forth, see [13] for more details. Besides the geometrical definitions some other important specifications are summarized below [9]:

- Nominal precharge pressure \(P_0\) at 20° C including tolerance and type of gas
- Inner volume of the accumulator
- Maximum flow rate
- Permissible pressure ratio
- Temperature ranges
- General operating and environmental condition

The most common types of gas pressure accumulators are the diaphragm accumulators, bladder accumulators, sleeve accumulators and the piston accumulators. Based on earlier studies, it has been chosen to use piston accumulators for this suspension system. These accumulators are especially suitable for high pressure ratios and high flow rates. The suspension system should have four accumulators, one for each two cylinders, as presented in Figure 6. The determination of \(P_0\) is not straightforward and requires together with some calculations a bit of parameter tuning which is explained in more detail in Subsection 7.2. The average loads acting over the suspension cylinders was determined by Equation (3) and (4) and are here used to calculate \(P_0\) as

\[
P_{0,\text{front}} = k_f \frac{F_{\text{front,static}}}{A_p \cdot 10^5} \tag{7}
\]

\[
P_{0,\text{rear}} = k_r \frac{F_{\text{rear,static}}}{A_p \cdot 10^5} \tag{8}
\]

where \(k_f\) and \(k_r\) are correction factors to be tuned until a static equilibrium is found for the vehicle. However, for now it will only stated that the \(P_0\) for the accumulators in the front and rear are \(P_{0,\text{front}} = 48 \text{ bar}\) and \(P_{0,\text{rear}} = 82 \text{ bar}\), respectively. While \(P_2\) is assumed to be around 350 bar, the minimum working pressure \(P_1\) can be determined using the following rule of thumb

\[
P_{1,\text{front}} = \frac{P_{0,\text{front}}}{0.9} \approx 53 \text{ bar} \tag{9}
\]
\[ P_{\text{rear}} = \frac{P_{0,\text{rear}}}{0.9} \approx 91 \text{bar} \quad (10) \]

The required accumulator volume depends mainly on the oil volume the accumulator should be able to store, but also the temperature range the vehicle operates in due to expansion and contraction of the nitrogen gas as function of the temperature. Consider the vehicle in normal drive position, the volume of discharged liquid \( \Delta V \) when the piston completely retracts can be determined as

\[ \Delta V = 2A_p \frac{S_p}{2} = 0.0009 m^3 = 0.9 \text{litre} \quad (11) \]

By assuming that no interchange of heat takes place due to the quick operation of the suspension system, the process can be defined as adiabatic and the accumulator volumes can according to [9] be determined as

\[
\begin{align*}
V_{0,\text{front}} &= \frac{\Delta V}{(\frac{P_{0,\text{front}}}{P_{1,\text{front}}})^{1/n} - (\frac{P_{0,\text{front}}}{P_2})^{1/n}} = 1.31 \text{litre} \quad (12) \\
V_{0,\text{rear}} &= \frac{\Delta V}{(\frac{P_{0,\text{rear}}}{P_{1,\text{rear}}})^{1/n} - (\frac{P_{0,\text{rear}}}{P_2})^{1/n}} = 1.57 \text{litre} \quad (13)
\end{align*}
\]

where \( n = 1.4 \) here. This volume is defined at room temperature, but as mentioned before the external temperature variations has to be considered. According to the specifications in [4], the highest and lowest operating temperatures are \( T_2 = +35^\circ \text{C} \) and \( T_1 = -32^\circ \text{C} \), respectively. The new accumulator volumes is calculated as

\[
\begin{align*}
V_{0T,\text{front}} &= \frac{V_{0,\text{front}}(T_2 + 273)}{T_1 + 273} = 1.67 \text{litre} \quad (14) \\
V_{0T,\text{rear}} &= \frac{V_{0,\text{rear}}(T_2 + 273)}{T_1 + 273} = 2 \text{litre} \quad (15)
\end{align*}
\]

The formulas used here refers to ideal gases, but industrial nitrogen used in accumulators does not behave according to ideal gas laws when pressure increases. It is convenient to keep in mind this characteristics for pressure \( P_2 > 200 \text{ bar} \). A correction factor \( C_a \) from Figure 10 can be used to compensate for that.

\[\text{Figure 10: Graph for determining the correction factor } C_a \quad [14]\]
The pressure ratio $P_2/P_1$ in the front and rear are

$$\frac{P_2}{P_{1,\text{front}}^\text{front}} = 6.56$$

(16)

$$\frac{P_2}{P_{1,\text{rear}}^\text{rear}} = 3.84$$

(17)

As can be seen, the value of $P_2/P_{1,\text{rear}}$ is in the range presented in Figure 10 and $C_{a,\text{rear}}$ can be estimated to be 1.15. This is not true for the front mounted accumulators but the value of $C_{a,\text{front}}$ has here been roughly estimated to 1.1. This gives the volumes

$$V_{0TC,\text{front}} = \frac{V_{0T,\text{front}}}{C_{a,\text{front}}} = 1.53\text{litre}$$

(18)

$$V_{0TC,\text{rear}} = \frac{V_{0T,\text{rear}}}{C_{a,\text{rear}}} = 1.70\text{litre}$$

(19)

Recalling that these calculations are based on the vehicle starting from its normal drive position, additional volume has to be added to consider the oil volume that is transported to the cylinders if the pistons completely extracts from their normal position. The new accumulator volumes becomes

$$V_{0,\text{front,n}} = V_{0TC,\text{front}} + 2Ap\frac{s_p}{2}10^3 = 2.43\text{litre}$$

(20)

$$V_{0,\text{rear,n}} = V_{0TC,\text{rear}} + 2Ap\frac{s_p}{2}10^3 = 2.60\text{litre}$$

(21)

Based on the standard accumulator volumes on the market, an accumulator with a $V_0$ of 2.5 litre should be chosen to be used in the front part of the vehicle. A possible piston accumulator that could be used is the SK350 2.5/2112 U-350 ... ... 08 48 from Hydac [14]. The dots marks where the terms for the fittings should typed which is not so important at this stage. This accumulator has a piston diameter of 80 mm, an external diameter of 180 mm and weighs approximately 50 kg. For the accumulators in the rear part of the vehicle, Hydac should be contacted to discuss the possibility of manufacturing accumulators with a $V_0$ of 2.6 litre for instance. The accumulators should be located in protected positions, but also as close as possible to the suspension cylinder for low resistance of the connecting line. The spring force is calculated as

$$F_S = P_{\text{acc}}A_p$$

(22)

### 4.4 Flow Resistor

The damping created in lines and fittings is non-adjustable and its share to the total damping is recommended to be around 10 %. The rest of the damping can be created by flow resistors whose restriction is selectable and gives therefore a selectable damping. Important to keep in mind when designing the flow resistors is that they can be subjected to very high hydraulic flow forces due to high pressure losses over them. Therefore they have to be designed and sized accordingly to avoid any damages that can have a negative influence on the flow and system behaviour. Usually a flow resistor is defined by the required pressure loss vs. flow curve, otherwise it needs to be defined by its geometry. Some of the most important specifications are [9]:

- Dimension of the cross-sectional constriction.

- Tolerances.
• Edges in the flow path: possibly detailed definition of the edges.

• Maximum flow rate. Especially important if the flow resistor is made of low strength material.

Due to the suspension movement, the fluid flows between the cylinder and accumulator with regularly changing direction. As the fluid passes through a resistor (see Figure 11), the kinetic energy of the fluid is transformed into heat. A pressure loss $\Delta P$ is generated, which causes, via the active area of the piston a damping force that counteracts the piston motion.

$$F_D = \Delta P A_p$$

where

$$\Delta P = \frac{q^2 \rho}{4(C_q A_{O.f})^2}$$

The value $A_{O.f}$ for the orifices mounted in connection to the suspension cylinders was analytically determined to be $5 \cdot 10^{-6}$ m$^2$. It is however more preferable to choose this value and gradually tune it based on the results from the co-simulation. This was not possible due reasons that is discussed in Section 9. Although the calculated value of $A_{O.f}$ turned out to be suitable, it is not an optimal value due to reasons that is discussed in Section 8.

Figure 11: Flow resistor with fixed orifice area
5 Co-Simulation Interface

The difficulties encountered in the simulation of a coupled hydro-mechanical system are slightly different from those encountered in the simulation of pure mechanical or hydraulic systems. Using only one simulation program can be very demanding and it can be more convenient to model the different subsystems in the most appropriate simulation environment. Today, there are several sophisticated programs on the market providing engineers the ability to simulate and analyse products under development. The HFT has already been modelled in Adams for dynamic analysis where forces, moments, accelerations etc. can be studied. The dimensioned suspension system is in this project modelled in Simulink and has to be connected to the vehicle model in Adams for a co-simulation. This implies that the whole system is simulated simultaneously and the impact of the designed suspension system on the vehicle vibration levels can be studied. The two programs communicates during the entire process by exchanging data with an communication interval $\tau$. The procedure of how to establish an environment for co-simulation with Simulink and Adams is explained, step by step, in Subsection 7.1. The process of how the interface between the programs was determined is described here.

One of the challenges in this kind of co-simulation, especially when there is physical models in both programs, is to determine the interface between them. In other words, to determine the in- and output data for the respective program. Simplified models was therefore used for this purpose. These models are presented in Figure 12 and Figure 13.
The preliminary idea was to apply a time dependent force on the "track assembly" in Adams and to use that force as input to the Simulink model. This force could then be applied on the suspension system in Simulink to put the piston into motion. The pressure drop over the orifice and the accumulator pressure was measured and used to calculate $F_D$ and $F_S$. These forces, together with the piston displacement and velocity was then sent as inputs to the adams_sub block.

It became very soon obvious that this interface was not appropriate. The produced results was not at all as expected and the interface had to be changed. The problem was that this interface restricted the computations in Adams by prohibiting it from determining $x_p$ and $v_p$ by itself. After more research and trying, a more suitable interface was found which can be described as Figure 14 presents. The dynamics are completely handled by Adams while the Simulink model is only used to determine $F_S$ and $F_D$ and send it back to Adams.

The Simulink model was modified accordingly and the final version is presented in Figure 15. The interface to be used for the co-simulation was thus determined and could now be applied on the real models.

![Figure 14: Interface between Simulink and Adams](image1)

![Figure 15: Final version of the simplified Simulink model](image2)
Simulink Model

Simulink provides a graphical editor, customizable block libraries, and solvers for modelling and simulating dynamic systems. Moreover, it is integrated in MATLAB which enables the user to incorporate MATLAB algorithms and export results for further analysis [3]. The Simulink add-on product that was used to model the suspension system is called SimHydraulics. It provides a library that includes models of hydraulic components, such as, cylinders, accumulators and flow resistors. References for modelling the suspension system is the scheme in [15], the component dimensions previously determined and the desired interface between the programs. Before presenting the suspension system model, some of the component blocks used in the model is presented to discuss some of their functionalities and limitations. More details can be found in the SimHydraulics documentation.

6.1 Single-acting Hydraulic Cylinder

The suspension cylinders are modelled with the block presented in Figure 16. Only the basic cylinder functionalities are reproduced to favor of better numerical efficiency which makes it especially suitable for real-time and HIL simulations. Some of the factors neglected here are fluid compressibility, friction and leakages [3].

![Simulink block for the simplified version of the single-acting hydraulic cylinder](image)

**Figure 16: Simulink block for the simplified version of the single-acting hydraulic cylinder**

6.2 Gas-charged Accumulator

The accumulators are modelled by the block presented in Figure 17. The accumulator is divided into two chambers, one with gas and the other with fluid, separated by a piston. As the fluid pressure on the accumulator inlet side becomes higher than the gas pressure, fluid enters the accumulator and compresses the gas. If the pressure at the accumulator inlet drops below $P_0$, the gas chamber gets isolated from the systems by the inlet valve. It remains in that state until the pressure at the inlet builds up to $P_0$ or higher [3].

![Simulink block for the gas-charged accumulator](image)

**Figure 17: Simulink block for the gas-charged accumulator**

Some of the basic assumptions and limitations are:

- The gas compression is calculated using the thermodynamic laws for ideal gases.
- The process is polytropic
- No loading on the separator, such as friction and so on is considered.
- Fluid compressibility is neglected
6.3 Constant Area Hydraulic Orifice

The flow resistor with constant orifice area has been modelled with the block presented in Figure 18. The positive direction is from port A to B, which implies that \( q \) is positive in that direction. The flow rate through the orifice is also proportional to the pressure differential \( P_A - P_B = \Delta P \) across the orifice which causes the damping in the system. One of the settings is the critical Reynolds number \( Re_{cr} \) at where transition from laminar to turbulent flow takes place. The recommended value of this parameter can be found in hydraulic textbooks and depends on the orifice geometrical profile. Here, the default value has been used which corresponds to a round orifice in thin material with sharp edges. No fluid inertia is taken into account and the transition between laminar and turbulent regimes is assumed to take place exactly at \( Re = Re_{cr} \) [3].

![Simulink model for the constant area hydraulic orifice](image)

6.4 Variable Area Hydraulic Orifice

This orifice is presented in Figure 19 and has exactly the same functionality and limitations as the fixed area hydraulic orifice besides that the area is variable and has to be computed from outside the block.

![Simulink model for the variable area hydraulic orifice](image)

Here it is used to model the end-of-stroke damping. As previously determined, \( s_p \) is \( \pm 90 \) mm from its normal position and the end-of-stroke damping is only supposed to influence the system during the last 10% of the stroke in respective direction. Figure 20 presents a simple cylinder model where the area where the end-of-stroke damping should be inactive is marked by the dashed box. As the piston is in this area, the orifice is assigned a relatively large area value to "deactivate" it, in the sense that there will not be any pressure drop over it. As the piston approaches the end stops (outside the dashed box), the variable orifice is activated by being assigned an area value \( A_{O,v} \) that decreases linearly to zero as the piston approaches the end.

![Cylinder where the dashed box illustrates the area at where the end-of-stroke damping is inactive](image)
6.5 Suspension System Model

The suspension system for the HFT can be divided into four subsystems as presented in Figure 21. The subsystems are almost identical and contain the same components, the only difference is the accumulator parameter values depending on if the subsystem correspond to the quarter of the suspension system in the front or the rear.

![Figure 21: Complete suspension system divided in subsystems](image)

Figure 21: Complete suspension system divided in subsystems

Figure 22 presents the content of the subsystems presented in Figure 21. See Appendix A for clearer figures.

![Figure 22: Quarter model of the suspension system](image)
The part in Figure 22 which is marked with a dashed box is here enlarged and presented with its content in Figure 23.

**Figure 23: Subsystem with its content, used for calculating the spring and damping forces**
7 CO-simulation

7.1 Preprocessing

In general there are five main steps that have to be processed before the co-simulation can be started:

1. **Identify Adams inputs and outputs**: These were identified when the interface between the programs was determined and are presented in Tables 3 and 4. The template file for the HFT model is opened in the template builder in Adams and the inputs and outputs is defined as state variables in the following way:

   - Go to Build→System Elements→State Variable→New...
   - The window shown in Figure 24 appears

   ![Figure 24: Menu for creating state variables]

   - When all state variables are defined, the template file is saved and the template builder is closed.

<table>
<thead>
<tr>
<th>Variable name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>.HFT.v2.HFT_body.damper_force_front_1_left</td>
<td>$F_D$, first cylinder, left side in the front</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.damper_force_front_1_right</td>
<td>$F_D$, first cylinder, right side in the front</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.damper_force_front_2_left</td>
<td>$F_D$, second cylinder, left side in the front</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.damper_force_front_2_right</td>
<td>$F_D$, second cylinder, right side in the front</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.damper_force_rear_1_left</td>
<td>$F_D$, first cylinder, left side in the rear</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.damper_force_rear_1_right</td>
<td>$F_D$, first cylinder, right side in the rear</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.damper_force_rear_2_left</td>
<td>$F_D$, second cylinder, left side in the rear</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.damper_force_rear_2_right</td>
<td>$F_D$, second cylinder, right side in the rear</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.spring_force_front_1_left</td>
<td>$F_S$, first cylinder, left side in the front</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.spring_force_front_1_right</td>
<td>$F_S$, first cylinder, right side in the front</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.spring_force_front_2_left</td>
<td>$F_S$, second cylinder, left side in the front</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.spring_force_front_2_right</td>
<td>$F_S$, second cylinder, right side in the front</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.spring_force_rear_1_left</td>
<td>$F_S$, first cylinder, left side in the rear</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.spring_force_rear_1_right</td>
<td>$F_S$, first cylinder, right side in the rear</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.spring_force_rear_2_left</td>
<td>$F_S$, second cylinder, left side in the rear</td>
</tr>
<tr>
<td>.HFT.v2.HFT_body.spring_force_rear_2_right</td>
<td>$F_S$, second cylinder, right side in the rear</td>
</tr>
</tbody>
</table>
Table 4: Output variables from Adams

<table>
<thead>
<tr>
<th>Variable name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>.HFT_v2.HFT_body.sdl_suspension_length_front_first</td>
<td>$l_{cyl}$, first cylinder, left side in the front</td>
</tr>
<tr>
<td>.HFT_v2.HFT_body.sdl_suspension_length_front_second</td>
<td>$l_{cyl}$, second cylinder, left side in the front</td>
</tr>
<tr>
<td>.HFT_v2.HFT_body.sdl_suspension_length_rear_first</td>
<td>$l_{cyl}$, first cylinder, left side in the rear</td>
</tr>
<tr>
<td>.HFT_v2.HFT_body.sdl_suspension_length_rear_second</td>
<td>$l_{cyl}$, second cylinder, left side in the rear</td>
</tr>
<tr>
<td>.HFT_v2.HFT_body.sdl_suspension_speed_front_first</td>
<td>$v_p$, first cylinder, left side in the front</td>
</tr>
<tr>
<td>.HFT_v2.HFT_body.sdl_suspension_speed_front_second</td>
<td>$v_p$, second cylinder, left side in the front</td>
</tr>
<tr>
<td>.HFT_v2.HFT_body.sdl_suspension_speed_rear_first</td>
<td>$v_p$, first cylinder, left side in the rear</td>
</tr>
<tr>
<td>.HFT_v2.HFT_body.sdl_suspension_speed_rear_second</td>
<td>$v_p$, second cylinder, left side in the rear</td>
</tr>
</tbody>
</table>

2. **Import the Adams model**: The HFT is loaded into Adams/Car and should be complete and include all necessary geometry, constraints, forces and measures. Furthermore, the track system has to be wrapped and the road surface has to be loaded which is done in the following way:

- Go to ATV → Tracked Vehicle Setup → Track System Wrapping...
- The window shown in Figure 25 appears

![Figure 25: Track systems setup window with the settings and data used in this project](image)

- The necessary information, such as road data and starting position for the vehicle is entered here.
3. **Generate files:** Necessary files containing model properties, initialization commands and other information has to be generated which is done in the following way:

- In Adams/Car go to Simulate→Tracked Vehicle Analysis→Submit...
- The window shown in Figure 26 appears

![Figure 26: Tracked vehicle analysis window with the settings used in this project](image)

- The required information, such as, simulation time and vehicle velocity is entered here. This will generate three files: `Co-sim.adm`, `Co-sim.acf` and `Co-sim.log`. The `Co-sim.adm` file contains detailed information about the vehicle model, such as geometry, masses, constraints and so on. The `Co-sim.acf` contains both initializations commands and commands for how the simulation should be performed. For instance, the simulation should be started by a static analysis to find an equilibrium state for the vehicle. Next, the track pads are assigned the correct track tension and finally the dynamic simulation commands are executed. The `Co-sim.log` contains general information, such as the analysis name, assembly info and subsystem.

4. **Generate the Adams plant block:** After the Adams/control plug-in has been loaded, the Adams plant block is generated in the following way:

- Go to Controls→Plant Export
- The window shown in Figure 27 appears.
- The necessary information, such as, input- and output data to the Adams block is defined. When selecting Apply, a Matlab file called `Co_sim.m` will be generated and stored in the working directory of Adams. A small modification has to be made in the `Co_sim.m` file. The value of the variable ADAMS_init has to be manually changed from `'file/command=Co_sim_controls.acf'` to `'file/command=Co_sim.acf'`. Important to note here is that the Adams Host Name is the name of the computer with the network path. This information is stored under ADAMS_host in the m-file.
and has to be changed if the files is to be used for co-simulation on an other computer or network.

![Figure 27: Adams/Controls plant export window with the data and settings used in this project](image)

5. **Building the block diagram**: At this stage it is time to connect the Simulink and Adams models which is done in the following way:

   - In the Matlab command window, `Co_sim` is typed which will load variables, such as current working directory, solver type for Adams, Adams host name and so on. These variables can be seen in the MATLAB workspace.
   - Next, `adams.sys` is typed in the MATLAB command window which will open the window shown in Figure 28.

![Figure 28: Adams plant block](image)

   - The Adams_sub block is copied and pasted to the Simulink model as presented in Figure 29.
Figure 29: Simulink suspension system model and Adams vehicle model connected
7.2 Static Co-Simulation

In Subsection 4.3, \( P_0 \) was given without any details of how it was determined. This is presented here since a static co-simulation is required for this purpose. As soon as the co-simulation process is started, an Adams terminal window opens and the execution of the initialization commands in the `Co_sim.acf` starts. Adams is chosen to lead the process and since no data exchange occurs between the blocks in Figure 29 in this phase, the vehicle will have trouble finding a static equilibrium due to the absence of a suspension system in the Adams model. Hence, fixed joints are placed in the cylinders to enable the initialization commands to be successfully executed, see Figure 30. On the other hand the joints must be deactivated as soon as the simulation starts and the programs begins to exchange data. Therefore the `Co_sim.acf` file had to be modified.

The original `Co_sim.acf` file:

```
file/model=Co_sim
!
preferences/solverbias=cxx
!
equilibrium/error=100.000000,maxit=250,stability=10.000000,imbalance=10.000000
simulate/static
!
equilibrium/error=10.000000,maxit=250,stability=10.000000,imbalance=1.000000
simulate/static
!
equilibrium/error=1.000000,maxit=250,stability=1.000000,imbalance=0.100000
simulate/static
! The next line will perform a static simulation and set the tensioners
control/function=USER(1,204,205,208,209)\ routine=atv_solver::con001
!
simulate/dynamics,end=20.0,steps=1000
!
stop
```
When the initialization commands are successfully executed, the simulation starts and Simulink and Adams initializes their solution at \( t=0 \). Then, Adams simulates for a duration of \( \tau \), holding the Simulink inputs constant. Next, Simulink simulates for a duration of \( \tau \) holding the Adams inputs fixed. Finally, Simulink and Adams exchanges updated input/output state values and the process repeats for another \( \tau \) duration.

As mentioned, the purpose of this static co-simulation is to determine \( P_0 \). In the first simulation, \( P_{0,\text{front}} \) and \( P_{0,\text{rear}} \) was assigned the values obtained with Equations (7) and (8), given that \( k_f \) and \( k_r \) are equal to 1. The vehicle velocity is set to zero and its vertical motion during the simulation is analysed. The static equilibrium found during the execution of the initialization commands is valid before the fixed joints is deactivated. Together with the fact that there is a time delay of \( \tau \) from the moment these joints are deactivated until the programs exchanges data, it is almost impossible to have an ideal scenario where the vehicle is completely still. However, a certain level of oscillation, in terms of piston displacement of up to \( \approx \pm 10 \text{ mm} \) is considered as acceptable and works as a reference when determining \( P_{0,\text{front}} \) and \( P_{0,\text{rear}} \). It turned out from the first co-simulation that the chosen precharge values was too high. The correction factors was therefore gradually reduced until a satisfying result was achieved with \( k_r = 0.2 \) and \( k_f = 0.22 \).

### 7.3 Dynamic Co-Simulation

Whole-body vibrations is the most important health and safety issue for the forest truck drivers. In order to measure and compare emerging vibration levels in different forest trucks, a standardized test track was developed by Skogforsk and Hultdins AB [16]. The same test track has been modelled in Adams to be used for simulations during the development of the HFT. The time it takes for the vehicle to pass the test track with a velocity of 7 km/h is estimated to 20 s, which is also used as the stop time for the co-simulation here. \( \tau \) is set to 0.0005 s which is relatively low and makes the co-simulation time demanding. However, this values is chosen to
avoid any sudden changes in the input signals to the blocks that could disturb the simulation due to numerical problems. It turned out to be more challenging than expected to run a dynamic co-simulation with physical models in two different programs. The co-simulation had a tendency to suddenly stop before it was finished and it was not possible to run a co-simulation for 20 s. A lot of effort was put on tuning numerical parameters, such as, relative and absolute tolerances, consistency tolerance, max/min step and so on. Finally, after some troubleshooting and parameter tuning it was possible to run a co-simulation for 17 s. Since the vehicle at that time had almost passed the whole test track, the results from that co-simulation was sufficient for drawing some conclusions regarding the dimensioned suspension system.

The encountered problems is discussed in Section 9, together with suggestions for future work and improvements. The final solver settings used in Simulink for the co-simulation are presented in Figure 31. The solver used in Adams is called HHT (recommended for stiff problem) and the error tolerance there was set to 1e-5.

Figure 31: Solver settings in Simulink for the co-simulation
8 Results

As mentioned in the introduction of this report, the main objective of this work was to dimension a suspension system, such that, the standards for whole-body vibration exposure could be met, but also to reduce the emerging accelerations in the vehicle cab in comparison to the ones obtained from the simulations performed with a simplified suspension system in [17]. The accelerations measured in that project [17] are presented in Figures 32 and 33.

As can be seen, the acceleration values post-processed according to ISO 2631-1 (1985) is just on the limit for what is considered as not uncomfortable according to Table 2. Moreover, they are slightly higher than the ones post-processed according to AFS 2005:15 (2005). It should here be mentioned that the ISO 2631-1 (1985) and ISO 2631-1 (1997) are similar, despite some parts that has been updated through the years.
Here, Equation (2) is used to post-process the acceleration data obtained from the co-simulation in this project. The result is presented in Figure 34.

![Acceleration data post-processed according to Equation (2)](image)

As can be seen, the highest value of $a$ is 0.2 m/s$^2$ which implies that not only the criteria and standards for whole-body vibration exposure are met, but also that the maximum value of $a$ has been reduced in comparison to the values from [17], see Figures 32 and 33. Moreover, the sensor for measuring the accelerations was placed on the top of the vehicle cab. Thus, the vibration level the driver is exposed to is even lower thanks to the additional damping in the seat. Important to mention here, is that the simulations in [17] was performed with a vehicle weight of 26000 kg while the co-simulations in this project was performed with a vehicle weight of 28000 kg.

The angular acceleration $\alpha$ around the three axis x, y and z is presented in Figure 35. One could clearly see that $\alpha$ is remarkably higher around the x-axis than the other axis. This information is important and should be considered in the future if one aims to further reduce the vibration levels.

![Angular acceleration $\alpha$ around the three coordinate axis x, y and z](image)

The highest flow rate in the system was measured to 60 l/min. This value can be used as a reference, together with $A_{p,f}$ to estimate the order of magnitude of the forces the flow resistors are subjected to. The solidity of the component should be one of the specifications when choosing flow resistors.
The piston displacement $\Delta s$ is presented in Figure 36. As can be seen, not the full stroke is used during operation which implies that the system is a bit too stiff. The orifices area are too small and should be slightly increased to reduce the high damping in the system.

![Figure 36: Piston displacement](image-url)
9 Discussion

Setting up a working co-simulation turned out to be more challenging than expected due to the complexity and high stiffness of the models. Moreover, the programs was very sensitive to the numerical settings used. Not only that they were independently sensitive for their respective solver settings, but also they had a large influence on each other and was model dependent. For instance, certain settings that worked moderately well for a certain model configuration appeared to be completely inappropriate if the orifice area was slightly changed. For this reason $A_{o,f}$ could not be gradually tuned until an optimal value was found. Much effort was put on troubleshooting and adjusting numerical parameters until finally a co-simulation of 17 s could be performed.

The short simulation step time made the co-simulations very time demanding. It took the computer in average 13 minutes to simulate a second. Despite that, there was also no guarantee for how long a co-simulation would run due to the sensitivity of the programs. The problem was that there were no physical explanation for why the simulation suddenly stopped at some random time. All quantities of interest that are coming in/out to the Adams model was plotted to analyse what was happening around the time of failure, but nothing strange was observed. The main problem was that the Adams integrator HHT could suddenly, and for an unknown reason, reduce the time step rapidly to the minimum allowable time step value HMIN which resulted in convergence problem. The value of HMIN was adjusted together with other numerical settings that could have an impact on this behaviour but without success.

At this stage, MSC Adams support was contacted to discuss this problem and it appeared to be a bug in the solver. Their first recommendation was to upgrade Adams where the bug has been fixed in the newer versions. This was discussed with one of the engineers at BAE Systems and it turned out that he was aware of this problem but the reason for why they were still using an older Adams version had to do with the HFT model. They had previously tried to update Adams but not managed to run a simulation with the HFT model. Since the bug in the integrator becomes noticeable at the very end of a simulation, they went back to the older version and tackled the bug problem by increasing the simulation time.

This might be a solution for simulations where only Adams is used and where the simulation time can be slightly increased so that all results of interest can be obtained before the simulation crashes. However, this can not be done in a co-simulation where the programs runs "new" simulations at each time progression but with updated initial values. Thus, increasing the simulation time was not a solution, nor upgrading the program which would lead to new problems. The second recommendation was to change the integrator from HHT to GSTIFF. This did not help and the co-simulations did not run for more than four seconds on average. Although there was a bug with the HHT integrator it worked a lot better and might be the best solver option for co-simulations of this kind.

Regarding the suspension system modelling, the preliminary idea was to model it without any orifices. This is to study the flow rate levels in the system and based on them dimension the orifices. It turned out to not be so proper since the absence of any damping in the system allowed the piston to hit the cylinder hard stops which resulted in high forces in directions that made the vehicle to completely loose contact with the ground. This is of course not a realistic scenario and an attempt to solve this problem was made by reducing the hard stop stiffness in the cylinders. Still, the upcoming forces was too high and the problem persisted. Due to time
restriction, orifices was directly added to the suspension system, and with a certain damping level the problem was avoided.

One of the assumptions made in the very beginning of this project was that the vehicle load was equally distributed over the front and rear parts of the vehicle. This was based on the assumption made in [8] when the track frame was designed. The suspension cylinders was dimensioned according to that, which turned out to not be completely true (see Figure 4) and the cylinders had to be re-dimensioned. This also implies that the calculations made in [8] have to be checked to see if the results still holds for the real load distribution, or if anything have to be updated.
10 Conclusions and Recommendations

The results from the co-simulation clearly show that with the dimensioned suspension system, the ISO 2631-1 (1997) and AFS 2005:15 (2005) standards can be met. Moreover, the vibration levels is reduced compared to the ones obtained in earlier studies. Although no full co-simulation could be run due to the numerical problems and a software bug, the obtained results from the most successful co-simulation was enough to draw this conclusion. The time and effort put on troubleshooting and changing numerical parameters is most likely due to the bug in Adams which caused convergence problems.

One of the thoughts that arose during the project was that maybe having physical models in two different programs for co-simulation is not suitable. However, when it has been managed to run a co-simulation for 17 of 20 seconds, with realistic results and despite the bug in one of the programs, such conclusion can not be drawn before new co-simulations have been performed with proper working programs. Therefore, it is highly recommended to upgrade Adams and invest some time and effort on making the HFT model work in the new version as well. In any case, Adams has to be upgraded sooner or later. If the co-simulations can be successfully performed with an upgraded program, then there already exist a suspension system model that can be used for further development. On the other hand, if a program upgrading do not solve the problem, then the conclusion can be drawn that it is probably more suitable to model all physics in the same program. In that case, the suspension system should be moved from Simulink into Adams. Simulink can still be used, but in a later stage when the suspension system is extended with active valves and a control system needs to be modelled to control these valves.

If the suspension system has to be modelled in Adams, the use of the Adams/Hydraulics plug-in should be considered. The reasons for why it was not used in this project is that BAE Systems did not have the required license, but also, the Simulink plug-in SimHydraulics is more powerful for simulating hydraulic systems and provides the possibility to model the suspension system on a very detailed level. However, regardless of what program is used in the future for modelling the suspension system, it has to be extended. Hoses and fittings has to be added to study their impact on the system, especially on the damping. The end-of-stroke damping is today represented by a variable area orifice. This part have to be modelled in more detail.

The suspension system modelled in this project can be considered as passive. Due to the unexpected problems that arose, there was no time to develop it further to be of active type. For the future this should be done since the vehicle have to operate in different configurations, speeds and terrains. To be competitive on a market where the driver comfort is one of the highest prioritized requirements, the suspension system must be optimized for best performance. A recommendation is to use CES valves [18], one at each suspension accumulator. These valves are ideal and can be used to control the damping of the suspension systems as function of the vehicle weight, operation speed, slope and terrain for instance.

Finally, one should use the CAD-model for determining the weight of the different vehicle parts in Adams more exactly. This will change the centre of gravity position, as well as the moments of inertia, thus also the vehicle dynamic response during simulations. Today, the vehicle mass in Adams is adjusted by changing the mass of the spheres added to the model, see Figure 5. They should be removed and the vehicle components masses should be updated more accurately, which would have an influence on the results.
References


Appendix B

Attachment flange on the chassis beam for the suspension cylinder
Scale 1:1 [mm]
Attachment flange on the chassis beam for the control arms

Scale 1:1 [mm]