Torque Control Strategy for Off-Road Vehicle Mobility

By
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In Mechanical Engineering

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Dedicated to my parents, my wife and my kids
They mean a lot to me, here and hereafter!
Abstract

Multi-wheeled off-road vehicles behavior depend not only on the total provided power by the engine but also on the power distribution among the drive axles/wheels. In turn, this distribution is primarily regulated by the drivetrain layout and the torque distribution devices. At the output of the drivetrain system, the torque is constrained by the interaction between the wheels and the soft terrain. For off-road automotive applications, the construction of drivetrain system has usually been largely dominated by the mobility requirements. With the growing demand to have a multi-purpose on/off road vehicle with improved maneuverability over soft soil particularly at higher speed, the challenges confronting car designers have become more sophisticated.

A number of simulation studies, during longitudinal and cornering maneuvers, are conducted to investigate the contribution of typical significant parameters. In addition, the influences of different drivetrain arrangements are presented. The obtained results defined that both traction and cornering response of multi-wheeled off-road vehicles are highly affected by the driving torque distributed between axles/wheels.

In this thesis, the main challenge is to develop an effective torque distribution control strategy to improve both directional dynamics and safety of the vehicle. The developed torque vectoring control strategy can be widely applied to vehicles of two or more axles. In this research work, the application to multi-wheeled combat vehicles is extensively investigated. An advanced fuzzy slip control and a yaw moment control systems designed, and both performance and effectiveness of the developed controllers evaluated using different standard test maneuvers. Finally, the integrated control systems investigated to verify the proposed control strategy effectiveness on the vehicle direction stability and mobility based on some predefined standard test maneuvers.
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Finally, I would like to express my deepest gratitude to my wife Randa, who allowed me to focus on my research through her love and looked after our kids Mohamed and Hana.
## Nomenclature

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<th>Symbol</th>
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<tbody>
<tr>
<td>$a_y$</td>
<td>Vehicle lateral acceleration, g</td>
</tr>
<tr>
<td>B</td>
<td>Vehicle wheelbase, m</td>
</tr>
<tr>
<td>$b$</td>
<td>Characteristic dimension in Bekker's pressure-sinkage equation, m</td>
</tr>
<tr>
<td>$b_f$</td>
<td>Footing width, m</td>
</tr>
<tr>
<td>C</td>
<td>Soil cohesion, kN/m$^2$</td>
</tr>
<tr>
<td>$C_a$</td>
<td>Cornering stiffness of the tire (averaged per axle)</td>
</tr>
<tr>
<td>$C_{10}$, $C_{01}$</td>
<td>Mooney-Rivlin constants</td>
</tr>
<tr>
<td>D</td>
<td>Outside tire diameter, m</td>
</tr>
<tr>
<td>$D'$</td>
<td>Diameter of the substitute circle</td>
</tr>
<tr>
<td>$dW$</td>
<td>Work done during an isothermal displacement</td>
</tr>
<tr>
<td>$dl$</td>
<td>Isothermal displacement</td>
</tr>
<tr>
<td>$dS$</td>
<td>Entropy change</td>
</tr>
<tr>
<td>$F_y$</td>
<td>Cornering force</td>
</tr>
<tr>
<td>j</td>
<td>Shear displacement</td>
</tr>
<tr>
<td>K</td>
<td>Soil shear deformation modulus, m</td>
</tr>
<tr>
<td>$K_c$</td>
<td>Cohesive modulus of vertical soil deformation, kN/m$^{n+1}$</td>
</tr>
<tr>
<td>$K_{\phi}$</td>
<td>Frictional modulus of vertical soil deformation, kN/m$^{n+2}$</td>
</tr>
<tr>
<td>$K_{us}$</td>
<td>Understeer gradient of the vehicle</td>
</tr>
<tr>
<td>L</td>
<td>Vehicle wheel track, m</td>
</tr>
<tr>
<td>n</td>
<td>Soil exponent in Bekker's pressure-sinkage equation</td>
</tr>
<tr>
<td>$N_c$</td>
<td>Mobility number for clay, dimensionless</td>
</tr>
<tr>
<td>$N_s$</td>
<td>Mobility number for sand, dimensionless</td>
</tr>
<tr>
<td>$N_{q,c,\gamma}$</td>
<td>Bearing capacity factors</td>
</tr>
<tr>
<td>P</td>
<td>Ground pressure</td>
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<tr>
<td>$P_c$</td>
<td>Pressure produced by the stiffness of the carcass</td>
</tr>
<tr>
<td>$P_i$</td>
<td>Tire inflation pressure, kPa</td>
</tr>
<tr>
<td>$P_f$</td>
<td>Terzaghi’s bearing capacity</td>
</tr>
<tr>
<td>$q'$</td>
<td>Effective surcharge</td>
</tr>
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r  tire rolling radius, m
R  tire radius, m
R_1  Outer clutch disk radii, m
R_2  Inner clutch disk radii, m
R_c  rolling resistance due to soil compaction, kN
R_m  Wheel motion resistance
S  tire slip, percent
T  torque, kN.m
\(\sigma_{1,2,3}\)  the principal stresses
u  Vehicle longitudinal speed
W  vertical load on the tire, kN
Z  sinkage of any point on the tire-soil interface, m
Z_o  maximum sinkage, m

**Greek Letters:**

\(\alpha\)  Shape factor
\(\alpha_{1,2}\)  First and second axles slip angle
\(\beta\)  vehicle sideslip angle
\(\gamma\)  Unit weight
\(\phi\)  soil friction angle, deg
\(\phi_s\)  Angle of terrain shearing resistance, deg
\(\omega\)  angular velocity of the tire, rad/s
\(\omega_s\)  Sun gear angular speed
\(\omega_r\)  Ring gear angular speed
\(\omega_c\)  The planet carrier angular speed
\(\sigma\)  normal stress on the soil, kPa
\(\tau\)  soil shear stress, kPa
\(\tau_m\)  maximum soil shear stress, kPa
\(\lambda_{1,2,3}\)  principal extension ratios
\(\lambda_i\)  extension ratio
\(\varepsilon\)  Strain
$\delta_i$    Inner steering angle

$\delta_o$    Outer steering angle

$\Delta T$    differential corrective torque

$e_{a_y}, e_r$    errors corresponding to lateral acceleration and yaw rate control respectively.
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Chapter 1

Introduction

1.1 Research Overview

Multi-wheeled vehicles that are used mainly for military or for special purposes have to fulfill several main requirements. One of these requirements concerns is the off-road vehicle mobility, which is the ability of the vehicle to cope with challenging cross-country terrains. Off-road terrains characterized by deformable irregular surfaces with abrupt slopes and obstacles of the distinctive nature. The interaction between wheeled vehicles and soft terrain is complex and strongly dominated by the terrain’s mechanical properties. Furthermore, some soils can behave excessively in terms of sinkage and slippage according to the applied vertical load and driving moment on the wheel.

Nowadays, many researchers are interested in enhancing the vehicle mobility over a wider range of terrains. Initially, the rigid four-wheel drive layout was assumed such that both the front and rear axles were coupled to the transfer-case without speed differential offering better tractive performance. However, during cornering maneuvers on rigid terrains serious problems still need more research work.

The primary objective now is to design a multi-purpose on/off road vehicles with high traction, acceleration performance, and improved maneuverability especially over soft terrains. Off-road vehicles are more sensitive to these requirements in comparison to the passenger cars due to the high ground clearances that represent an essential requirement for off-road operations.

1.2 Problem Definition

The striking challenge is to design an efficient electronically controlled torque management system to distribute the available driving torque between the axles/wheels independently to
enhance vehicle traction performance and cornering stability. The concept of all-wheel-drive (AWD) enhanced vehicle performance and mobility. The principal components that are widely used in AWD powertrain layouts are mechanical differentials (open and locked), limited slip differentials and electronically controlled differentials.

Multi-wheeled off-road vehicle modeling including vehicle body dynamics, powertrain configuration, multi-axle steering systems, suspensions, and tires for different terrain conditions is a very complex task. Even, the probable model should provide the designers with the capability to investigate the vehicle components that will be a significant step in developing control systems.

An extensive work to investigate multi-wheeled off-road vehicle performance based on different powertrain configurations has been performed. However, the tire was characterized by empirical on-road tire models, and the road conditions were approximated and represented by the coefficient of adhesion. This approach should not be extended to off-road vehicles due to the complication of tire-soil interaction characteristics such as multi-pass sinkage.

Consequently, having an accurate pneumatic tire and soft soil models is essential for improving the mathematical modeling representation of off-road vehicle dynamics. Traction, braking performance and handling properties of the vehicle are affected by the tire-terrain interaction characteristics. However, the tire-rigid terrain interaction is fully understood, tire-soil interaction still need extensive work from researchers.

For the reason that tire-soil interaction field tests are both inherently costly and difficult to control, the cost efficient finite element analysis method (FEA) has been used for decades for conducting such tests. Likewise, FEA has been used to study a variety of aspects of terramechanics with great success. Non-linear tire look-up tables for rigid and soft terrain obtained from FEA off-road tire models has been integrated with full 8x8 vehicle model to investigate the vehicle maneuverability and directional control stability on soft ground as a tuning process for control strategy development. The tuning process contains torque
distribution characteristics, sensitivity analysis for different powertrain configurations and vehicle parameters to understand its effect on vehicle off-road performance.

1.3 Overall Aims and Objectives

The aim and objectives of the current research follow directly from the problems stated in the preceding sections. A set of well-defined tasks have been performed and are outlined below:

- Development of FEA off-road tire model based on a real combat vehicle tire, 12.00R20 XML TL 149J, dimensions and material data to present the terramechanical phenomena between elastic tires and soft soils. Then, this tire model experiences validation tasks to check whether it follows the similar behaviors of the available measured data.
- Development of a multi-wheeled combat vehicle dynamic model based on a real combat vehicle dimensions and weights. Then, this vehicle model experiences validation tasks to check whether it follows the same behaviors of the available measured data.
- Integrating the developed off-road tire model with the multi-wheeled combat vehicle model. Carrying out a comprehensive investigation of traction and handling performance during typical maneuvers under different operating conditions.
- Development of a controller in a typical programming language environment (MATLAB, Simulink), to enhance vehicle mobility performance based on actively torque distribution control according to terrain conditions and other environmental conditions.
- Carrying out an investigation to evaluate the tractive performance and cornering stability of the multi-wheeled combat vehicle, fitted with the controller, as well as ordinary drivetrain systems in different powertrain configurations (8x4 and 8x8).
Chapter 2

Review of Literature

2.1 Introduction

Multi-wheeled off-road vehicles behavior depend not only on the total provided power by the engine but also on the power distribution among the drive axles/wheels. In turn, the drivetrain layout and the torque distribution devices primarily regulate this distribution. The drivetrain system output torque depends on the tire-soil interaction characteristics.

In this chapter, the issues of off-road tire modeling, off-road vehicle dynamic simulation, and various torque management devices implemented in multi-wheeled vehicles are reviewed. Attention is paid to the use of active control devices in AWD vehicles. The following sections critically analyze the most appropriate reported work.

The review is divided into the following areas:

1. Off-road vehicle mobility.
3. Off-road vehicle dynamic simulation.
4. Torque management devices.

2.2 Off-Road Vehicle Mobility

Wheeled vehicles that are used in military or for special purposes have to satisfy several requirements and mobility is one of the most important concerns. Off-road terrains characterized by deformable irregular surfaces with abrupt slopes and obstacles of the distinctive nature. The interaction between wheeled vehicles and soft terrain is complex and strongly dominated by the terrain’s mechanical properties. Furthermore, some soils can behave excessively in terms of sinkage and slippage according to the applied vertical load and driving moment on the wheel.

The available publications related to the off-road vehicle mobility evaluation show
significant and useful efforts in this area. These efforts brought to light some methods and techniques that can be used in vehicle mobility evaluation.

The mobility of the vehicle is influenced by many parameters, Figure 2.1, which make the evaluation process complicated, the main factors affecting vehicle mobility are:

- Vehicle design and construction parameters.
- Soil parameters.
- Environmental parameters.

![Factors Affecting Vehicle Mobility](image)

Figure 2.1 Factors affecting vehicle mobility [2]

In the present work, climate conditions and driver's skill are assumed in satisfactory condition. Hereafter, only vehicle and soil parameters are to be considered when studying the parameters influencing the off-road vehicle mobility evaluation.

### 2.2.1 Vehicle parameters affecting vehicle mobility

The vehicle parameters have considerable influence on vehicle mobility. Figure 2.2 shows the vehicle parameters affecting vehicle mobility that include; vehicle performance, geometric configuration, vehicle construction and economy of operation [2].

#### 2.2.1.1 Vehicle performance

The vehicle performance can be evaluated based on the study of; engine characteristics, transmission characteristics, climbing ability, acceleration, towing ability, crossing of obstacles, crossing of trenches, and flotation. The transmission may be divided into two
groups; axled and H-shaped as shown in Figure 2.3 and Figure 2.4.

Figure 2.2 Vehicle parameters affecting vehicle mobility [2]
Figure 2.3 Axle designs of transmission [1]

Figure 2.4 H-shaped and combined designs of transmission [1]
Axle designs are used with dependent and independent suspensions as well; the primary transmitters and inter-wheel differentials located on the axles. Power distribution between the axles is effected by either one or more distributor cases while H-shaped transmissions usually used on vehicles with high off-the-road mobility with an independent suspension. The use of H-shaped transmission provides greater road clearance and better utilization of the inner volume of the body [1].

2.2.1.2 Vehicle geometric configuration
Vehicle geometric configuration refers mainly to the vehicle shape and dimensions including vehicle overall height, width and length, wheelbase, ground clearance, angle of approach, angle of departure, longitudinal vaulting radius and transversal vaulting radius as shown in Figure 2.5 and Figure 2.6 [2].

![Figure 2.5 Geometrical properties of a wheeled off-road vehicle](image1)

![Figure 2.6 Vaulting radii, (a) Longitudinal and (b) Transversal](image2)

2.2.1.3 Vehicle construction
Vehicle construction deals with some design parameters of the vehicle such as vehicle
weight and payload, handling characteristics, tire forces and self - recovery means.

(a) Vehicle weight and payload

The ability of a low-weight vehicle to carry greater loads indicates higher vehicle performance. On soft terrain, the optimum load carrying capacity varies with the mechanical properties of the soil. Rolling resistance increases with increasing vehicle weight due to increased soil sinkage [2].

(b) Tires

The primary functions of tires are supporting the weight of the vehicle, cushions the vehicle over surface irregularities, provides sufficient traction of driving and braking, and provides adequate steering control and directional stability [2].

Vehicle mobility performance depends on several tire parameters, the following items are to be investigated; tire types, inflation pressure and rigidity with relative to the soil, ground pressure, tire tread pattern, and tire pressure control.

1) Tire types:

According to the construction, there are two main types of tires that are commonly used; bias-ply tires and radial-ply tires. Radial-ply tires show the following advantages over the bias-ply tires [2]:

- Less slippage.
- Increased drawbar pull.
- Less tread wear.
- Better distribution of torque.
- Less rolling resistance.
- Excellent upholding during cornering.

Dwyer et al. [3] investigated the performance of five different agricultural tractor tires on thirty-two different terrain conditions to compare the obtained results with a predictive approach valid for different range of tire sizes, load, and soil conditions.
Hetherington and Littelton [4] studied the effect of dual wheel configuration on both rolling resistance and sinkage of towed rigid wheels on sand. The conducted study stated that using dual tires instead of single one reduces both sinkage and rolling resistance.

2) **Inflation pressure:**
The increase of tire inflation pressure increases the tire stiffness and reduces the contact area. Czako [5] found that the drawbar pull increases with reduction of inflation pressure. Figure 2.7 and Figure 2.8 shows the drawbar pull-slip curves for fine and coarse sand respectively.

![Figure 2.7](image)

Figure 2.7 (Drawbar pull / weight) - slip curves in fine sand [5]

3) **Specific ground pressure:**
Specific ground pressure is known as the weight per unit contact area between tire and ground. In addition, low specific ground pressure, especially for soft soils, is recommended for higher mobility performance.

4) **Tire tread pattern:**
It is the appropriate arrangement of ribs, grooves, lugs and sips in the tread. Road grip, wear and driving noise are dependent on the type of tread pattern and its condition. The pattern itself is chosen according to the tire application. All wheels of a vehicle should be equipped with tires of the same tread pattern [6].
Figure 2.8 (Drawbar pull / weight) - slip curves in coarse sand [5]

Tread configuration, as shown in Figure 2.9, affects the performance of off road tires. In soft soils, the lugs will increase the operative tire radius, as it will be clogged with soil. While on rigid terrain, smooth tires will provide the same drawbar pull. In the case of high moisture terrains, traction aids will not provide sufficient traction [7].

Figure 2.9 Tread configuration [7]
5) **Tire pressure control:**
Adjusting the inflation pressure according to the kind of soil is necessary to improve the tire-soil interaction. Vehicles equipped with pressure control systems have an increased off-road performance, as the tire pressure can be adjusted according to load and terrain conditions even during vehicle motion. This system is suitable for vehicles operating on a wide range of terrain types [2].

### 2.2.2 Soil parameters affecting vehicle mobility

The word "soil" is widely known as the surface layer of earth that supports our plant life [8 and 9]. This definition is incomplete from the point of view of researchers and specialists such as terrain-vehicle engineers who design off-road vehicles capable of negotiating different kinds of soils. The soil parameters affecting vehicle mobility could be permanent or transient parameters and soil behavior under loading as shown in Figure 2.10.

![Soil parameters affecting vehicle mobility](image)

**Figure 2.10** Soil parameters affecting vehicle mobility [2]
The main soil parameters affecting vehicle mobility may be summarized as follows:

- Grain size distribution.
- Bulk density.
- Moisture content.
- Shear strength.
- Bearing capacity.

2.2.2.1 Soil grain size distribution

Particle size distribution in soil and its density influences the soil strength and compressibility, both of which are necessary for the consideration of flotation for vehicle mobility. Therefore, the grain size distribution of the soil influences mechanical, physical, and biological properties of soils. The effect of grain size distribution on the output drawbar pull of the tested vehicles in different soil types like loam, fine sand, and coarse sand is investigated by Czako [5] as shown in Figure 2.11.

![Graph showing drawbar pull/weight versus slip curves in hard loam](image)

Figure 2.11 Drawbar pull / weight versus slip curves in hard loam [5]

2.2.2.2 Soil bulk density

Soil bulk density can be defined as the solids weight per unit of the total soil volume. Soil compaction will increase shear strength, increase bearing capacity, and decrease permeability.
2.2.2.3 Soil moisture content

The moisture content has a significant effect on wheeled vehicles traction and resistance coefficient as shown in Figure 2.12 and Figure 2.13 [10]. Yusu and Dechao [11] deduced the soil friction resistance per unit area and the moisture content relationship which presented that the soil has single peak close to the plastic limit as shown in Figure 2.14.

![Figure 2.12 Net traction coefficient - water content at different inflation pressures](image1.png)

Figure 2.12 Net traction coefficient - water content at different inflation pressures [10]

![Figure 2.13 Resistance coefficient -water content at different inflation pressures](image2.png)

Figure 2.13 Resistance coefficient -water content at different inflation pressures [10]
2.2.2.4 Soil shear strength

It can be defined as the soil maximum resistance to shearing stresses and depends on moisture content, soil type, and grain size distribution of the soil. The soil shear strength can be determined using Equation (2.1) depending on two parameters, soil cohesion (C) and internal friction angle (Φ). The two parameters are obtained based on the Mohr-coulomb failure criterion as shown in Figure 2.15:

\[ \tau_m = C + \sigma \tan \Phi \]  

Where:

- \( \tau_m \) .......... the maximum shear stress.
- \( \sigma \) .......... the normal stress.
- \( C \) .......... the soil cohesion.
- \( \Phi \) .......... the angle of internal friction.

Figure 2.14 Experimental relation between friction and soil water content [11]
There are two types of shear stress curves; the first one presents the maximum shear stress $\tau_{\text{max}}$ and a part of residual shear stress $\tau_r$ after yielding as shown by curve 1 in Figure 2.16. The second one is the shear stress-displacement curve as shown by curve 2 in Figure 2.16 [12].

![Mohr-Coulomb relationship](image1)

Figure 2.15 The Mohr-coulomb relationship [12]

![Shear stress-displacement curves](image2)

Figure 2.16 Shear stress-displacement curves [12]

### 2.2.2.5 Soil bearing capacity

The bearing capacity is the required average load per unit area on the contact area to reach the supporting soil mass failure [13]. The bearing capacity theory estimates the maximum load that the vehicle can exert on the terrain without failure. The pressure sinkage relation
of terrain, assuming homogeneous characteristics, can be determined using Equation (2.2) [12].

\[ P = (\frac{K_c}{b} + K_\phi)Z^n \]  

(2.2)

Where:

- \( P \) ................. the ground pressure.
- \( b \) ................. the width of contact area.
- \( Z \) ................. the sinkage.
- \( n \) .................. the exponent of deformation.
- \( K_c, K_\phi \) ............. the terrain constants.

Terzaghi’s bearing capacity formula is given by the following equation [14].

\[ P_f = \alpha C N_c + q' N_q + \frac{1}{2} \gamma b_f N_\gamma \]  

(2.3)

Where:

- \( P_f \) ................ Bearing capacity
- \( \alpha \) ................. the shape factor.
- \( C \) ................. the cohesion.
- \( q' \) ................ the effective surcharge.
- \( \gamma \) ................. the unit weight.
- \( b_f \) ................. the footing width.
- \( N_c, N_q, N_\gamma \) ..... the bearing capacity factors.

### 2.3 Mechanics of Wheel-Soil Interaction

#### 2.3.1 Introduction

Mechanics of wheel-soil interaction is one of the essential aspects in off-road vehicle studies. Tire-soil interaction is one of the most complex tasks for researchers as it includes many features such as sinkage, multi-pass and slip sinkage. Driven wheel performance is usually characterized by its thrust, resistance to motion, sinkage, slip, driving torque and angular speed. One of the prime interest to all researchers and designers of off-road vehicles is how to predict these parameters accurately. Different approaches have been suggested to
investigate the tire-soil interaction characteristics starting from empirical approaches to highly theoretical ones as shown in Figure 2.17.

![Diagram of tire-soil interaction approaches]

**Figure 2.17 Common approaches used to study tire-soil interaction**

### 2.3.2 Empirical approach

This approach was introduced for the first time in the Second World War by the U.S. Army Waterways Experiment Station (WES) to support the military with a simple and quick tool to determine the terrain mobility on the basis of (go/no go) [15]. This method is based on measuring the soil penetration resistance to describe the soil properties using a standard cone penetrometer device as shown in Figure 2.18. The developed models based on this approach are applicable for in-situ decision-making during field operations [16].

Based on the WES approach, Ahlvin and Haley [17] developed a mobility model which is called the NATO Reference Mobility Model ‘NRMM’. The NRMM is a set of equations that predict an individual vehicle’s mobility performance in a given terrain based on the
vehicle characteristics and the terrain properties. The primary objective of NRMM is vehicle's *speed-made-good* per terrain unit. Therefore, speed prediction and limiting force calculations can be determined for on-road, off-road, and obstacle crossing maneuvers.

![Figure 2.18 Cone penetrometer, (a) standard (b) electronic](image)

WES and TACOM [18] (Tank Automotive Command) developed another mobility model known as ‘NRMM-II’ to include improved mobility processes. Sullivan worked on having a better-organized modular structure and a more flexible user interface. NRMM-II is used to determine on-road/off-road mobility characteristics based terrain characteristics, vehicle attributes, and scenario parameters, e.g. to predict vehicle speeds over terrains, often used to compare two vehicles over a given terrain.

### 2.3.3 Analytical approach

Analytical (or semi-empirical) models are very common and are computationally very useful. Most of the basic knowledge regarding Tire-soil interaction analytical modeling is accessible in textbooks by Bekker and Wong. In 1950, Bekker developed different tire-soil analytical models. He supposed for the same sinkage ($z$) that the normal ground pressure ($P_n$) will be equivalent to the pressure under a plate. This equation is called the *Bekker pressure-sinkage equation*, and founds the basis for the off-road analytical tire models.

$$P_n = K_c \left( \frac{K_c}{b} + K_\phi \right) \cdot z^n = K \cdot z^n$$  \hspace{1cm} (2.4)

Where:

- $K_c$, $K_\phi$ ............ the cohesive and frictional moduli of soil deformation.
Based on this assumption, Bekker established a formula for predicting the resistance to the wheel motion \(R_m\) and its sinkage \(z\) as follows:

\[
R_m = b \cdot K \cdot \frac{z^{n+1}}{n+1}
\]

(2.5)

\[
z = \left( \frac{3 \cdot W}{b \cdot (3 - n) \cdot K \cdot \sqrt{D}} \right)^{\frac{2}{2n+1}}
\]

(2.6)

Far ahead, Bekker established an equation to define the tire critical inflation pressure at which the tire may be considered to be in elastic mode. Based on this equation; if the total inflation pressure \(p_i\) and the carcass pressure \(p_c\) is less than the pressure that the terrain can support. The terrain is considered rigid, and the tire contact area would be flattened and could no longer be modeled as a rigid rim as seen in Equation (2.7).

\[
p_i = \frac{W \cdot (n + 1)}{b \cdot \left( \frac{3 \cdot W}{(3 - n) \cdot b \cdot K \cdot \sqrt{D}} \right)^{\frac{1}{2n+1}} \cdot \sqrt{D - \left( \frac{3 \cdot W}{(3 - n) \cdot b \cdot K \cdot \sqrt{D}} \right)^{\frac{2}{2n+1}}} - p_c
\]

(2.7)

Bekker established a test facility that can be used to characterize soil shear strength known by ‘Bevameter’. This device was used to obtain shearing torque versus displacement curves using a shear annulus head at different vertical loads. The well-known “shear stress-shear displacement equation” proposed by Janosi and Hanamoto is used to fit the shearing torque-displacement data and predict the shear stress at the tire contact area with terrain by using the following equation:

\[
\tau = \left( C + P_n \cdot \tan \varphi_s \right) \cdot \left( 1 - e^{-j/k} \right)
\]

(2.8)

The first term in Equation (2.8) consists of two parts; the first part corresponds to the apparent terrain cohesion \(C\) and the second part is due to the frictional portion of the shear strength \(P_n \cdot \tan \varphi_s\), where \((\varphi_s)\) is the shearing resistance angle \([19]\). \((j)\) is the shear
displacement and (K) is the shear deformation modulus.

Schmid [20] presented the state of the art in the field of tire-terrain interaction. Schmid and Ludewig [21] proposed a parabolic shape to present the contact area between tire and terrain using the circle-section (D*) as shown in Figure (2.19).

![Figure 2.19 Contact geometry models proposed by Schmid [21]]

The proposed circle diameter (D*) is obtained based on the equilibrium condition between the tire vertical load and ground vertical reaction. Furthermore, Harnisch et al. [22] optimized the off-road tire model for use in MATLAB/Simulink dynamics simulation environment (S-function). Currently, this tire model is a commercially available software tool and known by AS²TM “AESCO Soft Soil Tire Model”.

### 2.3.4 Finite element method (FEM) approach

Perumpral et al. [23] used Finite Element Method (FEM) for his study of tire-terrain interactions to calculate the stress distributions and soil deformation under a tractor tire. This method requires the contact area geometry and the stress distributions to be specified accurately. It can only be used to analyze the strain, stress and displacement within the soil mass.
Yong et al. [24] investigated the stress and strain fields in the soil underneath the tire an advanced FEA model. The presented model assumed that the tire is a linear elastic body, and the soil is a linear elastic finite element. Normal and shear stress data were used as inputs, and the length of the contact area was predicted using modified Hertzian theory. Nakashima and Wong [25] developed a finite element tire model based on the available data from the tire manufacturers (generalized deflection, load, and contact area charts) to determine the Young’s moduli of elasticity for both sidewall and tread of the tire.

Aubel [26] developed a full FEM model known by ‘VENUS’, ‘VEhicle-NatUre Simulation’ as shown in Figure 2.20. The model contains three main sub-modules to present the soil, tire and tire-soil interaction. Furthermore, the FEM-soil model was adapted to consider the cohesive properties as well. The tire was modeled using three concentric rings; tread, carcass and wheel-rim. The primary output of the model was the deformation of the soil and the tire.

![Simulation of the Tire-Soil Interaction using FEM](image)

(a) pressure-distribution  
(b) Shear stress

Figure 2.20 Simulation of the Tire-Soil Interaction using FEM [26]

Liu and Wong [27] developed different tire-soil interaction models based on soil mechanics and finite element analysis using a finite element program known by MARC as shown in Figure 2.21.
Guan Yanjin et al. [29] developed a non-linear FEM model using MSC.MARC software to investigate the tire rolling performance. Several results, such as tire deformation at different condition, strain distribution, and the normal stress distribution. In addition, Kaiming Xia [30] developed a three-dimensional finite element model for tire/terrain interaction for modeling of rubber materials as shown in Figure 2.22.

Figure 2.21 Finite element mesh and the distribution of vertical stress on loose sand [28]

Figure 2.22 Finite element model of tire-soil interaction [30]
2.4 Off-road Vehicle Dynamic Simulation

Development in vehicle mobility over different types of terrain has encouraged a great interest in the simulation of vehicles over off-road terrain. Commonly, there are two goals for off-road vehicle simulation [16]:

- The first one is to describe the behavior of an off-road vehicle and soil mechanical properties. Predicting vehicle performance under different operating situations is the main challenge to the designer and users of off-road vehicles [31].
- The second one is to study the multi-pass effect on soft terrain and how it can affect on vehicle mobility performance [32].

The primary structures of some of the well-known off-road vehicle dynamics studies will be discussed in the following sections. In addition, it should be mentioned that all the previous research presented in this chapter based on the analytical approach of tire-soil mechanics was originally introduced by Bekker, ([33], [34], and [35]).

2.4.1 The Canadian school

Wong and Preston-Thomas [36] developed a computer-aided methodology for multi-axle wheeled vehicles tractive performance over off-road terrains. In addition, they have investigated the effect of different parameters; tire configuration, inflation pressure, and static load distribution over two types of terrain on vehicle tractive performance.

Wu [37] developed a 17-DOF model to simulate handling performance of off-road vehicles of a 6WD military vehicle on both rigid and soft terrain based on a computer-aided simulation program known by “AUTOSIM ”. The handling characteristics on soft terrain verified low tire lateral forces and a significant time lag with respect to the steering input.

Wong and Huang ([38] and [39]) compared the thrust produced by a multi-axle Light Armored Vehicle (LAV, 8x8). Their comparison carried out based on using different models like RTVPM, NTVPM and NWVPM.
NWVPM, ‘Nepean Wheeled Vehicle Performance Model’ is a computer program for predicting off-road vehicles performance based on using two modules; the first one predicts the operating mode of the tire in the form of thrust, motion resistance and sinkage. The second one predicts the dynamic load transfer and the multi-pass effects.

2.4.2 The British school

Crolla [40] over 20 years of research in the field of off-road vehicle dynamics presented many research work in the various aspects; improvement of off-road vehicle ride, steering behavior and lateral stability of tractor, braking, slope stability and tire modeling. Some of Crolla’s contributions, which are related to the current research, will be presented as follows.

Crolla [41] developed a computer program to investigate an agricultural tractor performance under different loading conditions. Various features of tractor design were discussed and design criteria were suggested to control the variations in load. Furthermore, Crolla and Hales [42] found that lateral forces were related to the slip angle by an exponential relationship and the lateral force characteristic at small slip angles was found to be non-linear. In addition, lateral force coefficient reduced with an increase in tire vertical force and the presence of braking or tractive force reduced the lateral force.

Crolla and Horton [43] suggested suitable approaches for off-road vehicle steering systems modeling and simulation including the role of tire/soil interaction in tire forces generation, effect of tire dynamic response, hydrostatic system characteristics and articulated-frame steer vehicles. Since all the analytical models are subjected to some limitations, Crolla and El-Razaz [44] proposed a tire model that can be used to determine the generated forces at the tire-soil contact are in both longitudinal and lateral directions. Furthermore, this tire model was adapted to study the tire-soil interaction characteristics for different assumptions and to investigate the effect of several factors ([45], [46], [47], and [48]).

2.4.3 The German school

Ruff et al. ([49], [50], and [31]) developed an interactive simulation program for off-road vehicles mobility performance known by ‘ORIS’ (Off Road Interactive simulation). The
developed program consists of different sub-models to present the tire-soil interaction, motion resistance and driveline power transmission as shown in Figure 2.23. Furthermore, Harnisch [51] investigated the effect of increasing the number of axles from the perspective of efficient off-road truck design. The results of the simulated multi-axle vehicle presented a notable reduction in rolling resistance due to the multi-pass effect.

![Diagram of ORIS program main structure](image)

Figure 2.23 ORIS program main structure [49]

Additionally, Harnisch [52] investigated the multi-pass effect on the process of cornering of multi-axle-steer vehicles considering the ruts of the wheels. The outcomes presented that, the multi pass-effect was reduced during lateral maneuvers of multi-axle vehicle causing a higher rolling resistance. Furthermore, this negative effect could be reduced by using multi-axle-steering layout especially for the case of symmetric all-wheel steering systems (AWS).
Harnisch [53] improved the abilities of the ORIS program and added more features to the tire model itself, such that the new version was able to simulate multi-drive-axles and multi-steer-axles. Furthermore, it is also possible to include test stands, Hardware in the Loop, as well as driving simulators with motion systems. The new version of the program is known by ‘ORSIS’ (Off Road Systems Interactive Simulation) as shown in Figure 2.24.

![ORSIS Program Main Structure](image)

**Figure 2.24 ORSIS Program Main Structure [54]**

### 2.5 Torque Management Devices Implemented in AWD Vehicles

Off-road vehicles have different running abilities; higher traction, tractive efficiency and improved mobility, which depend not only on total tractive effort available by the power, plant but also on its distribution between the driving wheels. Which can be determined by actuating vehicle systems and characteristics of the power dividing mechanisms e.g. inter-wheel, inter-axle reduction gear and transfer cases. The locking features of these mechanisms control the force distribution between driving wheels. Consequently, they can control both vehicle longitudinal performance and handling characteristics [55]. Mohan and Williams [56] organized different AWD traction control systems, including passive and active devices, by the used general principles and their strategies as shown in Figure 2.25.
Lanzer [58] suggested a torque split factor to evaluate the impact of tractive force on drivability, handling, ease of operation, cost, and compatibility with the ABS system for different 4WD systems based on a the performed comparison between permanent and part time 4WD systems.

2.5.1 Mechanical differential (open and locked)

The conventional open differential has been the standard device for an automotive powertrain for a long time. This device is simple and effective in providing the necessary speed differential between the driving wheels during vehicle turning, Figure 2.26.
However, it cannot take full advantage of the available traction at the driving wheels on roads with different levels of adhesion. Consequently, the vehicle’s maximum driving power is limited to twice the torque at the low friction side of the driving wheels which means that any increase in the engine throttle makes the low friction side wheels to spin more, which would increase the slip sinkage in case of driving on an off-road terrain [60].

The ordinary bevel-gear differential can be presented as a set of planetary gears, the gear attached to the left half-axle can be considered as the sun gear with angular speed ($\omega_s$), the other gear attached to the right half-axle can be considered as the ring gear with an angular speed ($\omega_r$). The crown wheel is considered as the planet carrier with an angular velocity ($\omega_c$) [61]. In addition, the driving speed and torque along the lateral axis can be calculated as shown in Equation (2.9):

$$\omega_c = \left( \frac{\omega_r + \omega_s}{2} \right) \quad \text{and} \quad T_s = T_r = \left( \frac{T_c}{2} \right) \quad (2.9)$$

Where:
- $T_s$ .............. sun gear torque
- $T_r$ .............. ring gear torque
- $T_c$ .............. carrier gear torque
The locked differential has the ability to lock the two output together using an electric, pneumatic and hydraulic or frictional mechanism. This mechanism can be selected manually, and when the differential is locked, the wheels will have the same speed as shown in Equation (2.10).

\[
\omega_c = \omega_r = \omega_s \quad \text{and} \quad T_c = (T_s + T_r)
\]  

(2.10)

2.5.2 Clutch-Type LSD

Torque bias can be introduced only by adding friction clutch to the system as shown in Figure 2.27. The clutch type LSD has the same mechanical parts used in the open differential, but it has a set of clutches and springs.

![Figure 2.27 Clutch type limited slip differential [62]](image)

The clutches objective is to keep both wheels at the same rotating speed. The springs stiffness combined with the clutch friction regulates how much torque required to overcome the clutch resistance. The main disadvantage is the frictional clutches wear, which result in deterioration of differential performance. The biased torque based on the applied force in the friction disc is given by Equation (2.11).

\[
C_f = n \cdot f \cdot N \cdot \left( \frac{R_1 + R_2}{2} \right) \cdot \text{sgn} \left( \Delta \omega \right)
\]  

(2.11)

Where:

- \( n \) ............... the number of slipping surfaces.
- \( f \) ............... the clutch dynamic coefficient of friction.
- \( N \) ............... the normal load applied on the clutch disc.
R₁, R₂ ........... the outer and inner clutch disc radii.
Δω ............. the differential angular speed of the rotating discs.

2.5.3 Torsen LSD

Torsen differential has been involved in the powertrain driveline since 1983, and they are frequently used in high-performance AWD vehicles. Torsen (Torque sensing) differential is a purely mechanical device that perform as an open differential in the case of having same driving torque for both wheels as shown in Figure 2.28. While, in the case of losing traction of one of the wheels, the differential gears will use torque difference between the wheels to bind them together.

Figure 2.28 Torsen limited slip differentials [63]

Chocholek [63] studied and compared the operating principles and performance of the Torsen differentials with open differentials. In addition, Shih and Bowerman [64] compared the torque bias ratio and the efficiency of friction clutch based LSD, Torsen differentials and Lockable differential devices. It should be stated that LSD differential biases torque based on the available torque at the slipping wheel. Several differentials are designed with a preload to ensure that there will be some torque available to the wheel with good traction. In addition, this preload must be limited to prevent opposing handling effects in the vehicle, [65].

31
2.5.4 Visco-Lock Devices

Viscous coupling consists of a sealed housing and a splined hub. A set of thin plates are alternately connected to the housing and the hub. The intervening space between the plates and the housing is partially filled with high viscosity silicone oil as shown in Figure 2.29. If one set of wheels attempts to spin faster, the adjacent plates will rotate faster in comparison with the others. The fluid follows the faster plates and drag the slower plates with it. This action will add additional torque to the slower set of wheels.

![Viscous coupling diagram](image)

Figure 2.29 Viscous coupling characteristics [66]

Taureg and Herrmann [66] introduced several applications of viscous coupling in all-wheel drive vehicles. In addition, they developed a simple empirical equation to calculate the transmitted viscous torque \( T \) based on the speed difference \( \Delta n \) and the friction torque \( T_{FR} \) as shown in Equation (2.12):

\[
T = T_{FR} + a \cdot (\Delta n)^b
\]

Their method of calculation has been supported by several experimental measurements to predict the empirical constants \( a, b \) as shown in Equation (2.13):

\[
a = \frac{\log \left( \frac{T_2 - T_{FR}}{T_1 - T_{FR}} \right)}{\log \left( \frac{\Delta n_2}{\Delta n_1} \right)} \quad \text{and} \quad b = \frac{(T_2 - T_{FR})}{(\Delta n_2)^a}
\]

MOHAN ([67] and [68]) developed a theory to define the conditions necessary for initiating and sustaining STA in rotary viscous couplings. In addition, he verified the processes that
produce STA by proposing a sequence of events that are qualitatively viable and consistent with one another.

### 2.5.5 Electronically Controlled LSD

The ordinary controlled limited slip differential has limited capabilities due to its design while both traction and handling can be directly optimized by electronically controlling the differential’s output. In addition, if the vehicle is equipped with one of the advanced traction or braking control systems, the differential can resist by applying a torque to the wheel that is slowing down. This reduces the effectiveness of both the differential and the control systems. Optimal mobility and handling can easily be achieved by programming the differential to react differently to specific external conditions. Figure 2.30 shows the torque transfer range of an electronically controllable differential compared with an ordinary viscous coupling LSD [69].

![Figure 2.30 Passive versus electronically controlled LSD](image)

A Proportional-Integral-Differential (PID) controller is used to calculate the engagement force based on using various inputs to determine the vehicle operating condition. Inputs include individual wheel speeds, steering angle, throttle position, vehicle speed, brake status, transfer case mode, and temperature. The controller determines how much correction is needed based on the difference between the actual and theoretical wheel speeds.
Gradu [70] investigates different coupling solutions by employing a magnetic particle clutch, coupled, in a quasi-static torque split arrangement with a planetary gear system. The proposed arrangement increases the torque capacity of the coupling by directing only a fraction of the torque through the magnetic particle clutch.

The term “Torque vectoring” is defined as a driveline device capable of controlling both the magnitude and direction of torque to influence traction and vehicle dynamics. Such devices may be applied between wheels of the same axle or between axles in AWD applications. As torque vectoring can deliver power to any wheel instantly without using either the brakes or engine management. Torque vectoring depends on using advanced differentials that can distribute power to the wheels that have traction, which means that wheels do not need to be stopped.

Park and Kroppe [71] presented a novel torque vectoring called ‘Differential System Dynamic Trak’, which can be applied to both the inter-axle and the inter-wheel differential.
systems. The ‘Dynamic Trak’ has three multi-plate clutches as shown in Figure 2.31. The main clutch either offers a limited-slip or complete lock-up ability based on the driving conditions. The two exterior clutches regulate the torque delivered to the left or right shafts/wheels. An electronic control unit control the three clutches actively to manage the torque delivered to the two output shafts/wheels. The ‘Dynamic Trak’ can provide a maximum of 100% torque bias.

Mitsubishi Super All Wheel Control (S-AWC) integrates its Active Center Differential (ACD), Active Stability Control (ASC), Active Yaw Control (AYC), and ABS control as shown in Figure 2.32. The feedback control depends on a direct yaw moment control strategy that affects left-right torque vectoring and controls cornering maneuvers based on the desired yaw rate during different vehicle driving states. S-AWC succeeded in enhancing vehicle stability performance at different driving situations.

Ricardo’s Torque Vectoring technology used in Audi A6 4.2l V8 Quattro Avant allows the driving torque to be redistributed based vehicle speed and road conditions is shown in Figure 2.33. In addition, Debowski and Zardecki [73] developed a simplified model of center differential control containing: the equations, which describe the vehicle, the model structure, important values and parameters for the simulation. In addition, the authors described a concept of a simplified torque of split control system.
Jianhua Guo et al. [74] introduced a control system to enhance vehicle stability and controllability performance based on two control systems; Electronic Stability Program (ESP) and Variable Torque Distribution (VTD). The control strategy depends on the identifying the driving situations based on the vehicle slip angle as shown in Figure 2.34.

In the case of steady-state conditions, the VTD system is used, while ESP controller is primarily used for emergency maneuvers. To solve this difference, an individual subsystem should be activated depending on operating conditions as shown in Figure (2.35).
Qin Liu et al. [75] developed a torque-vectoring control strategy based on using a 2-DOF linear Parameter Varying (LPV) control to enhance the vehicle performance as shown in Figure 2.36.

Kaiser et al. [76] developed a torque vectoring control strategy using a PID and LQR controllers for longitudinal and lateral dynamics respectively for hybrid electric vehicle as shown in Figure 2.37. Simulation results presented enhancements in the vehicle performance.
2.6 Summary

The discussion above has covered the following aspects: mechanics of wheel-soil interaction, off-road vehicle simulation and various strategies to control torque distribution in multi-wheeled vehicles. These are critically analyzed and summarized as follows:

In the field of wheel soil mechanics and off-road vehicle simulation:
- Among the different reported approaches of wheel-soil mechanics, the finite element analysis approach, which is initiated by Perumpral et al. [23] and a lot of research has been done ending with the three-dimensional finite element model developed by Kaiming Xia [30].
- It is observed that some improvement could be achieved using the Mooney-Rivlin material for tire modeling. In addition, multi-pass effect for off-road vehicle dynamic simulation can be discussed besides the effects of the other parameters.

In the sections describing torque management devices and their effect on vehicle behavior, the following overall conclusions can be made:
- It is obvious that, the concepts of AWD-powertrains developed or under developments range from types activated manually, automatically, or permanently applied, with different kinds and degrees of differential locks. More sophisticated theories use data monitored from driving conditions to control the transmission properties using various electronic systems.
- The majority of research work carried out on torque vectoring differentials has been
carried out on slippery roads and using tire force representation mechanisms that has been based on on-road empirical maps as functions of vertical load, slip angle and coefficient of friction. At this point, it should be emphasized that, this approach should not be extended to off-road vehicles due to the complexity of the tire-soil interaction characteristics. However, more research still required investigating the effectiveness of using the active torque distribution strategy on off-road vehicles.
Chapter 3

FEA Tire and Soft Soil Modeling

3.1 Introduction

Tires are usually required to support the vehicle weight and cushion road surface irregularities to provide a comfortable ride to driver and passengers in ground vehicles. As a result, tire companies spend a lot of money to perform physical tests such as vertical stiffness, damping constant tests, cornering tests, and durability tests in order to inspect and enhance the tire performance. Therefore, many investigators have tried to construct another tire testing environment. Fortunately, current computer technology facilitates new tire model simulations that can be used to replicate most of the laboratory tire tests including that cannot be performed in the laboratory.

Many researchers investigated and developed several full FEA models since 1970’s that can reflect real operating conditions of tires. FEA tire models require high computational power and longer computational time. However, the FEA model method can predict tire performance and characteristics accurately and cost-effectively.

Kao and Muthukrishnan [77] developed and verified a simple tire test by using FEA software. For the first time, an FEA tire model incorporated geometry, material properties of different parts, layout, and other features of a commercial passenger car radial-ply tire P205/65R15. Kamoulakos and Kao verified the same setup as Kao and Muthukrishnan [78] by finite element software, PAM-SHOCK.

Chang and El-Gindy [79] developed tire-drum model to predict tire standing waves and tire free vibration modes. The determination of the tire in-plane free vibration modes was achieved by recording the reaction force histories of the tire axle at longitudinal and vertical directions when the tire rolling over a cleat on the road. The results showed good agreement when compared to more than ten previous studies.
In this chapter, a detailed, full three-dimensional off-road tire, 12.00R20 XML TL 149J, is modeled in association with nonlinear FEA software, PAM-CRASH. Tread patterns of the 4-groove truck tire developed by Chae [80] have been modified to represent the off-road tire tread. The developed FEA tire has an asymmetric tread pattern to prevent the tire from trapping and holding stones in the tread.

The developed FEA off-road tire model will be validated statically and dynamically by comparing predicted tire responses with available measurement data. For the validation of the FEA tire model, basic characteristic responses such as load-deflection curve, free vertical vibration mode and cornering characteristics will be virtually conducted.

### 3.2 Tire Structure, Components, and Materials

Tire generally can be defined as a flexible cord-rubber structure filled with compressed air. Rubber material has excellent flexibility properties to be used for building tire. While the rubber still need some flexible reinforcement to avoid extreme tire deformations upon loading. Mainly tire consists of a carcass, belts, beads, tread, and tread base as shown in Figure 3.1.

![Figure 3.1](image)

Figure 3.1 (a) Components of radial tire; and (b) tire section in detail
3.2.1 Carcass

The carcass sustains vertical load and absorbs ground vertical reactions. Therefore, the carcass should provide some requirements as strong anti-fatigue and stretching characteristics. Flexible but high modulus cords are embedded in a low modulus rubber matrix to form the carcass. The number of plies is determined by; tire type, tire size, inflation pressure, and loads in service.

There are two types of tires, bias-ply and radial-ply tires, as shown in Figure 3.2. In bias-ply tires, the reinforcing cords extend diagonally across the tire from bead to bead as shown in Figure 3.2(a). The bias-ply tires are used for bicycles, motorcycles, racing cars, aircraft, agricultural machinery, and some military machinery.

![Bias-ply tire and Radial-ply tire](image)

(a) Bias-ply tire  (b) Radial-ply tire

Figure 3.2 Typical Tire Constructions [81]

In radial-ply tires, the carcass cords are inclined in a radial direction as shown in Figure 3.2(b). The flexing of the carcass involves relatively small motion of the belt cords reducing the wiping motion between the tire and the road is small.

3.2.2 Belts

Belts are located between the carcass and the tread base. The belt restricts deformation of the carcass plies and provides additional stiffness to the tread. They also absorb the impacts due to road surface irregularities.
3.2.3 Tread and Tread Base

Tread is the most important part in the tire structure as it is the one in contact with the operating terrain in normal conditions. Generally, tread is built up from solid rubber with addition of carbon black to enhance the tire wear resistance during operation. Tread has another critical function as a protection to the remaining tire parts and provides the required friction with terrain to transmit driving, braking, and cornering forces. The primary function of the tread patterns is to transmit traction and can be considered as a group of ribs, grooves, rugs, and sipes. Figure 3.3 shows basic examples of these tread patterns of tires.

![Tread Patterns](image)

(a) Highway rib  (b) Highway rug  (c) On/off highway  (d) Off-highway

Figure 3.3 Basic Tread Patterns of Tires [82]

3.2.4 Beads

The beads reinforce the tire-rim assembly on the rim and prevent the tire slippage on the terrain. Hard drawn steel wires, flat or round, are grouped in a different arrangement based on the required strength and rigidity. Different bead groups used in radial tires are shown in Figure 3.4.

![Bead Configurations](image)

Figure 3.4 Bead configurations [82]
3.2.5 Aspect ratio

The section width is defined as the width of a new tire from sidewall to sidewall. Protective side ribs, bars, and decorations in the section width are not included, Figure 3.5. Distance from sidewall to sidewall is defined as the overall tire width. Distance from crown to the beads is defined as tire section height. Tire overall diameter is the outer diameter which is double the tire section height plus the rim diameter.

Different factors affects the tire performance characteristics like; load, inflation pressure, tread geometry, and compound and reinforcement properties. Modeling the rubber material in the simulation of the tire is based on using Moony-Rivlin coefficient that explained in details in Chae [84].

![Definitions of a tire cross-sectional shape](image)

Figure 3.5 Definitions of a tire cross-sectional shape [83]

3.3 FEA Tire Modeling

A four-groove Finite Element Analysis (FEA) truck tire, which was originally developed by Chae [84], has been developed to represent the off-road tire, 12.00R20 XML TL 149J. The Off-road 12.00R20 XML TL 149J tire has an asymmetric tread pattern to prevent the tire from trapping and holding stones in the tread. The complicated design was simplified to contain the fundamental elements while minimizing modeling and processing time.
Straight edges were used wherever possible to replace curves for the shape of the lugs and the grooves between the lugs. In addition, the max tread depth is modeled as 30 mm. Each lug was simplified as rectangular with angled sides. Solid tetrahedron elements with Mooney-Rivlin material properties were chosen for the tread. Figure 3.6 shows the final FEA model tread design.

Figure 3.6  Tread design as viewed from different views

The material property for two different layers (one for rubber and the other for steel) and the orientation of each layer is assigned appropriately to model the rubber tire carcass and steel belts. In this case, the belts run radially in the carcass from bead to bead.

The tire model is constructed using the following finite element components:

- 25 Parts,
- 9,920 nodes,
- 1,800 layered membrane elements,
- 13,280 solid elements,
- 120 beam elements,
- 25 material definitions, and
- One rigid body definition.

The advantages of this tire model are its computational efficiency and stability. Figure 3.7 shows the basic dimensions of the finite element tire model. Figure 3.8 shows a comparison between the actual tire and the FEA tire model. Technical data for the off-road tire model is shown in Table (3.1). The Mooney- Rivlin material properties for the solid tread and undertread elements is shown in Table (3.2).

Figure 3.7  Tire basic dimensions

Figure 3.8  Comparison of actual (a) and FEA model (b) combat vehicle tires
Table 3.1 FEA tire model technical data

<table>
<thead>
<tr>
<th></th>
<th>Max. Tread depth</th>
<th>Rim Width</th>
<th>Rim Weight</th>
<th>Tire Weight</th>
<th>Total Tire Weight</th>
<th>Overall Width</th>
<th>Overall Diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30 mm</td>
<td>283.4 mm</td>
<td>31.2 kg</td>
<td>55.3 kg</td>
<td>86.5 kg</td>
<td>309 mm</td>
<td>1130 mm</td>
</tr>
<tr>
<td></td>
<td>1.181 in</td>
<td>11.16 in</td>
<td>68.78 lbs.</td>
<td>121.92 lbs.</td>
<td>190.7 lbs.</td>
<td>12.16 in</td>
<td>44.48 in</td>
</tr>
</tbody>
</table>

Table 3.2 Mooney-Rivlin material properties for tread and undertread elements

<table>
<thead>
<tr>
<th>Tire Component</th>
<th>Under-tread</th>
<th>Tread</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m³)</td>
<td>596.2</td>
<td>693.3</td>
</tr>
<tr>
<td>1st Mooney-Rivlin coeff. (C₁₀)</td>
<td>0.51</td>
<td>0.67</td>
</tr>
<tr>
<td>2nd Mooney-Rivlin coeff. (C₀₁)</td>
<td>1.86</td>
<td>2.46</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.49</td>
<td>0.49</td>
</tr>
</tbody>
</table>

Figure 3.9 shows in detail the tire construction and the element types for each of the tire parts. These tire parts and materials include layered membrane elements for the tire carcass (grey) and Mooney-Rivlin elements for the bead fillers (purple), shoulders (yellow), tread (green), and the undertread (gray). The layered membrane elements allow for different material properties and orientations for three different layers in the same part.

In this case, the tire carcass includes the rubber tire carcass and the steel belts and cords. The steel cords run radially within the carcass from bead to bead. A circular beam element with a defined cross-sectional area and steel-like properties are chosen to represent the tire bead. The bead elements are attached directly to the bottom of the bead fillers. The complete tire is formed by copying and rotating this section 20 times.
3.4 FEA Tire Model Validation

The developed FEA off-road tire model needs to be validated by checking whether it shows real tire characteristics. For the validation, different tire simulations are conducted at various operating conditions (load, inflation pressure and slip angles). The results of the validation tests are compared with physical measurements.

3.4.1 Vertical stiffness

The tire model was subjected to extensive sensitivity analysis to tune up the mechanical properties of various material components in order to achieve reasonable load-deflection characteristics in comparison with measured data. In order to obtain the correct model characteristics, it is necessary to adjust the thickness (h), the Mooney-Rivlin coefficients of rubber compounds of the tread and under-tread ($C_{10}$ and $C_{01}$), and the modulus of elasticity (E) of both the sidewall and the under-tread of the tire model. The final tire model with adjusted material parameters under a 55 kN static load with an inflation pressure of 6 bars is shown in Figure 3.10.
Figure 3.10  FEA Off-road tire model under 55 kN load and 6 bar inflation pressure

Figure 3.11 shows the static deflection curve from actual tire data and the predicted results using the FEA tire model over a wide range of loads and inflation pressures. The actual tire data was obtained from published measurement data for a tire similar to the Off-road 12.00R20 XML TL 149J. Reasonable agreement can be observed, and this data is presented as model validation.

![Load-Deflection Curve](image)

Figure 3.11 Load - Deflection curve at different inflation pressure

### 3.4.2 First mode of vibration test

A tire and cleat-drum test was conducted to determine the first mode of vertical free vibration. Figure 3.12 shows the tire running on the virtual cleat drum test rig. A test was run for a tire load of 26.7 kN and an inflation pressure of 7.58 bars.

![Tire Running on Virtual Cleat Drum](image)
A Fast Fourier Transform (FFT) procedure was applied to the vertical reaction force at the tire spindle to obtain the frequency analysis shown in Figure 3.13. Peaks in the figure represent free vibration modes. The drum rotates at an angular velocity of 15 rad/sec, which results in about a 2.5 Hz excitation due to the cleat impact. The first peak shows this impact from around 1 to 4 Hz in the FFT. The second peak at approximately 46 Hz corresponds to the first vertical free vibration mode.

The available experimental data for the first vertical free vibration mode for passenger cars tires lies in the range of 60-80 Hz [85]. For the developed FEA off-road tire which has larger diameter and softer materials comparing to passenger car tires, its sidewalls will absorb more vibrations instead of transferring it to the tire center. So, it can be expected to have values lower than 60 Hz.
3.4.3 Cornering characteristics on flat surface

The cornering test is virtually conducted to examine the characteristic cornering performances of the FEA off-road tire model. The tire model is inflated at a pressure of 7.2 bars and loaded vertically up to 63.75 kN at the spindle of the tire model. Then, the tire model is steered at slip angles (α) up to 6°. A flat road is moving at constant speed of 10 km/h under the tire to rotate the tire model. Figure 3.14 shows the cornering simulation at slip angles of 2°, 4° and 6° and the lateral deformation of the tire at the contact area with the road surface.

The predicted cornering forces at different slip angles up to 6° at vertical loads of 15.94 kN, 31.88 kN, and 63.75 kN are presented in Figure 3.15 and compared with the published measurement data from the tire manufacturer. Aligning moment is one of the important cornering characteristic parameters. It is also predicted at various slip angles (α) and compared with published measurement data as seen in Figure 3.16.

Figure 3.14 Cornering simulation for the FEA off-road tire at slip angles of 2°, 4° and 6°
In the regions of slip angles from 0° to 6°, the predicted aligning moments show good agreement with the measurements at the lower two tire load cases. For slip angles (α)> 3°, considerable discrepancies are observed. The discrepancies are considered to be due to the differences in cross-sectional shapes, contact areas, and tread patterns between the FEA and real off-road tire.

3.4.4 Tire-slip characteristics

A tire and drum model was conducted to determine the normalized longitudinal force at different road friction coefficient (μ). A test was run for a tire load of 18 kN and an inflation pressure of 7.58 bars and road friction coefficient (μ) 0.2, 0.4, 0.6, and 0.8 as seen in
Figure 3.17. These results show good agreement with the published experimental data, [86], as the peaks reach the road friction coefficient value and then decrease with different rates depending on road friction coefficient, i.e. higher rates for higher friction coefficient.

![Normalized longitudinal force versus slip](image)

**Figure 3.17** Normalized longitudinal force versus slip

### 3.5 Tire Model Development in TruckSim

Non-linear tire look-up tables were developed based on FEA off-road tire simulation results and implemented in 8x8 combat vehicle model used for vehicle simulation using the multi-body dynamics code TruckSim. The predictions of vehicle handling characteristics and transient response during lane change test on rigid road at different vehicle speeds were compared with simulation results for same vehicle configuration using real experimental tire data. Simulation results are compared based on vehicle steering, yaw rates and accelerations. The published US Army validation criteria has been used to validate simulation results.

The vehicle model was tested during lane change maneuver at different speeds using the developed FEA tire model and the tire model based on experimental measurements. Figure 3.18 shows how a lane-change maneuver was performed.
Sample of the results of the simulation responses during the lane change maneuvers are given in the figures below. In this figures the vehicle speed was maintained approximately at 90 km/h as shown in Figure 3.19. The vehicle yaw rate and lateral acceleration are given in Figure 3.20 and Figure 3.21. As it can be seen excellent agreement between the measurement and simulation.

Figure 3.18 Lane change test course

Figure 3.19 Vehicle input speed versus time
The results obtained from set of tests at 30, 60 and 90 km/h were used to validate the model using US army validation criteria. Tables (3.3) to (3.5) show the simulation results for FEA and measured tire Kurtosis, Skewness and RMS at each speed. The FEA simulation values are within the US army criteria range. That means they are in excellent agreement with measured tire simulation results from point of the magnitude and the shape. It should be noted that the RMS is calculated only for the lateral acceleration as specified by US army.
Table 3.3 Validation of predicted and measured responses at 30 km/h

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<th>Yaw Rate</th>
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Table 3.4 Validation of predicted and measured responses at 60 km/h

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Table 3.5 Validation of predicted and measured responses at 90 km/h

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3.6 Soil Modeling

Soil modeling is a very complicated issue. Most soil is composed of a nonhomogeneous mixture of particles causing it to act in a different way from well-understood elastic plastic materials. Standards have been set for measuring soil properties and different soils have been characterized as possible.

According to The Idaho Association of Soil Conservation Districts, soil is classified based on the relative proportions of silt, sand, and clay [87]. The different soil types, which result from the various composition ratios, are shown by the triangle in Figure 3.22. For this thesis, the soil type being modeled is a “Clayey sand”.

A new type of soil was created using an elastic-plastic solid material (PAM-CRASH Material 1). The meshing is performed in PAM-CRASH by splitting a large solid block into 25mm by 25mm by 25mm elements. The tire-to-soil contact is defined as a node to segment contact with a friction coefficient of 0.8. The new soil modeled is a clayey soil. The material properties for this new soil are listed in Table (3.6). It should be noted that the material properties are chosen by using the mean value of the ranges given by the U.S. Department of Transportation, Federal Highway Administration.
Table 3.6 Material properties for the new soil

<table>
<thead>
<tr>
<th>Soil Type</th>
<th>Elastic Modulus, E (MPa)</th>
<th>Bulk Modulus, K (MPa)</th>
<th>Shear Modulus, G (MPa)</th>
<th>Yield Stress, Y (MPa)</th>
<th>Density, ρ (ton/mm³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clayey Soil</td>
<td>24</td>
<td>15</td>
<td>9</td>
<td>0.016</td>
<td>1.60E-09</td>
</tr>
</tbody>
</table>

Soil characteristics can be compared and validated by looking at the relationship between applied pressure and soil sinkage. This type of testing is discussed in detail by Wong [81]. The pressure-sinkage test is done by applying a known pressure over a circular plate placed on the soil and observing how far the plate sinks into the soil. The new soil is compared to the terrain values, given in Table 2.3 from Wong [81] using the Bekker formula, Equation (3.1).

\[
p = \left(\frac{k_c}{b} + k_v\right)z^n = k_z^n
\]  

(3.1)

Figure 3.23 shows the pressure-sinkage simulation of the soil with a rigid 15 cm circular plate. Figure 3.24 depicts the effect of normal pressure on tire sinkage. As can be seen in the figure a comparison between the predicted and previously published measurements confirm the validity of the proposed model.

Figure 3.23 Virtual measurements of pressure-sinkage using a 15 cm circular plate on the new soil with a pressure of 2 bars
3.7 FEA Tire Model on Soft Soil

After validation of the new FEA off-road tire model, as well as the soil model, it was used to evaluate tire performance on soft soil to facilitate the development of a set of look-up tables that can be used to represent the tire-soil interaction characteristics.

In addition, the FEA off-road tire models used to investigate the multi-pass behavior of the wheels running on soft terrain and its effect on vehicle mobility performance. The steering characteristics namely cornering forces and self-aligning moments versus slip angles of the multi-wheels were also predicted:
- The equivalent tire vertical stiffness on soft soil.
- The rolling resistance on soft soil for multi-wheels.
- The steering characteristics on soft soil for multi-axle steering.
- The longitudinal tire force-slip characteristics.

3.7.1 Tire vertical stiffness on soft soil

The off-road tire model was inflated at three different inflation pressures of 3.79, 7.58 and 11.37 bar and loaded at the spindle of the tire model on soil surface instead of the flat road surface as seen in Figure 3.25 and Figure 3.26. After the tire model reaches stability, the steady-state vertical deflection of the tire model and soil was recorded to calculate tire and soil stiffness as seen in Figure 3.27.
Figure 3.25  FEA off-road tires on soil surface

Figure 3.26  FEA off-road tires on soil surface simulation

Figure 3.27  Load - Sinkage curve under different inflation pressure
3.7.2 Rolling resistance on soft soil for multiple wheels

For the rolling resistance of multi-wheels (4 tires) running on soil surface, the off-road tire model is inflated at three different inflation pressures of 4, 6 and 8 bar and loaded with three vertical loads of 6, 18 and 48 kN at the spindle of the tire model on soil surface as seen in Figure 3.28.

![Figure 3.28 FEA off-road tires (4 tires) running on soil](image)

As soon as the tire model stabilizes, the steady-state tire model sinkage and rolling resistance coefficient are recorded to clarify the multi-pass effect on vehicle mobility performance as shown in Figure 3.29 and Figure 3.30 for tire inflation pressure 6 bars.

![Figure 3.29 FEA off-road tires (4 tires) sinkage on soil](image)
Figure 3.30  FEA off-road tires (4 tires) rolling resistance coefficient on soil

3.7.3 Steering characteristics on soft soil for multi-axle steering

For the steering characteristics on soil surface, the off-road tire model was developed for two steered tires with different steering angles (δ) and it will be tested for different inflation pressures (4, 6 and 8 bar) and vertical loads (6, 18 and 48 kN) at 15 km/h, as seen in Figure 3.31 and Figure 3.32.

Figure 3.31  FEA off-road tires (2 steered tires) on soil
As soon as the tire motion is stabilized, the steady-state longitudinal and lateral forces acting on the tire are recorded to calculate tire-cornering characteristics. Lateral forces and aligning moments acting on steered tires are presented in separate 3D surfaces for the first and second steering axles for each inflation pressure as seen from Figure 3.33 to Figure 3.36.

Figure 3.32 FEA off-road tires (2 steered tires) on soil

Figure 3.33 Lateral forces acting on the first FEA off-road tire on soil
(Inflation pressure 6 bars)
Figure 3.34  Lateral forces acting on the second FEA off-road tire on soil (Inflation pressure 6 bars)

Figure 3.35  Aligning moment acting on the first FEA off-road tire on soil (Inflation pressure 6 bars)
3.7.4 *Longitudinal tire force-slip characteristics on soft soil*

Figure 3.37 shows the traction test of the off-road tire on soft soil to determine the longitudinal slip characteristics. In this test, two longitudinal tires under different inflation pressures (4, 6 and 8 bar) and vertical tire loads (6, 18 and 48 kN), are rapidly accelerated to a rotational velocity of 30 km/hr and is allowed to roll forward. Initially the tires longitudinal slip were nearly 100% slip before the tires began to move forward due to the excessive tractive torque applied at the center of the tires. Then as the tires move forward, the slip is reduced gradually and the slip approached about 20% as the tires asymptotically near a linear velocity of 30 km/h.

Figure 3.36 Aligning moment acting on the second FEA off-road tire on soil

(Inflation pressure 6 bars)
Figure 3.38 First tire normalized longitudinal force-slip characteristics on soil
(Inflation pressure 6 bar)

Figure 3.38 and Figure 3.39 show sample result of the predicted normalized force at different slip percentages for both first and second tire at inflation pressure 6 bar and different vertical loads (6, 18, 48 kN).

Figure 3.39 Second tire normalized longitudinal force-slip characteristics on soil
(Inflation pressure 6 bar)
Chapter 4

Multi-Wheeled Combat Vehicle Modeling and Validation

4.1 Introduction

Validated vehicle models can be comprehensively used instead of field experimental testing especially for specific severe maneuvers. The developed vehicle models need to be validated for the acceptance and confident of the simulation results [88]. The variations between the virtual and the real test can be attributed to many issues such as virtual modeling, programming, and experimental data quality during experimental tests. Experimental testing has many causes of variation due to randomness and human error. These sources are absent in the simulation models and can contribute towards the inconsistency in results.

In this study, once the experimental test data and simulation results are compared, the virtual model could be tuned depending upon the varying performance parameter. Virtual vehicle model should be tuned at the component level and care should be taken that the comparison is made at the linear as well as the non-linear range. Time domain and frequency domain correlation are recommended for steady and transient responses respectively [89]. In addition, LeBlanc and El-Gindy [90] offered the endings of an experimental and theoretical investigations on the self-steering axle effect on the directional stability of straight truck. The field tests were aimed at generating steady-state handling diagrams to evaluate the directional behavior under different operating conditions.

El-Gindy and Mikulcik [91] investigated the yaw rate response sensitivity of a three-axle single-unit heavy vehicle to sinusoidal steering input. The frequency response technique and first order standard and logarithmic sensitivity functions were applied which present a significant source of information for the researchers for further development in control systems.

Hillegass et al. [92] introduced an approach that can be used for evaluating and validating
a multi-wheeled combat vehicle model based on comparing simulation results with the actual field experimental measurements. The performed validation procedure was established on J-Turn and double lane change simulations at three speeds and one tire pressure. Authors defined a validation criteria based on performing some statistical measures; Kurtosis, Skewness and Root Mean Square. Furthermore, Hillegass et al. [93] extended the presented strategy for validating the multi-wheeled combat vehicle models to include its vertical dynamic performance based on vehicle weights, dimensions, tires and suspension characteristics. Authors compared the predicted vertical dynamics responses with the field experimental results for different speeds on different road.

The dynamic performance of multi-wheeled off-road vehicles on rigid and soft terrain was developed in multi-body dynamics software and validated by utilizing the measured data. Non-linear tire look-up tables for rigid and soft terrain were obtained from the developed three-dimensional non-linear FEA off-road tire model in PAM-CRASH. The predictions of the vehicle handling characteristics and transient response during a lane change on rigid road at different vehicle speeds were compared with field tests results. Measured and predicted results are compared based on vehicle steering, yaw rates and accelerations. Published US Army validation criteria have been used to validate simulations. The combat vehicle model was used to study vehicle lane-change maneuverability on rigid and soft terrain at different speeds and powertrain configurations.

4.2 Vehicle Modeling and Validation

![Actual vehicle configuration](image1.png) ![Simulation model](image2.png)

Figure 4.1 Actual vehicle configuration [94] (a) and the simulation model (b)
The actual vehicle configuration and simulation model of a multi-wheeled combat vehicle are shown in Figure 4.1. The vehicle is equipped with four axles, which can be operated in either 4WD or 2WD. The front two axles are steering axles ($\delta_1$ and $\delta_2$). The vehicle is equipped with independent suspensions. The vehicle model consists of 22 Degrees of freedom, namely pitch, yaw and roll of the vehicle sprung mass and spin and vertical motions of each wheel of the eight wheels.

### 4.2.1 Vehicle modeling

The vehicle model has been developed using TruckSim and based on the actual vehicle configuration for multi-wheeled combat vehicle design parameters, including non-linear tire/terrain interaction characteristics in form of look-up tables for both rigid and soft terrain. The tire/soft terrain characteristics were obtained from FEA off-road tire models developed using PAM-CRASH as explained in Chapter 3.

As it can be seen in Figure 4.2, the vehicle is equipped with two front steering axles. The individual steering angle according to Ackerman condition, for a specific turning radius, can be determined by plotting perpendicular lines on the four steering wheels and the rear two axles at their geometric center.

![Figure 4.2 Ackerman steering of eight-wheel vehicle with multi-axle steering](image)

69
The inner and outer steering angles ($\delta_i$ and $\delta_o$, respectively) for the first and second axles have been approximated and calculated using Equation (4.1).

$$\cot \delta_o - \cot \delta_i = \frac{B}{L}$$

(4.1)

Figure 4.3 shows the relationship between gearbox output and the steering angle at ground of each road wheel of the first and second axle, at the nominal suspension position and in the absence of tire forces, without accounting for speed effects.

The developed combat vehicle model is used to study vehicle maneuverability on rigid and soft terrain at different speeds and powertrain configurations (8x4 and 8x8). The predictions of the vehicle handling characteristics and transient response during a lane change on rigid road at different vehicle speeds were compared with field tests results. Measured and predicted results are compared based on vehicle steering, yaw rates and accelerations. Published US Army validation criteria have been used to validate the simulation results [93]. At each measurement location, the model predicted RMS value should agree with the measured RMS acceleration within ±10%. The model time domain data and measured time domain data Skewness, and kurtosis values should agree within ± 50% of the measured data values to provide a comparison on wave shape in the time domain. The Kurtosis, Skewness and RMS are defined as follows:
• **Kurtosis**, the measure of the peaks of the random data and was chosen as a statistical parameter because it is an excellent indicator of extreme values and how they relate to the general data. It is extremely useful in picking out wild points.

\[
Kurtosis = \frac{\sum (x_i - \mu)^4}{N\sigma^4} - 3
\]  

(4.2)

Where,
- \( X_i \) ............ is the \( i^{th} \) value
- \( \mu \) ............ is the mean
- \( N \) ............ number of data points
- \( \sigma \) ............sample standard deviation

• **Skewness**, a measure of the probability distribution of random variables, Skewness is a measure of one-sidedness.

\[
Skewness = \frac{\mu_3}{\sigma^3}
\]  

(4.3)

• **Root Mean Square (RMS)** is the magnitude of varying quantity of data. It is relatively insensitive to wild points, and it does not provide an indication of variation about the mean.

\[
RMS = \sqrt{\frac{1}{N} \sum_{i=1}^{N} x_i^2} = \sqrt{\frac{x_1^2 + x_2^2 + \ldots + x_N^2}{N}}
\]  

(4.4)

4.2.2 Vehicle model validation

The vehicle model was tested in four different test courses, Double Lane Change, Constant Step Slalom, J-Turn with 8x4 powertrain drive and Turning circle test with two different powertrain configurations (8x4 and 8x8). All the test courses have been conducted on rigid road with tire inflation pressure of 0.72 MPa. Table 4.1 shows the test courses and vehicle speeds used to validate the vehicle model. In the following sections, sample of the performed validation tests of each test course will be demonstrated.
### Table 4.1 Test Courses Matrix

<table>
<thead>
<tr>
<th>No.</th>
<th>Test Course</th>
<th>Vehicle Speed</th>
<th>Additional Test Data</th>
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<tbody>
<tr>
<td>1</td>
<td>Double Lane Change (NATO AVTP-1 03-160W)</td>
<td>40, 53, 72, 81 km/h and maximum</td>
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<tr>
<td>2</td>
<td>Constant Step Slalom (NATO AVTP-1 03-30)</td>
<td>40, 53, 60 km/h and maximum</td>
<td>30 m cone spacing</td>
</tr>
<tr>
<td>3</td>
<td>J-Turn (75ft radius)</td>
<td>30, 35, 40, 45, 50 km/h</td>
<td>----------</td>
</tr>
<tr>
<td>4</td>
<td>Turning Circle (4x8 &amp; 8x8)</td>
<td>Crawling</td>
<td>Maximum cramping angle = 34 deg</td>
</tr>
</tbody>
</table>

#### 4.2.2.1 Double Lane Change (NATO AVTP-1 03-160W)

This maneuver is designed to examine the vehicle transient response. The vehicle was tested during Lane-change maneuver at different speeds; Figure 4.6 shows schematic drawing of the lane change test course.

![Figure 4.4 NATO (AVTP 03-160) lane change test course [95]](image)

(a) **NATO Lane Change - 53 km/h**

This test was performed using the simulation speed as shown in Figure 4.5 which is simulated to replicate what was measured during the experimental testing. As can be seen, the simulation speed and measured speed are constant of the approximate value of 53 km/h. The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 4.6. The vehicle lateral acceleration and yaw acceleration are given in Figure 4.7 and Figure 4.8.
Figure 4.5 Vehicle speed time history

Figure 4.6 Vehicle steering angle time history for measured and simulation tests at a speed of 53 km/h

Figure 4.7 Vehicle lateral acceleration time history at a speed of 53 km/h
US Army validation criteria (section 4.2.1) has been used to validate the simulation results of this test. Table 4.2 shows that the lateral acceleration validation criteria are found to be within the recommended range (minimum and maximum values) of the Kurtosis, Skewness and RMS. In case of the yaw acceleration, the skewness and kurtosis values are found to be within the recommended range, while the predicted RMS value found to be outside the recommended range, due to the high noise level of the supplied measured data.

Table 4.2 Validation results for left lane change at 53 km/h

<table>
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<tr>
<th></th>
<th>Lateral Acceleration</th>
<th>Yaw Acceleration</th>
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<tr>
<td></td>
<td>US Army Validation Criteria</td>
<td>Min.</td>
</tr>
<tr>
<td>Kurtosis</td>
<td>Measured (3.081)</td>
<td>Simulation (2.426)</td>
</tr>
<tr>
<td>Skewness</td>
<td>Measured (1.208)</td>
<td>Simulation (0.747)</td>
</tr>
<tr>
<td>RMS</td>
<td>Measured (0.369)</td>
<td>Simulation (0.031)</td>
</tr>
<tr>
<td>Kurtosis</td>
<td>Measured (11.265)</td>
<td>Simulation (8.006)</td>
</tr>
<tr>
<td>Skewness</td>
<td>Measured (2.715)</td>
<td>Simulation (2.418)</td>
</tr>
<tr>
<td>RMS</td>
<td>Measured (199.168)</td>
<td>Simulation (172.325)</td>
</tr>
</tbody>
</table>
(b) **NATO Lane Change - 85 km/h**

This test was performed using the simulation speed as shown in Figure 4.9 which is simulated to replicate what was measured during the experimental testing. As can be seen, the simulation speed and measured speed are constant of the approximate value of 85 km/h. The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 4.10. The vehicle lateral acceleration and yaw acceleration are given in Figure 4.11 and Figure 4.12. As it can be seen there is a good agreement between the measurement and simulation in both shape and peaks’ locations.

![Vehicle speed time history](image1)

*Figure 4.9 Vehicle speed time history*

![Vehicle steering angle time history](image2)

*Figure 4.10 Vehicle steering angle time history for measured and simulation tests at a speed of 85 km/h*
Figure 4.11 Vehicle lateral acceleration time history at a speed of 85 km/h

Figure 4.12 Vehicle yaw acceleration time history at a speed of 85 km/h

Table 4.3 Validation results for left lane change at 85 km/h

<table>
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<tr>
<th></th>
<th>Lateral Acceleration</th>
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<th>Yaw Acceleration</th>
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</thead>
<tbody>
<tr>
<td>Skewness</td>
<td>1.045</td>
<td>0.842</td>
<td>0.523</td>
</tr>
<tr>
<td>RMS</td>
<td>0.076</td>
<td>0.053</td>
<td>0.068</td>
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</table>
US Army validation Criteria (section 4.2.1) has been used for validation. Table 4.3 shows that the lateral acceleration validation criteria found to be within the recommended range of the Kurtosis and Skewness, while the predicted RMS value found to be outside the recommended range due to the high noise level of the supplied measured data. In the case of the yaw acceleration, the Kurtosis and skewness values found to be within the recommended range, while the predicted RMS value found to be outside the recommended range. In addition, the simulation results are compared with additional eight different tests. The calculated Skewness and Kurtosis values found to be within the recommended range. While the model prediction of RMS values of the lateral acceleration and yaw acceleration did not agree with some of the measured ones within ±10% due to the high noise level of the measured lateral acceleration and yaw acceleration data.

**4.2.2.2 Constant Step Slalom (NATO AVTP-1 03-30)**

This maneuver is designed to examine the vehicle transient response. The vehicle was tested during constant step slalom maneuver at different speeds. Figure 4.13 shows schematic drawing of the constant step slalom test course.

Figure 4.13 NATO (AVTP-1 03-30) constant step slalom test course [95]

(a) 30m slalom 40 km/h

This test was performed using the simulation speed as shown in Figure 4.14, which is simulated to replicate what was measured during the experimental testing. As can be seen, the simulation speed and measured speed are constant of the approximate value of 40 km/h. The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 4.15. The vehicle lateral acceleration and yaw acceleration are given in
Figure 4.14 Vehicle speed time history

Figure 4.15 Vehicle steering angle time history for measured and simulation tests

Figure 4.16 Vehicle lateral acceleration time history at a speed of 40 km/h
US Army validation Criteria (section 4.2.1) has been used for validation. Table 4.4 shows that the lateral acceleration validation criteria found to be within the recommended range of the Kurtosis and RMS while the predicted Skewness found to be outside the recommended range. In the case of the yaw acceleration, the Kurtosis and Skewness values found to be within the recommended range, while the predicted RMS values found to be outside the recommended range.

![Vehicle yaw acceleration time history at a speed of 40 km/h](image)

Figure 4.17 Vehicle yaw acceleration time history at a speed of 40 km/h

Table 4.4 Validation results for constant step slalom at 40 km/h

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<td>Min.</td>
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<td>Max.</td>
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<tr>
<td></td>
<td></td>
<td>Measured</td>
<td>4.017</td>
<td>1.739</td>
<td>2.008</td>
<td>6.025</td>
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<tr>
<td></td>
<td></td>
<td>Skewness</td>
<td>1.213</td>
<td>0.242</td>
<td>0.606</td>
<td>1.819</td>
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<td></td>
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<td>RMS</td>
<td>0.057</td>
<td>0.052</td>
<td>0.051</td>
<td>0.062</td>
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<td>US Army Validation</td>
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<td>Measured</td>
<td>16.496</td>
<td>11.516</td>
<td>8.248</td>
<td>24.745</td>
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<tr>
<td></td>
<td></td>
<td>Kurtosis</td>
<td>3.031</td>
<td>2.417</td>
<td>1.515</td>
<td>4.546</td>
<td></td>
<td></td>
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<td></td>
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<td>Skewness</td>
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</tbody>
</table>

79
(b) 30m slalom 60 km/h

This test was performed using the simulation speed as shown in Figure 4.18, which is simulated to replicate what was measured during the experimental testing. As can be seen, the simulation speed and measured speed are constant of the approximate value of 60 km/h. The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 4.19. The vehicle lateral acceleration and yaw acceleration are given in Figure 4.20 and Figure 4.21.

Figure 4.18 Vehicle speed time history

Figure 4.19 Vehicle steering angle time history for measured and simulation tests
US Army validation criteria (section 4.2.1) has been used for validation. Table 4.5 shows that the lateral acceleration validation criteria found to be within the recommended range of the Kurtosis and Skewness, while the predicted RMS found to be outside the recommended range, but still very close to it. In the case of the yaw acceleration, the Kurtosis, Skewness and RMS values found to be outside the recommended range.
Table 4.5 Validation results for constant step slalom at 60 km/h

<table>
<thead>
<tr>
<th></th>
<th>Lateral Acceleration</th>
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<tbody>
<tr>
<td></td>
<td></td>
<td>US Army Validation Criteria</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Measured</td>
<td>Simulation</td>
<td>Min.</td>
</tr>
<tr>
<td>Kurtosis</td>
<td>1.766</td>
<td>1.889</td>
<td>0.883</td>
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<tr>
<td>Skewness</td>
<td>0.466</td>
<td>0.472</td>
<td>0.233</td>
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<tr>
<td>RMS</td>
<td>0.187</td>
<td>0.092</td>
<td>0.169</td>
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<table>
<thead>
<tr>
<th></th>
<th>Yaw Acceleration</th>
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<th></th>
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<tbody>
<tr>
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<td>US Army Validation Criteria</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Measured</td>
<td>Simulation</td>
<td>Min.</td>
</tr>
<tr>
<td>Kurtosis</td>
<td>27.132</td>
<td>1.889</td>
<td>13.566</td>
</tr>
<tr>
<td>Skewness</td>
<td>4.521</td>
<td>0.472</td>
<td>2.260</td>
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<tr>
<td>RMS</td>
<td>218.464</td>
<td>302.704</td>
<td>196.618</td>
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4.2.2.3 J-Turn (22m radius)

(a) 75ft J turn - 25 km/h

This test was performed using the simulation speed as shown in Figure 4.22 which is simulated to replicate what was measured during the experimental testing. As can be seen, the simulation speed and measured speed are constant of the approximate value of 25 km/h. The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 4.23. The vehicle lateral acceleration and yaw acceleration are given in Figure 4.24 and Figure 4.25.

![Vehicle speed time history](image1)

Figure 4.22 Vehicle speed time history
Figure 4.23 Vehicle steering angle time history for measured and simulation tests

Figure 4.24 Vehicle lateral acceleration time history at a speed of 25 km/h

Figure 4.25 Vehicle yaw acceleration time history at a speed of 25 km/h
US Army validation criteria (section 4.2.1) has been used for validation. Table 4.6 shows that the lateral acceleration validation criteria found to be within the recommended range of the Kurtosis and RMS while the predicted Skewness value found to be outside the recommended range. In the case of the yaw acceleration, the Skewness and RMS found to be within the recommended range, while the predicted Kurtosis value found to be outside the recommended range but still very close to it.

Table 4.6 Validation results for right J Turn at 25 km/h

<table>
<thead>
<tr>
<th></th>
<th>Lateral Acceleration</th>
<th>US Army Validation Criteria</th>
</tr>
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<tbody>
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<td>Measured</td>
<td>Simulation</td>
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<tr>
<td>Kurtosis</td>
<td>1.705</td>
<td>1.450</td>
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<tr>
<td>Skewness</td>
<td>-0.008</td>
<td>-0.401</td>
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<tr>
<td>RMS</td>
<td>0.037</td>
<td>0.035</td>
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</table>

<table>
<thead>
<tr>
<th></th>
<th>Yaw Acceleration</th>
<th>US Army Validation Criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Measured</td>
<td>Simulation</td>
</tr>
<tr>
<td>Skewness</td>
<td>2.824</td>
<td>1.997</td>
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<tr>
<td>RMS</td>
<td>12.444</td>
<td>11.926</td>
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</tbody>
</table>

(b) 75ft J turn - 45 km/h

This test was performed using the simulation speed as shown in Figure 4.26 which is simulated to replicate what was measured during the experimental testing. As can be seen, the simulation speed and measured speed are constant of the approximate value of 45 km/h. The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 4.27. The vehicle lateral acceleration and yaw acceleration are given in Figure 4.28 and Figure 4.29.
Figure 4.26 Vehicle speed time history

Figure 4.27 Vehicle steering angle time history for measured and simulation tests

Figure 4.28 Vehicle lateral acceleration time history at a speed of 45 km/h
Figure 4.29 Vehicle yaw acceleration time history at a speed of 45 km/h

Table 4.7 Validation results for right J Turn at 45 km/h

<table>
<thead>
<tr>
<th></th>
<th>Lateral Acceleration</th>
<th>Yaw Acceleration</th>
</tr>
</thead>
<tbody>
<tr>
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<td>US Army Validation</td>
<td>US Army Validation</td>
</tr>
<tr>
<td></td>
<td>Criteria</td>
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</tr>
<tr>
<td>Measured</td>
<td>Simulation</td>
<td>Min.</td>
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<tr>
<td><strong>Kurtosis</strong></td>
<td>1.924</td>
<td>1.651</td>
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<td><strong>Skewness</strong></td>
<td>0.286</td>
<td>0.215</td>
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<td><strong>RMS</strong></td>
<td>0.236</td>
<td>0.224</td>
</tr>
<tr>
<td></td>
<td>11.997</td>
<td>12.238</td>
</tr>
<tr>
<td><strong>Skewness</strong></td>
<td>2.997</td>
<td>3.157</td>
</tr>
<tr>
<td><strong>RMS</strong></td>
<td>227.727</td>
<td>195.675</td>
</tr>
</tbody>
</table>

US Army validation criteria (section 4.2.1) has been used for validation. Table 4.7 shows that the lateral acceleration validation criteria found to be within the recommended range of the Kurtosis, Skewness and RMS. In the case of the yaw acceleration, the Kurtosis and Skewness found to be within the recommended range, while the predicted RMS value found to be outside the recommended range but still very close to it. In addition to the demonstrated results, the simulation results are compared with additional eight different tests. The calculated Skewness and Kurtosis values found to be within the recommended
range. While the model prediction of RMS values of the lateral acceleration and yaw acceleration did not agree with some of the measured ones within ±10% due to the high noise level of the measured lateral acceleration and yaw acceleration data.

4.2.2.4 Turning Circle (8x8 & 8x4)

(a) Turning Circle (8x4) Right

This test was performed using the simulation speed as shown in Figure 4.30 which is simulated to replicate what was measured during the experimental testing (crawling speed). The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 4.31. The vehicle lateral acceleration and yaw acceleration are given in Figure 4.32 and Figure 4.33.

![Vehicle speed time history](image1)

Figure 4.30 Vehicle speed time history

![Steering angle time history](image2)

Figure 4.31 Vehicle steering angle time history for measured and simulation tests
Figure 4.32 Vehicle lateral acceleration time history

Table 4.8 Validation results for turning circle (8x4) _right

<table>
<thead>
<tr>
<th></th>
<th>Measured</th>
<th>Simulation</th>
<th>Min.</th>
<th>Max.</th>
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<tbody>
<tr>
<td><strong>Lateral Acceleration</strong></td>
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<td>US Army Validation Criteria</td>
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<tr>
<td>Kurtosis</td>
<td>1.932</td>
<td>1.832</td>
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<td>Skewness</td>
<td>0.546</td>
<td>0.535</td>
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<tr>
<td>RMS</td>
<td>0.091</td>
<td>0.094</td>
<td>0.082</td>
<td>0.101</td>
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</table>

<table>
<thead>
<tr>
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<th>Min.</th>
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</tr>
</thead>
<tbody>
<tr>
<td><strong>Yaw Acceleration</strong></td>
<td></td>
<td></td>
<td>US Army Validation Criteria</td>
<td></td>
</tr>
<tr>
<td>Kurtosis</td>
<td>30.746</td>
<td>21.334</td>
<td>15.373</td>
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<tr>
<td>Skewness</td>
<td>4.890</td>
<td>3.887</td>
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<td>RMS</td>
<td>23.239</td>
<td>10.847</td>
<td>20.915</td>
<td>25.563</td>
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</table>

US Army validation criteria (section 4.2.1) has been used for validation. Table 4.8 shows that the lateral acceleration validation criteria found to be within the recommended range of the Kurtosis, Skewness and RMS. In the case of the yaw acceleration, the Kurtosis and Skewness found to be within the recommended range while the predicted RMS value found to be outside the recommended.
(b) Turning Circle \((8\times8)\) left and Right

This test was performed using the simulation speed as shown in Figure 4.34 which is simulated to replicate what was measured during the experimental testing (crawling speed). The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 4.35. The vehicle lateral acceleration and yaw acceleration are given in Figure 4.36 and Figure 4.37.
Figure 4.35 Vehicle steering angle time history for measured and simulation tests

Figure 4.36 Vehicle lateral acceleration time history

Figure 4.37 Vehicle yaw acceleration time history
US Army validation criteria (section 4.2.1) has been used for validation. Table 4.9 shows that the lateral acceleration validation criteria found to be within the recommended range of the Kurtosis, Skewness and RMS. In the case of the yaw acceleration, the Skewness, Kurtosis and RMS found to be outside the recommended range.

In addition to the demonstrated results, the simulation results are compared with additional two different tests. The calculated Skewness and Kurtosis values were found to be within the recommended range. The model prediction of RMS values of the lateral acceleration and yaw acceleration did not agree with some of the measured ones within ±10% due to the high noise level of the measured lateral acceleration and yaw acceleration data.

<table>
<thead>
<tr>
<th>Table 4.9 Validation results for turning circle (8x8) _left &amp; right</th>
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<td>RMS</td>
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<tr>
<td><strong>Yaw Acceleration</strong></td>
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<td>US Army Validation Criteria</td>
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<tr>
<td>Measured</td>
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<tr>
<td>Skewness</td>
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<tr>
<td>RMS</td>
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</table>

4.3 Summary

Based on the tests simulation results for the four test maneuvers, twenty-seven runs have been performed and validated in comparison with the experimental data provided by GDLS-Canada. It should be mentioned here that the measured vehicle speed and steering wheel angle time history have been used as input parameters for all the test maneuvers to increase the accuracy of the simulation output results. In addition, the vehicle parameters
delivered by GDLS-Canada have been used for developing a full TruckSim combat vehicle model.

Obtained validation results based on the published US Army Criteria (section 4.2.1) led to the following conclusions:

- 81% of the calculated Kurtosis of the predicted lateral accelerations of all the tests are passed the US Army validation criteria.
- 52% of the calculated Skewness of the predicted lateral accelerations of all the tests are passed the US Army validation criteria.
- 33% of the calculated RMS of the predicted lateral accelerations of all the tests are passed the US Army validation criteria based on ±10% and 51% based on ±20%.
- 59% of the calculated Kurtosis of the predicted yaw accelerations of all the tests are passed the US Army validation criteria.
- 88% of the calculated Skewness of the predicted yaw accelerations of all the tests are passed the US Army validation criteria.
- 22% of the calculated RMS of the predicted yaw accelerations of all the tests are passed the US Army validation criteria based on ±10% and 41% based on ±20%.

Finally, based on the above conclusions the developed vehicle model with its current design parameters and tire characteristics is considered to be suitable for the design of an active torque distribution control system for both 8X4 and 8x8 powertrain configurations. It has been demonstrated that it will enhance the multi-wheeled combat vehicle maneuverability and mobility performance on both rigid and soft terrain.
Chapter 5

Active Torque Distribution Control System

5.1 Introduction

In passenger vehicles, the rapidly increasing applications of all-wheel drive (AWD) requires the development of vehicles not only with higher traction capability but also with better maneuverability. Although improving traction performance is of prime concern for off-road vehicle applications, handling behavior is an important aspect of new vehicles, which requires the capability to undergo high lateral accelerations, while maintaining a proper level of directional stability. The desired increase in mobility must be reached without making any compromises regarding safety or ease of operation or driver comfort. It is expected that, the performance of off-road vehicles depend not only on the total tractive effort available by the power plant, but also on its distribution between the driving wheels.

The advancement in the field of road vehicles is the use of active torque distribution control systems to fulfill the function of torque split and transfer among all the driving wheels. The primary objective of this chapter is to develop an active torque distribution control strategy for a multi-wheeled combat vehicle with (8x4) powertrain configuration. The developed vehicle model, presented in chapter 4, is used to investigate different control strategies for torque distribution on rigid road at different operating conditions.

An active torque distribution control strategy will be presented in the following sections, and comparison between the vehicle directional stability and performance with and without the developed control strategy will be performed and discussed.

5.2 Vehicle Dynamics Control

The primary objective of vehicle dynamics control (VDC) system is to enhance vehicle directional stability based on limiting the deviation of the vehicle states from its desired states by utilizing different types of actuators; engine management, Braking system and vectoring differentials as shown in Figure 5.1.
5.2.1 Actual vehicle responses

The actual vehicle responses can be obtained based on real-time measurements using different sensors for; wheel speed, yaw rate, steering angle and lateral acceleration. The non-linear vehicle model developed and validated in chapter 4 is utilized to generate the actual vehicle responses required for the proposed control strategy.

5.2.2 Desired vehicle responses

Simplified vehicle model could be used to obtain the desired vehicle responses based on the driver responses; steering input, torque and braking inputs [97]. In this research, the desired responses are obtained from a developed four-axle vehicle bicycle model and the considered vehicle states are the yaw rate and lateral acceleration. The primary goal of the proposed control system is to minimize the driver required action in difficult driving situations. Accordingly, the driver has been excluded from all analysis of the control systems. The state space representation of the bicycle model used to generate desired or target responses as given by [98] and [99]. In most cases, the desired responses of the state variables are chosen from steady state values of the bicycle model. For a given road wheel
steering angle $\delta$, the desired states are defined as follows:

**The slip angles:**

First axle:

$$\alpha_1 = \delta_1 - \tan^{-1}\left[\frac{V + ar}{u}\right] \quad (5.1)$$

Assume small slip angles: $\tan(\alpha) = \alpha$ and $\cos(\alpha) = 1$

$$\alpha_1 = \delta_1 - \left[\frac{V + ar}{u}\right] \quad (5.2)$$

Second axle:

$$\alpha_2 = \delta_2 - \left[\frac{V + br}{u}\right] \quad (5.3)$$

For simplification ($\delta_a$, $\alpha_a$) will be used to present the first and second axle as follows:

$$\delta_a = \frac{\delta_1 + \delta_2}{2} \quad (5.4)$$

$$\alpha_a = \frac{\alpha_1 + \alpha_2}{2} \quad (5.5)$$

$$\alpha_a = \delta_a - \left[\frac{V + a_ar}{u}\right] \quad (5.6)$$

Third axle:

$$\alpha_3 = -\left[\frac{V - cr}{u}\right] \quad (5.7)$$

Fourth axle:

$$\alpha_4 = -\left[\frac{V - dr}{u}\right] \quad (5.8)$$

Cornering forces calculations:

$$F_ya = C_{\alpha_a} \alpha_a \quad (5.9)$$

Where:

$$C_{\alpha_a} = \frac{c_{\alpha_1} + c_{\alpha_2}}{2} \quad (5.10)$$
In addition, for third and fourth axle:

\[ F_{y3} = C\alpha_3 \alpha_3 \]  
\[ F_{y4} = C\alpha_4 \alpha_4 \]  

**Equation of motion for the model:**

The lateral and yaw equations of motion can be expressed as follows:

\[ m(\dot{V} + ur) = F_{ya} + F_{y3} + F_{y4} \]  
\[ I\dot{r} = a_F y_a - cF_{y3} - dF_{y4} \]
Substituting cornering forces in the equation of motion:

\[ m(\ddot{V} + ur) = C_{\alpha a} \delta_a - C_{\alpha a} \left[ \frac{V + a\alpha r}{u} \right] - C_{\alpha 3} \left[ \frac{V - cr}{u} \right] - C_{\alpha 4} \left[ \frac{V - dr}{u} \right] \]  \hspace{1cm} (5.15)

\[ m(\ddot{V} + ur) = -(C_{\alpha a} + C_{\alpha 3} + C_{\alpha 4}) \left[ \frac{V}{u} \right] - (a_a C_{\alpha a} - c C_{\alpha 3} - d C_{\alpha 4}) \left[ \frac{r}{u} \right] + C_{\alpha a} \delta_a \]  \hspace{1cm} (5.16)

\[ I \dot{r} = -(a_a C_{\alpha a} - c C_{\alpha 3} - d C_{\alpha 4}) \left[ \frac{V}{u} \right] - (a_a^2 C_{\alpha a} + c^2 C_{\alpha 3} + d^2 C_{\alpha 4}) \left[ \frac{r}{u} \right] + a_a \alpha_a \delta_a \]  \hspace{1cm} (5.17)

**Stability Criteria:**

\[ \dot{V} = PV \hspace{1cm} \dot{r} = Pr \hspace{1cm} P = \frac{d}{dt} \]

\[ \left[ mp + \frac{c_{\alpha a} + c_{\alpha 3} + c_{\alpha 4}}{u} \right] V + \left[ mu + \frac{a_a c_{\alpha a} - c C_{\alpha 3} - d C_{\alpha 4}}{u} \right] r = C_{\alpha a} \delta_a \]  \hspace{1cm} (5.18)

\[ \left[ lp + \frac{a_a^2 c_{\alpha a} + c^2 C_{\alpha 3} + d^2 C_{\alpha 4}}{u} \right] r + \left[ mu + \frac{a_a c_{\alpha a} - c C_{\alpha 3} - d C_{\alpha 4}}{u} \right] V = a_a C_{\alpha a} \delta_a \]  \hspace{1cm} (5.19)

For steady state response:

\[ P = 0 \hspace{1cm} \dot{V} = \dot{r} = 0 \]

\[ \left[ A_1 \quad B_1 \right] \begin{bmatrix} V/\delta_a \\ r/\delta_a \end{bmatrix} = \begin{bmatrix} C_{\alpha a} \\ a_a C_{\alpha a} \end{bmatrix} \]

\[ \left[ V/\delta_a \right]_{ss} = \frac{[F_1 \quad B_{s1}]}{|A|} \]  \hspace{1cm} (5.20)

\[ \left[ r/\delta_a \right]_{ss} = \frac{[B_{s2} \quad F_1 \quad T_1]}{|A|} \]  \hspace{1cm} (5.21)

Where:

\[ A_{s1} = \frac{C_{\alpha a} + C_{\alpha 3} + C_{\alpha 4}}{u} \]

\[ A_{s2} = \frac{a_a C_{\alpha a} - c C_{\alpha 3} - d C_{\alpha 4}}{u} \]
\[ B_{s1} = mu + \frac{a_a C_{\alpha a} - c C_{\alpha a} - d C_{\alpha a}}{u} \]

\[ B_{s2} = \frac{a_a^2 C_{\alpha a} + c^2 C_{\alpha a} + d^2 C_{\alpha a}}{u} \]

\[ |A| = A_{s1} B_{s2} - A_{s2} B_{s1} \]

\[ |A| = \frac{C_{\alpha a} C_{\alpha a} L_4^2 + C_{\alpha a} C_{\alpha a} (c + a_a)^2 + C_{\alpha a} C_{\alpha a} (d - c)^2 + mu^2 (c C_{\alpha a} + d C_{\alpha a} - a_a C_{\alpha a})}{u^2} \]

\[ \left[ \frac{V}{\delta_a} \right]_{ss} = \frac{u[L_4 d C_{\alpha a} C_{\alpha a} + L_3 c C_{\alpha a} C_{\alpha a} - mu^2 a_a C_{\alpha a}]}{L_4^2 C_{\alpha a} C_{\alpha a} + L_3^2 C_{\alpha a} C_{\alpha a} + C_{\alpha a} C_{\alpha a} (d - c)^2 + mu^2 (c C_{\alpha a} + d C_{\alpha a} - a_a C_{\alpha a})} \] (5.22)

\[ r / \delta_a_{ss} = \frac{u[L_4 C_{\alpha a} C_{\alpha a}]}{L_4^2 C_{\alpha a} C_{\alpha a} + L_3^2 C_{\alpha a} C_{\alpha a} + C_{\alpha a} C_{\alpha a} (d - c)^2 + mu^2 (c C_{\alpha a} + d C_{\alpha a} - a_a C_{\alpha a})} \] (5.23)

Where:

\[ L_4 = a_a + d \quad \& \quad L_3 = a_a + c \]

The steady state acceleration and the curvature response will be as follows:

\[ \left[ \frac{A_y}{\delta_a} \right]_{ss} = \left[ \frac{r}{\delta_a} \right]_{ss} u \]

\[ \left[ \frac{1/R}{\delta_a} \right]_{ss} = \left[ \frac{r/\delta_a}{u} \right]_{ss} \]

The Ackerman steering angle at \( u=0 \)

\[ \left[ \frac{1/R}{\delta_a} \right]_{ssu=0} = \frac{L_4 C_{\alpha a} C_{\alpha a} + L_3 C_{\alpha a} C_{\alpha a}}{L_4^2 C_{\alpha a} C_{\alpha a} + L_3^2 C_{\alpha a} C_{\alpha a} + C_{\alpha a} C_{\alpha a} (d - c)^2} \] (5.24)

\[ \delta_a = \frac{L_4^2 C_{\alpha a} C_{\alpha a} + L_3^2 C_{\alpha a} C_{\alpha a} + C_{\alpha a} C_{\alpha a} (d - c)^2}{R(L_4 C_{\alpha a} C_{\alpha a} + L_3 C_{\alpha a} C_{\alpha a})} \]
Let:

\[ L_a = \frac{L_4^2 C_{\alpha_4} C_{\alpha_4} + L_3^2 C_{\alpha_3} C_{\alpha_3} + C_{\alpha_3} C_{\alpha_4} (d-c)^2}{L_4 C_{\alpha_4} C_{\alpha_a} + L_3 C_{\alpha_3} C_{\alpha_a}} \]

\[ \delta_a = \frac{L_a}{R} \]

\[ K_{us} = -\left[ \frac{L_a/R - \delta_a}{A_y/g} \right] \]

\[ \delta_a = \frac{L_a}{R} + K_{us} \frac{A_y}{g} \]

1. Desired yaw rate \( (r_d) \):

\[ r_d = \frac{u \delta_a [L_4 C_{\alpha_4} C_{\alpha_a} + L_3 C_{\alpha_3} C_{\alpha_a}]}{L_4^2 C_{\alpha_4} C_{\alpha_4} + L_3^2 C_{\alpha_3} C_{\alpha_3} + C_{\alpha_3} C_{\alpha_4} (d-c)^2 + \mu u^2 (c C_{\alpha_3} + d C_{\alpha_4} - a C_{\alpha_a})} \]  \hspace{1cm} (5.25)

2. Desired lateral acceleration \( (A_{y_d}) \)

\[ A_{y_d} = \frac{u^2 \delta_a [L_4 C_{\alpha_4} C_{\alpha_a} + L_3 C_{\alpha_3} C_{\alpha_a}]}{L_4^2 C_{\alpha_4} C_{\alpha_4} + L_3^2 C_{\alpha_3} C_{\alpha_3} + C_{\alpha_3} C_{\alpha_4} (d-c)^2 + \mu u^2 (c C_{\alpha_3} + d C_{\alpha_4} - a C_{\alpha_a})} \]  \hspace{1cm} (5.26)

Desired yaw rate and lateral acceleration for the combat vehicle used in the study are evaluated using the following vehicle dimensions:

\[ a=d=1930 \text{ mm} \quad \& \quad b=c=710 \text{ mm} \quad \& \quad a_a=1320 \text{ mm} \]

\[ L_3=2030 \text{ mm} \quad \& \quad L_4=3250 \text{ mm} \]

For rigid road:

\[ C_{\alpha_a} = C_{\alpha_3} = C_{\alpha_4} = 7.68 \text{ kN/degree} \quad \text{(For two tires)} \]

\[ r_d = \frac{\delta_a u}{3.063 + K_{us} u^2} \quad \text{rad/sec} \]

\[ A_{y_d} = \frac{\delta_a u^2}{3.063 + K_{us} * u^2} \quad \text{m/sec}^2 \]
For soft soil (Clayey soil):

\[ C_{\alpha_1} = 2.902 \, kN/\text{degree} \]
\[ C_{\alpha_2} = C_{\alpha_3} = C_{\alpha_4} = 3.116 \, kN/\text{degree} \]
\[ C_{\alpha_{a}} = 3.01 \, kN/\text{degree} \]

\[ r_d = \frac{\delta_a u}{3.075 + K_u u^2} \text{ rad/sec} \]

\[ A_{y_d} = \frac{\delta_a u^2}{3.075 + K_u u^2} \text{ m/sec}^2 \]

Where: \( \delta_a \) in rad and \( u \) in m/sec

The respective errors in some desired variables are defined as follows. The lateral acceleration error is:

\[ e_{ay} = A_y - A_{y_d} \]  \hspace{1cm} (5.27)

The yaw rate error is:

\[ e_r = r - r_d \]  \hspace{1cm} (5.28)

\( A_y \) and \( r \) are the actual values of the corresponding vehicle states (lateral acceleration and yaw rate respectively) obtained from actual vehicle model. The lateral acceleration error, \( e_{ay} \) and yaw rate error \( e_r \) are the feedback variables used in the controller design as will be detailed in the following sections.

**5.2.3 Architecture of the proposed control**

This sub-section describes the control structure adopted as shown in Figure 5.3.

![Figure 5.3 Schematic of control architecture](image-url)
5.2.3.1 Development of the upper controller

The upper controller utilizes the developed four-axle bicycle model, and the actual vehicle response; yaw rate, lateral acceleration, and longitudinal speed to prepare the desired vehicle responses as a first step in the upper controller. Then, Three PID controllers are used to develop the needed corrective yaw moment based on the differences between the actual and desired vehicle responses to enhance vehicle directional stability. The corrective yaw moment is then passed to the management system (the lower controller) as shown in Figure 5.4.

![Block diagram of the upper controller](image)

Figure 5.4 Block diagram of the upper controller

5.2.3.2 Development of the lower controller

Generally, the lower controller objective is to produce the needed action to generate the required corrective yaw moment by the upper controller by means of either braking, driving or steering effort. In the proposed control system strategy, the lower controller is the torque distribution management system (torque vectoring differentials) that manages the torque distribution between all wheels independently to achieve the desired yaw moment. In addition, the physics description of the yaw moment control through torque distribution as achieved by vectoring differentials is described as follows.

(a) Inter-axle torque distribution

More torque transfer to the front axle wheels will increase longitudinal slip of the front axle wheels while the rear axle wheels will drop and decrease the lateral forces generated by the front axle wheels compared to the rear ones. Accordingly, torque transfer from the rear to the front wheels induces an understeering effect.
(b) *Left to Right torque distribution:*
Reducing the driving torque delivered to the outer wheel in comparison to the inner one generates a yaw moment in the opposite direction of the turn that will induce understeering effect on the vehicle. The differences in longitudinal forces produce a significant yaw moment while the differences in lateral forces, being partially compensating, lead to the generation of small positive yaw moments. Thus, a net positive yaw moment in the opposite direction of motion is generated, leading to understeer.

Active torque distribution systems utilize the physics described above for yaw moment control by varying the torques on individual wheels. In this research, yaw moment control is based on left to right torque distribution strategy and various torque distribution approaches are considered and analyzed as follows.

(a) *Torque ratios variations approach*
Osborn and Shim [100] introduced a torque distribution strategy based on two torque ratios; front-rear ration and left-right ratio. The front-rear ratio, $r_{fr}$, is determined based on the calculated yaw rate error, while the left-right ratio, $r_{lr}$, is determined based on the calculated lateral acceleration error. The front-rear torque ratio can be defined as the ratio of the front left wheel torque to the sum front left and rear left wheel torques. In addition, the left-right torque ratio can be defined as the ratio of the front left wheel torque to the sum of the front left and front right wheel torques. These ratios could be expressed as shown in the following equations:

$$r_{fr} = \frac{T_{fl}}{T_{fl} + T_{rl}} = \frac{T_{fr}}{T_{fr} + T_{rr}}$$

$$r_{lr} = \frac{T_{fl}}{T_{fl} + T_{fr}} = \frac{T_{rl}}{T_{rl} + T_{rr}}$$

Given a total driveline torque $T$, using the above definitions of torque distribution ratios, the four individual torques on the wheels can be evaluated from the following equations:

$$T_{fl} = T r_{fr} r_{lr}$$

$$T_{fr} = T r_{fr} (1 - r_{lr})$$

$$T_{rl} = T (1 - r_{fr}) r_{lr}$$

$$T_{rr} = T (1 - r_{fr})(1 - r_{lr})$$
The presented simulation response based on using the ‘torque-ratio’ approach in [100] are promising in achieving an adequate stability control system. The torque distribution ratios are constrained by the two ratios and the total torque on the vehicle always remains constant. Consequently, this approach reduced the control variables from four (each of four individual wheels) to two (two torque ratios) which reduces the torque distribution independence by limiting the total torque.

(b) **Differential torque distribution approach**

This approach utilizes differential torque distribution by either addition or subtraction of corrective torque which the already produced by the upper controller. In addition, this approach does not limit the total torque as in the torque ratio variations approach which allow independent torque control of each wheel. In this research, this approach is implemented in simulations based on the selected control variables; yaw rate and lateral acceleration.

The torque distribution strategies are analyzed and implemented with and without controlling vehicle speed. Therefore, different standard maneuvers are performed at constant or nearly constant speed. Consequently, speed control is introduced as a PID speed controller. The speed error, \( e_v \), is defined as the difference between the actual forward velocity, \( V_x \), and the desired (test) forward velocity of the vehicle, \( V_{xd} \).

\[
e_v = V_x - V_{xd} \tag{5.29}
\]

In all the performed simulations at constant speed, the total torque \( \Delta T_v \) is considered to be equally distributed between all wheels. Therefore, the speed control torque is added to the corrective torques of each wheel. On the other hand, in the case of no speed control, constant torques ‘base torques’ are delivered to each wheel and added to the corrective torques of each wheel. The total base torques on the left and right sides of the vehicle are given as follows:

\[
T_L = T_{fl} + T_{rl} \\
T_R = T_{fr} + T_{rr}
\]

Where \( T_{fl}, T_{rl}, T_{fr}, \) and \( T_{rr} \) are the individual base torques acting on the individual wheels. The proposed control strategy used in this research was interfaced with the developed
vehicle model in TruckSim as shown in Figure 5.5.

1- **Yaw rate control:**

A proper controller can be developed to generate the necessary corrective yaw moment based on the yaw rate differences between the actual and desired values. The necessary corrective torque, $\Delta T_r$, that will be added or subtracted to the base torques (in case of no speed control) or speed control torques of the individual wheels for generating the desired yaw moment is evaluated using a PID controller. In this research, half the corrective torques are added to the left wheels and half of them are subtracted from the right wheels for both the driving axles.

\[
T_{l3\_new} = T_{l3} + \frac{\Delta T_r}{2} \tag{5.30}
\]
\[
T_{l4\_new} = T_{l4} + \frac{\Delta T_r}{2} \tag{5.31}
\]
\[
T_{r3\_new} = T_{r3} - \frac{\Delta T_r}{2} \tag{5.32}
\]
\[
T_{r4\_new} = T_{r4} - \frac{\Delta T_r}{2} \tag{5.33}
\]

2- **Lateral acceleration control:**

For the lateral acceleration as a feedback variable, the required differential torque, $\Delta T_{ay}$ can be evaluated from the PID controller based on the lateral acceleration error in a similar way as was done for yaw rate control.

\[
T_{l3\_new} = T_{l3} + \frac{\Delta T_{ay}}{2} \tag{5.34}
\]
\[
T_{l4\_new} = T_{l4} + \frac{\Delta T_{ay}}{2} \tag{5.35}
\]
\[ T_{r3,new} = T_{l3} - \frac{\Delta T_{ay}}{2} \] (5.36)

\[ T_{r4,new} = T_{l4} - \frac{\Delta T_{ay}}{2} \] (5.37)

3- Combined lateral acceleration and yaw rate control:

This approach combines the corrective torques being added to left wheels and subtracted from right wheels based on yaw rate and lateral acceleration errors. The final wheel driving torques on the individual wheels is calculated by the following equations:

\[ T_{l3,new} = T_{l3} + \frac{\Delta T_r}{2} + \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \] (5.38)

\[ T_{l4,new} = T_{l4} + \frac{\Delta T_r}{2} + \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \] (5.39)

\[ T_{r3,new} = T_{r3} - \frac{\Delta T_r}{2} - \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \] (5.40)

\[ T_{r4,new} = T_{r4} - \frac{\Delta T_r}{2} - \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \] (5.41)

Where \( \Delta T \) is the corrective differential torque to be transferred according to the error function for yaw rate, lateral acceleration, and longitudinal vehicle speed as follows:

\[ \Delta T_r = K_p \cdot r e_r + K_i \cdot r \int e_r \, dt + K_d \cdot r \frac{d}{dt} (e_r) \] (5.42)

\[ \Delta T_{ay} = K_p \cdot ay e_{ay} + K_i \cdot ay \int e_{ay} \, dt + K_d \cdot ay \frac{d}{dt} (e_{ay}) \] (5.43)

\[ \Delta T_v = K_p \cdot v e_v + K_i \cdot v \int e_v \, dt + K_d \cdot v \frac{d}{dt} (e_v) \] (5.44)

5.2.3.3 MATLAB/Simulink – TruckSim Co-Simulator

Co-simulator that consists of the TruckSim combat vehicle model and MATLAB/Simulink controller was developed to verify the proposed control strategy as shown in Figure 5.6. The vehicle module in Matlab/Simulink represent the vehicle as specified in the TruckSim software and to fit with the signal requirements of the Simulink control model.
5.2.4 Results and Discussion

Different standard simulation maneuvers have been performed to demonstrate the effectiveness of the proposed design of the torque distribution control strategy and its effect on 8x4 combat vehicle performance as shown in Table 5.1. The next sections will show a
comparison between the vehicle maneuverability performance with and without controller.

Table 5.1 Test Course Matrix

<table>
<thead>
<tr>
<th>No.</th>
<th>Test course</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>FMVSS 126 ESC Test Maneuver</td>
</tr>
<tr>
<td>2</td>
<td>J-turn (Step Steer)</td>
</tr>
<tr>
<td>3</td>
<td>Fish-Hook Maneuver</td>
</tr>
<tr>
<td>4</td>
<td>Constant Step Slalom (NATO AVTP-1 03-30)</td>
</tr>
<tr>
<td>5</td>
<td>J-turn (22m radius)</td>
</tr>
<tr>
<td>6</td>
<td>Constant radius lateral acceleration (30m radius)</td>
</tr>
</tbody>
</table>

5.2.4.1 FMVSS 126 ESC Test

The Federal Motor Vehicle Safety Standard (FMVSS) No. 126 test has been modified and applied for evaluating the proposed control strategy performance [101].

In this test, a “Slowly Increasing Steer” angle is defined as the steering wheel angle associated with a vehicle lateral acceleration about 0.3 g. The vehicle speed was maintained at approximately 80 km/h as shown in Figure 5.7. The test consists of a "Sine with Dwell" test conducted with a steering pattern of a sine wave at 0.7 Hz frequency with a 400 ms delay beginning at the second peak amplitude, Figure 5.8. The vehicle lateral acceleration and yaw rate responses with and without controller are given in Figure 5.9 and Figure 5.10.

![Figure 5.7 Vehicle speed time history](image-url)
Figure 5.8 FMVSS 126 VDC test steering input

Figure 5.9 Vehicle lateral acceleration time history

Figure 5.10 Vehicle yaw rate time history
From the simulation results, it can be noticed that the proposed controller did not affect mostly the vehicle performance as both lateral acceleration, and yaw rate are smaller than the desired values obtained from the bicycle model. However, the vehicle yaw rate has been reduced in comparison with the vehicle without controller.

5.2.4.2 J-turn (Step Steer)

A standard J-turn test [102] has been performed to investigate the vehicle performance characteristics like its tracking ability in a sudden steer angle change (step steer). In this test, the vehicle speed was maintained at approximately 80 km/h as shown in Figure 5.11. The steering wheel input used in the simulation as shown in Figure 5.12. The vehicle lateral acceleration and yaw rate are given in Figure 5.13 and Figure 5.14.
From the simulation results, it can be noticed that the controller succeeded to reduce both lateral acceleration and yaw rate by generating the required corrective yaw moment with acceptable reduction in vehicle speed.

5.2.4.3 Fish-Hook Maneuver

A standard fish-hook maneuver test [101, 102] designed by National Highway Traffic Safety Administration (NHTSA) for prompting and analyzing dynamic rollover has been modified to investigate the proposed control strategy.

(a) Modified Fish-Hook Maneuver

In this test, the vehicle speed was maintained at approximately 80 km/h as shown in Figure 5.15. The steering wheel input used in the simulation was calculated to produce
about 0.3g lateral acceleration as shown in Figure 5.16. The vehicle lateral acceleration and yaw rate are given in Figure 5.17 and Figure 5.18.

Figure 5.15 Vehicle speed time history

Figure 5.16 NHTSA Fish hook maneuver test steering input

Figure 5.17 Vehicle lateral acceleration time history
From the simulation results, it can be noticed that the controller succeeded to reduce both lateral acceleration and yaw rate by generating the required corrective yaw moment with acceptable reduction in vehicle speed.

(b) **Severe Fish-Hook Maneuver**

A standard fish-hook maneuver test for prompting and analyzing dynamic rollover has been performed to investigate the proposed control strategy. In this test, the vehicle speed was maintained at approximately 80 km/h as shown in Figure 5.19. The steering wheel input used in the simulation as shown in Figure 5.20.
The vehicle lateral acceleration and yaw rate are given in Figure 5.21 and Figure 5.23. Figure 5.22 shows the combat vehicle with and without controller during the simulation and how the developed controller prevents the vehicle from rollover.
From the simulation results, it can be noticed that during the first two sec the proposed controller did not affect the vehicle performance as both lateral acceleration and yaw rate are below the desired values obtained from the bicycle model. While, before reaching the vehicle dynamic rollover threshold (0.56 g), Figure 5.21, the controller succeeded to reduce both lateral acceleration and yaw rate by generating the required corrective yaw moment with acceptable reduction in vehicle speed.

5.2.4.4 Constant Step Slalom (NATO AVTP-1 03-30)

In this test, the vehicle speed was maintained at approximately 65 km/h as shown in Figure 5.24. The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 5.25.
The vehicle lateral acceleration and yaw rate are given in Figure 5.26 and Figure 5.27. Figure 5.28 shows how the developed controller prevents the vehicle from rollover.
From the simulation results, it can be noticed that the proposed controller succeeded to reduce both lateral acceleration and yaw rate in comparison with the vehicle without controller. In addition, keeping both lateral acceleration and yaw rate below the desired values obtained from the bicycle model before reaching the vehicle dynamic rollover threshold (0.56 g) and preventing vehicle rollover as shown in Figure 5.26.

5.2.4.5 J-Turn (22m radius)

In this test, the vehicle speed was maintained at approximately 45 km/h as shown in Figure 5.29. The steering wheel input used in the simulation was obtained from the measurements as shown in Figure 5.30.
The vehicle lateral acceleration and yaw rate are given in Figure 5.31 and Figure 5.32. Figure 5.33 shows the combat vehicle with and without controller during the simulation and how the developed controller prevents the vehicle from rollover.
From the simulation results, it can be noticed that the proposed controller succeeded to reduce both lateral acceleration and yaw rate to be within the desired values obtained from the bicycle model and before reaching the vehicle dynamic rollover threshold (0.56 g).

5.2.4.6 Constant radius lateral acceleration

In this test, the vehicle test course of 100ft. radius, Figure 5.34, was used to verify the effectiveness of the proposed control strategy and its effect on vehicle directional stability.

(a) Constant radius lateral acceleration - 40 km/h

In this test, the vehicle speed was maintained at 40 km/h as shown in Figure 5.35. The vehicle lateral acceleration and yaw rate are given in Figure 5.36 and Figure 5.37. Figure 5.38 shows vehicle trajectory with and without controller during the simulation.
Figure 5.35 Vehicle speed time history

Figure 5.36 Vehicle lateral acceleration time history

Figure 5.37 Vehicle yaw rate time history
In this test, the vehicle speed was maintained at 45 km/h as shown in Figure 5.39. The vehicle lateral acceleration and yaw rate are given in Figure 5.40 and Figure 5.41. Figure 5.42 shows vehicle trajectory with and without controller during the simulation.

From the simulation results, it can be noticed that with increasing vehicle speed the proposed controller succeeded to reduce both lateral acceleration and yaw rate to be below the desired values obtained from the bicycle model. In addition, the vehicle without controller was not able to complete the test at 45 km/h as the one with the developed controller did.
Figure 5.40 Vehicle lateral acceleration time history

Figure 5.41 Vehicle yaw rate time history

Figure 5.42 Vehicle trajectory
Figure 5.43 Vehicle model without controller (Green) and with the controller (Red)

5.3 Summary

This chapter presents the development of a torque distribution control strategy based on three PID controllers to enhance the directional stability and mobility of a multi-wheeled combat vehicle. Comparison between vehicle directional performance with and without the proposed control strategy was performed using different standard maneuvers as mentioned in Table 5.1.

From these tests, it can be concluded that:

- The developed PID controllers were effective in preventing rollover during severe Fish-Hook maneuver at 80km/h.
- In the case of Constant Step Slalom (NATO AVTP-1 03-30) and J-Turn (22m radius), the proposed controller enhanced both yaw rate and lateral acceleration and succeeded in preventing rollover in both testing maneuvers.
- In the case of Constant radius lateral acceleration test, the proposed controller enhanced both lateral acceleration during all the performed tests at 35 and 40 km/h. In addition, the completed test at 45 km/h helped the vehicle with controller to remain in the desired path.
Chapter 6

Advanced Fuzzy Slip Control System

6.1 Introduction

Slip control system, such as ABS or TCS, are developed to enhance the longitudinal dynamics of a vehicle by preventing the tires from locking up when braking or spinning out when accelerating to improve the vehicle directional stability. Monash University Accident Research Centre investigated the effect of using the ABS control system and how it could affect the vehicle directional stability and safety. The conducted study concluded that the risk of multiple vehicle crashes reduced by 18% and the risk of run-off-road crashes reduced by 35% [103]. In addition, National Highway Traffic Safety Administration (NHTSA) conducted more investigations that leads to the same outcomes [104]. Therefore, both the European Automobile Manufacturers Association and United States suggested using of the ABS control system in the new vehicles [105].

6.1.1 Anti-lock braking system

The anti-lock braking system (ABS) is based on preventing the wheels from lock-up by sensing the wheel speeds to calculate the longitudinal slip. The directional stability during braking can be enhanced when the vehicle is equipped with an ABS system [106]

6.1.2 Traction control system

The first traction control system was introduced by the Buick division of GM based on detecting the rear wheel spin and using engine management procedure to reduce the delivered power to those wheels in order to provide the maximum available traction. Tire slip can also be controlled during acceleration using integrated brake system and engine management controller.

The objective of the traction control system depends on the vehicle configuration such that; In the case of front-wheel-drive, the objective is to maximize the traction force while retaining controllability while in the case of rear-wheel-drive, the objective is to maintain
vehicle stability while maximizing the traction force.

6.1.3 Methods of adjusting the tire slip ratio

The first strategy depends on adjusting the tire slip ratio in a slip control system depends on limiting the maximum possible slip ratio to a fixed value which can be modified as desired as shown in Figure 6.1, where the longitudinal force ($F_x$) and lateral force ($F_y$) of the tire are plotted as functions of the longitudinal slip ratio of the tire [107]. In addition, when the tire slip angle increases, the longitudinal tire force decreases and the lateral force potential increases as well which enhances the vehicle lateral stability.

![Figure 6.1 Characteristics of the tire longitudinal and lateral forces as a function of tire slip ratio used for limited slip ratio control system [107]](image)

The second strategy depends on adjusting the tire slip ratio in a manner that maximize the traction force at all slip angles. This procedure orders the longitudinal tire force over the lateral tire force to ensure achieving the maximum possible traction force at all slip angles [107] and the lateral force potential will not increase as shown in Figure 6.2. Where the upper bold-dashed line indicates the peak tire longitudinal forces at every slip angle and the lower bold-dashed line indicates the corresponding tire lateral force.
Figure 6.2 Characteristics of the tire longitudinal and lateral forces as a function of tire slip ratio used for adjustable slip ratio control system [107].

6.2 Advanced Fuzzy Slip Control System for 8x4 Drivetrain

The multi-wheeled combat vehicle powertrain has been modified to represent 8x4 powertrain configuration using two twin clutch differentials for both the 3rd and 4th axle and have been connected mechanically using full-time limited slip differentials as shown in Figure 6.3.

![Figure 6.3 Powertrain configuration (8x4)]
6.2.1 Slip control system design

Based on the adhesion coefficient versus tire slip ratio, as shown in Figure 6.4, it is recommended that the maximum adhesion coefficient for different road conditions can be achieved at a slip ratio of about 20%. While this limit corresponds to the position of the peak adhesion coefficient for dry roads. Although on soft soil, the target value for the slip controller has been selected to be about 65% based on the tire slip characteristics on soft soil. With this in mind, and noting that higher vehicle stability is more beneficial than maximum traction when driving in a curve, the limited tire slip ratio strategy is chosen in this research to develop the advanced fuzzy slip controller.

![Figure 6.4 Typical adhesion coefficient characteristics as a function of tire slip ratio for different road conditions](image)

The actual slip ratio of each tire is calculated as a positive number using the following equations for brake and acceleration modes, respectively:

\[
\lambda_{\text{brake},i} = \frac{v_{w,i} - \omega_{w,i}r_{d,i}}{v_{w,i}} \quad \text{if} \quad v_{w,i} \geq 0, \quad \omega_{w,i} \geq 0, \quad \text{and} \quad \omega_{w,i}r_{d,i} \leq v_{w,i} \quad (6.1)
\]

\[
\lambda_{\text{accel},i} = \frac{\omega_{w,i}r_{d,i} - v_{w,i}}{\omega_{w,i}r_{d,i}} \quad \text{if} \quad v_{w,i} \geq 0, \quad \omega_{w,i} \geq 0, \quad \text{and} \quad \omega_{w,i}r_{d,i} \geq v_{w,i} \quad (6.2)
\]

Where, \( v_{w,i} \) is the speed of the wheel center along the wheel plane, \( r_{d,i} \) is the dynamic tire radius, and \( \omega_{w,i} \) is the angular velocity of the tire. All of the mentioned variables must be
measured or estimated in real life. Although the dynamic tire radius has to be estimated. Fuzzy logic control systems are robust and flexible inference methods that are well suited for tackling complicated nonlinear dynamic control problems. Consequently, they are the ideal selection for controlling the highly nonlinear behavior in vehicle dynamics.

The rule base of the developed fuzzy slip controller was established based on using the slip ratio error, \(e(\lambda)\), and the rate of change of the slip ratio error, \(\dot{e}(\lambda)\) as input variables to the controller and using the corrective torque, \(T_{\text{corr}}\), to represent the controller as shown in Table 6.1. The tire slip ratio error is calculated instantaneously by comparing the actual tire slip with the desired one. The rate of change of the slip ratio error is calculated by subtracting the previous slip ratio error from the current one, and dividing the result by the sample time of the controller. In addition, the controller inputs and output are normalized to simplify the fuzzy sets definition. Two seven fuzzy sets are used for the slip ratio error and the rate of change of the slip ratio error in order to provide enough rule coverage and nine fuzzy sets are used to describe the output of the fuzzy slip controller.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input 1</td>
<td>(e(\lambda) = \lambda_{\text{lim}} - \lambda_{\text{act}})</td>
</tr>
<tr>
<td>Input 2</td>
<td>(\dot{e}(\lambda) = \frac{e(\lambda)<em>k - e(\lambda)</em>{k-1}}{\text{sample time}})</td>
</tr>
<tr>
<td>Output</td>
<td>(T_{\text{corr}})</td>
</tr>
</tbody>
</table>

The fuzzy inference system processes the list of rules in the knowledge base using the fuzzy inputs obtained from the previous time step of the simulation, and produces the fuzzy output, which, once defuzzified, is applied in the next time step. The Mamdani fuzzy inference method is used, which is characterized by the following fuzzy rule schema:

\[
IF \ e(\lambda) \text{ is } A \text{ AND } \dot{e}(\lambda) \text{ is } B \text{ THEN } T_{\text{corr}} \text{ is } C
\]  \tag{6.3}

Where A, B, and C are fuzzy sets defined on the input and output domains. The control rule base of the proposed fuzzy slip controller is developed based on expert knowledge and
extensive investigation. Figure 6.5 illustrates the control rule base and control surface of the fuzzy slip controller. The linguistic terms that have been used in this table are listed in Table 6.2. The shape and distribution of the membership functions used for the input and output variables of the fuzzy slip controller are shown in Figure 6.6.

Table 6.2 Linguistic variables used in the fuzzy rules

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Linguistic Variable</th>
</tr>
</thead>
<tbody>
<tr>
<td>NVL</td>
<td>Negative Very Large</td>
</tr>
<tr>
<td>NL</td>
<td>Negative Large</td>
</tr>
<tr>
<td>NM</td>
<td>Negative Medium</td>
</tr>
<tr>
<td>NS</td>
<td>Negative Small</td>
</tr>
<tr>
<td>ZE</td>
<td>Zero</td>
</tr>
<tr>
<td>PS</td>
<td>Positive Small</td>
</tr>
<tr>
<td>PM</td>
<td>Positive Medium</td>
</tr>
<tr>
<td>PL</td>
<td>Positive Large</td>
</tr>
<tr>
<td>PVL</td>
<td>Positive Very Large</td>
</tr>
</tbody>
</table>

Figure 6.5 Control rule base (a) and control surface (b) of the fuzzy slip control system
6.2.2 Results and Discussion

Various vehicle maneuvers have been performed to demonstrate the effectiveness of the proposed control strategy design and its effect on the combat vehicle performance as shown in Table 6.3. The next sections will show a comparison between the vehicle maneuverability performance with and without controller.
Table 6.3 Test Course Matrix

<table>
<thead>
<tr>
<th>No.</th>
<th>Test Course</th>
<th>Vehicle Speed</th>
<th>Additional Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Straight-line acceleration</td>
<td>Ramp to full throttle in 1.0 sec</td>
<td>Rigid and soft soil</td>
</tr>
<tr>
<td>2</td>
<td>Split Mu maneuver (0.1L/1.0)</td>
<td>Ramp to full throttle in 1.0 sec</td>
<td>Only on rigid surface</td>
</tr>
<tr>
<td>3</td>
<td>FMVSS 126 ESC Test</td>
<td>40 and 80 km/h</td>
<td>Only on rigid surface</td>
</tr>
<tr>
<td>4</td>
<td>J-turn (Step Steer)</td>
<td>40 and 80 km/h</td>
<td>Only on rigid surface</td>
</tr>
<tr>
<td>5</td>
<td>Fish-Hook Maneuver</td>
<td>30 and 50 km/h</td>
<td>Only on rigid surface</td>
</tr>
<tr>
<td>6</td>
<td>Constant radius lateral acceleration</td>
<td>10 and 45 km/h</td>
<td>30m (100ft) radius (Rigid and soft soil)</td>
</tr>
</tbody>
</table>

6.2.2.1 Straight-line acceleration maneuver

In this test, the vehicle initial speed was zero with a ramp to full throttle in 1.0 sec as shown in Figure 6.8. In addition, there is no steering wheel input during the simulation.

![Figure 6.8 Throttle position time history](image)

(a) Test maneuver on a rigid surface (road friction 0.2)

The vehicle initial speed was zero with a ramp to full throttle in 1.0 sec as shown in Figure 6.8. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.9. The wheel longitudinal slip for the third and fourth axles for the vehicle with controller are shown in Figure 6.10.
(b) Test maneuver on soft soil

In this test, the vehicle initial speed was zero with a ramp to full throttle in 1.0 sec as shown in Figure 6.8. The developed FEA tire model has been used to represent the tire-soil interaction characteristics. The total wheel driving moment for the third and fourth axles
with and without the integrated yaw-slip controller are shown in Figure 6.11. The wheel longitudinal slip for the third and fourth axles with controller are shown in Figure 6.12.

From the simulation results, the developed integrated yaw-slip controller succeeded in controlling the total driving moment for all the driving wheels to prevent slippage increase.
over the target value, which is 0.2 and 0.6 for both rigid road and soft soil respectively. Moreover, it is expected that better results can be achieved with 8x8-powertrain configuration especially on soft soil.

6.2.2.2 Split Mu maneuver (0.1L/1.0R)

In this test, one side of the vehicle is on a high-coefficient of friction surface (1.0) and the other side is on a low-coefficient of friction surface (0.1), the vehicle initial speed was zero with a ramp to full throttle in 1.0 sec as shown in Figure 6.8. Moreover, there is no steering wheel input during the simulation.

The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.13. The wheel longitudinal slip for the third and fourth axles with controller are shown in Figure 6.14. Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.15. Figure 6.16 shows combat vehicle with and without the controller during the simulation.

![Figure 6.13 Total wheel driving moment time history](image-url)
From the simulation results, the vehicle equipped with the developed integrated yaw-slip controller succeeded in completing the test in a straight line and kept the slip within an accepted range from the slip controller target value (0.2).
6.2.2.3 **FMVSS 126 ESC TEST**

The Federal Motor Vehicle Safety Standard (FMVSS) No. 126 test has been modified and applied for evaluating the proposed control strategy performance. In this test, a “Slowly Increasing Steer” angle is defined as the steering wheel angle associated with a vehicle lateral acceleration about 0.3 g. The test consists of a "Sine with Dwell" test conducted with a steering pattern of a sine wave at 0.7 Hz frequency with a 400 ms delay beginning at the second peak amplitude as shown in Figure 6.17.

![Figure 6.16 Vehicle model without controller (Green) and with the controller (Red)](image)

(a) **Test maneuver on low friction surface (0.2)**

1. **FMVSS 126 ESC at 40 km/h:**

The vehicle speed was maintained at approximately 40 km/h as shown in Figure 6.18.
Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.19. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.20. The wheel longitudinal slip for the third and fourth axles with and without controller are shown in Figure 6.21.

Figure 6.18  Vehicle speed time history

Figure 6.19  Vehicle trajectory
The vehicle speed was maintained at approximately 80 km/h as shown in Figure 6.22. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.23. The wheel longitudinal slip for the third and fourth axles are shown in Figure 6.24.
fourth axles for the vehicle with controller are shown in Figure 6.24. Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.25.

![Vehicle speed time history](image)

**Figure 6.22** Vehicle speed time history

![Total wheel driving moment time history](image)

**Figure 6.23** Total wheel driving moment time history
The vehicle speed was maintained at approximately 40 km/h as shown in Figure 6.26. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.27. The wheel longitudinal slip for the third and fourth axles for the vehicle with controller are shown in Figure 6.28. Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.29.
Figure 6.26  Vehicle speed time history

Figure 6.27  Total wheel driving moment time history
The vehicle speed was maintained at approximately 80 km/h as shown in Figure 6.30. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.31. The wheel longitudinal slip for the third and fourth axles with controller are shown in Figure 6.32. Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.33.
Figure 6.30  Vehicle speed time history

Figure 6.31  Total wheel driving moment time history
From the simulation results on rigid surface (road friction 0.8 and 0.2), the developed controller succeeded in controlling the tire longitudinal slip with increasing the vehicle speed in comparison with the vehicle without controller.

### 6.2.2.4 J-TURN (STEP STEER)

A standard J-turn test has been performed to investigate the vehicle performance characteristics like its tracking ability in a sudden steer angle change (step steer). The steering wheel input used in the simulation as shown in Figure 6.34.
Figure 6.34  J-turn test steer input

(a) J-Turn test at 40 km/h on low friction surface (0.2)

The vehicle speed was maintained at approximately 40 km/h as shown in Figure 6.35. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.36. Figure 6.38 shows combat vehicle with and without the controller during the simulation. The wheel longitudinal slip for the third and fourth axles for the vehicle with controller are shown in Figure 6.37. Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.39.

Figure 6.35  Vehicle speed time history
Figure 6.36 Total wheel driving moment time history

Figure 6.37 Wheel Longitudinal slip time history
Figure 6.38 Vehicle model without controller (Green) and with the controller (Red)

Figure 6.39 Vehicle trajectory

(b) J-Turn test at 80 km/h on low friction surface (0.2)

The vehicle speed was maintained at approximately 80 km/h as shown in Figure 6.40. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.41. The wheel longitudinal slip for the third and fourth axles with controller are shown in Figure 6.42. Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.43.
Figure 6.40 Vehicle speed time history

Figure 6.41 Total wheel driving moment time history
From the simulation results, the developed controller succeeded in preventing the vehicle from spinning on the performed maneuver at both test speeds 40 and 80 km/h.

**6.2.2.5 Fish-Hook Maneuver**

A standard fish-hook maneuver test designed by National Highway Traffic Safety Administration (NHTSA) for prompting and analyzing dynamic rollover has been modified to investigate the proposed control strategy. In this test, the steering wheel input used in the simulation was calculated to produce about 0.3g lateral acceleration, Figure 6.44.
Figure 6.44 NHTSA Fish-hook maneuver test steering input

(a) Fish-hook maneuver at 30 km/h on low friction surface (0.2)

The vehicle speed was maintained at approximately 30 km/h as shown in Figure 6.45. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.46. The wheel longitudinal slip for the third and fourth axles with controller are shown in Figure 6.47. Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.48.
Figure 6.46 Total wheel driving moment time history

Figure 6.47 Wheel Longitudinal slip time history
The vehicle speed was maintained at approximately 50 km/h as shown in Figure 6.49. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.50. The wheel longitudinal slip for the third and fourth axles for the vehicle with controller are shown in Figure 6.51. Vehicle trajectory of the combat vehicle with and without the controller is shown in Figure 6.52.
Figure 6.50 Total wheel driving moment time history

Figure 6.51 Wheel Longitudinal slip time history
From the simulation results, the developed integrated yaw-slip controller did not show any difference in vehicle performance as the road friction is very small for the torque distribution to affect the vehicle performance.

### 6.2.2.6 Constant radius lateral acceleration

In this test, the vehicle test course of 100ft. radius, Figure 6.53, was used to verify the effectiveness of the proposed control strategy and its effect on vehicle directional stability.

#### (a) Test maneuver on low friction surface (0.2)

**1- Constant radius lateral acceleration at 20 km/h**

The vehicle speed was maintained at approximately 20 km/h as shown in Figure 6.54. The
vehicle trajectory with and without controller are given in Figure 6.56. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.55. The wheel longitudinal slip for the third and fourth axles for the vehicle with controller are shown in Figure 6.57.

Figure 6.54 Vehicle speed time history

Figure 6.55 Total wheel driving moment time history
The vehicle speed was maintained at approximately 30 km/h as shown in Figure 6.58. The vehicle trajectory with and without controller are given in Figure 6.60. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.59. The wheel longitudinal slip for the third and fourth axles for the vehicle with controller are shown in Figure 6.61.
Figure 6.58 Vehicle speed time history

Figure 6.59 Total wheel driving moment time history
The vehicle speed was maintained at approximately 30 km/h as shown in Figure 6.62. The vehicle trajectory with and without controller are given in Figure 6.64. The total wheel driving moment for the third and fourth axles with and without the integrated yaw-slip controller are shown in Figure 6.63. The wheel longitudinal slip for the third and fourth axles for the vehicle with controller are shown in Figure 6.65.
Figure 6.62 Vehicle speed time history

Figure 6.63 Total wheel driving moment time history

Figure 6.64 Vehicle trajectory
From the simulation results, the developed integrated yaw-slip controller did not show any difference in vehicle performance on rigid road as the road friction is very small for the torque distribution to affect the vehicle performance. While on soft soil, the controller succeeded in increasing the tire longitudinal slip to the recommended range on soft soil (0.6±0.05) and it is recommended to extend the control strategy for 8x8 powertrain configuration especially on soft soil.

6.3 Advanced Fuzzy Slip Control System for 8x8 Drivetrain

The previously developed integrated yaw-slip controller (Two-axle torque vectoring) will be extended to become four-axle torque vectoring (8x8 powertrain configuration). The validated vehicle model using TruckSim has been used to investigate the proposed controller on both rigid surface and soft soil to verify the integrated controllers effectiveness.

6.3.1 Combat Vehicle Model Modifications

The vehicle model consists of 22 Degrees of freedom, namely pitch, yaw and roll of the vehicle sprung mass and spin and vertical motions of each wheel of the eight wheels. The TruckSim vehicle model has been developed based on the actual vehicle configurations of
multi-wheeled combat vehicle. The model is using the measured tire lateral force versus slip angle and aligning moment versus slip angle as well as the FEA predicted longitudinal force versus slip ratio. All powertrain components starting from engine to the axle’s differentials have been modeled in Matlab/Simulink to represent the 8x8-powertrain configuration of a multi-wheeled combat vehicle. Figure 6.66 shows the powertrain assembly screen from TruckSim.

![Powertrain assembly screen from TruckSim](image)

Figure 6.66 Powertrain assembly for 8x8 drive system

### 6.3.2 Controller Design for 8x8 configuration

As described in chapter 5, three PID controllers are used as the upper controller to develop the corrective yaw moment which is then passed to the lower controller. While, the speed controller was disabled as the used driver control in all the performed simulation was starting with the initial speed and applying constant throttle position during all the simulation test courses.

The final wheel driving torques on the individual wheels can be given by:

\[
T_{l1, new} = T_{l1} + \frac{\Delta T_r}{2} + \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2}
\]

\[
T_{l2, new} = T_{l2} + \frac{\Delta T_r}{2} + \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2}
\]
\[ T_{l3,\text{new}} = T_{l3} + \frac{\Delta T_r}{2} + \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \]  
(6.6)

\[ T_{l4,\text{new}} = T_{l4} + \frac{\Delta T_r}{2} + \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \]  
(6.7)

\[ T_{r1,\text{new}} = T_{r1} - \frac{\Delta T_r}{2} - \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \]  
(6.8)

\[ T_{r2,\text{new}} = T_{r2} - \frac{\Delta T_r}{2} - \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \]  
(6.9)

\[ T_{r3,\text{new}} = T_{r3} - \frac{\Delta T_r}{2} - \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \]  
(6.10)

\[ T_{r4,\text{new}} = T_{r4} - \frac{\Delta T_r}{2} - \frac{\Delta T_{ay}}{2} + \frac{\Delta T_v}{2} \]  
(6.11)

Where \( \Delta T \) is the corrective differential torque to be transferred according to the error function for yaw rate, lateral acceleration, and longitudinal vehicle speed as follows:

\[ \Delta T_r = K_{p,r} e_r + K_{i,r} \int e_r \, dt + K_{d,r} \frac{d}{dt} (e_r) \]  
(6.12)

\[ \Delta T_{ay} = K_{p,ay} e_{ay} + K_{i,ay} \int e_{ay} \, dt + K_{d,ay} \frac{d}{dt} (e_{ay}) \]  
(6.13)

\[ \Delta T_v = K_{p,v} e_v + K_{i,v} \int e_v \, dt + K_{d,v} \frac{d}{dt} (e_v) \]  
(6.14)

Co-simulator that consists of the TruckSim combat vehicle model and MATLAB/Simulink controller was developed to verify the proposed control strategy effectiveness.

### 6.3.3 Results and Discussion

Various vehicle maneuvers have been performed to demonstrate the effectiveness of the proposed control strategy design and its effect on the multi-wheeled combat vehicle performance as shown in Table 6.4. The next sections will show a comparison between the vehicle maneuverability performance with and without controller.
Table 6.4 Test Course Matrix

<table>
<thead>
<tr>
<th>No.</th>
<th>Test Course</th>
<th>Vehicle Speed</th>
<th>Additional Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>FMVSS 126 ESC Test</td>
<td>40 and 80 km/h</td>
<td>Rigid road and soft soil</td>
</tr>
<tr>
<td>2</td>
<td>J-turn (Step Steer)</td>
<td>40 and 80 km/h</td>
<td>(road friction 0.2)</td>
</tr>
<tr>
<td>3</td>
<td>Fish-Hook Maneuver</td>
<td>30 and 50 km/h</td>
<td>(road friction 0.2)</td>
</tr>
<tr>
<td>4</td>
<td>Constant radius lateral</td>
<td>10 and 45 km/h</td>
<td>Rigid road and soft soil</td>
</tr>
<tr>
<td></td>
<td>acceleration</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Acceleration test on uniform</td>
<td>Ramp to full throttle in 1.0 sec</td>
<td>(road friction 0.2)</td>
</tr>
<tr>
<td></td>
<td>low friction surface</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Acceleration test on Split Mu</td>
<td>Ramp to full throttle in 1.0 sec</td>
<td>(road friction 0.2L/ 0.8R)</td>
</tr>
</tbody>
</table>

6.3.3.1 **FMVSS 126 ESC Test**

The Federal Motor Vehicle Safety Standard (FMVSS) No. 126 test has been modified and applied for evaluating the proposed control strategy performance. The test consist of a "Sine with Dwell" test conducted with "a steering pattern of a sine wave at 0.7 Hz frequency with a 400 ms delay beginning at the second peak amplitude as shown in Figure 6.17.

(a) **Test maneuver on low friction surface (0.2)**

1- **FMVSS 126 ESC at 40 km/h**

The initial vehicle speed was 40 km/h with constant throttle control from the driver of 1.0 as shown in Figure 6.67. The vehicle trajectory with and without controller is given in Figure 6.68. The total wheel driving moment of the four axles with and without the integrated yaw-slip controller are shown in Figure 6.69. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.70. The vehicle yaw rate and lateral acceleration responses with and without controller are given in Figure 6.71 and Figure 6.72.
Figure 6.67 Vehicle speed time history

Figure 6.68 Vehicle trajectory

Figure 6.69 Total wheel driving moment time history
Figure 6.70 Wheel Longitudinal slip time history

Figure 6.71 Vehicle yaw rate time history

Figure 6.72 Vehicle lateral acceleration time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be within the desired values obtained from the bicycle model. In addition, the vehicle traction performance has been improved as the vehicle speed increased from 67 km/h to 80 km/h.

2- **FMVSS 126 ESC at 80 km/h**

The initial vehicle speed was 80 km/h with constant throttle control from the driver of 0.7 as shown in Figure 6.73. Vehicle trajectory with and without controller are shown in Figure 6.74. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.75. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.76. The vehicle yaw rate and lateral acceleration response with and without controller are given in Figure 6.77 and Figure 6.78.

![Figure 6.73 Vehicle speed time history](image)

![Figure 6.74 Vehicle trajectory](image)
Figure 6.75 Total wheel driving moment time history

Figure 6.76 Wheel Longitudinal slip time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be within the desired values obtained from the bicycle model. In addition, the developed controller improved the vehicle traction performance and directional stability at high speeds.

(b) **Test maneuver on high friction surface (0.8)**

1. **FMVSS 126 ESC at 40 km/h**

The initial vehicle speed was 40 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.79. Vehicle trajectory with and without controller are shown in Figure 6.81. The total wheel driving moment of the four axles with and without the integrated yaw-slip controller are shown in Figure 6.80. The wheel longitudinal slip for the
four axles with and without controller are shown in Figure 6.82. The vehicle yaw rate and lateral acceleration response with and without controller are given in Figure 6.83 and Figure 6.84.

Figure 6.79 Vehicle speed time history

Figure 6.80 Total wheel driving moment time history
Figure 6.81 Vehicle trajectory

Figure 6.82 Wheel Longitudinal slip time history

Figure 6.83 Vehicle yaw rate time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce the yaw rate to be within the desired values obtained from the bicycle model. In addition, the lateral acceleration increased in comparison with the vehicle without controller but still below the desired values obtained from the bicycle model.

2- **FMVSS 126 ESC at 80 km/h**

The initial vehicle speed was 80 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.85. Vehicle trajectory with and without controller are shown in Figure 6.88. The total wheel driving moment of the four axles with and without the integrated yaw-slip controller are shown in Figure 6.86. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.87. The vehicle yaw rate and lateral acceleration with and without controller are given in Figure 6.89 and Figure 6.90.
Figure 6.86 Total wheel driving moment time history

Figure 6.87 Wheel Longitudinal slip time history
Figure 6.88 Vehicle trajectory

Figure 6.89 Vehicle yaw rate time history

Figure 6.90 Vehicle lateral acceleration time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be below the desired values obtained from the bicycle model. In addition, the developed controller improved the vehicle traction performance and directional stability at high speeds.

(c) **Test maneuver on soft soil**

1- **FMVSS 126 ESC at 60 km/h**

The initial vehicle speed was 60 km/h with constant throttle control from the driver of 1.0 as shown in Figure 6.91. The vehicle trajectory for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.92. The total wheel driving moment of the four axles with and without the controller are shown in Figure 6.93. The wheel longitudinal slip for the four axles with and without the controller are shown in Figure 6.94. The vehicle yaw rate and lateral acceleration responses with and without controller are given in Figure 6.95 and Figure 6.96.

![Vehicle speed time history](image1)

![Vehicle trajectory](image2)

Figure 6.91 Vehicle speed time history

Figure 6.92 Vehicle trajectory
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be within the desired values obtained from the bicycle model. In addition, the vehicle traction performance has been improved as the controller succeeded in maintaining the tire longitudinal slip within the recommended range for soft soil (0.6 ± 0.05) and the vehicle speed increased from 80 km/h to 107 km/h.
The initial vehicle speed was 80 km/h with constant throttle control from the driver of 0.7 as shown in Figure 6.97. Vehicle trajectory of the combat vehicle with and without controller are shown in Figure 6.98. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.99. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.100. The vehicle yaw rate and lateral acceleration response with and without controller are given in Figure 6.101 and Figure 6.102.
Figure 6.97 Vehicle speed time history

Figure 6.98 Vehicle trajectory

Figure 6.99 Total wheel driving moment time history
Figure 6.100 Wheel Longitudinal slip time history

Figure 6.101 Vehicle yaw rate time history

Figure 6.102 Vehicle lateral acceleration time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be below the desired values obtained from the bicycle model. In addition, the vehicle traction performance has been improved as the controller succeeded in maintaining the tire longitudinal slip within the recommended range for soft soil (0.6 ± 0.05) and the vehicle speed increased from 84 km/h to 120 km/h.

6.3.3.2 J-TURN (STEP STEER)

A standard J-turn test has been performed to investigate the vehicle performance characteristics like its tracking ability in a sudden steer angle change (step steer). The steering wheel input used in the simulation as shown in Figure 6.34

(a) J-Turn test at 40 km/h on low friction surface (0.2)

The initial vehicle speed was 40 km/h with constant throttle control from the driver of 1.0 as shown in Figure 6.103. Vehicle trajectory with and without controller are shown in Figure 6.106. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.104. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.105. The vehicle yaw rate and lateral acceleration response with and without controller are given in Figure 6.107 and Figure 6.108.

![Vehicle speed time history](image_url)
Figure 6.104 Total wheel driving moment time history

Figure 6.105 Wheel Longitudinal slip time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be within the desired values obtained from the bicycle model.
(b) J-Turn test at 80 km/h on low friction surface (0.2)

The initial vehicle speed was 80 km/h with constant throttle control from the driver of 1.0 as shown in Figure 6.109. Vehicle trajectory with and without controller are shown in Figure 6.110. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.111. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.112. The vehicle yaw rate and lateral acceleration response with and without controller are given in Figure 6.113 and Figure 6.114.

![Vehicle speed time history](image1)

**Figure 6.109 Vehicle speed time history**

![Vehicle trajectory](image2)

**Figure 6.110 Vehicle trajectory**
Figure 6.111 Total wheel driving moment time history

Figure 6.112 Wheel Longitudinal slip time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be within the desired values obtained from the bicycle model.

### 6.3.3.3 FISH-HOOK MANEUVER

A standard fish-hook maneuver test designed by National Highway Traffic Safety Administration (NHTSA) for prompting and analyzing dynamic rollover has been modified to investigate the proposed control strategy. In this test, the steering wheel input used in the simulation was calculated to produce about 0.3g lateral acceleration as shown in Figure 6.44.
(a) *Fish-hook maneuver at 30 km/h on low friction surface (0.2)*

The initial vehicle speed was 30 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.115. Vehicle trajectory with and without controller are shown in Figure 6.116. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.117. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.118. The vehicle yaw rate and lateral acceleration response with and without controller are given in Figure 6.119 and Figure 6.120.

![Figure 6.115 Vehicle speed time history](image)

![Figure 6.116 Vehicle trajectory](image)
Figure 6.117 Total wheel driving moment time history

Figure 6.118 Wheel Longitudinal slip time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be within the desired values obtained from the bicycle model.

**(b) Fish-hook maneuver at 50 km/h on low friction surface (0.2)**

The initial vehicle speed was 50 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.121. Vehicle trajectory with and without controller are shown in Figure 6.122. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.123. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.124. The vehicle yaw rate and lateral acceleration response with and without controller are given in Figure 6.125 and Figure 6.126.
Figure 6.121 Vehicle speed time history

Figure 6.122 Vehicle trajectory

Figure 6.123 Total wheel driving moment time history
Figure 6.124 Wheel Longitudinal slip time history

Figure 6.125 Vehicle yaw rate time history

Figure 6.126 Vehicle lateral acceleration time history
From the simulation results, it can be noticed that the developed controller succeeded to reduce both lateral acceleration and yaw rate to be within the desired values obtained from the bicycle model.

6.3.3.4 Constant radius lateral acceleration

In this test, the vehicle test course of 100ft. radius, Figure 6.53, was used to verify the effectiveness of the proposed control strategy and its effect on vehicle directional stability.

(a) Constant radius lateral acceleration on rigid surface

1- Constant radius lateral acceleration at 20 km/h

The initial vehicle speed was 20 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.127. Vehicle trajectory with and without controller are shown in Figure 6.128. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.129. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.130.

![Figure 6.127 Vehicle speed time history](image1)

![Figure 6.128 Vehicle trajectory](image2)
From the simulation results, it can be noticed that both vehicles with and without the controller succeeded in completing the simulation test course.
2- Constant radius lateral acceleration at 30 km/h

The initial vehicle speed was 30 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.131. Vehicle trajectory with and without controller are shown in Figure 6.132. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.133. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.134.
From the simulation results, it can be noticed that the vehicle without controller was not able to complete the simulation test course correctly, while the vehicle equipped with the controller succeeded in completing the simulation test course but in larger circle diameter.
3- Constant radius lateral acceleration at 40 km/h

The initial vehicle speed was 40 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.135. Vehicle trajectory with and without controller are shown in Figure 6.136. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.137. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.138.

![Figure 6.135 Vehicle speed time history](image)

![Figure 6.136 Vehicle trajectory](image)
From the simulation results, it can be noticed that both vehicles with and without controller was not able to complete the simulation test course correctly.
(b) Constant radius lateral acceleration on soft soil

1- Constant radius lateral acceleration at 20 km/h

The initial vehicle speed was 20 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.139. Vehicle trajectory with and without controller are shown in Figure 6.140. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.141. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.142.

Figure 6.139 Vehicle speed time history

Figure 6.140 Vehicle trajectory
From the simulation results, it can be noticed that both vehicles with and without the controller succeeded in completing the simulation test course successfully. In addition, the vehicle speed increased from 25 km/h to 34 km/h.
2- Constant radius lateral acceleration at 30 km/h

The initial vehicle speed was 30 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.143. Vehicle trajectory with and without controller are shown in Figure 6.144. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.145. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.146.

Figure 6.143 Vehicle speed time history

Figure 6.144 Vehicle trajectory
From the simulation results, it can be noticed that both vehicles with and without the controller succeeded in completing the simulation test course successfully. In addition, the vehicle speed increased from 28 km/h to 34 km/h.
3- Constant radius lateral acceleration at 40 km/h

The initial vehicle speed was 40 km/h with constant throttle control from the driver of 0.2 as shown in Figure 6.147. Vehicle trajectory with and without controller are shown in Figure 6.148. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.149. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.150.

![Vehicle speed time history](image)

**Figure 6.147** Vehicle speed time history

![Vehicle trajectory](image)

**Figure 6.148** Vehicle trajectory
From the simulation results, it can be noticed that both vehicles did not succeed in completing the original test as the starting speed was 40 km/h. In addition, it can be concluded that the maximum speed for the vehicle equipped with the controller to complete the test course successfully is 34 km/h.
6.3.3.5 Acceleration test on uniform low friction surface

In this test, the vehicle initial speed was zero with a ramp to full throttle in 1.0 sec as shown in Figure 6.151 and Figure 6.152. In addition, there is no steering wheel input during the simulation. The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.153. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.154.

![Figure 6.151 Throttle position time history](image1)

![Figure 6.152 Vehicle speed time history](image2)
From the simulation results, it can be noticed that the controller succeeded in limiting the driving wheels longitudinal slip. Moreover, the combat vehicle traction performance improved as the vehicle speed increased from 80 km/h to 106 km/h.
6.3.3.6 Acceleration test on Split Mu (0.2L/0.8R)

In this test, one side of the vehicle is on a high-coefficient of friction surface (0.8) and the other side is on a low-coefficient of friction surface (0.2), the vehicle initial speed was zero with a ramp to full throttle in 1.0 sec as shown in Figure 6.151 and Figure 6.155. In addition, there is no steering wheel input during the simulation.

The total wheel driving moment of the four axles for the combat vehicle with and without the integrated yaw-slip controller are shown in Figure 6.156. The wheel longitudinal slip for the four axles with and without controller are shown in Figure 6.157.

![Figure 6.155 Vehicle speed time history](image)

![Figure 6.156 Total wheel driving moment time history](image)
From the simulation results, it can be noticed that the controller succeeded in distributing the driving torque to maintain the driving wheels longitudinal slip within the recommended range for better traction performance on rigid surfaces (0.2 ± 0.05).

6.3 Summary

This chapter presents the integration of slip controller with the developed active yaw controller design for two powertrain configurations (8x4 and 8x8) of the multi-wheeled combat vehicle. The developed and validated FEA tire and soft soil models have been used to represent the tire-soil interaction characteristics for the simulations performed on soil.

All the powertrain components starting from engine to the axle differentials have been externally modeled in Matlab/Simulink in case of 8x8-powertrain configuration while in the 8x4-powertrain configuration, the powertrain components were internally modeled within TruckSim.

The advantage of externally molded powertrain will lead to improvement and flexibility in making modifications to the powertrain design and characteristics. The developed and validated vehicle model using TruckSim and the developed yaw controller have been used...
to investigate the developed slip controller integration on both rigid road and soft soil to verify its effectiveness.

The newly design control strategy (four-axle torque vectoring) succeeded in improving vehicle directional stability whenever the tire slip exists with high values. In addition, the combat vehicle traction performance has been improved.
Chapter 7

Conclusions and Future Work

7.1 Motivations

Active torque distribution (torque vectoring) is a new technology that has been developed based on the standard differential. Torque vectoring differentials act like standard differentials while allowing the torque to be transmitted between wheels/axles independently at the same time. In addition, they were initially used for racing and passenger cars applications and the technology is now being used in most of the all-wheel drive vehicles. Till now, there has been a lack of a control strategy approach for enhancing off-road vehicles mobility performance on soft terrain.

The primary goal for vehicle designers now is to increase vehicle speed over soft terrains and maintaining the vehicle directional stability and enhancing its mobility performance. Off-road vehicles can be more sensitive to these demands than passenger cars due to its high ground clearances. Consequently, during cornering maneuvers, large lateral weight transfers can cause significant changes in tire-soil contact conditions such as sinkage and longitudinal slip. Furthermore, the vehicle sideslip and yaw motion are dependent on vehicle design parameters, tire characteristics and the mechanical properties of the terrain.

In this research, a multi-wheeled combat vehicle is represented by a 22-DOF model in TruckSim software package including all relevant subsystems such as vehicle body, suspension, steering system and wheel dynamics. The vehicle model also incorporates body dynamics for different drivetrain systems (8x4 and 8x8) including sources of torsional damping and stiffness in axles/shafts. Within the drivetrain model, the driving torque is transmitted/regulated by the various torque distribution devices. At the output end of the system, the torque is regulated by the interaction between the tires and the terrain.

The employment of a detailed model of any drivetrain system would be worthless without similarly representation of the tire-soil interaction. Therefore, FEA tire-soft soil model was
developed to represent the tire-soil interaction characteristics and the selected soil type for the current research was clayey soil. In order to validate the developed combat vehicle model, the vehicle model was tested in four different test courses, Double Lane Change, Constant Step Slalom, J-Turn with 8x4 powertrain drive and Turning circle test with two different powertrain configurations (8x4 and 8x8). All the test courses have been performed on rigid road. The measurements were carried out by General Dynamics Land Systems-Canada (GDLS-Canada) to capture vehicle responses; yaw rate and lateral acceleration. Experimental results were compared with the simulation results, showing satisfactory predictions.

Active torque distribution control (yaw controller) has been developed for the multi-wheeled combat vehicle with 8x4-powertrain drive. The controller generate correcting corrective yaw moment based on the deviation of desired and actual vehicle responses and passes this information to the torque vectoring differentials. In addition, an advanced fuzzy slip control system has been developed and integrated with the yaw controller. The integrated yaw-slip controller has been presented for different powertrain configurations (8x4 and 8x8) and tested on rigid surface and soft soil to investigate its effectiveness on vehicle traction performance and directional stability.

The integration of all control systems resulted in a complex model, which was implemented in MATLAB/Simulink environment. This modeling approach can be used to support the multi-wheeled combat vehicle design engineers and manufacturers in the following manner:

- Simulation of a wide variety of conditions including ride, traction and handling tests using high degree of sophistication for multi-wheeled vehicle models. However, the main strength of the model is the inclusion of a detailed drivetrain model for 8x8-powertrain configuration.
- Investigation of components’ selection, particularly those related to drivetrain gearing/coupling design, which would produce the characteristics, best suited to the proposed vehicle.
- Possibility for the future integration of advanced control strategies and automatic
optimization techniques for multi-wheeled vehicles.

- Possibility for future presenting different types of soil to investigate the multi-wheeled vehicle mobility performance using FEA modeling to characterize tire-soil interaction rather than using complex/commercial off-road tire models, which often require special know-how in order to be adapted to particular design requirements.

7.2 Findings and Conclusions

Measured non-linear tire cornering characteristics look-up tables were used for the simulations on rigid terrains. While, on soft terrain three-dimensional non-linear Finite Element Analysis (FEA) tire-soil models were developed and its interaction characteristics were predicted using PAM-CRASH and used in the vehicle model. The predictions of the vehicle handling characteristics and transient response during lane change on rigid road at different vehicle speeds were compared with field tests results obtained from the industry partner. Measured and predicted results were compared based on vehicle steering, yaw rates and accelerations. Published US Army validation criteria have been used to validate the simulation models. The models showed very good agreement with the measurement.

The developed vehicle simulation model was successfully employed in order to predict both traction and handling characteristics for different powertrain configurations (8x4 and 8x8). The traction performance is evaluated based on vehicle traction capabilities for a specific power plant system in terms of maximum speed. The handling characteristics have been examined under both transient and steady state conditions during different standard cornering maneuvers.

Torque distribution strategy was developed based on both yaw rate and lateral acceleration deviations from the desired values to achieve the required corrective yaw moment. Standard test maneuvers such as fish-hook maneuver, FMVSS 126 ESC test, J-turn were appropriately modified and used for evaluating the effectiveness of the developed torque distribution strategy on both rigid surface and soft soil.
The main conclusions of this work can be summarized as:

- The developed torque vectoring control strategy can be widely applied to vehicles of two or more axles. In this research work, the application to multi-wheeled combat vehicles is extensively investigated.

- The simulation model of the combat vehicle was extended to simulate the vehicle performance on both rigid and soft terrains by integrating the developed three-dimensional non-linear Finite Element Analysis (FEA) tire model and the heavy clayey soil model. Other soil types can be easily added to the model. The yaw rate controller was found to be effective in enhancing yaw rate and lateral acceleration of the vehicle on dry and slippery surface conditions.

- The torque distribution strategy (corrective torques being added to left wheels and subtracted from right wheels in case of left turn) was found to be effective when considering various control parameters and its ability to achieve the realistic results as presented in Chapter 5 of this thesis.

- The observed rollover of the 8x4 vehicle configuration during the Constant Step Slalom (NATO AVTP-1 03-30), J-Turn (75ft radius) and severe Fish-Hook maneuvers could be prevented using the developed yaw rate controller.

- It was found that the directional performance of the multi-wheeled vehicles directional stability is highly influenced by the way the driving torque is distributed between the axles/wheels. This is particularly true where the lateral forces are regulated by the longitudinal slip and the tractive forces at the tires.

- The ability to simulate the complex 8x8 combat vehicle configuration was enhanced for the first time by modeling the powertrain components starting from engine to the axle differentials which was externally modeled in Matlab/Simulink.

- The newly developed fuzzy slip controller succeeded in keeping the driving wheels longitudinal slip within the recommended range on rigid surface (0.2±0.05) and on soft soil (0.6±0.05) to achieve the max available traction performance of the vehicle.

- The newly design integrated controller (yaw-fuzzy slip) strategy (four-axle torque vectoring) succeeded in improving vehicle directional stability whenever the tire slip exists with high values.

- The developed torque vectoring controller has improved the combat vehicle traction
performance on both rigid and soft terrain.

7.3 Future Work and Recommendation

In this research, FEA technique has been used to represent the selected soil for off-road simulation (Clayey soil) which has some limitations to replicate the soil shear strength and a large soil sinkage accurately. However, the meshless modeling method of Smooth Particle Hydrodynamics (SPH) may be a viable approach to more accurately simulation of large soil deformations and complex tire-soil interactions. Conducting field test on soft soil and measuring vehicle behavior in both longitudinal and lateral directions during typical maneuvers, aimed at verification of the entire vehicle model, can be another avenue for further research.

In this research, the upper controllers were simple PID controllers that require fine-tuning to handle different operating conditions. Therefore, more robust and non-linear controllers can be designed instead, to control non-linearities and uncertainties in the model more effectively.

It is very important to examine the robustness of the developed control systems against internal and external disturbances. This should be done both in the simulation environment and, later, through various field testing on both rigid terrain and soft soil.

It is also recommended to integrate a friction coefficient estimator, since the bicycle model and the maximum torque estimator of the developed torque vectoring controller require knowledge of the current friction coefficient between the tire and the road in order to adequately adapt different road conditions.

Finally, the results in this research establish first steps towards the selection of a combination of torque-distribution strategy and feedback controller for ensuring vehicle stability with independent drives. As an extension of the work, the computed torque magnitudes and time responses can be utilized into the design or selection of the electric motors or hydraulic motors for independent drive systems.
References


