System modelling and evaluation of main battle tank fire precision

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Abstract

This master thesis describes a study of the main battle tank dynamics in order to investigate the fire precision when a tank is driving in terrain. A model has been developed to simulate the dynamic interaction between the tank's hull and the ground irregularity in MATLAB and SIMULINK through the modelling of the tank's dynamics. Two different models of suspension system have been analysed. One linear model and one hydro-pneumatic model. Further the contribution from the cannon's recoil has been modelled to investigate its contribution to the dynamics of the vehicle. The models developed are on system level and is to be implemented in a larger model. Therefore are the models simplified and the thesis investigates to what degree of simplification the models will accurately predict the movement of the tank.
Sammanfattning

Denna examensarbete beskriver en studie på stridsvagns-
dynamik för att undersöka precision av målträff när stridsvagnen framförs i terräng. En modell har utvecklats för
att simulera hur stridsvagnar påverkas av underlagets va-
ration i MATLAB och SIMULINK genom att modellera
stridsvagnens dynamik. Två olika former av stötdämpare
har undersökts, en linjär modell samt en hydro-pneumatisk
modell. Även bidraget från kanonen’s avfyraning har mo-
dellerats för att se hur rekylens bidrag påverkar rörelsen
av stridsvagnen. Målet med studien var att ta fram en så
förenklad modell som möjligt. Flera modeller har därför
utvecklats för att jämföra förenklingsgraden.
Acknowledgement

I would like to say a really big thank you to Ekaterina Fedina for all the support during this thesis and being an excellent supervisor for the project. I would also like to thank Arvid Carlstedt and Mikael Lyth for all the good discussions and help with the progress of the thesis. Last but not least I would like to thank Niclas Stensbäck, Anders Lindström and Mikael Nybacka for the good discussions regarding the thesis.

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Abbreviations

MBT - Main Battle Tank
HGS - Hydro-Gas Suspension
CoG - Centre of Gravity
IFV - Infantry Fighting Vehicle
Variable declaration

Table 0.1: Variable declaration.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Explanation of variable</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>z</td>
<td>Vertical displacement of the hull</td>
<td>m</td>
</tr>
<tr>
<td>\dot{z}</td>
<td>Vertical speed of the hull</td>
<td>m s(^{-1})</td>
</tr>
<tr>
<td>\ddot{z}</td>
<td>Vertical acceleration of the hull</td>
<td>m s(^{-2})</td>
</tr>
<tr>
<td>z_{ti}</td>
<td>Vertical displacement of respective tyre</td>
<td>m</td>
</tr>
<tr>
<td>z_{li}</td>
<td>Vertical displacement of respective track component</td>
<td>m</td>
</tr>
<tr>
<td>z_{gi}</td>
<td>Vertical displacement of ground</td>
<td>m</td>
</tr>
<tr>
<td>\phi</td>
<td>Pitch angle of the hull</td>
<td>rad</td>
</tr>
<tr>
<td>\dot{\phi}</td>
<td>Pitch speed of the hull</td>
<td>rad s(^{-1})</td>
</tr>
<tr>
<td>\ddot{\phi}</td>
<td>Pitch acceleration of the hull</td>
<td>rad s(^{-2})</td>
</tr>
<tr>
<td>\gamma</td>
<td>Roll angle of hull</td>
<td>rad</td>
</tr>
<tr>
<td>\dot{\gamma}</td>
<td>Roll speed of hull</td>
<td>rad s(^{-1})</td>
</tr>
<tr>
<td>\ddot{\gamma}</td>
<td>Roll acceleration of hull</td>
<td>rad s(^{-2})</td>
</tr>
<tr>
<td>x_{i}</td>
<td>Wheel position wrt CoG</td>
<td>m</td>
</tr>
<tr>
<td>y_{i}</td>
<td>Wheel position wrt CoG (width)</td>
<td>m</td>
</tr>
<tr>
<td>m</td>
<td>Mass of the vehicle</td>
<td>kg</td>
</tr>
<tr>
<td>m_{ti}</td>
<td>Mass of the wheel</td>
<td>kg</td>
</tr>
<tr>
<td>m_{li}</td>
<td>Mass of the track component</td>
<td>kg</td>
</tr>
<tr>
<td>J_{xx}, J_{yy}</td>
<td>Moment of inertia</td>
<td>kg m(^2)</td>
</tr>
<tr>
<td>J_{xy}</td>
<td>Moment of deviation</td>
<td>kg m(^2)</td>
</tr>
<tr>
<td>i</td>
<td>Index</td>
<td>–</td>
</tr>
<tr>
<td>N</td>
<td>Quantity of wheels/track components</td>
<td>–</td>
</tr>
<tr>
<td>k_{d}</td>
<td>Spring stiffness of main spring</td>
<td>N m(^{-1})</td>
</tr>
<tr>
<td>k_{i}</td>
<td>Spring stiffness of wheel to track</td>
<td>N m(^{-1})</td>
</tr>
<tr>
<td>k_{l}</td>
<td>Spring stiffness of track to ground</td>
<td>N m(^{-1})</td>
</tr>
<tr>
<td>c_{d}</td>
<td>Damping coefficient main damper</td>
<td>N s m(^{-1})</td>
</tr>
<tr>
<td>c_{i}</td>
<td>Damping coefficient wheel</td>
<td>N s m(^{-1})</td>
</tr>
<tr>
<td>c_{l}</td>
<td>Damping coefficient track to ground</td>
<td>N s m(^{-1})</td>
</tr>
<tr>
<td>P_{i}</td>
<td>Pressure of nitrogen chamber</td>
<td>Pa</td>
</tr>
<tr>
<td>V_{i}</td>
<td>Volume of nitrogen chamber</td>
<td>m(^3)</td>
</tr>
<tr>
<td>P_{0}</td>
<td>Initial nitrogen pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>V_{0}</td>
<td>Initial nitrogen chamber volume</td>
<td>m(^3)</td>
</tr>
<tr>
<td>n</td>
<td>Polytropic index</td>
<td>–</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
<td>------</td>
</tr>
<tr>
<td>Q</td>
<td>Volume flow rate</td>
<td>m³ s⁻¹</td>
</tr>
<tr>
<td>K</td>
<td>Piston position</td>
<td>m</td>
</tr>
<tr>
<td>K</td>
<td>Piston speed</td>
<td>m s⁻¹</td>
</tr>
<tr>
<td>Aₚ</td>
<td>Piston area</td>
<td>m²</td>
</tr>
<tr>
<td>Aₒ</td>
<td>Orifice area</td>
<td>m²</td>
</tr>
<tr>
<td>Cₜ</td>
<td>Discharge coefficient</td>
<td>–</td>
</tr>
<tr>
<td>ρ</td>
<td>Density</td>
<td>kg m⁻³</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

This chapter starts of with an introduction to the thesis. First the background to the problem formulation is discussed then the problem description is given and defined. Further the purpose of the thesis is described and what the goals of the project are. Last the delimitations of the thesis are presented and the ethical considerations are mentioned.

1.1 Background

When designing tanks many factors need to be taken into consideration. More often than not the tank will need to operate under varying circumstances and hence be able to work well in multiple different scenarios. Urban areas can in some cases facilitate the movement of a tank where good roads are available, but can also be limiting if the tank will not be able to drive up narrow alleys and have a limited line of sight. It is not uncommon for tanks to be deployed in multiple different parts of the world with a large variations in terrain and temperature. Even locally may the circumstances change rapidly as for example heavy rain may drastically change soil properties of the ground. In order to utilize the tank as effectively as possible means using the terrain to an advantage, taking routes that gives added protection from the enemy. This demands a good understanding of the tanks ability and limitations in a large span of different terrains and also how to maneuver the terrain given the conditions that are present.

The development of the MBT is affected by the changes in operational environment, the changing threats and the development of technology. Tanks will continue to face more advanced threats in the future and hence will need to be updated and improved in order to continue to be the best vehicle that the military has in its arsenal. The next generation of tanks are currently in design and with more advanced features than ever before. But in order to do the correct design decisions a proper understanding of the fundamentals of the tank is necessary.
Overall are there three major areas to consider when studying tanks. The tanks protection, its mobility and its firepower, the sum of which can be used to estimate the tank’s ability to accomplish missions in given scenarios [1].

This thesis will primarily focus on the mobility of the tank. How to model and evaluate the mobility of the tank in order to get a better understanding of the dynamic behaviour of the tank and also its interaction with the environment. The thesis will also investigate how the mobility can affect the firing precision of the tank.

The models that are to be developed are on system level and therefore the models do not need to be modelled in every detail. A large part of factors that contribute to the dynamics of the tank on detail level will not noticeably change the general behaviour of the model. To what degree of detail the model will need to be in order to predict the movement of the tank is to be investigated in the thesis.

1.2 Problem formulation

To what degree of simplification can a model of a MBT reasonably accurately predict the movement and firing accuracy of a main battle tank in a range of driving scenarios?

1.3 Purpose and goal

The purpose of this thesis is to gain understanding of an MBT’s dynamic behaviour in running condition. To investigate how the ground that the tank drives on affects the movement of the hull and the barrel of the tank in order to be able to evaluate the tanks performance.

The goal is to create a dynamic system model of a modern tank in MATLAB/SIMULINK that given a road profile and a speed will approximate the movement of the tank’s hull, i.e chassis, turret and barrel. Given this, the firing accuracy of the tank while driving will be investigated.

1.4 Delimitations and ethics

The ground will be considered as a rigid non-moveable object. The model can be further developed with the addition of terramechanics but this will not be considered. Further, the model will be a simplified model of the real tank hence details of the tank that can be considered as not relevant or not giving a substantial impact on the results will be neglected as this is a part of the goals of the thesis.

The tank is a military vehicle and hence a lot of ethical discussion could be done regarding the warfare part but this is an entirely different subject in itself. Since
this thesis is done only working with a model of a tank the ethical aspects of war are not considered in this report. The intent of the model is only to evaluate the tank’s mobility performance and the results from the model will only be used to predict outcomes of the mobility of the tank and its accuracy. As a parable a similar model could be used to predict the movement and vibrations of a camera mounted on a frame on a car or other vehicle.
Chapter 2

Background and theory

A MBT is a tracked vehicle made for the purpose of combat. The purpose of the tank is to have the best protection, the best mobility and the best firepower possible. The targets are most often ground targets or other tanks. The tank consist of the chassis which is housing the drive-train and most of the crew, the turret which is placed on top of the chassis which can be either manned or unmanned and the cannon which is attached to the turret. In order for the tank to have the best protection the tank is fitted with heavy armour and active anti tank countermeasures. The best mobility is given by the very powerful engine upwards of 1500 horsepower that the tank is equipped with giving the tank a high top-speed and the ability to traverse difficult terrain. The best firepower is given by its main armament, which is a cannon of around 120 mm in diameter. The combination of all these factors makes the tank a very fearsome vehicle.

The chapter starts with providing some background on last generations damper systems will be presented since the topic in focus is the mobility of the tank. Thereafter the mathematics of the half vehicle are introduced. Staying in the half-plane the half vehicle model is then extended to more degrees of freedom (DoF) to investigate the tracks added DoF influence. This is followed by introducing mathematical representation of the full vehicle model. Given the theory of the movement of the full tank the dampers are investigated further and the nonlinear hydrogassystem (HGS) is presented, which gives the possibility to improve the ride dynamics of the vehicle. Proceeding, the parts affected by the dynamics of the tank’s hull is investigated. This includes the motion of the turret and barrel assembly. Last the contribution of firing of the tanks cannon is shown and how the tanks firing accuracy is affected by the motion of the tank.
2.1 Dynamics of tracked vehicles

The design of the suspension will affect the mobility of the tank. The main purpose of the suspension system on a tank is to give the crew of the tank a tolerable ride minimizing fatigue. A proper tuned suspension will also give minimal vibrations, which gives the added benefit of needing less control input on order to stabilize the turret in the move, improving the accuracy of the cannon in a dynamic scenario. In worst case scenario a tank that is unable to move is an easy target and a unwanted occurrence, as the tank will need support and rescue from other units to get moving again. Hence a well working and well tuned suspension system is of great importance for the tanks mobility and ultimately its survivability [2].

Tanks are very heavy vehicles overall. They can weigh upwards of 70 tons and above. This is largely due to the tank’s armour that needs to be very thick in order to protect the vehicle from threats. Adding to the weight of the tank is also the main armament of the tank which, in itself can weigh multiple tons. This also means that the tank needs a heavy and powerful engine to give the tank fast acceleration and have the ability to climb steep gradients. The suspension and track layout of tanks are usually different in configuration compared to other tracked vehicles such as in agriculture since the tank needs to be able to run at higher speeds over rough ground. The engine power and weight also greatly affect the overall agility of the tank [3].

Looking back at the last generation of tanks, the most commonly used a torsion-bar suspension. The torsion-bar is a very simple type of suspension where a long bar of metal creates a spring force through a twisting motion of the bar. The torsion-bar acts as the spring for the running wheels in contact with the tracks. On the tank, the first and last pair of wheels are fitted with a damper in order to improve the ride but the wheels in between only have the torsion bar and hence no specific damper in order to smooth the motion. Because the tanks substantial weight, the torsion-bar needs to be quite large in order to accommodate the large forces that a tank would experience during manoeuvring. The construction would protrude though the hull of the vehicle taking up a substantial amount of space. Figure 2.1 gives a visual representation of how the torsion-bar suspension is fitted to the chassis of the tank. The benefits of running a torsion-bar suspension is the simplicity of the system and the absence of maintenance requirements since there are very few moving parts in the construction.
Figure 2.1: Visual representation of the torsion-bar suspension implementation in the chassis of the tank.

However, with increasing demands on the ability to go faster over rough terrain and have overall better mobility the trend is to go towards more advanced suspension systems such as a hydro-gas suspension. This because they tend to be lighter, more compact, improve ride quality for the crew and be more versatile [4].

2.1.1 Mathematical representation of half-plane suspension excluding track

The simplest way of modelling the motion of a vehicle is the Quarter-car model. This model represents the vehicle by looking at one quarter of the vehicle i.e one wheel of a car. This model has very few degrees of freedom and will not give much information about the motion of the tank as only the bounce motion can be observed. To start as simple as possible with representing the motion i.e. vertical displacement of the tank the half-car model is used. The half-car model effectively cuts the vehicle in half and observes one side of the vehicle in a 2D plane as can be seen in figure 2.3. The torsion bar suspension from the last generation of tanks produces a force from the twisting motion of the bar as formerly described. The force from the twisting motion is not precisely linear but quite close [5]. The tank’s suspension system can be modelled as a combination of masses, springs and dampers. The motion of a, simplified in plane, traditional suspension can be described mathematically by Lagrange equations [5]. As a simplification the tracks contribution has been neglected. Figure 2.2 shows the notations of the respective rotation roll, pitch and yaw around the axes, bounce is denoted Z and is a displacement.
Figure 2.2: The respective axes and the rotation notation.

\[
\frac{d}{dt} \left( \frac{\delta E_k}{\delta \dot{q}} \right) - \frac{\delta E_k}{\delta q} + \frac{\delta E_d}{\delta \dot{q}} + \frac{\delta E_p}{\delta q} = F(t) \quad (2.1)
\]

Where the respective terms are the following: \( E_k \) - kinetic energy, \( E_p \) - potential energy, \( E_d \) - dissipation energy, \( q \) - generalized coordinates, \( F(t) \) - generalized forces depending on time.

The equations of motion are derived using Lagrange equation and the result is a second order linear differential equation as follows.

\[
m\ddot{z} + c\dot{z} + kz = F(t) \quad (2.2)
\]

Where the constants are: \( m \) - mass, \( c \) - damping coefficient, \( k \) - spring stiffness.

As the suspension system of a tracked vehicle is a combination of many wheel-pairs the equation for the motion of the hull can be written as the sum of all forces acting on the hull of the vehicle. The mathematical description of the motion can be written as equation 2.3 and equation 2.4 for bounce and pitch motion of the hull. The respective variables and units can be seen in Table 0.1.
The vertical deviation for the hull is described by equation 2.3.

$$m \ddot{z} + \sum_{i=1}^{N} k_d(z + x_i\varphi - z_{ti}) + \sum_{i=1}^{N} c_d(\dot{z} + x_i\dot{\varphi} - \dot{z}_{ti}) = 0 \quad (2.3)$$

The pitch motion of the hull can in a similar way be described with equation 2.4

$$J_{yy}\ddot{\varphi} + \sum_{i=1}^{N} k_d x_i(z + x_i\varphi - z_{ti}) + \sum_{i=1}^{N} c_d x_i(\dot{z} + x_i\dot{\varphi} - \dot{z}_{ti}) = 0 \quad (2.4)$$

The equation of motion for each independent wheel is described with equation 2.5.

$$m_{ti}\ddot{z}_{ti} - k_d(z + x_i\varphi - z_{ti}) - c_d(\dot{z} + x_i\dot{\varphi} - \dot{z}_{ti}) + k_t(z_{ti} - z_g) = 0 \quad (2.5)$$

Where: $z$ - Vertical displacement of the hull, $\dot{z}$ - Vertical speed of the hull, $\ddot{z}$ - Vertical acceleration of the hull, $z_{ti}$ - Vertical displacement of respective tyre, $z_g$ - Vertical displacement of ground, $\varphi$ - Pitch angle of the hull, $\dot{\varphi}$ - Pitch speed of the hull, $\ddot{\varphi}$ - Pitch acceleration of the hull, $x_i$ - Wheel position wrt CoG, $m$ - Mass of the vehicle, $m_{ti}$ - Mass of the wheel, $J_{yy}$ - Moment of inertia, $J_{xy}$ - Moment of deviation, $i$ - Index, $N$ - Quantity of wheels/track components, $k_d$ - Spring stiffness of main spring, $k_t$ - Spring stiffness of wheel to track, $c_d$ - Damping coefficient main damper, $c_t$ - Damping coefficient of wheel.

A visual representation of the simplified model can be seen in figure 2.3. The figure gives a general representation of the system, a damper is represented on each wheel but this might not always be the case depending on how the suspension is configured.

![Figure 2.3: Visual representation of the simplified half-vehicle model.](image-url)
2.1.2 Mathematical representation of half-plane suspension including track

In order to understand how the contribution of the tracks affects the dynamics of the tank the tracks are included into the model. Because the track is not behaving linearly along the length of the track some different methods of modelling the track’s contribution have been made. One way of modelling the track is to introduce an imaginary running-wheel between the actual physical wheels of the tank as done by [6]. According to the author this seems to be a viable way of modelling the tracks but the results seems to deviate somewhat from the actual measured data in the report.

In this thesis the method for modelling track involves the vertical movement of the track and hence the horizontal stiffness is neglected. The equations of motion for the half-plane model with the track component included are derived in equations 2.6 to 2.9 where each track component is represented by a mass, spring and damper.

The equations for bounce and pitch seen in equations 2.6 and 2.7 are very similar to the simplified model.

\[
m\ddot{z} + \sum_{i=1}^{N} k_d(z - z_{ti} + \varphi x_i) + \sum_{i=1}^{N} c_d(\dot{z} - \dot{z}_{ti} + \dot{\varphi} x_i) = 0 \quad (2.6)
\]

\[
J\ddot{\phi} + \sum_{i=1}^{N} k_d x_i(z - z_{ti} + \varphi x_i) + \sum_{i=1}^{N} c_d x_i(\dot{z} - \dot{z}_{ti} + \dot{\varphi} x_i) = 0 \quad (2.7)
\]

The equation of motion for each individual wheel can be seen in equation 2.8.

\[
m_{ti}\ddot{z}_{ti} = k_d(z + x_i\varphi - z_{ti}) + c_d(\dot{z} + x_i\dot{\varphi} - \dot{z}_{ti}) - k_l(z_{ti} - z_t) - c_l(\dot{z}_{ti} - \dot{z}_t) \quad (2.8)
\]

And last the equation of motion for the track component can be seen in 2.9.

\[
m_{li}\ddot{z}_{li} = k_l(z_{li} - z_l) + c_l(\dot{z}_{li} - \dot{z}_l) - k_l(z_{gi} - z_{li}) - c_l(\dot{z}_{gi} - \dot{z}_l) \quad (2.9)
\]

Where: \( z_{ti} \) - Vertical displacement of respective track component, \( m_{ti} \) - Mass of the track component, \( k_l \) - Spring stiffness of track to ground, \( c_l \) - Damping coefficient track to ground.

A general visual representation of the model can be seen in figure 2.4.
2.1.3 Mathematical representation full vehicle suspension

The half-plane model only accommodates the bounce and pitch motion of the hull but since the goal is to simulate the firing precision a full vehicle model is necessary in order to be able to include the horizontal deviation due to roll and yaw. In the same way that the Lagrange equation is used in equation 2.1 to describe the in-plane motion of the tank, the same equation can be used in order to describe the full vehicle motion [7].

Applying equation 2.1 to the full vehicle it can be seen that.

\[
E_k = \frac{1}{2} \left( m\dot{z}^2 + J_y k_2 \dot{\phi}^2 + J_z k_2 \dot{\gamma}^2 - 2 J_{xy} \dot{\phi} \dot{\gamma} \right) \tag{2.10}
\]

\[
E_p = \frac{1}{2} \sum_{i=1}^{N} k_i (z - x_i \dot{\phi} + y_i \dot{\gamma} - z_i)^2 \tag{2.11}
\]

\[
E_d = \frac{1}{2} \sum_{i=1}^{N} c_i (\dot{z} - x_i \ddot{\phi} + y_i \ddot{\gamma} - \dot{z}_i)^2 \tag{2.12}
\]

Given the above equations the full vehicle model can be derived as the following:

\[
M \ddot{z} + C \dot{z} + Kz = Fw \tag{2.13}
\]
Where: \(M\) - inertia matrix, \(C\) - damping matrix, \(K\) - stiffness matrix, \(F\) - force matrix, \(\ddot{z}\) - acceleration vector, \(\dot{z}\) - velocity vector, \(z\) - displacement vector.

\[
\ddot{z} = \begin{bmatrix} \ddot{\varphi} \\ \ddot{\gamma} \end{bmatrix}, \quad \dot{z} = \begin{bmatrix} \dot{\varphi} \\ \dot{\gamma} \end{bmatrix}, \quad z = \begin{bmatrix} \varphi \\ \gamma \end{bmatrix}, \quad w = \begin{bmatrix} \dot{z}_i \\ \ddot{z}_i \end{bmatrix}
\] (2.14)

\[
M = \begin{bmatrix} m & 0 & 0 \\ 0 & J_{yy} & -J_{xy} \\ 0 & J_{xy} & J_{xx} \end{bmatrix}
\] (2.15)

\[
C = \begin{bmatrix} \sum_1^n C_i - \sum_1^n C_i x_i & -\sum_1^n C_i y_i \\ -\sum_1^n C_i x_i & \sum_1^n C_i (x_i)^2 - \sum_1^n C_i x_i y_i \\ \sum_1^n C_i y_i & -\sum_1^n C_i x_i y_i & \sum_1^n C_i (y_i)^2 \end{bmatrix}
\] (2.16)

\[
K = \begin{bmatrix} \sum_1^n K_i & -\sum_1^n K_i x_i & \sum_1^n K_i y_i \\ -\sum_1^n K_i x_i & \sum_1^n K_i (x_i)^2 - \sum_1^n K_i x_i y_i \\ \sum_1^n K_i y_i & -\sum_1^n K_i x_i y_i & \sum_1^n K_i (y_i)^2 \end{bmatrix}
\] (2.17)

\[
F = \begin{bmatrix} 0 & \sum_1^n C_i & \sum_1^n K_i \\ 0 & -\sum_1^n C_i x_i & -\sum_1^n K_i x_i \\ 0 & \sum_1^n C_i y_i & \sum_1^n K_i y_i \end{bmatrix}
\] (2.18)

And: \(y_i\) - Wheel position wrt CoG (width), \(J_{xx}\) Moment of inertia.

This can then be expanded for the respective equations of motion for bounce, pitch, and roll. A simplified version of the full vehicle model is represented in figure 2.5 with all the respective axes.
2.1.4 Hydro-gas suspension

In order to improve the ride comfort and the motion of the tank, a so called hydro-gas suspension system (HGS), seen in figure 2.6, can be implemented. The HGS is similar to an advanced shock absorber on a road going vehicle in that it has adjustable high and low speed compression and rebound. But instead of a traditional coil spring, the system relies on compression of a gas to act as the spring element. Also, the suspension system is more compact than the conventional torsion-bar suspension and the HGS units can be mounted outside of the hull freeing up space in the hull. As each independent wheel will be fitted with an HGS module the tank receives full damping on each independent wheel. The spring element of the HGS system have lower spring rate at lower wheel displacement and stiffer spring rate at high displacement giving a better ride for the crew and preventing hitting a hard stop at maximum travel in compression [8]. The HGS also has the possibility to give the vehicle road levelling capabilities [4]. With the possibility to adjust the level of each individual wheel height comes also the ability to adjust the tension in the track as well as the ground clearance of the vehicle. The tension in the track has been shown to have a significant effect on the vehicles mobility over soft soil [9].
The unit is constructed of a nitrogen chamber with a given preset pressure and volume shown as the top part in figure 2.7. The nitrogen chamber is connected to a fluid chamber, with a floating piston in between. The fluid chamber has a small orifice that is restricting the flow of fluid from below to above. When the wheel of the tank moves the lower piston moves displacing the fluid forcing it to flow through the orifice to the other chamber where the floating piston is located and compresses the nitrogen gas. In order to control the flow and have the pressure due to the movement of the wheel not exceed to high pressures, a bleed valve is located parallel to the orifice. The bleed valve can be designed in different ways but the main purpose is to allow increased flow between the chambers regulating the pressure differential.

The principle of HGS unit is working on can be seen in figure 2.7.
There are two parts adding to the force of the suspension unit. The first part acts as a spring when the nitrogen chamber is compressed. The second part is the viscous damping given by the fluid passing through the orifice and the bleed valve. This gives the non linear spring-damper action and with the adjustable bleed valve the characteristics of the damper can be changed in order to get the best performance [10].

The following equations explain the relation between the motion of the piston and the reacting force of the HGS unit. Starting of with the compression of the nitrogen gas. The gas of the chamber is considered to be an ideal gas undergoing an polytropic process hence the relation between the pressure and volume is given by equation 2.19 [11].

The process of compressing the nitrogen gas is assumed to be an adiabatic process meaning that no heat exchange is occurring since the process of compression and expansion is quite rapid. The adiabatic index $n$ is depending on temperature and pressure. From [11, 4] the adiabatic index can be seen as in proximity of 1.3 to 1.4 in temperatures around 20 degrees Celsius and pressures below 200 bar. $P_i$ represents the instantaneous pressure of the nitrogen chamber and $V_i$ is the instantaneous volume of the gas and $P_0$ and $V_0$ is the initial pressure and volume.

$$ P_0 V_0^n = P_i V_i^n \Rightarrow P_i = \frac{P_0 V_0^n}{V_i^n} \quad (2.19) $$

The instantaneous volume of the nitrogen chamber is only depending on the dis-
placed volume of the piston, assuming that the fluid is incompressible. Therefore the volume of the nitrogen chamber can be written as equation 2.20 where $K$ is the position of the lower piston and $A_p$ is the piston area.

$$V_i = V_0 - A_p K$$  \hspace{1cm} (2.20)

The volumetric flow rate $Q$ though the orifice can be seen in equation 2.21 and originates from Bernoulli’s equation. $C_d$ is the discharge coefficient and describes the losses in the orifice plate. The value for the coefficient is usually between 0.6 and 0.85 and is affected by geometry of the orifice [12]. $A_o$ is the orifice area, $\rho$ is the density of the oil and $\Delta P$ is the pressure difference between the oil chambers.

$$Q = C_d A_o \sqrt{\frac{2\Delta P}{\rho}}$$  \hspace{1cm} (2.21)

Since the only component that is inducing any flow though the orifice is the movement of the piston the flow rate of can also be written as a function of the piston speed.

$$Q = A_p \dot{K}$$  \hspace{1cm} (2.22)

Solving for the pressure differential between the two chambers gives the following expression.

$$(2.21) \& (2.22) \Rightarrow \Delta P = \frac{\rho A_p^2 \dot{K}^2}{2C_d^2 A_o^2}$$  \hspace{1cm} (2.23)

Last, the two component of force can be derived. The spring force from the nitrogen compression denoted $F_s$ and the damper force from the pressure difference denoted $F_d$.

$$F_s = P_i A_p$$  \hspace{1cm} (2.24)

$$F_d = A_p \Delta P$$  \hspace{1cm} (2.25)

Each wheel of the tank is fitted with an individual HGS unit. The force from equation 2.24 and 2.25 is from a single unit and hence the total force from the combined system with multiple units will be the sum of all the forces that the units produce.

### 2.2 Firing of cannon

The tank must be able to fire its cannon while moving. In order to make this possible the cannon will need to be controlled and stabilized in order to be able to aim at a given target. The control part will not considered in the thesis. The firing of the cannon will impose a large force on the turret and the hull of the tank during a short timespan and will affect the entire vehicle. The magnitude and duration of the
force from the cannon’s recoil is depending on the bore of the cannon and the type of ammunition used. The orientation will also affect how the force is transmitted to the vehicle. As the firing of the tank’s cannon is a common procedure the recoil’s contribution to the dynamics of the tank’s hull will be investigated.

### 2.2.1 Firing sequence from a force perspective

The main armament of modern tank is usually a cannon with a calibre of approximately 120 mm. The force experienced from the recoil of the cannon is large enough to affect the entire vehicle. Because of this, tanks are fitted with a recoil mechanism in order to relieve some of the force from the hull and insure that the large impulse of the cannon will not permanently damage the vehicle. The recoil mechanism works by enabling the cannon to move in its axial direction. When the cannon is fired it moves backwards and compresses either a set of springs or containers of air to absorb some of the force [13].

The firing sequence is a very complex process to model in detail. The general behaviour of the recoil force is as follows. The cannon is fired and the projectile leaves the barrel, during that time the tank experiences a large increase in force, which decays in about the same time as the force was generated. The entire event is over within around one tenth of a second. As the purpose of the thesis is to investigate the tank’s dynamics on a system level, a simplified model of the recoil force is utilized. Vallier’s formula is used for calculating the recoil and is valid under the assumption that the force from the recoil acting on the vehicle is a rectangular impulse of a set time [13].

\[
R = \frac{0.5M_0U_{max}^2}{L_{max} - L_k + U_{max}t_p}
\]  
(2.26)

\[
U_{max} = \frac{M_p + \beta m_m}{M_0}U_0
\]  
(2.27)

\[
\beta = \left(\frac{700 + U_0}{U_0}\right)^{1.1}
\]  
(2.28)

Where: R - recoil force resistance, \(U_{max}\) - maximum velocity of the free recoil, \(L_{max}\) - maximum recoil path, \(L_k\) – free recoil path (at \(R = 0\)), \(t_p\) – duration of gunpowder gases exhaust activity, \(M_p\) – projectile mass, \(U_0\) – initial velocity of the projectile, \(m_m\) – mass of propellant charge, \(M_0\) – mass of recoiling assembly, \(\beta\) – activity of gunpowder gases factor.

With data from the a given cannon and its ammunition a recoil force can be calculated. A few examples of force and recoil time can be seen in Table 2.1.
Table 2.1: Tank main armament recoil force [13, 14].

<table>
<thead>
<tr>
<th>Cannon</th>
<th>Recoil force [kN]</th>
<th>Duration [ms]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2A46, 125 mm</td>
<td>524</td>
<td>56</td>
</tr>
<tr>
<td>L44, 120 mm</td>
<td>411</td>
<td>58</td>
</tr>
<tr>
<td>L7A3 105 mm</td>
<td>460</td>
<td>43</td>
</tr>
<tr>
<td>2A28, 73 mm</td>
<td>126</td>
<td>123</td>
</tr>
</tbody>
</table>

The whole course of firing the cannon is over within milliseconds. The forces involved are extremely high as the projectile is often approximately 10 kg in mass and is accelerated during the length of the barrel, often below 8 meters, to a muzzle velocity of 700 to 1700 m s\(^{-1}\). In order to evaluate if the values for the recoil force are reasonable a simplified check is done. Assuming constant acceleration of the projectile, the acceleration to a muzzle velocity of 1 km s\(^{-1}\) over the course of a 8 m barrel, the acceleration would be 62 km s\(^{-2}\). With Newton’s second law, the force required to achieve this constant acceleration would be in the order of hundredths of kN with a projectile of above 1.6 kg which, would be much lower than the mass of the projectiles seen normally. And according to Newton’s third law every action there is an equal and opposite reaction, the tank would experience a recoil force of the roughly same magnitude.

### 2.2.2 Accuracy depending on barrel movement

The accuracy of the tank’s cannon is influenced by the movement of the entire tank. That is the performance of the firing control system and the stabilization of the barrel among other things. If the control of the tank’s cannon is neglected and the cannon is stationary relative to the turret and hull, then the accuracy of the tank will mainly be influenced by the motion depending on the surface the tank is running on. The pure movement at the muzzle is a good indicator of what performance the control system will need in order to stabilize the barrel.

The vertical displacement from the target aim point is given by the following equations [15]. A visual representation can be seen in figure 2.8.

\[
Z + H \leq Y \tag{2.29}
\]

Where: \(Y\) - Half height of target, \(Z\) - contribution of vertical deviation at target area distance \(L\) away from the cannon. \(Z\) is given at the time the shell leaves the barrel and is affected by the entire movement of the tank, i.e pitch motion of the vehicle. \(H\) - the deviation contribution at the target, which is given by the movement of the barrel itself.

\[
Z = \varphi_{uz}L + \dot{\varphi}_{uz}l_z \frac{L}{V_0} \tag{2.30}
\]
\[ H = \varphi_{uh}L + \varphi_{uh}l_h \frac{L}{V_0} \]  \hfill (2.31)

Where: \( \varphi_{uz} \) - angle deviation of the tank, \( \dot{\varphi}_{uz} \) - angular velocity of the tank, \( L \) - direct distance to target, \( l_z \) - length between the cannons CoG and the muzzle, \( V_0 \) - projectile muzzle velocity, \( \varphi_{uh} \) - angle deviation of cannon, \( \dot{\varphi}_{uh} \) - angular velocity of cannon, \( l_h \) - length between the recoiling parts’ CoG and the muzzle.

Given the displacement of the projectile impact-point to the original aim-point on the target is it possible to use this information to calculate the probability of hitting the target given the above mentioned assumptions.

Figure 2.8: A visual representation of firing accuracy.
Chapter 3

Modelling and simulations

This chapter will go through the different models and its implementations. Due to difficulties obtaining open research data for model validation the model is developed aiming at producing comparable behaviour to the available research data [4] [8] [10]. One of the goals when designing the suspension system is to have minimal vibrations transferred to the hull and hence this goal will be taken into consideration when comparing the models.

3.1 Half tank model

3.1.1 9 Degrees of freedom

The first model to be designed is the simplified model of the tank. The tank is represented by a half car model without the track link. The hull is represented as a mass with two degrees of freedom, and the track and running wheels of the tank are also represented as a mass with one degree of freedom each. The model is constructed in SIMULINK and consists of simple blocks such as gain and integrators.

The road is considered as stiff and the reason for this is that a road profile, considering road damping or terramechanics, is adding a damping factor to the motion of the tank and could hence give the model a better performance than if the road is stiff, since all of the vibration from the ground are transferred into the hull of the tank.

Some different variations of road profile are considered. First a step impulse of 100 millimeter in order to check that the model is settling in a correct way and that the response of the hull is behaving in a reasonable manner. Further a periodic road with amplitude of 60 millimeter and a few different frequencies. Finally a vibration course seen in figure 3.1, with is represented of bumps of varying size and distance is tested in order to compare the performance of the models in a mobility challenging scenario more like the real driving conditions that the tank will encounter. The
track is roughly 150 m long with bumps in sizes of 0.3, 0.15, 0.12 and 0.075 m with a length of 7.6, 6.1, 0.9 and 0.6 m.

![Vibration Course profile](image)

Figure 3.1: The vibration course track profile.

The parameters used for the tank are presented in Table 3.1. The tank is equipped with seven running-wheels with a wheelbase of 5.7 m and equally spaced wheels.

Table 3.1: Half vehicle 9 DOF parameters.

<table>
<thead>
<tr>
<th>m</th>
<th>25000 kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>m_t</td>
<td>875 kg</td>
</tr>
<tr>
<td>J_{yy}</td>
<td>200000 kg m^2</td>
</tr>
<tr>
<td>k_d</td>
<td>280 kN m$^{-1}$</td>
</tr>
<tr>
<td>c_d</td>
<td>39.2 kN s m$^{-1}$</td>
</tr>
<tr>
<td>k_t</td>
<td>613 kN m$^{-1}$</td>
</tr>
</tbody>
</table>

The reasoning for choosing the variables in table 3.1 comes from a comparison of the first un-damped eigenfrequency of the sprung mass. Since no data on MBT models where found where data from infantry fighting vehicles (IFV) used as it is available in public articles [4] [6][10]. Knowing the mass of the vehicle and the spring constant of the model one can calculate the first undamped eigenfrequency of the model with the following equation.

$$\omega = \sqrt{\frac{k}{m}} \quad (3.1)$$

By calculating the first frequency from other models in articles the spring constant could be determined for a tank model such that the first eigenfrequency would be within proximity of the article models. The relation between spring stiffness and damping coefficient was also checked and determined for the model in a similar way.

The stiffness of the wheels was also chosen with data from the articles. The reasoning for the chosen value where that the stiffness is the metal wheel should be very stiff and the damping is more or less absent.
3.1.2 16 Degrees of freedom

To investigate the influence of adding more degrees of freedom on the tank model, the tank tracks are put into the model. The tracks are represented in the same way as the wheels with a mass and with one degree of freedom each. This simplification does not take into account the tracks horizontal impact on the dynamics but will give valuable information regarding if the extra degrees of freedom will add any essential information before developing the full vehicle model. The parameters used in the extended half vehicle model can be seen in Table 3.2.

Table 3.2: Half vehicle sixteen DOF parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m$</td>
<td>25000 kg</td>
</tr>
<tr>
<td>$m_t$</td>
<td>875 kg</td>
</tr>
<tr>
<td>$m_l$</td>
<td>285 kg</td>
</tr>
<tr>
<td>$J_{yy}$</td>
<td>200000 kg m$^2$</td>
</tr>
<tr>
<td>$k_d$</td>
<td>280 kN m$^{-1}$</td>
</tr>
<tr>
<td>$c_d$</td>
<td>39.2 kN s m$^{-1}$</td>
</tr>
<tr>
<td>$k_l$</td>
<td>613 kN m$^{-1}$</td>
</tr>
<tr>
<td>$c_l$</td>
<td>2 kN s m$^{-1}$</td>
</tr>
<tr>
<td>$k_l$</td>
<td>8000 kN m$^{-1}$</td>
</tr>
<tr>
<td>$c_l$</td>
<td>0 kN s m$^{-1}$</td>
</tr>
</tbody>
</table>

The same parameters are used for the main springs in the 16 DoF model as the 9 DoF model. The tracks are given a value for stiffness of at least ten times higher than the wheels because the tracks are mainly composed of steel and hard rubber plates and would most likely not deflect at all. The wheels are given a small damping part in an attempt to include some damping that might be present in the wheel assembly. Both models are simulated with a road profile step of 100 mm denoted test 1 as well as a periodic road profile of 60 mm and frequency of 1 Hz denoted test 2 as seen in figures 3.2 to 3.4. It can be observed that more frequencies are present at the tracks but that there is no major impact on the dynamics of the hull. This result is to be expected as the added degrees of freedom will result in further detail in the motion of the wheels but the lower frequencies corresponding to the motion of ground will be dominating and the higher frequencies will not add to the motion of the hull.
Figure 3.2: Test 1, comparing the vertical deviation of each of the tank’s 7 wheels between the 9 and the 16 DoF model.

Figure 3.3: Test 1, comparison of the hull vertical displacement and the pitch angle between the 9 and the 16 DoF model.
Taking into consideration the complexity that is added to the model with increasing the degrees of freedom, in combination with small influence that the extra degrees of freedom gives the model. Proceeding, the full vehicle model will not include the track’s contribution.

### 3.1.3 HGS model

The model for the gas damper is implemented and tested in SIMULINK to replace the linear spring damper in the model. The model consists of two parts. The pressure chamber corresponding to the spring and the oil chamber with the small channel corresponding to the damper.

The orifice connecting the two chambers is also fitted with a bypass valve in order to regulate the pressure difference. The valve is opened when the speed is reaching a preset value corresponding to the pressure differential opening the valve, which is forced shut by a preloaded spring. The force form the HGS in relation to the displacement of the wheel is show in figure 3.5 and the damping component related to the speed of the wheels is shown in figure 3.6 where positive speed is corresponding to compression. In the physical HGS system the wheel and the piston are connected on a pivoting leaver. The piston and the arm connecting to the wheel have different lengths and hence there will be a ratio between the movements. The non-linearity between the motion of the piston and the wheel can vary greatly depending on the
geometry of the HGS system. Attempting to keep the model as simple as possible as well as the lack of data to confirm the model against, the relation between the piston and wheel was set to a constant value of 0.2 corresponding to having the wheels rotating on a leaver of 750 mm length and the piston on a leaver of 150 mm. The arm lengths of 500 mm and 137 mm is seen in [8] and [10] corresponding to a ratio of 0.25 but as the geometric relation will give a reduction in force the slightly smaller ratio selected.

![Spring force in relation to wheel displacement.](image)

Two models of the dampers are implemented. The first model seen in figure 3.6 is based on an increase of the area that the fluid can flow through after a set value. The area will increase linearly in relation to the speed of the piston up to a set maximum combined opening of the canal.
As can be seen in figure 3.6 the bypass valve is opening on the compression and 
rebound stroke taking down the force from the damper by increasing the area that 
the oil can flow through.

The second model of the damper is based on the same force build-up in the low 
speed region. The high speed on the other hand is set to be linear in relation to the 
piston speed. The result of the linearised damper can be seen in figure 3.7
3.2 Full vehicle model

The half car model is expanded to a full vehicle model in order to accommodate more degrees of freedom of the tank and give a more complete picture of the movement. The second pair of wheels are added and with this the roll axle is also introduced. The second pair of tracks is given its own road profile input that can be given a time delay. This gives the possibility to have the tracks see the same road profile and hence little to no roll should be present, or have the tracks experience completely different profiles depending on what kind of excitation one want to test. The tank has the same placement of the wheels in longitudinal direction and the respective wheel-set are located 1.525 m from the CoG. The additional data used for the full vehicle model can be seen in table 3.3
Table 3.3: Full vehicle model parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>53370 kg</td>
</tr>
<tr>
<td>m_t</td>
<td>875 kg</td>
</tr>
<tr>
<td>m_I</td>
<td>285 kg</td>
</tr>
<tr>
<td>J_{yy}</td>
<td>200130 kg m^2</td>
</tr>
<tr>
<td>J_{xx}</td>
<td>224093 kg m^2</td>
</tr>
<tr>
<td>k_d</td>
<td>280 kN m^{-1}</td>
</tr>
<tr>
<td>c_d</td>
<td>39.2 kN s m^{-1}</td>
</tr>
<tr>
<td>k_t</td>
<td>613 kN m^{-1}</td>
</tr>
<tr>
<td>c_t</td>
<td>0 kN s m^{-1}</td>
</tr>
</tbody>
</table>

The HGS dampers are introduced into a separate model and replace the linear springs and dampers. The equations of motion change as can be seen in equations 3.2 to 3.4 due to the fact that the force now comes from the HGS unit.

\[ m \ddot{z} = - \sum F_{HGS} \]  
\[ J_{yy} \ddot{\phi} = - \sum F_{HGS}(x_i \phi) \]  
\[ J_{xx} \ddot{\gamma} = - \sum F_{HGS}(y_i \gamma) \]

Where \( F_{HGS} \) is the reacting force from the HGS unit.

The value for the precharged pressure was set according to commonly used pressures of either 11.4 MPa or 13 MPa [10]. The volume for the nitrogen chamber is preferably as small as possible. There are some physical constraints that need to be taken into consideration that the model might not take into account. The volume of the nitrogen pressure chamber is in relation to the piston position and its area (see equation 2.20), hence the volume need to be large enough such that the volume can not be less than or equal to zero as a result of the movement of any wheel. Also the area of the orifice needs to be significantly smaller than the area of the piston.

Simulations were made for different combinations of areas and volume with low frequency and high displacement of the wheels as well as the opposite in order to verify that the force from the HGS model was producing combinations of force in relation to speed and distance. The resulting parameters used for the HGS model can be seen in table 3.4.
Table 3.4: HGS parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_0$</td>
<td>11.4 MPa</td>
</tr>
<tr>
<td>$V_0$</td>
<td>$30.4 \cdot 10^{-9}$ m$^3$</td>
</tr>
<tr>
<td>$A_p$</td>
<td>$1.3 \cdot 10^{-3}$ m$^2$</td>
</tr>
<tr>
<td>$A_0$</td>
<td>$2.8 \cdot 10^{-6}$ m$^2$</td>
</tr>
<tr>
<td>$\rho$</td>
<td>850 kg m$^{-3}$</td>
</tr>
<tr>
<td>$C_d$</td>
<td>0.73</td>
</tr>
<tr>
<td>$n$</td>
<td>1.35</td>
</tr>
</tbody>
</table>

The combined damper unit was simulated with a periodic ground excitation using two frequencies of 0.1 and 1 hertz and increasing amplitudes between 0.1 and 0.45 m in order to make sure that the simulation would produce creditable resulting forces as can be seen in figure 3.8 and 3.9. The behaviour from the 0.1 hertz test in figure 3.8 is corresponding with what is to be expected from the HGS unit. With the low frequency excitation of the ground, the force is dominated by the gas spring with a small contribution from the damper giving the exponential behaviour of the force in relation to distance. The force in relation to speed seems reasonable with peak force at maximum force at where the damper goes from compression to rebound i.e. at zero speed. The behaviour in the 1 hertz ground excitation seen in figure 3.9, shows the same behaviour as the 0.1 hertz but more force from the damper present due to the higher ground disturbance speed. The force in relation to speed have a slight tilt as to be expected with the larger contribution of force from the damper’s with the behaviour seen in 3.6.
Figure 3.8: HGS-unit force in relation to speed and distance for 0.1 Hz sinusoidal excitation with amplitudes between 0.1 and 0.45 m from equilibrium position.

Figure 3.9: HGS-unit force in relation to speed and distance for 1 Hz sinusoidal excitation with amplitudes between 0.1 and 0.45 m from equilibrium position.
3.3 Firing sequence implementation

The force from the recoil is calculated and implemented into the SIMULINK model. The force is set to act on the attachment point where the barrel pivots. The implementation of the force from the recoil can be seen in figure 3.10. The recoil force is calculated using data from a kinetic energy penetrator round fired from a 120 mm cannon and the time that the impulse is active for is set to 70 ms corresponding to the average impulse duration of the example ammunition seen in table 2.1.

![Figure 3.10: The force from the recoil acting on the barrel attachment-point.](image)

A visual representation of the implementation can be seen in figure 3.11.

![Figure 3.11: Visual representation of the recoil acting-point.](image)
The implementation is tested on the linear half vehicle model. Assuming the cannon is oriented straight forward the contribution from the recoil will exclusively affect the pitch axis of the tank and therefore the pitch axis is observed. The pitch axis will also be the main contributor to the firing accuracy as most of the cases when driving the pitch axis will have the largest deviations in movement. The pitch acceleration and angle can be seen in figure 3.12. As can be seen the impulse from the force of the recoil gives a high acceleration impulse in pitch, which decays quickly. The pitch angle is relatively small as can be expected because even though the force is very high the time the force is acting on the hull is low and hence the total energy transferred is low. Also the lever arm that the force from the recoil has corresponding to the distance between the attachment point of the barrel and the CoG of the tank is not very large. This means that the large force that is produced does not give as large of a torque rotating the hull.

![Figure 3.12: The hull pitch angle and pitch acceleration due to firing of the cannon.](image-url)
Chapter 4

Results and analysis

In this chapter the results from the modelling are presented. The different models are compared to each other and the results from running the simulated tank. The influence of the recoil while driving is investigated and the performance gains from introducing the HGS system are analysed. Also the possibility to simplify the HGS model is investigated. Lastly the relation between the movement of the tank and it’s line of sight is analysed.

4.1 Model comparison

Three standard scenarios are tested on the tank as follows.

- test 1: 100 mm step,
- test 2: 60 mm 1 Hz sinusoidal road excitation,
- test 3: vibration course.

All the above mentioned scenarios where simulated for multiple speeds. The speed shown in all the following figures 4.1 to 4.7b are 40 km h⁻¹. A small time delay of 0.1 s was added on the left set of tracks in order to excite roll behaviour. Fire accuracy was only observed on the vibration course.

4.1.1 Damper model performance comparison

The first test of 100 mm step was simulated with three models tested: The linear model, the full HGS system and a simplified HGS model where the high speed compression and rebound force was set to a linear behaviour. The result of the simulations can be seen in figures 4.1 to 4.6.
Figure 4.1: Comparison of the vertical deviation for test 1 between the linear model and the 2 HGS models.

As can be seen from the peaks are lower in both of the pneumatic models compared to the linear model both in bounce and pitch. A somewhat over-damped behaviour is also seen in the HGS system especially the linearized version as the overshoot of the step is more or less absent. Test 2 can be seen in the following figures 4.3.
to 4.4 showing the pitch angle and the vertical deviation for the periodic ground irregularity. An improvement in both vertical deviation and pitch angle can be seen for the HGS models in comparison to the linear model.

Figure 4.3: Comparison of the vertical deviation for test 2 between the linear model and the 2 HGS models.

Figure 4.4: Comparison of the pitch angle for test 1 between the linear model and the 2 HGS models.

Last, test 3, the vibration course can be seen in the following figures 4.5 to 4.6. In order to evaluate the impact that the recoil has on the tank the simulation is set to fire at 7.5 s as at this point the tank is on its way up one of the larger obstacles. The results are showing that in comparison to the other obstacles of the same size no major change in either pitch nor bounce can be observed. This indicates that
the suspension is handling the recoil very well since the firing of the cannon is not affecting the hull in any significant way.

Figure 4.5: Comparison of the vertical deviation for test 3 between the linear model and the 2 HGS models.

Figure 4.6: Comparison of the pitch angle for test 3 between the linear model and the 2 HGS models.

Another thing that can be observed from the vibration course is that the linearized HGS model deviates from the non-linearized HGS model during high frequency movement of the wheels but not during low frequency movement to the same degree. This is to be expected as the force produced from the HGS models deviates from each other in the higher speeds hence the observed outcome.
4.1.2 Firing accuracy

Translating the movement from the hull of the tank to the aim-point displacement is of interest. It will give an indication on what performance requirements that the control system for the barrel and turret will need to achieve.

The aim-point displacement is simulated both for the linear model as well as the HGS model in order to be able to compare the influence of the different damper characteristic on the deviation at the target. As the information from the test is to translate into performance requirements of the control system the test scenario used for generating the aim-point displacement is the vibration track. The reason for this is that the vibration track includes a wide variety of frequencies similar to the regular operation of the vehicle.

The model is given a small variation between the two tracks in order to excite some roll behaviour as well. The aim-point displacement for the linear model can be seen in figure 4.7a. The HGS model of the full vehicle is given the same input for the tracks. The aim-point displacement can be seen in figure 4.7b.

(a) The aim-point displacement of the linear model. (b) The aim-point displacement of the HGS model.

Figure 4.7: Aim-point displacement.

Observing the behaviour of the aim-point displacement of the tank it can be seen that the HGS system has a more composed behaviour compared to the linear model and more of the time is spent in the area of 10 by 10 meters from the centre point, in figure 4.7b shown as 0 m in x and 0 m in y. To draw conclusions from the behaviour of the aim-point displacement one could assume that the more composed behaviour of the HGS would indicate that a less advanced control system would be needed or less effort for a given control-system to control the barrel of the tank as the movement is more predictable.

Looking at the overall behaviour of the different models we see that using a non-linear damper model gives benefits in terms of performance. The peaks i.e. maximum dis-
tance from centre are in general lower which is desirable. Especially in the pitch axis the improvements are more noticeable with the peaks almost 15 m lower than the linear model, which is a great improvement.

The reason for the right leaning bias of the results are due to the left tracks always being small amount before the right tracks and hence the excitation of roll is more biased to the right.

The linearized HGS model is showing promising results. Observing the pitch axis in figure 4.6 one can see a very similar results between the HGS models and only the peaks differ a small amount. The major difference between the full HGS model and the linearized model can be seen in bounce at very high frequency ground excitation. This would most likely not be a very common occurrence for the real system and hence the simplified model can be used in most situations. The advantage of the linearized model is that adjustments to the linear part of the damper is very simple to set whereas the full model needs understanding on how the orifice and bleed valve opening changes the behaviour of the damper.
Chapter 5

Conclusions and future work

In this chapter the conclusions of the work are discussed. What the limiting factors are and what improvements can be made in order to further develop the model.

5.1 Conclusions and discussion

In order to gain understanding of the dynamic behaviour of a MBT two half-vehicle models with different DoF and three full-vehicle models with different suspension systems were developed in MATLAB/SIMULINK. The models are all based on the equations of motion describing the dynamics of the system. Through the dynamics of the tank the motion is then connected to the barrel in order to investigate the fire precision of the tank. In order to evaluate the performance three different models for dampers are developed and compared to see the influence the suspension system has on the entire tank.

From the half-vehicle models can it be observed that adding more degrees of freedom corresponding to the tracks does not significantly change the behaviour of the hull for the preset conditions. The linear model is used as a representation for last generation of tanks fitted with torsion-bar suspension and dampers on a select few wheels. Although the linear model will not perfectly represent the last generation of tanks as the torsion bar is not completely linear in its behaviour it is only used as a reference of comparison. Also some damping will be present in the torsion bar but it will in comparison to the stiffness be low and hence is neglected. The HGS system commonly seen on modern tanks can be observed to have increased performance with lower average disturbance in all axes compared to the linear model.

Comparing the aim-point displacement of the tank with the different damper configurations significant changes in behaviour are observed. The behaviour of the linear system is showing a larger spread in aim-point. Where the pneumatic system on the other hand does still have the same tendencies but the spread is more composed in nature.
The results can be separated into two different parts. The first part is how the different suspension systems affects the performance of the tank. This is the aim accuracy of the tank which is the main focus of this thesis. The results are measuring the performance of the tank and will be used to determine minimum performance requirements of the control system for the barrel stabilization.

The next part, which is equally as important, is the transmission of vibrations to the crew of the tank, which is not investigated in this thesis. This is a subject that include the question of how long the crew can withstand operating the tank. Minimizing the transmission of vibrations is of great importance as to not make the crew motion sick or having the crew experience fatigue due to the operation of the tank to name a few examples.

The parameters for both the linear and the non-linear models are produced by comparison to models of other tracked vehicles and simulations of multiple settings. Both models are passive suspension and none of these are in any way optimized. This means that the result of the simulations are in no way optimal and could possibly be either better or worse than the real vehicle.

## 5.2 Limitations

The main limitation of the model is that it is not verified against the tank that is modelled. This fact makes the results of the tank model estimates. It is therefore uncertain how accurately the tank model correlates to the actual system. The model can be used comparatively to see tendencies of how changes to the tank design or environment will affect the tank but it is only an estimate and should not be considered as equal to the real system.

The model is assuming a few things regarding the world. The first one regarding the ground of which the tank is running. Assuming that the ground is infinitely stiff will give a good indication of the dynamics of the tank but the tank will most likely never run in such conditions. This would mean that the tank would receive some damping from the ground and hence the motion would be smoother than what the model would indicate. Also the model is feed the terrain profile but can not say anything about the tanks ability to overcome the terrain at all. This would modelling of the contact between the surface and the tank tracks and also would need to have information about the ground of, which the tank is moving on.

Further, the model is an idealized model. There are many parts that could be taken into consideration if one is trying to model the tank as accurately as possible. To name a few of the main areas that will add to the dynamics: more contributing vibration sources, structural dynamics, terramechanics, viscous and flow properties,
and changes due to external influences such as climate, wear and more.

5.3 Future work

The models produced is a relatively simple model and has many areas that could be improved on to better represent the tank as a complete system if that is what is desired. The most important thing that the model needs is verification. All the parameters the model is based on are chosen to the best knowledge and ability, but can not confirm or accurately represent the movement with any certainty of the tank if not verified and tuned according to the real system. Other parts that could help improve are if data became available for subsystems such as dampers.

Further, the only part that excites movement is the road profile. The real tank has more sources of vibrations that could introduce movement or disturbances. There are many parts that can be of interest to add in order to evaluate if the contribution affects the motion of the tank. This could include, for instance an extended model of the tracks. The tracks themselves have both the loaded part in contact with the ground as well as the unloaded parts above the wheels. The section of unloaded tracks could very possibly induce vibrations as the mass of the tracks is quite large and in the unloaded state can most certainly come into eigen-frequency if under the right circumstances. Further, the tanks engine is a big vibration source. The engine can both come in to eigen-frequencies with the drive-train and also induce disturbances for the control system of the turret.

The HGS also introduces the possibility to add a control system to further increase the performance. Controlling the pressure in real time to minimize the acceleration of the hull could be a interesting area to investigate. There are some different approaches that might be considered. The model produced has the same pressure on all the HGS units so it would be interesting to investigate if there would be a performance gain to give individual units different pressures i.e. varying the spring stiffness across the length of the tank. Depending on how the unit is constructed if there is possibility to control in real time both the pressure of the spring as well as the damping with control of the bypass valves restriction of flow in the damper. There are different ways of approaching the minimization of the disturbance seen by the hull, some simpler and some more advanced. Depending on the construction and possibilities the given system will bring to the table it could be interesting to see how large of a decrease in vibrations that could be realistically implemented. Last, there are some simplifications done to the model but it could most probably be further simplified. A part could be to see how many of the wheel pairs that are needed and how the model is impacted by reducing the contact points with the ground.
Bibliography


