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DEPARTMENT OF MECHANICAL ENGINEERING

**Implementation of power-steering in  
a multibody dynamic model of a  
tractor semi-trailer**

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Internship report

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# Abbreviations and Symbols

## Abbreviations

MBD – Multi Body Dynamics

ADAMS - Automatic Dynamic Analysis of Mechanical Systems

HPAS – Hydraulic Power Assist Steering

ADE - Algebraic Differential Equations

GSE – General State Equation

DLL – Dynamic Link Library

## Symbols

$k_{tb}$	Torsion bar stiffness	[Nm/rad]	$J_{sw}$	Steering wheel inertia	[kg*m <sup>2</sup> ]
$k_{sc}$	Steering column stiffness	[Nm/rad]	$\delta_{sw}$	Steering wheel angle	[rad]
$k_{ha}$	Steering house stiffness	[Nm/rad]	$J_{bb}$	Bevel box inertia	[kg*m <sup>2</sup> ]
$T_{fric,bb}$	Bevel box friction torque	[Nm]	$\delta_{bb}$	Bevel box angle	[rad]
$T_{fric,ha}$	Steering house friction torque	[Nm]	$P_a$	Pressure chamber a	[bar]
$T_{ps}$	Power steering torque	[Nm]	$P_b$	Pressure chamber b	[bar]
$d_{hua}$	Hydraulically unactuated damping	[Nm*s/rad]	$P_s$	Pressure supply	[bar]
$d_{ha}$	Hydraulically actuated damping	[Nm*s/rad]	$A_{(i)}$	Valve opening area	[m <sup>2</sup> ]
$i_{sh}$	Piston-sector shaft ratio	constant	$R_{(i)}$	Flow resistance	[N/ms]
$\Delta_{tb}$	Torsion bar free play	[rad]	$Q_{x(i)}$	Flow rate	[m <sup>3</sup> /s]
$T_{tb}$	Torsion bar torque	[Nm]	$V_{(i)}$	Cylinder volume	[m <sup>3</sup> ]
$x_{hp}$	Hydro pulse displacement	[m]			

## Acknowledgement

The author is immensely thankful to his external supervisor Paul Bakker for his guidance and support . Jan Loof for his insights into the project on numerous occasions. Also, Prof. Henk Nijmeijer and Dr. Igo Besselink for the opportunity to work on this project.

## Summary

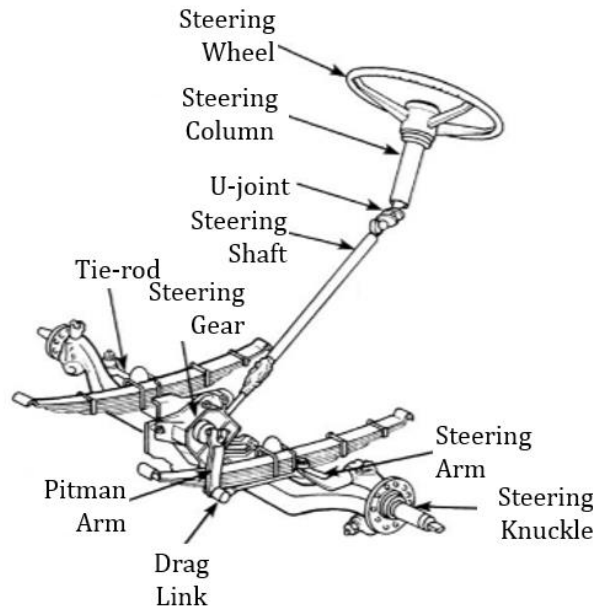
Hydraulic Power Assist Steering (HPAS) systems help the driver maneuver the vehicle by augmenting the effort at the steering wheel. The steering gear aims to achieve light steering operation and high driver comfort without compromising on the amount of road feedback. Commercial vehicles are driven for prolonged periods of time and factors like steering effort and fatigue are very relevant to evaluate performance of the power steering system.

The DAF vehicle multibody dynamic (MBD) model used for simulation represents the steering gear with a constant ratio joint between the steering column output and pitman arm, this gives unrealistic steering efforts in vehicle handling simulations. A detailed steering gear simulink model developed by Jan Loof at TU/e is included into the current steering template used at DAF by means of co-simulation between ADAMS and Simulink. A two stage verification process is carried out. The first stage checks the performance of the co-simulation methodology and the second stage verifies the behavior of the steering gear during co-simulation. At the end of the internship a vehicle model is obtained with higher fidelity in terms of steering wheel torque , the model can also be used to access the performance of the HPAS. A methodology is formulated which allows for simple implementation of electronic control logic in the ADAMS vehicle model.

# 1 Introduction

This document is an internship report as a part of the Automotive Technology Master's Program at Technical University of Eindhoven. The student is a member of the Dynamics and Control specialization group of the Department of Mechanical Engineering. The project was carried out from September 2016 to November 2016 at the Technical Analysis department of DAF Trucks NV under the supervision of Paul Bakker and with support from Jan Loof at the TU/e side.

1.1 **Background** - Multi body Dynamics (MBD) simulations are used to investigate the dynamic behavior of a system of rigid and flexible bodies. The bodies are constrained in both the kinematic and dynamic sense, they experience actuation based on the nature of simulation. MSC Adams is used at DAF Trucks to model and simulate vehicles in their operational environments. The most common goals of these analysis are studies regarding vehicle handling, load generation for durability tests, steering and suspension design optimization etc.



*Figure 1. Typical layout of a truck steering system.*

Figure 1 illustrates a typical truck steering system. Based on their functionality the steering system can be divided into two major groups of mechanisms. The first one includes the steering wheel and a number of links interconnected by universal and revolute joints. They convey the rotational motion actuated by the driver to the steering gearbox. The second group includes the pitman arm and draglink which convey rotational output motion from the steering gearbox to the wheels via the knuckles and tie-rod. The steering gearbox connects these two groups while amplifying the torque provided by the driver by hydraulic assistance.

Till date, the MBD models used at DAF are able to capture the kinematics of both the groups mentioned above. The hydraulic support of the steering gear was however not included in the

MBD models. A fixed ratio reduction gear is used to convey the motion of the steering column output to the input motion of the pitman arm.

A detailed Simulink model of a truck steering system model has been developed by Jan Loof, PhD candidate at Department of Mechanical Engineering, Technical University of Eindhoven. This model takes into account the hydraulic power assist steering (HPAS) feature. It has been validated based on test-rig measurements and full vehicle simulations with SimMechanics. The key features of the steering model are :

1. It captures the stiffness between the steering system components.
2. It uses a reset-integrator model to capture the dry and viscous friction at the bearings.
3. The model simulates working of the hydraulic power-steering gearbox through a Wheatstone bridge approximation.
4. The model validation has shown that it can accurately predict the steering wheel torque.

**1.2 Purpose** - The goal of the internship is to explore the possibilities to integrate the power-steering model into the MBD vehicle model and assess the performance of the combined model. This will involve co-simulation between Simulink and Adams. Successful integration of the power-steering system into the DAF vehicle model will increase the fidelity of the model. With this improvement the MBD model can be used to evaluate the power-steering system, driver steering effort and studies regarding active steering systems.

Objective of this project are

1. Implementation of power-steering model in ADAMS full vehicle model.
2. Investigate different methods of co-simulation if needed.
3. Verification of the co-simulated model against original hydraulic model.
4. Assess the calculation performance of the complete model.

**1.3 Report Outline** - The report starts with Literature study in Chapter 2. The modelling aspects are discussed in Chapter 3 while the test cases and verification results are presented in Chapter 4. Chapter 5 concludes the work done in this internship and suggests possible improvements and scope of further work.

## 2 Literature Review

### 2.1 Multi Body Dynamics

A mechanism is a part of a machine responsible for transmitting forces and motion in order to fulfill the functionality of the machine. Multibody dynamics is the study of the mechanism by modelling it as a collection of rigid and flexible linkages. These components are connected using kinematic constraints, force elements, friction elements and actuators. Msc ADAMS is a simulation software which systematically generates and solves the system equations so as to predict the motion of the bodies involved and the forces required to cause certain motions in the system [4]. The modelling approach typically involves assembly of the mechanism components in the desired location and orientation, definition of joints and other constraints and specification of displacement or force actuations as required for the analysis.

### 2.2 Co-simulation

Multiple domains and dependent variables describing different simulations are coupled systems when neither domain can be solved independently and the dependent variables cannot be eliminated at the system differential equation level [1]. [2] defines Co-Simulation as a coupled problem modelled and simulated in different environments through a segregated approach. This project involves a coupled problem between two different systems. The first system involves simulation of vehicle behavior in a MBD environment (ADAMS ) while the second one involves simulation of a hydraulic steering system in the simulink environment. Co-Simulation involves various challenges like integration between different software, accurate data exchange, coupling of systems with different dynamic behavior and guarantee of the accuracy of simulation [3]. This project requires a reliable modelling methodology which ensures correct data exchange between both the models. Also, interaction between the software environments should not cause any compromises in their respective computations.

### 2.3 Steering gear model by Jan Loof

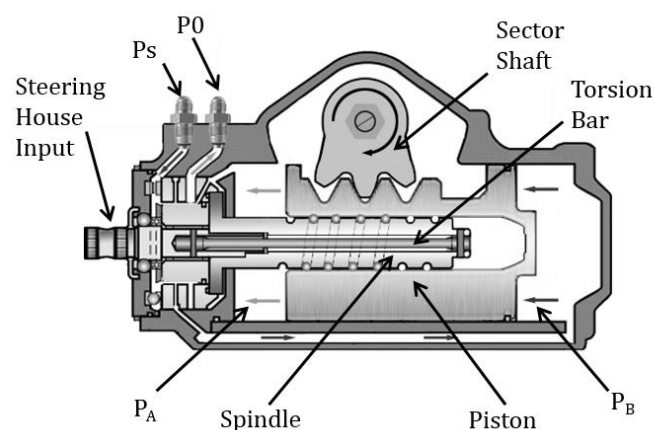


Figure 2. Cross-section of a typical hydraulically assisted steering gear-box.

Figure 2 shows the cross-section of a typical steering gear-box. The torsion bar receives the rotational input from the bevel-box, its motion is transferred to the spindle. The torsion bar has limited amount of twist, this end stop prevents the torsion bar from over actuating the hydraulic

assist. In the model this non-linear phenomenon is modelled as a parallel free-play. The torsion bar motion in the free-play range is utilized to actuate the valves controlling the pressures  $P_A$  and  $P_B$  on either sides of the piston. The spindle is a screw and it converts its rotational motion to translational motion of the piston through recirculating-ball gear arrangement. The piston receives only hydraulic actuation when the torsion bar is twisted. Beyond the free-play range the piston has the maximum hydraulic assist. The translational motion of the piston is converted to rotational motion of the sector shaft through a rack and pinion arrangement. The pitman arm is physically connected to the sector shaft via spline teeth.

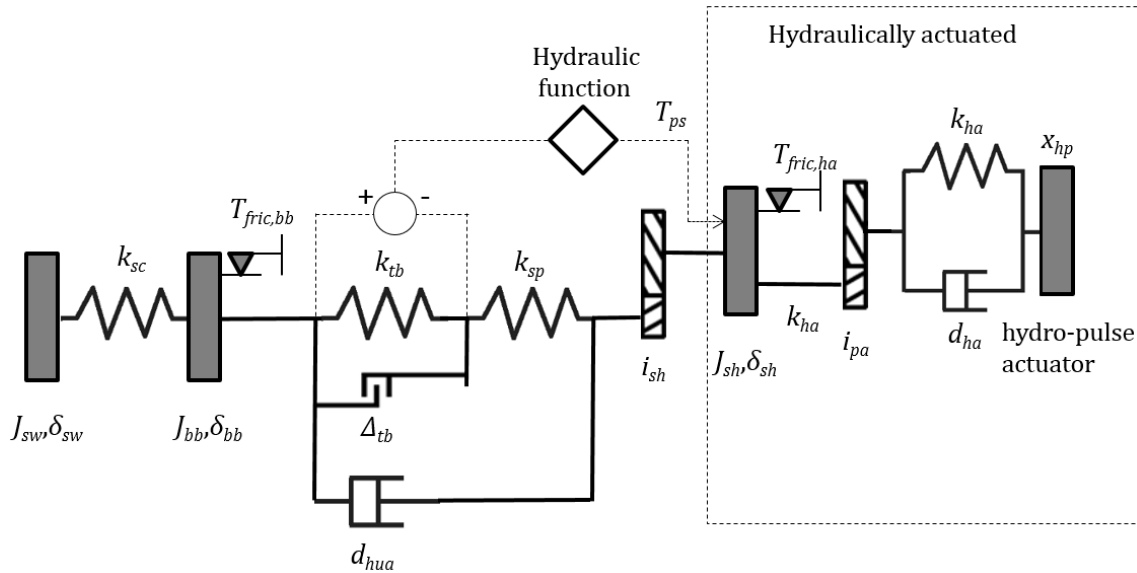


Figure 3. Steering system model.

Figure 3 represents the *original* steering system model. The steering wheel has inertia  $J_{sw}$  and an angle  $\delta_{sw}$ . The steering wheel is connected to the bevel-box through a stiffness  $k_{sc}$ . The bevel box has an inertia  $J_{bb}$  and its displacement  $\delta_{bb}$  is passed into the steering house block. A frictional torque  $T_{fric,bb}$  acts on the bevel box inertia. Both dry and viscous friction are modelled using a reset-integrator model, this method was chosen due to its straight-forward implementation and computational performance [6].

The Steering gear-box input passes through torsion bar stiffness  $k_{tb}$ , spindle stiffness  $k_{sp}$  and a damper of damping coefficient  $d_{hua}$ . The deflection of the torsion bar is limited to enable the vehicle to steer in the case of power-steering failure, this is modelled by a parallel free play  $\Delta_{tb}$  [5]. The ratio between the piston and the sector shaft  $i_{sh}$ . The output of the steering gearbox has an inertia  $J_{sh}$  and an angle  $\delta_{sh}$ . The equivalent inertia of the spindle, piston and sector shaft is represented by  $J_{sh}$ . A friction torque  $T_{fric,ha}$  resulting from the seals in-between the piston and the cylinder is applied to the steering gear-box inertia.



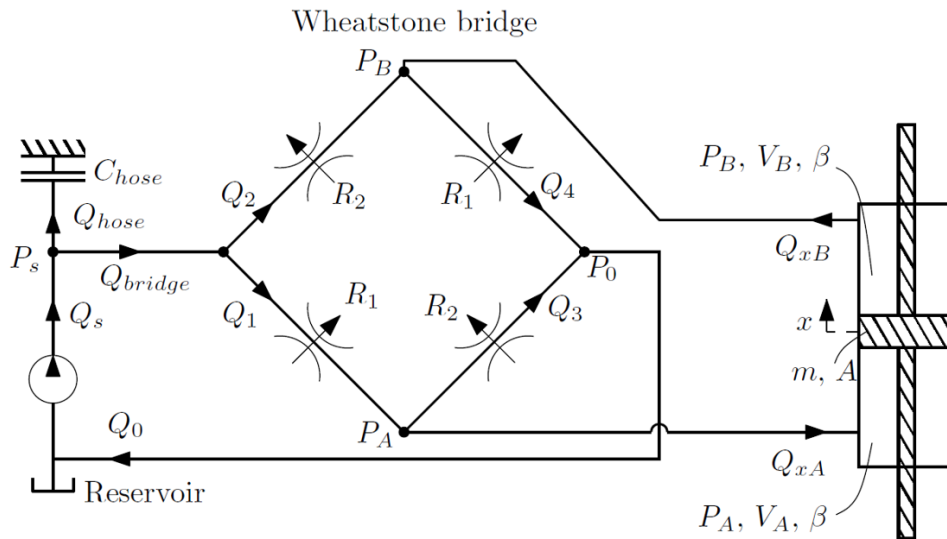


Figure 4. Wheatstone bridge approximation of flow in steering gear-box.(Source: [5])

The torsion bar controls the hydraulic flow which governs the pressure difference between the two faces of the piston. A rotary spool valve is used to regulate the flow. The valve inner housing rotates with the torsion bar while the valve outer housing is fixed, difference in their axial position causes a change in valve area of the different flow paths.

From figure 4 four different flow paths can be identified and these can be modelled together as four equivalent flows and the respective resistances. Due to the construction of the spool valve, while the piston moves due to a higher  $P_A$  the paths  $Q_{bridge}$  to  $Q_{xA}$  and  $Q_{xB}$  to  $Q_0$  open with the same amount. This also applies when the steering is in the opposite direction. Two unique flow resistances identified here are  $R_1$  and  $R_2$ . This explains the basic working of the hydraulic system. The various computations inside the hydraulic function block will be discussed at a later stage as they have been converted into simulink diagrams in order to accommodate the co-simulation process.

### 3 Steering System Model

Here by we shall discuss three different implementations of the steering model developed by Jan Loof at TU/e. First a simulink model which contains the power-steering model in combination with a test-rig environment used to validate the model, this is referred to as the *original* model. Thereafter this mode is altered in order to accommodate the co-simulation process, this is called the *modified* model. This power steering model is compiled such that it can be implemented in the MBD environment. In ADAMS a similar test-rig model is created, this is referred to as the *co-sim* model. Initial verification is carried out using these three models. At later stages the steering gear model is implemented in a vehicle model, this is the ADAMS model with power steering. This model shall be compared to the ADAMS vehicle model with fixed reduction ratio for the steering gear.

#### 3.1 Original Steering Test-rig model

The *original* model has been validated against measured data from a steering gear test-rig setup. The test-rig contains a steering model shown in figure 5. At the driver end, the steering column and intermediate shafts have been removed. The steering wheel is directly fixed to the input of the bevel box, it can be locked using a mechanical brake to constrain its motion. At the bottom the pitman arm is constrained with respect to the test-rig frame using a revolute joint and it is actuated by the torque available from the steering gear. The draglink is connected on one end to the pitman arm using a spherical joint. On the other end it is connected to the hydro-pulse actuator piston through an universal joint. The hydro-pulse actuator piston is connected to the frame using a prismatic joint and it experiences a force  $F_{hp}$  and displacement  $x_{hp}$ .

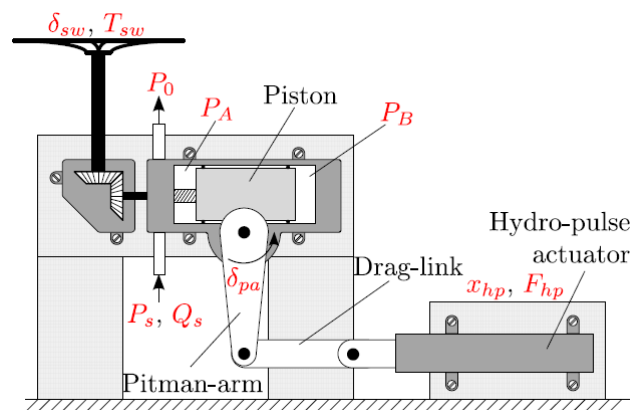


Figure 5. Diagram of the test-rig setup. (Source: [5])

Figure 6 is a schematic diagram which illustrates the data flow between the different component subsystems and broadly segregates the model into two categories based on the presence of hydraulic actuation. The MATLAB model of the power-steering system is built based on these basic subsystems of the steering system.

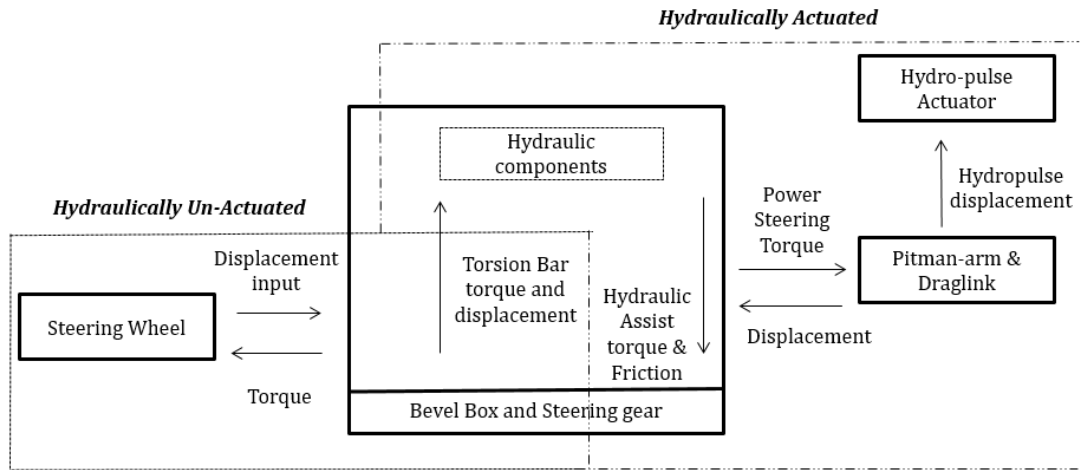


Figure 6. Data flow between different steering system components in SIMULINK.

### 3.2 Modified Steering Gear Model .

#### 3.2.1 Model composition and data flow.

In the MBD vehicle model currently used at DAF the steering gear is represented by a fixed gear ratio. This ratio will be replaced by the hydraulic power-steering model in a way that it requires minimal modifications to the existing vehicle model. Based on this criteria the original steering model was cut at two locations. The steering wheel, column and intermediate shafts were retained in the MBD environment, the first cut in the *original* steering model was made before the bevel box. On the other end the pitman arm was kept intact in the MBD model and the *original* model was cut between the steering-gearbox and the pitman arm.

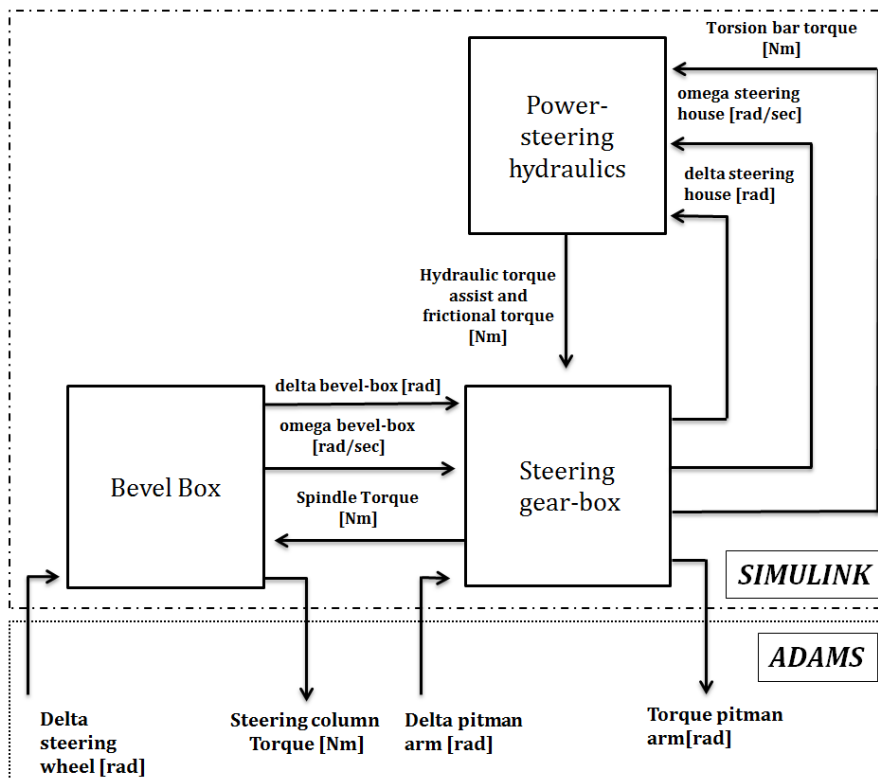


Figure 7. Data flow between ADAMS and SIMULINK in the co-simulation process.

The *modified* steering model consisted of three functional blocks bevel-box, steering gear and the hydraulic block. The signals previously provided by the steering wheel block to the bevel box block will be an output from the ADAMS model. The signals provided by the steering gear-box block to the pitman arm block will instead be an input into the ADAMS model. It should be mentioned that the computation inside the bevel box block accounts for the stiffness and friction of the steering column and intermediate shafts, the ADAMS model currently does not contain stiffness and friction in the steering column. Likewise the stiffness of the pitman arm and draglink have been included in the steering gear-box block and they shall be excluded in the ADAMS model. Figure 7 illustrates the flow of signals between the various simulink blocks and the signals exchanged between simulink and ADAMS.

### 3.2.2 Re-modelling of hydraulic block

Co-simulation between ADAMS and simulink is subjected to restrictions. The Real Time workshop (RTW) is a tool to convert simulink models to C language code and executable. The ADAMS Target in RTW is used to create Dynamic Link Library (DLL) that can be used together with the ADAMS Solver. This required modifications to the *original* steering model and a complete remodeling of the hydraulic sub block of the steering gear model. It should be mentioned that a considerable amount of time was spent trying to modify the *original* steering test-rig model to make it compatible with the co-simulation process. Many of these changes were done through trial and error process due to lack of adequate documentation. This report shall at a later stage document all the salient points which are essential for co-simulation and list the important checks which will eliminate the errors arising from the process. Some of the restrictions were as follows:

1. The simulink model cannot have structured variable parameters, all the parameter generation script files had to be modified such that the parameters were now simple MATLAB variables.
2. The simulink model cannot have MATLAB function blocks.
3. Valve area  $A_1$ ,  $A_2$  computation was modified to look up tables, this allows easy modification of valve characteristics.

The MATLAB function block accounting for the hydraulics of the steering gear was replaced by the simulink model, it was constructed in the same way as the computations in the MATLAB function. The following paragraphs describe the computation logic and flow of the new hydraulic block in simulink.

In order to determine the flow resistances  $R_1$  and  $R_2$  it is required to compute the available valve areas  $A_1$  and  $A_2$  for different positions of torsion bar torque. Due to the cross-sectional geometry of the valves,  $A_1$  and  $A_2$  when related to  $T_{tb}$  are non-linear by nature. The pressure drops to minimal values in the on-center region and this causes the change in behavior in this region. Curve fitting was carried out to estimate the equations of the valve areas. A scaled version of the actual graph represents the behavior of  $A_1$  and  $A_2$  as a function of the torsion bar torque in figure 8 .

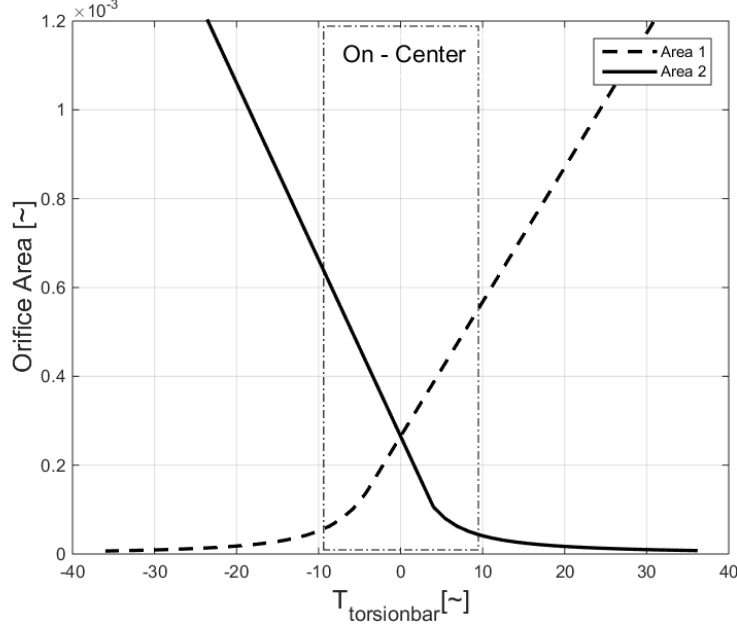


Figure 8. Valve area estimation as a function of  $T_{tb}$ .

The modelling in simulink was done to replace the computations performed in the embedded m-file of the original steering model. Equations (1),(2),(3) and (4) were used to calculate the valve areas as a function of  $T_{tb}$ ,  $a_i$  and  $b_i$  were coefficients used in estimating the curve while  $A_0$  is the valve opening at zero value of  $T_{tb}$ .

$$A_1 = \frac{1}{a_1 * |T_{tb}|^{b_1 + \frac{1}{A_0}}} \quad \text{for } T_{tb} \leq -T_{tbcenter} \quad (1)$$

$$A_1 = A_0 + c_1 T_{tb} \quad \text{for } T_{tb} > -T_{tbcenter} \quad (2)$$

$$A_2 = \frac{1}{a_2 * |T_{tb}|^{b_2 + \frac{1}{A_0}}} \quad \text{for } T_{tb} \geq T_{tbcenter} \quad (3)$$

$$A_2 = A_0 - c_2 T_{tb} \quad \text{for } T_{tb} < T_{tbcenter} \quad (4)$$

Using the values of the valve areas  $A_1$  and  $A_2$ , the resistance for the flow paths are calculated in Equations (5),(6) and (7).

$$R_1 = \frac{1}{C_d * \sqrt{\frac{2}{\rho} * A_1}} \quad (5)$$

$$R_2 = \frac{1}{C_d * \sqrt{\frac{2}{\rho} * A_2}} \quad (6)$$

$$R_{bridge} = 0.5 * \sqrt{R_1^2 + R_2^2} \quad (7)$$

At the next stage of the computation the flow rates  $Q_{bridge}$ ,  $Q_{xa}$  and  $Q_{xb}$ . Equations (8),(9) and (10) are calculated.  $Q_{bridge}$  refers to the fluid flow rate from the pump while  $Q_{xa}$  and  $Q_{xb}$  are the flow

rates to the cylinder chambers. These are calculated based on the pressure in the cylinder and the supply pressure. It must be noted that  $P_s$ ,  $P_a$  and  $P_b$  are not calculated, they are obtained by the integral action on the derivatives of the pressure  $\dot{P}_s$ ,  $\dot{P}_a$  and  $\dot{P}_b$ .

$$Q_{bridge} = \frac{sign(P_s) * \sqrt{|P_s|}}{R_{bridge}} \quad (8)$$

$$Q_{xa} = \frac{sign(P_s - P_a) * \sqrt{abs(P_s - P_a)}}{R_1} - \frac{sign(P_a) * \sqrt{abs(P_a)}}{R_2} \quad (9)$$

$$Q_{xb} = \frac{sign(P_s - P_b) * \sqrt{abs(P_s - P_b)}}{R_1} - \frac{sign(P_b) * \sqrt{abs(P_b)}}{R_2} \quad (10)$$

The rotational motion of the torsion bar is converted to the translational motion of the piston and this is in-turn used to calculate the volume on sides A , B of the piston and time derivatives in Equations (11),(12),(13) and (14).

$$V_a = v_{a0} + A * x \quad (11)$$

$$V_b = V_{b0} - A * x \quad (12)$$

$$\frac{dV_a}{dt} = A * \dot{x} \quad (13)$$

$$\frac{dV_b}{dt} = -A * \dot{x} \quad (14)$$

Equations (15),(16) and (17) are used to calculate the pressure rates which are integrated to obtain the supply pressure and pressure at both ends of the piston. Equation (18) defines the hydraulic assistance force due to the piston and equation (19) explains the frictional force between the piston and the cylinder.

$$\frac{dP_s}{dt} = \frac{Q_s - Q_{bridge}}{C_{supply}} \quad (15)$$

$$\frac{dP_a}{dt} = -\frac{B}{V_a} * (-Q_{xa} + \frac{dV_a}{dt}) \quad (16)$$

$$\frac{dP_b}{dt} = -\frac{B}{V_b} * (-Q_{xb} + \frac{dV_b}{dt}) \quad (17)$$

$$F_{ps} = (P_a - P_b) * A \quad (18)$$

$$F_{friction} = a * |P_a - P_b| + F_0 \quad (19)$$

### 3.3 ADAMS Models

#### 3.3.1 Test-rig Model

Figure 9 illustrates the manner in which the original test-rig model was divided for simulation between ADAMS and simulink. The simulink part of the model will be included into the ADAMS model using co-simulation and this part shall be referred to as the *simulink\_DLL*. The steering wheel and column were retained in the ADAMS environment, the steering-gear computations

were carried out by the *simulink\_DLL*. Pitman arm, drag-link and the hydro-pulse actuator were simulated in ADAMS. This section describes the ADAMS steering test-rig model.

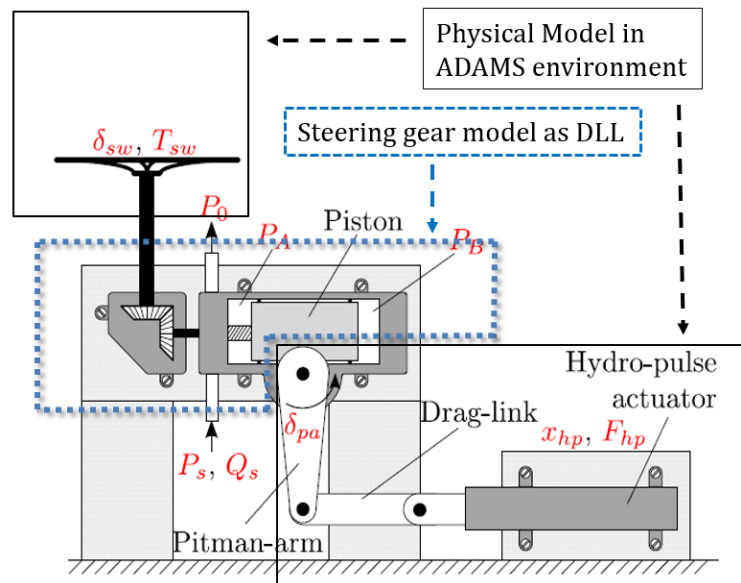


Figure 9. Schematic representation of the divided test-rig model for co-simulation.

All the components were modelled as rigid bodies with specific masses and inertias. Figure 11 shows the steering test-rig model in ADAMS. The steering wheel and column are modelled as one body and constrained with respect to the ground using a revolute joint. Ground in this context refers to the test-rig base. State variables are simulation run-time variables. In order to implement the co-simulation process the outputs and inputs for ADAMS. The functionality of these variables are reversed at the *simulink\_DLL* end of the process, thus the output variables of ADAMS are the input variables to simulink and vice versa. The angular displacement of the steering wheel and its time derivative are two of the output variables from ADAMS. The reaction torque acting on the steering wheel also called as the driver effort is an input from *simulink\_DLL*.

Functionality of the steering gear is part of the model in *simulink\_DLL*. The pitman arm is constrained with respect to the ground through a revolute joint, also connected to the drag-link through a spherical joint. The angular displacement of the pitman arm is an ADAMS output while the torque acting on it is computed and provided by the *simulink\_DLL*.

The drag-link is connected to the piston of the hydro-pulse actuator through a constant velocity joint. Ideally a spherical joint would suffice for the constraint, but that would cause a redundant constraint situation and complications with the ADAMS solver. The piston has one translational degree of freedom along the axis of the cylinder. The cylinder is fixed to the ground. The four bodies pitman-arm, drag link, piston and cylinder has 24 degrees of freedom (DOF). Fixed joint constraints 6 DOF, revolute and translational joints each constraints 5 DOF, constant velocity joint and spherical joints constraints 4 and 3 DOF respectively. Thus as expected, the mechanism has one DOF which is actuated by the hydro-pulse. Figure 10 is a schematic diagram of the model which illustrates the various bodies and their corresponding constraints.

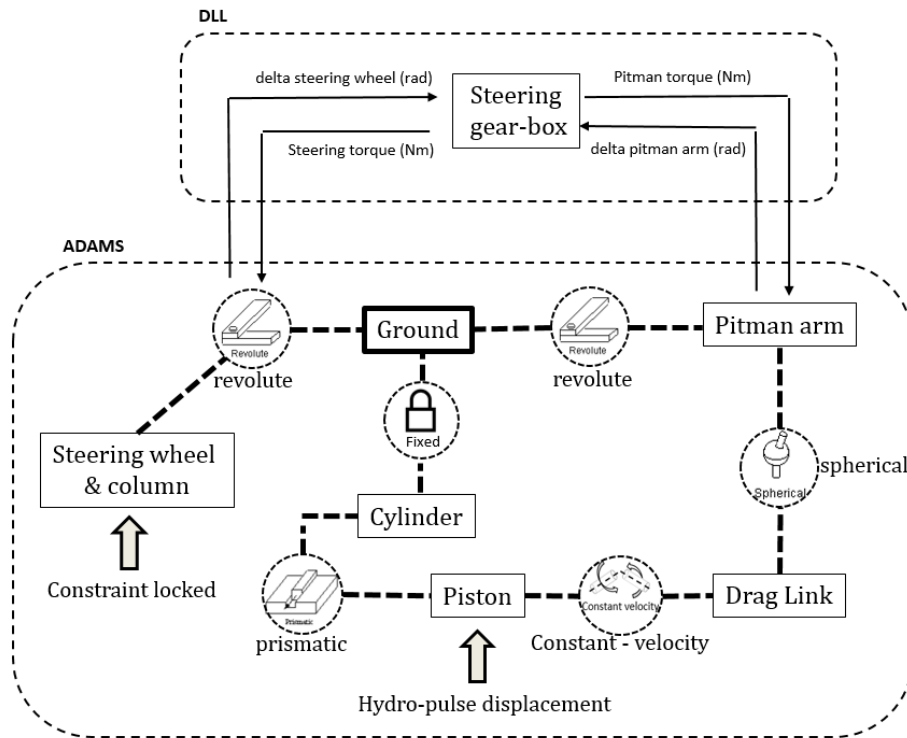


Figure 10. Schematic diagram of model components, connections and actuations.

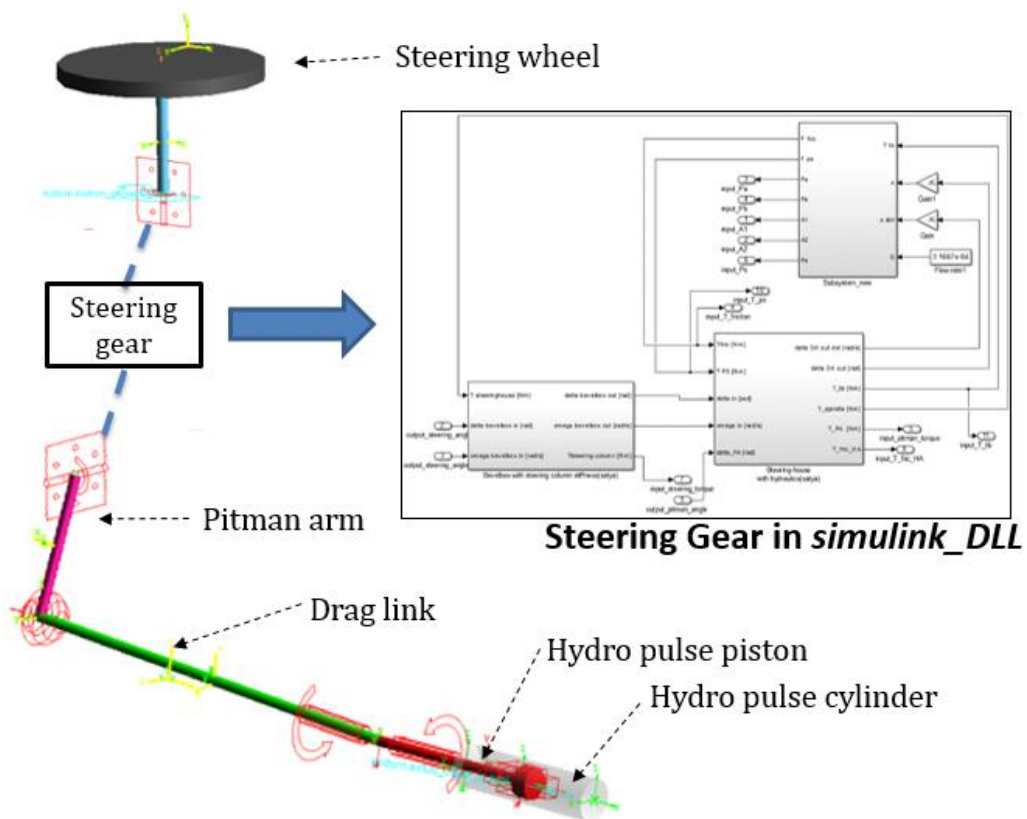


Figure 11. ADAMS test-rig model.



A number of test-cases are performed by Jan Loof, one of the test-cases used to verify the model is defined as follows. The steering wheel was locked with respect to the ground while a displacement set-point was applied to the hydro-pulse actuator. The ADAMS simulation is carried out in a similar manner. The rotational degree of freedom is locked for the steering-wheel. The hydro-pulse piston has a prescribed displacement along the axis of the cylinder. Results of the simulation are presented in the subsequent chapter.

### 3.3.2 Vehicle Model

Commercial vehicles have a more complicated steering system when compared to passenger cars. This is caused by numerous space constraints due to the size of the engine and related components. Also, the cab is suspended with respect to the chassis, this calls for a mechanism to accommodate for the relative motion between the cab and the chassis. ADAMS car works in a model hierarchy structure which integrates the numerous vehicle sub-systems into a full vehicle assembly, illustrated in figure 12. The lowest level of modelling is at the template level which defines vehicle sub-assembly topology. It contains all the relevant bodies, constraints and actuators. The steering system would be a template. The next level is the sub-system stage which references to a template with design data. The last stage is the assembly which brings together all the sub-systems and allows data communication between them through predefined communicators.

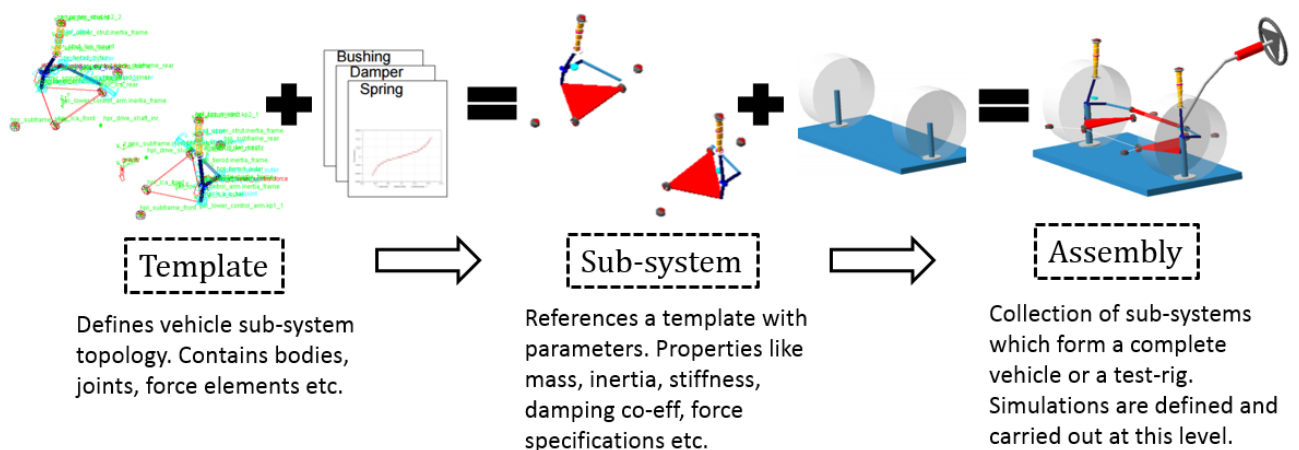


Figure 12. ADAMS Car data structure hierarchy.

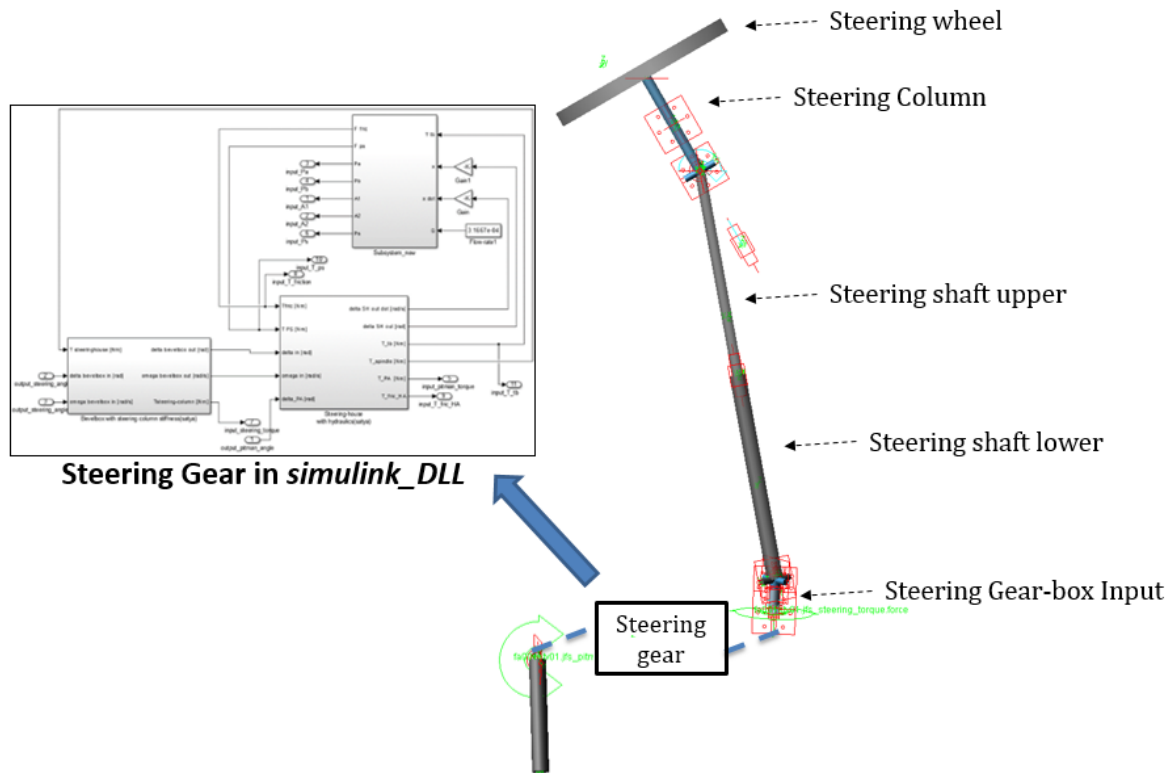


Figure 13. Vehicle steering-system ADAMS template.

Figure 13 is an image of the steering system template in ADAMS. The steering wheel is fixed to the steering column which is constrained by a revolute joint with respect to the cab. In order to accommodate the compliance between the cab and the chassis, the steering shaft upper is connected to the steering column via universal joint. A telescopic translational joint between the upper and lower steering shaft allows for a steering shaft of variable length. The lower steering shaft is connected to the input of the steering gear-box through another universal joint. The steering gear-box input is constrained rotationally to the chassis. At this point the model loses physical continuity. The angular displacement of the steering gear-box input is an ADAMS output to *simulink\_DLL*, the back torque acting on the same body which eventually reaches the driver as the steering effort is a *simulink\_DLL* output to ADAMS. The pitman arm is constrained with the chassis through a revolute joint, its angular displacement is an input to *simulink\_DLL* and the torque acting on it due to the driver effort and the hydraulic assist are computed and provided by the *simulink\_DLL* to ADAMS. The pitman is connected to the draglink through a spherical joint which is included in a different template.

### 3.3 Co-Simulation Process

Co-simulation between ADAMS and simulink is carried out using the ADAMS Control plug-in, this helps in defines the ADAMS inputs and outputs and convey this information to simulink. There are three methods for co-simulation. Discrete mode is one in which ADAMS solves the MBD algebraic differential equations (ADE) and simulink solves the part of the system modelled in it. The variables exchanged between the two programs may not be completely in-sync and require to be interpolated in between sample times. In the continuous mode the ADAMS plant is exported

to the simulink environment. The simulink solver solves both its preexisting equations and the ADAMS model differential equations. Adams integrators are specifically tuned to solve ADE which define the equations of motion, constraints and forces. In continuous mode these equations are formulated as compact ordinary differential equations with minimum independent states and passed on to simulink. Even though the states are easily communicated with simulink, it comes at a cost of computation performance and stability. The last method is the Control Systems Import method. This method was chosen due to the simplicity of implementation. The ADAMS vehicle model has a relatively high level of complexity due to the presence of the a flexible chassis and tire-road interaction. The idea was to perform the simulation process in ADAMS and let it decide the appropriate simulation solver attributes. In this method, we first identify the state variables which are used to communicate between the ADAMS and simulink models. This information is exported to simulink using the Control Plug-In toolbox in ADAMS. Figure 14 illustrates the co-simulation process and various important files which are generated at different stages of the process.

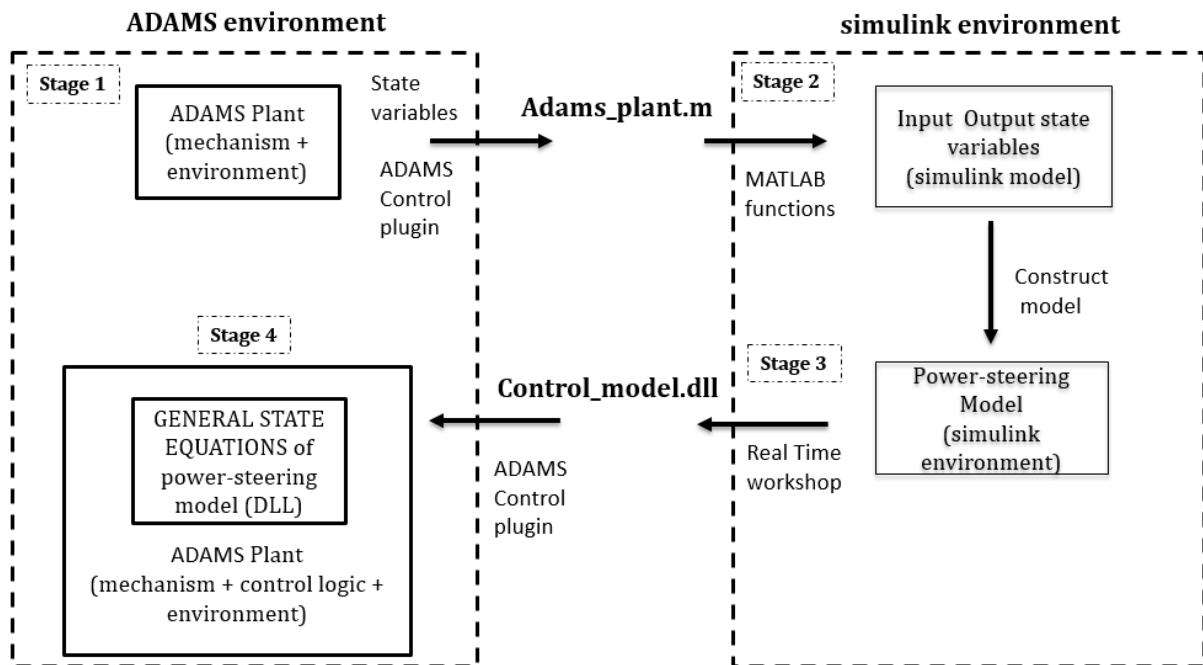


Figure 14. Schematic diagram of co-simulation process.

The first step is to define the state variables and to export the plant. An `adams_model.m` file is created in the working directory which contains the co-simulation inputs and output data. Pre-defined MATLAB functions from the ADAMS installation are called upon to create a simulink template which contains the specified inputs and outputs. The modified steering gear model is copied into this template and the inputs and outputs are connected accordingly. Using the Real-Time Workshop toolbox this model is built and exported as C-code and a `control_model.dll` file is created. The C-Code, also referred to as the External System Library is now imported into ADAMS using the Control Plug-In. General State Equations (GSE) are created in ADAMS with the state variables which account for the dynamics of the power-steering model in simulink. A detailed description of the process will be included in the Appendix.

## 4 Model Verification

### 4.1 Test-rig Verification

#### 4.1.1 Verification of SIMULINK C-code and GSE in ADAMS

The verification of the co-simulation was carried out in two stages as shown in figure 15. In the first stage the C-Code generation and the performance of the GSE in ADAMS are verified. A number of simulations were carried out here. The steering wheel was constrained and a hydro-pulse displacement actuation was provided to the hydro-pulse piston. A number of test cases were carried out with ramp, sinusoidal and staircase hydro-pulse displacements. As mentioned before the *modified* test-rig model is being co-simulated with the ADAMS test-rig model. Aside from co-simulation in ADAMS the pitman-arm and steering wheel angles are exported from ADAMS and used as inputs to simulate the modified test-rig model in simulink, which is not a co-simulation. As the input from ADAMS into its *simulink\_DLL* are same as the input to the modified test-rig model, this would verify the behavior of the GSE in ADAMS.

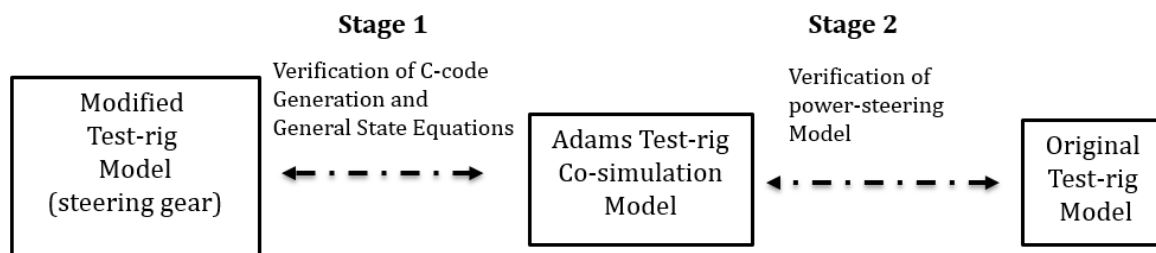


Figure 15. Verification process.

This report will present the verification results of the staircase signal due to its complex nature. In this co-simulation the steering wheel and pitman arm angles are generated by the rigid body dynamics in ADAMS while the steering wheel and pitman arm torques are provided by the *simulink\_DLL* built in ADAMS. Two different groups of signals are being verified here. Firstly, the signals which are being exchanged between the *simulink\_DLL* and ADAMS dynamic model, these are the inputs and outputs from ADAMS. Secondly it is important to verify internal states of the *simulink\_DLL*.

Figure 17 compares the behavior of the internal states as functions of time for both the simulations mentioned above while figure 16 compares the primary states. It must be mentioned that the results of the simulation are scaled for the sake of confidentiality. It can be concluded from the graphs that the error in computation of the GSE in ADAMS and the modified test-rig in simulink is acceptable. Thus it was decided to continue to the verification of the ADAMS test-rig model performance.

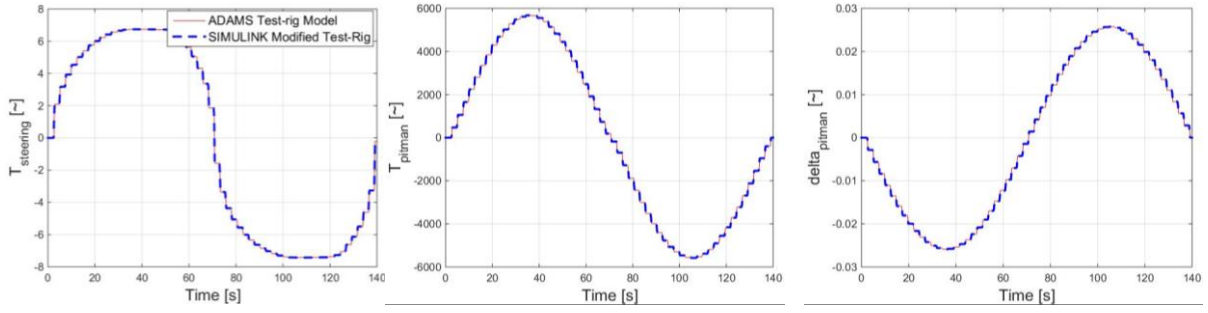


Figure 16. Verification of simulink C-code and GSE in ADAMS – Primary ADAMS Inputs and Output

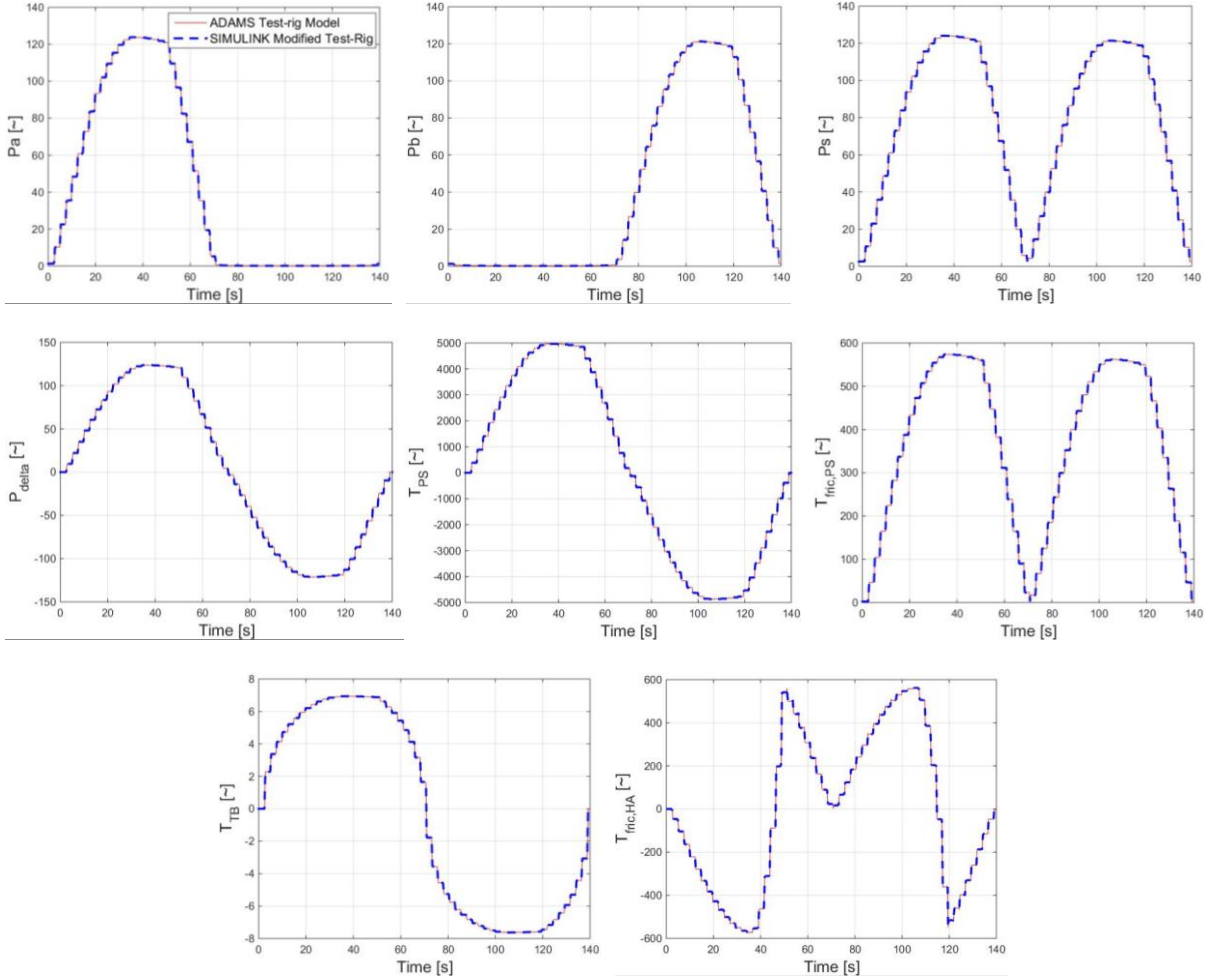


Figure 17. Results – Verification of simulink C-code and GSE in ADAMS – Internal states of Steering Gear

Error is computed for all the primary and internal states. The maximum relative error is defined in equation 20. Error between the two signals is checked for the whole simulation span, the highest error value is divided by the signal range and converted to a percentage. Table 1 contains the maximum relative error percentage for the first stage of verification.

$$Max Error_{relative} x (\%) = \frac{\max(\Delta x)}{range_x} * 100 \quad (20)$$

Table 1. Relative Error Percentages – Verification of C Code and General State Equations

Signal	Max Absolute Error	Max Error (%)
$T_{pitman}$ (Nm)	0.0100	1.76e-4
$T_{steering}$ (Nm)	0.0001	1.50e-3
$P_A$ (bar)	0.0005	4.03e-4
$P_B$ (bar)	0.0001	8.24e-5
$P_S$ (bar)	0.0005	4.03e-4
$T_{fric,PS}$ (Nm)	0.0020	4.48e-4
$T_{fric,HA}$ (Nm)	0.0200	3.70e-3
$T_{PS}$ (Nm)	0.0200	4.02e-4
$T_{TB}$ (Nm)	0.0001	1.40e-3

#### 4.1.2 Verification of ADAMS Test-rig model

In the second stage of the verification process results of the co-simulation are compared to the original test-rig model in simulink. Similar to the previous stage, three types of hydro-pulse displacement signals are used, a ramp, sinusoidal and staircase signal. Results of the staircase signal are presented in figures 18 and 19.

Table 2 contains the maximum relative error percentage between the ADAMS test-rig and *original* simulink test-rig model. The more important data to the co-simulation process are  $T_{steering}$  and  $T_{pitman}$  with maximum error percentages of 0.56 and 0.64 respectively. This is considered to be acceptable, the final step towards a full vehicle with power-steering is therefore taken in the next section.

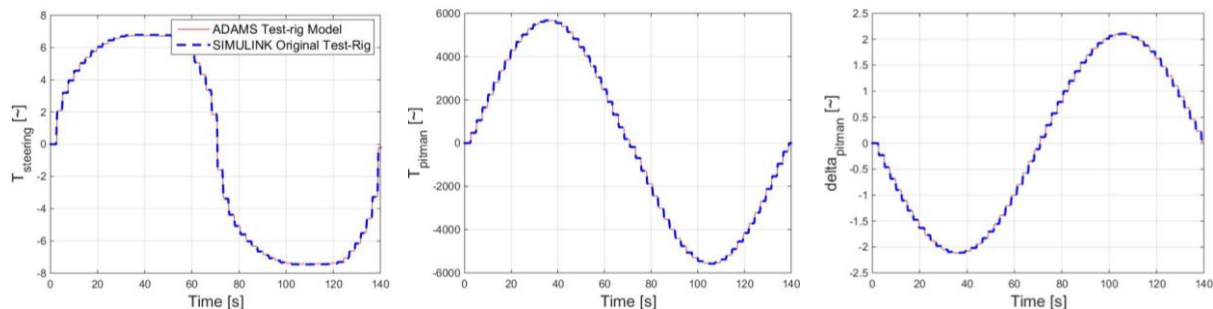


Figure 18. Results - Verification of ADAMS test-rig model – Primary ADAMS Inputs and Output

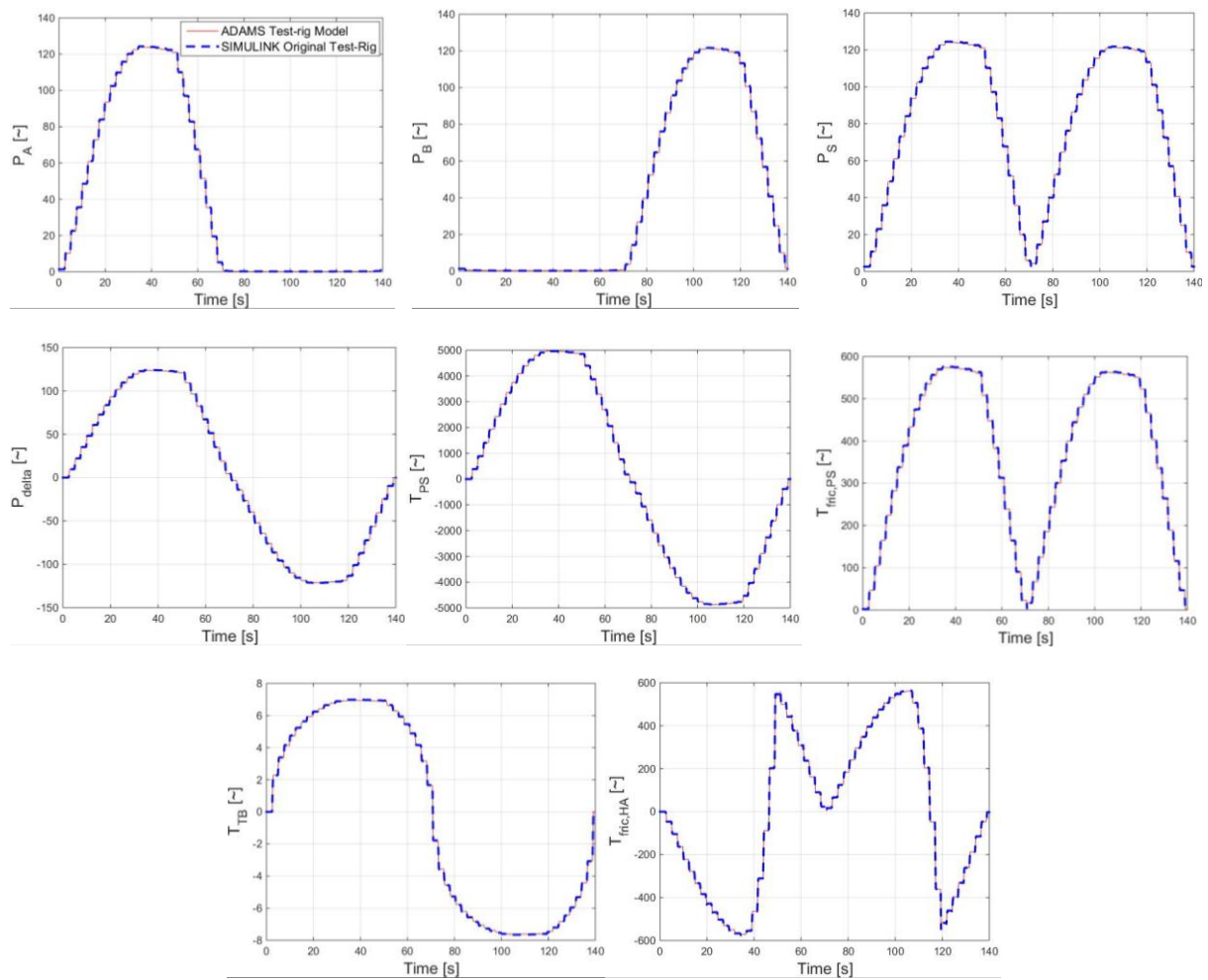


Figure 19. Results - Verification of ADAMS test-rig model – Internal states of Steering Gear

Table 2. Relative Error Percentages – Verification of ADAMS test-rig model.

Signal	Max Absolute Error	Max Error (%)
<b>T<sub>pitman</sub></b> (Nm)	2.10	0.64
<b>T<sub>steering</sub></b> (Nm)	0.04	0.56
<b>P<sub>A</sub></b> (bar)	0.54	0.57
<b>P<sub>B</sub></b> (bar)	0.35	0.33
<b>P<sub>S</sub></b> (bar)	0.40	0.32
<b>T<sub>fric,PS</sub></b> (Nm)	2.10	0.53
<b>T<sub>fric,HA</sub></b> (Nm)	6.80	1.24
<b>T<sub>PS</sub></b> (Nm)	5.10	0.10
<b>T<sub>TB</sub></b> (Nm)	0.04	0.57

## 4.2 ADAMS vehicle simulations

Two models of the DAF tractor semi-trailer are simulated in the ADAMS environment. One of them has the *Simulink\_DLL* implemented, it therefore lacks a physical connection between the steering gear input and pitman arm. The GSEs of the hydraulic power steering control the torque acting on the pitman-arm based on the steering angle actuated by the driver. The other model has a reduction gear with a fixed ratio between the steering gear-box input and the pitman arm. The two primary differences between both the models are the presence of compliance, friction in the steering sub-system and hydraulic assist torque. These models were subjected to a step steer simulation, in both the cases the steering wheel motion is prescribed identically. 3 similar simulations are carried out with increasing steering angles for both the models. Figure 20 shows the tractor behavior as a function of time. For confidentiality reasons all the results are scaled.

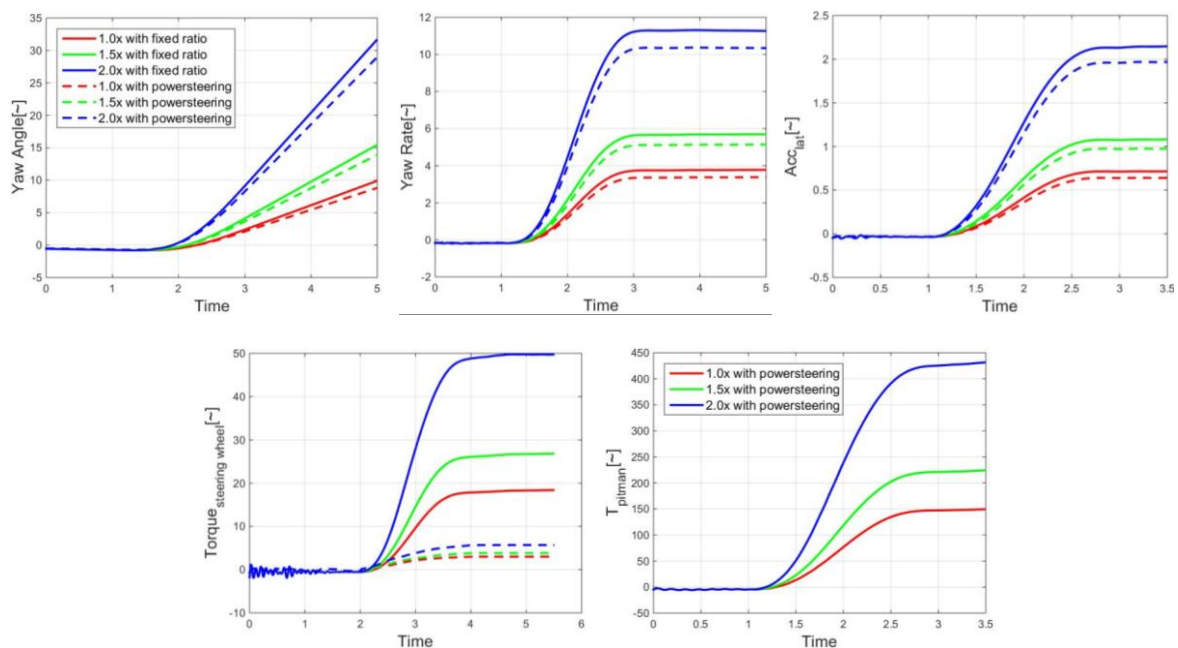


Figure 20. Results. Step-steer simulation.

In the tractor semi-trailer model with power-steering, lower lateral motion is noticed, this reflects upon the vehicle yaw angle, yaw rate and lateral acceleration. The behavior is explained by the fact that this model has compliance in its steering system. A certain steering angle causes higher wheel angles in the case of the fixed-ratio steering model. Steering effort in case of the hydraulically assisted model is much more realistic due to the fact that majority of the torque required to move the pitman arm is provided by the hydraulic assist. A more complex double lane change simulation was carried out, the vehicle models and the results are shown in figure 21. Similar results are observed in this simulation. More realistic values of steering effort in case of the vehicle model with power-steering are very clearly visible.



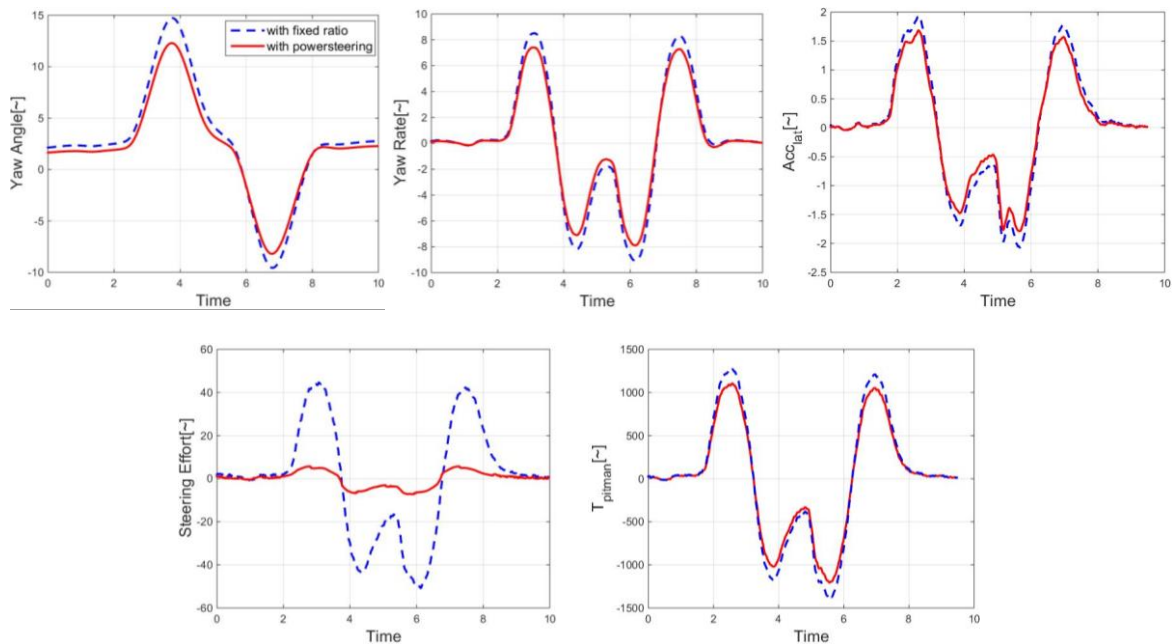


Figure 21. Results – Double lane change simulation.

Table 3 contains the simulation time for the various ADAMS simulations with and without power steering. There is a relatively small increase in simulation time for an increase in accuracy and fidelity of the vehicle model steering system.

Table 3. Simulation and CPU computation time

Simulation	Simulation time (sec)	Original vehicle Model – CPU time (sec)	Co-sim vehicle Model – CPU time (sec)
Double lane change	13	225.02	342.59
Step Steer (1.0x) angle	10	140.41	233.58
Step Steer (1.5x) angle	10	145.41	240.42
Step Steer (2.0x) angle	10	149.11	256.22

As the MBD model now has a high level of complexity, it is ensured that the GSE correctly mimics the behavior of the steering gear model in simulink. Steering wheel and pitman angles from the double lane change co-simulation were extracted. This data is provided as time-series to the steering gear-box model in simulink, the internal states from this simulation and the ADAMS vehicle are presented in figure 22. It is noticed that the simulink\_DLL behavior is correct once it is implemented in the vehicle model.

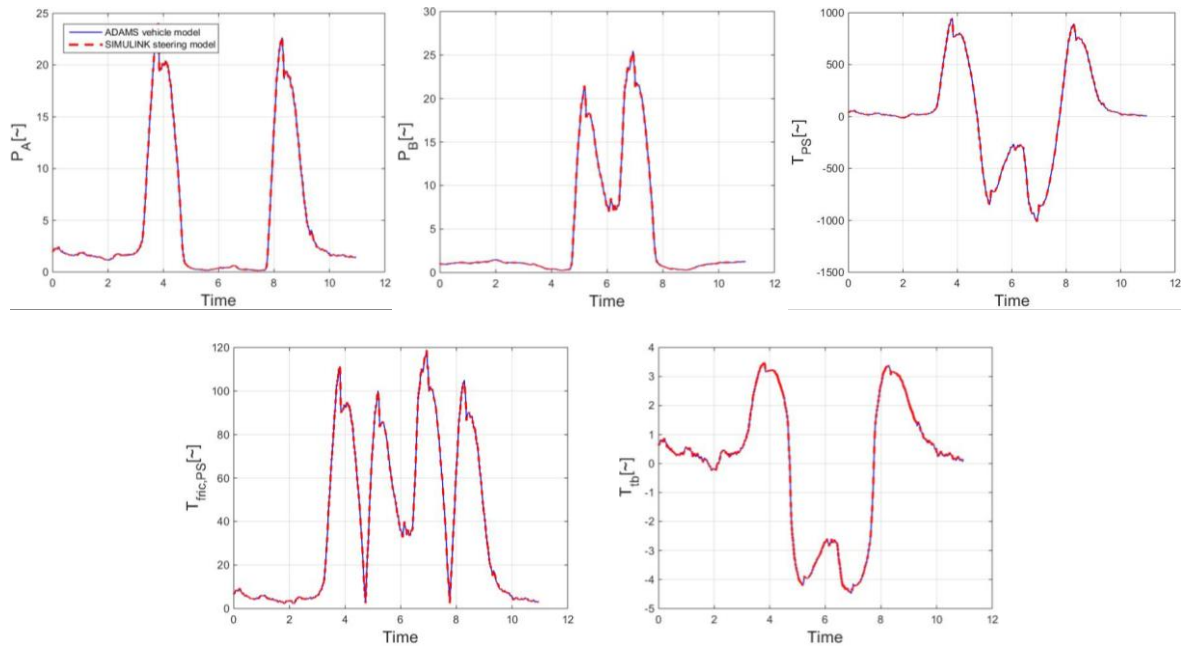


Figure 22 – Internal state variables from GSE compared to SIMULINK steering-model simulation.

## 5.0 Conclusion and Recommendations

The goal of the internship was to integrate the power-steering model into the MBD vehicle model used by the technical analysis team at DAF Trucks. Different methods of co-simulation between ADAMS and simulink were investigated. The original steering system developed at TU/e had to be modified to allow usage with the MATLAB real time workshop. The original power-steering model was maintained to make further developments of the model easily implementable for co-simulation. The next step was to create an ADAMS template to represent the test-rig situation. The verification performed for the co-simulation process was done systematically and at two different stages. After having performed several simulations with realistic inputs it was concluded that the ADAMS test-rig model was representing the simulink original test-rig model with good accuracy and the process was a success. Thereafter it was decided to implement the same process in the full vehicle model. The DAF truck model in ADAMS has been modified to accommodate the steering gear functionality. After having implemented the co-simulation process, step steer and double lane change situations were simulated to study the behavior of the vehicle with power-steering. The results show a successful implementation of the power-steering model into the existing ADAMS model with realistic results. Simulation time of the more complex co-simulation model increased as expected, but it was in acceptable limits. The power-steering system model adds to the functionality of the MBD vehicle model and broadens the possibilities of new analysis with the model. It is recommended that the current vehicle model with power-steering is validated against actual vehicle test measurements.

At the end of the internship, the new vehicle model shows more realistic results due to the inclusion of compliance in the steering system, it also includes the functionality of the HPAS. The performance of the hydraulic system can now be evaluated. Effects of changes in the power-steering hydraulic design can be studied from within ADAMS.

In order to increase the safety of both passenger and commercial vehicles today, active safety systems are widely implemented in commercial vehicles. The performance of such systems typically modelled with simulink can be analyzed through addition of the control model to the current ADAMS model. Also, due to a successfully working co-simulation methodology, electronic control in areas like suspension, propulsion and braking can be studied using the approach used in this project.

## References

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- [2] S. A. Sicklinger, "Stabilised Co-simulation of coupled problems," Technische Universität München, 2014.
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- [5] J. Loof, I. Besselink, W. post and H. Nijmeijer, "Development of a truck steering system including hydraulics to predict the steering wheel torque," in *Graz Symposium virtual vehicle*, Graz, 2016.
- [6] D.A. Haessig Jr, B. Friedland, "On modeling and simulation of friction," *Dynamic Systems - ASME*, September 1991.

## A. Appendix - Co-simulation process.

This section presents a detailed description of the co-simulation process which can be used as a guide to perform similar activities. It is assumed that the reader has a MBD model with state variables which represent the inputs and outputs of the ADAMS model, they are used to form the GSE. The input state variables are to be left at a value of 0, the output state variables have the relevant function monitoring any model property, E.g. displacement, force, torque etc. The reference co-ordinate systems for these state variables must be checked to avoid erroneous results.

### A.1 ADAMS Plant export

The first step is the generation of the ADAMS plant file named Controls\_Plant.m. Click on the *Controls* icon and then select *Plant Export*. The plant export dialogue box opens up, fill in the name for the file and the required inputs and outputs for the process, from ADAMS perspective. Note to choose 'MATLAB' as the target software and 'non\_linear' as Analysis Type. Note that the files will be generated in the ADAMS working directory.

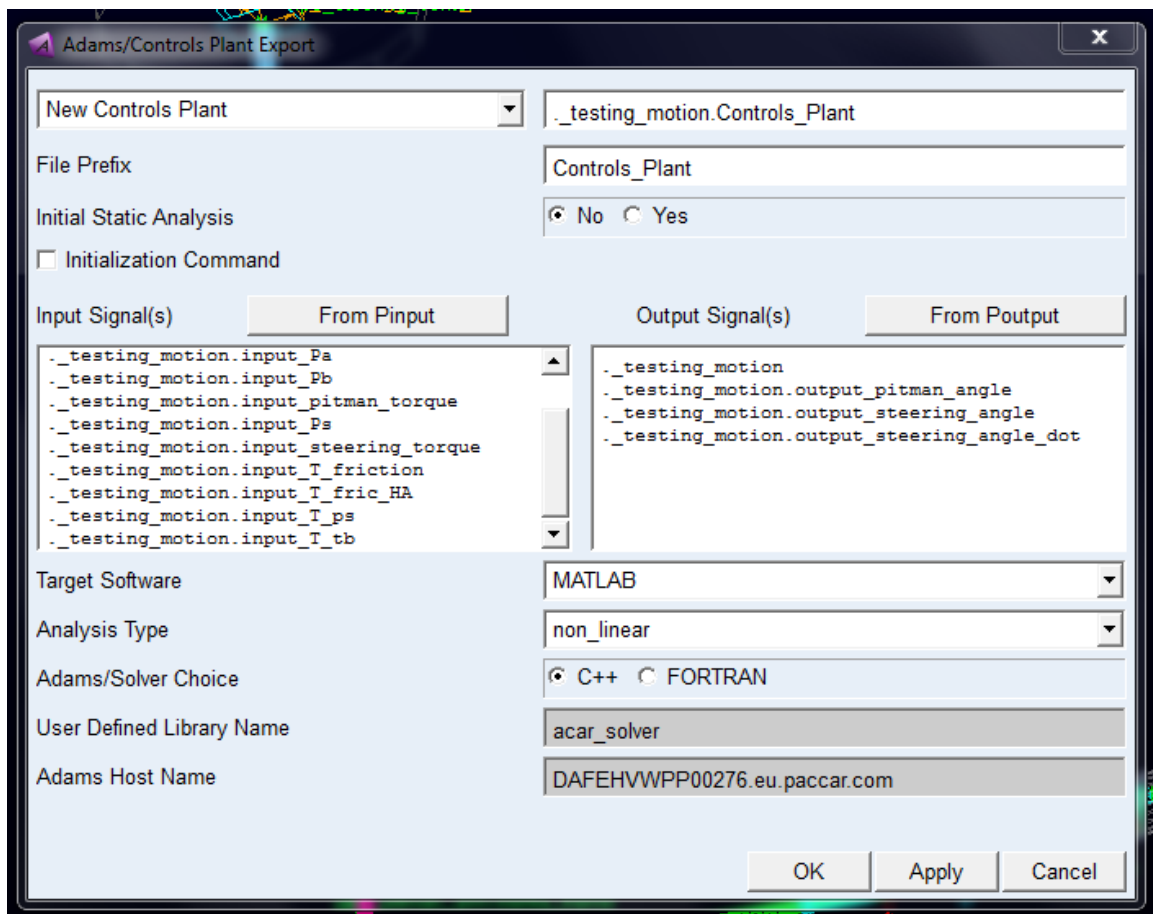


Figure 23. Plant export dialogue box.

## A.2 MATLAB C-Code generation

Set the current folder for MATLAB to the ADAMS working directory. Type the command **mex -setup** and confirm that mex is set to 'Microsoft Windows SDK 7.1 ©' for C language compilation. Type **Controls\_Plant.m** and it should display all the specified input and outputs as during ADAMS plant export. Then type **setup\_rtw\_for\_adams**, this detects the version of MATLAB, creates the make file and source code for ADAMS. Type the command **setio** to open a simulink template with just the inputs and outputs as defined in ADAMS. The MATLAB command sequence is illustrated in figure 24. "Setup\_rtw\_for\_adams.m" and "setio.m" are MATLAB function files in the ADAMS installation directory, the process is automated and these files do not require any modifications.

```
Command Window

ans =

30-Nov-2016 12:30:24

*** Warning : missing ADAMS plant model file(.adm) for Co-simulation or Function Evaluation.
*** If necessary, please re-export model files or copy the exported plant model files into the
*** working directory. You may disregard this warning if the Co-simulation/Function Evaluation
*** is TCP/IP-based (running Adams on another machine), or if setting up MATLAB/Real-Time Workshop
*** for generation of an External System Library.

*** INFO : ADAMS plant actuators names :
1 input_A1
2 input_A2
3 input_Pa
4 input_Pb
5 input_pitman_torque
6 input_Ps
7 input_steering_torque
8 input_T_friction
9 input_T_fric_HA
10 input_T_ps
11 input_T_tb
*** INFO : ADAMS plant sensors names :
1 output_pitman_angle
2 output_steering_angle
3 output_steering_angle_dot

>> setup_rtw_for_adams

*** Your MEX compiler setting is the following: Microsoft Windows SDK 7.1 (C)Microsoft Windows SDK 7.1 (C++)
*** Please confirm that your selected compiler is a C/C++ compiler compatible with Adams.
*** See Adams Hardware & Software Requirements for more details.
*** If necessary, use "mex -setup" to select a new compiler.

*** Using local process.py found in working directory.
*** Running the following command:
C:\cadcae\MSC~1\SOF\ADAMS_~1\2015_1\common\mdi python C:\cadcae\MSC~1\SOF\ADAMS_~1\2015_1\controls\utils\process.py -v 8.4

*** Successfully created files in the working directory needed for Adams library export.
>> setio
fx >> |
```

Figure 24. MATLAB commands to generate empty template with system inputs and outputs.

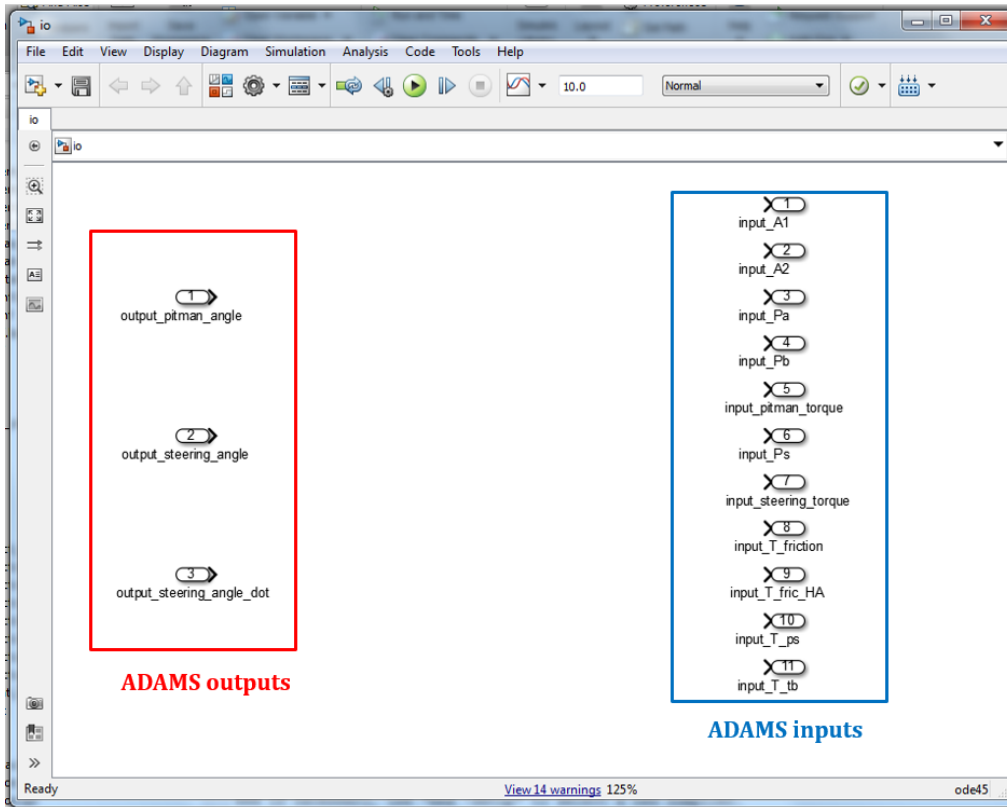


Figure 25. Empty template without steering model.

Copy the relevant simulink block diagram to the template as shown in figure 25 and connect the inputs and outputs to complete the model as in figure 26.

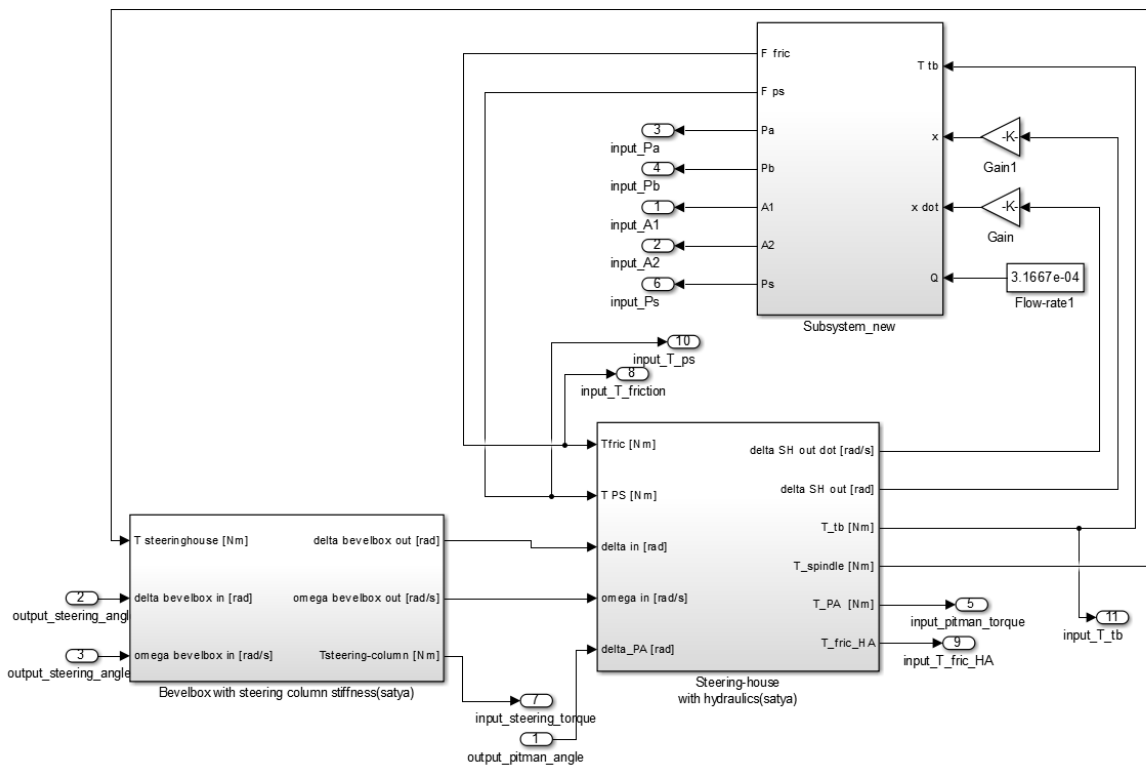
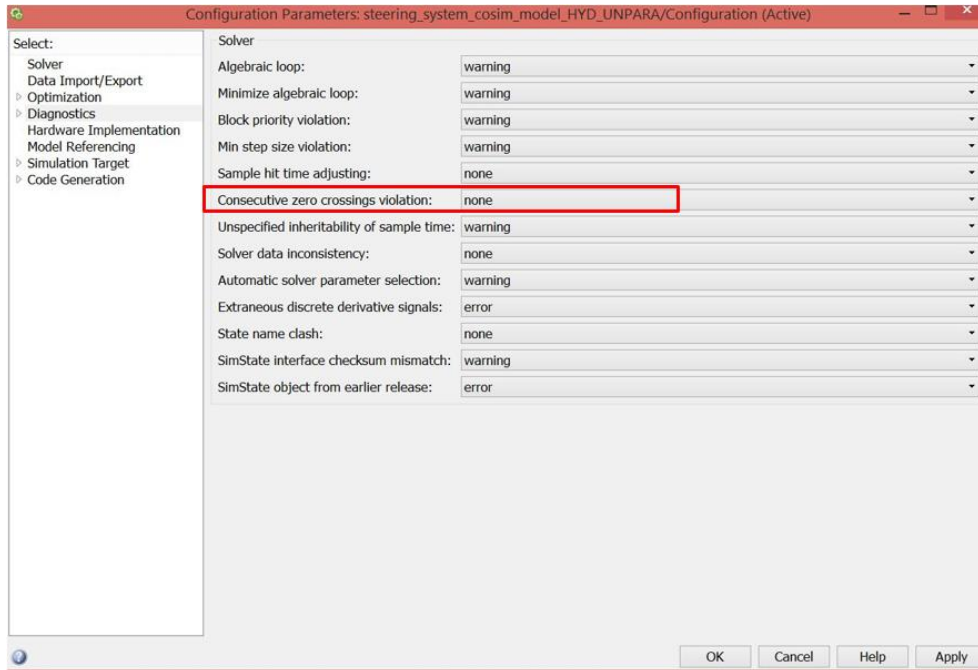


Figure 26. Complete simulink model before exporting C-Code.

Before building the model a few checks must be done in the Model Configuration Parameters. In the option Diagnostics, ensure that *Consecutive Zero Crossing Violation* is set to *none*. This avoids creating zero-crossing sensors in ADAMS which cause simulation to stop in case a state variable reaches the value zero.



In the solver settings ensure that *Solver Option Type* is set to *Variable-Step*. Integrators are solved by the ADAMS solver and this lets ADAMS choose the optimum time steps. Also, *Disable All* the *Zero-crossing control*.

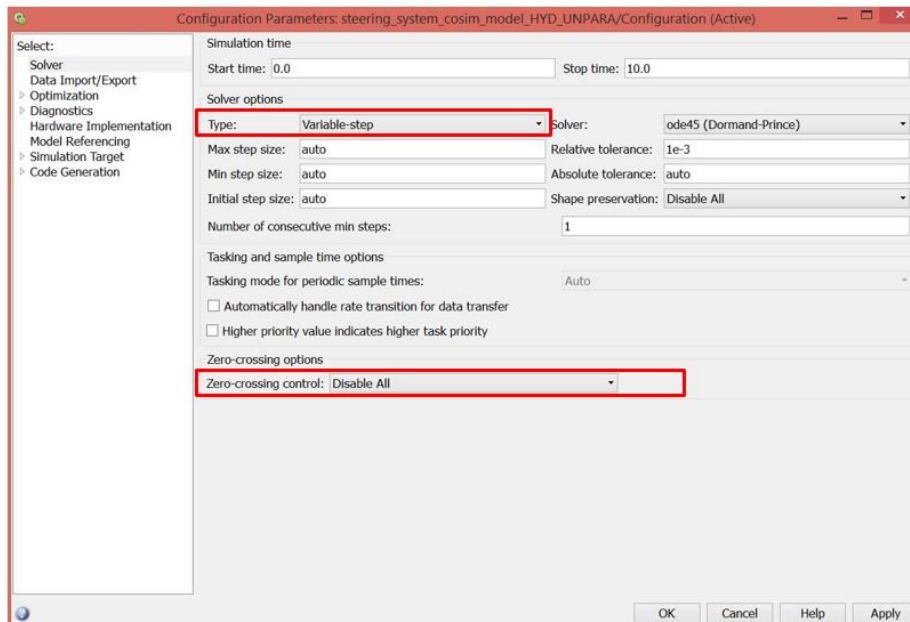
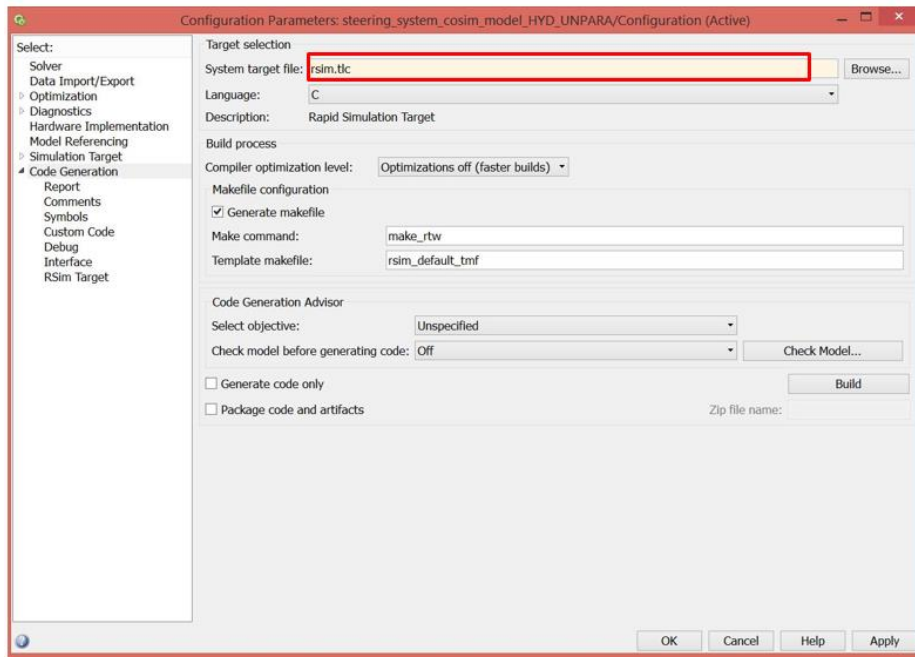


Figure 27. Solver setting.

And finally, in the Code Generation option, ensure that *System target File* type is set to *rsim.tlc*





Now build the model and a io.dll file is generated in the working directory.

### A.3 Creating GSE from simulink Model.

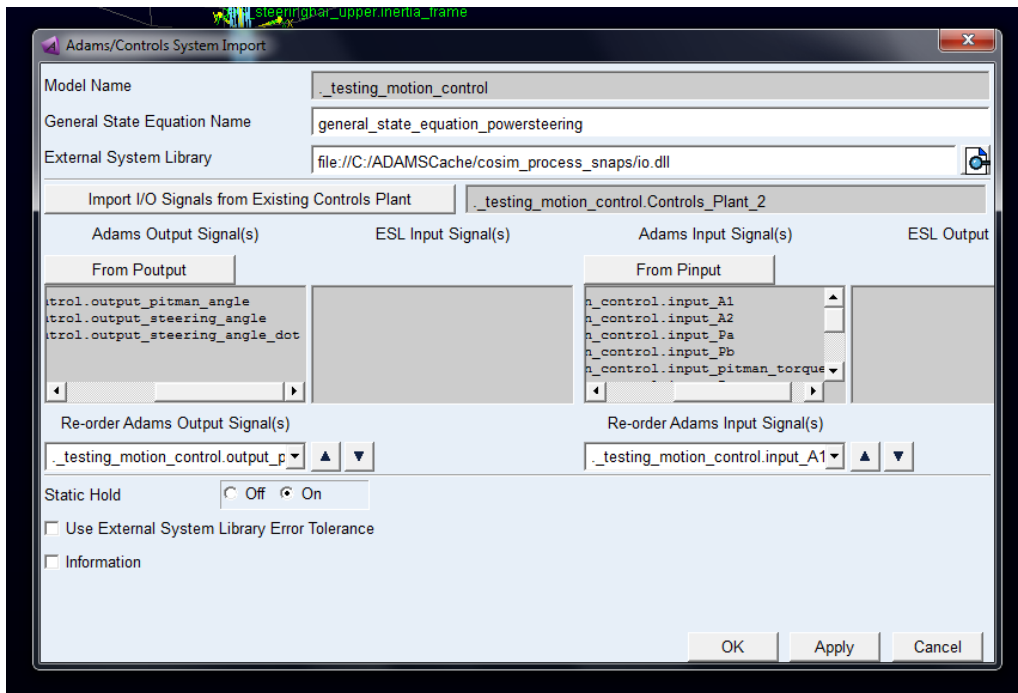


Figure 28. GSE generation

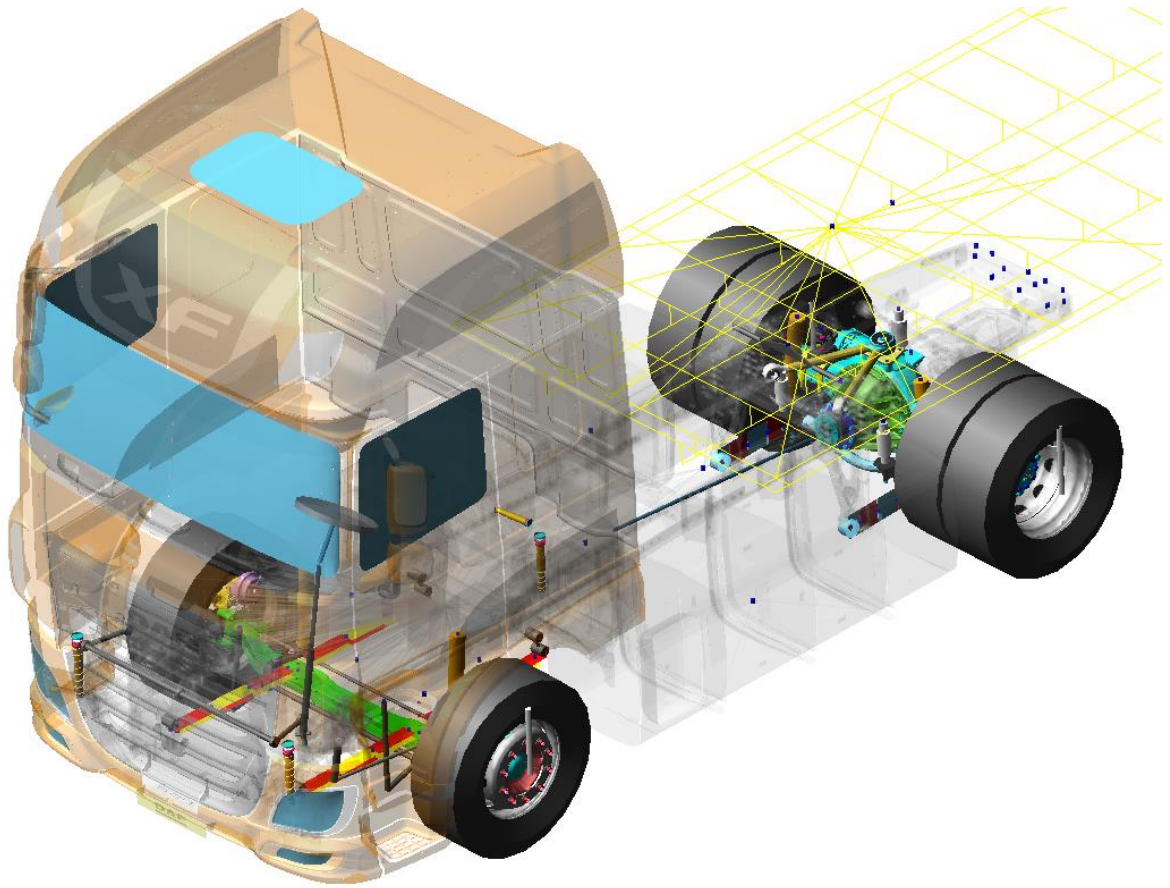
Go back to ADAMS and perform the Control System Import as shown in figure 28. Keep in mind to choose the correct *External System Library* from the working directory and the specific *Control\_Plant* file which was used to generate it, the ADAMS inputs and outputs are automatically updated. This step creates the GSE. Going into Data Base navigator and looking at design variables under the *Controls\_Plant*, you will be able to see all the variables from the simulink model and these can be modified here in case of design changes. Once the io.dll file has been imported, the only file required for co-simulation is the io.dll, this should always remain in the ADAMS working directory. It must be noted that generating a second set of GSE will cause the values of each of the state variables to add up the outputs from each the GSE. This causes erroneous results, only one GSE should be present for the same system during the co-simulation.

## B. Appendix - Models.

### B.1 ADAMS vehicle model.



Figure 29. ADAMS vehicle model used for vehicle simulations.



*Figure 30. Close up of the transparent tractor showing steering and suspension templates.*

## B.2 SIMULINK block for m-function replacement

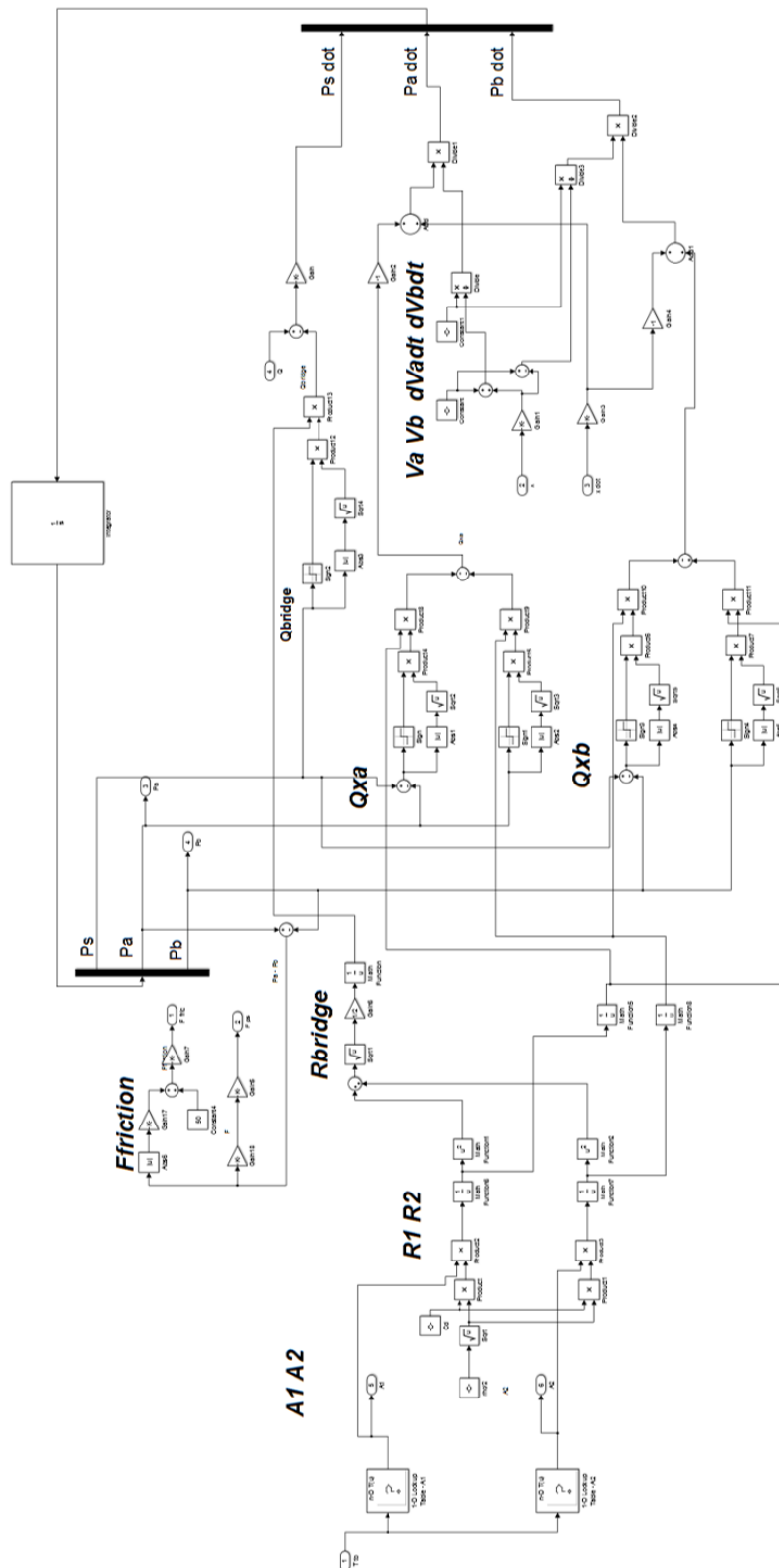


Figure 31. Simulink hydraulic block replacing embedded m-file in modified steering model.