LANE CHANGE MANEUVER OF VIRTUAL HEAVY VEHICLE EQUIPPED WITH YAW MOMENT CONTROL

Jan Fojtášek[1]

Summary: The goal of this article is to present a method for preparing a virtual heavy commercial vehicle for the analysis of the right-left torque vectoring technology effect. Lateral torque vectoring systems are nowadays usually used in sports cars or in luxury personal cars to improve vehicle stability and handling performance. In this paper the heavy commercial vehicle equipped with the direct yaw moment control system is simulated by using the full vehicle ADAMS model.

Key words: Multibody model, heavy commercial vehicle, yaw moment control, single lane change maneuver.

INTRODUCTION

With the increasing performance of the modern vehicles, manufacturers need to ensure vehicle stability and controllability during fast maneuvers. A lot of vehicle dynamic control systems have been developed for sports cars to improve their performance and for personal cars to enhance their active safety (1). From the viewpoint of heavy commercial vehicles, it is obvious that the manufacturers have the primarily target in systems reliability and operational efficiency. The new technologies development is very conservative. However, there are several technologies which are very useful not only for personal cars but for heavy trucks too.

Lateral torque vectoring system creates driving and braking force differences between left and right wheels and thereby can directly control the yaw moment acting on a vehicle (Fig. 1). That means that tire longitudinal forces are controlled and because of this the vehicle's cornering performance is enhanced (2, 3). The advantages of these kinds of systems are that they are not based on the vehicle brakes so they are not in violation with the driver's acceleration and braking demands (4). Equally, there is nearly no change in the total driving or braking forces of both wheels. For this purpose, a lot of mechanisms were developed which are based on the additional gears between differential cage and the output shafts. For the smooth control of the torque difference usually two friction or electromagnetic clutches are used.

[1] Ing. Jan Fojtášek, Brno University of Technology, Faculty of Mechanical Engineering, Institute of Automotive Engineering, Technická 2896/2, 616 69 Brno, Tel.: +420 54114 2275, E-mail: jan.fojtasek@vutbr.cz

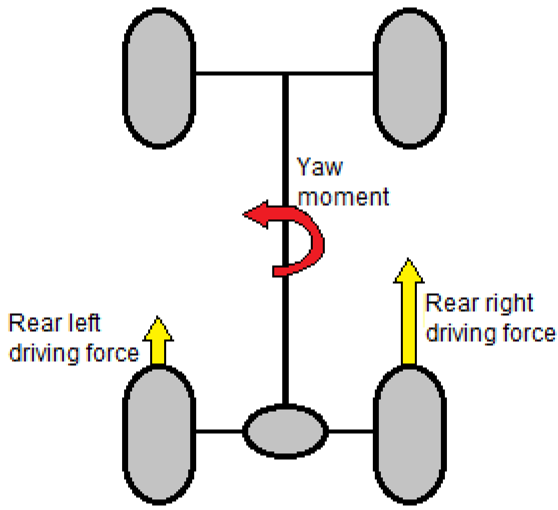

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Fig. 1 - Rear wheel drive torque vectoring scheme

1. VEHICLE MODEL

The example vehicle for virtual modeling is a two-axle off-road tipper with permanent rear-wheel drive. The chassis consists of rigid “backbone” tube with independent swinging half-axles. Full vehicle model is based on the Adams/Car Truck templates and components. The whole model consists of these individual templates: front suspension, rear suspension, body, powertrain, front wheels, rear dual wheels, steering and brakes. Based on these templates, Adams/Car subsystems were created and then the full vehicle model was assembled together with MDI SDI Testrig. The individual templates used for this model were created on the base of default templates from the acar and atruck shared databases.

1.1 Front suspension

This part of the model was created by modification of *msc truck steer suspension template* (5). Besides the changes of hard-points and construction frames locations, were the main differences in remodeling the rigid axle to the independent swinging half-axles. To specify the half-axle location, one center axle hard-point for each half-axle (left and right) is created. In the template builder interface the front suspension is modeled as symmetric and the asymmetrical longitudinal displacement of the half-axles is accomplished in standard user interface. Two revolute joints between half-axles and body are also added and the body cushioning is ensured by air-springs with specified characteristics. Other changes are on damper attachments, connections of steering, geometry and mass properties of single components. Fig. 2 shows the front suspension model topology and the model is in Fig. 3.

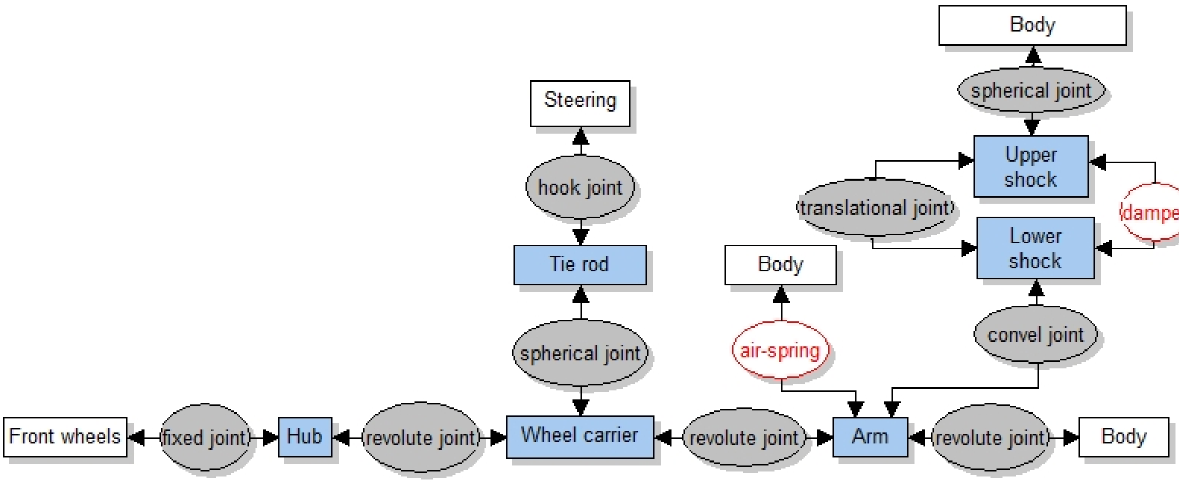

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Fig. 2 - One half of front suspension model topology

1.2 Rear suspension

This template is based on *msc truck airspring drive axle* template (5). The main modifications are similar to front suspension template except steering attachments. Topology of this template is shown in Fig. 4 and model is in Fig. 5. To connect hubs and the powertrain drive-shafts perpendicular joints are used. These attachments provided the proper connection between two rotating parts without over-constraining the model.

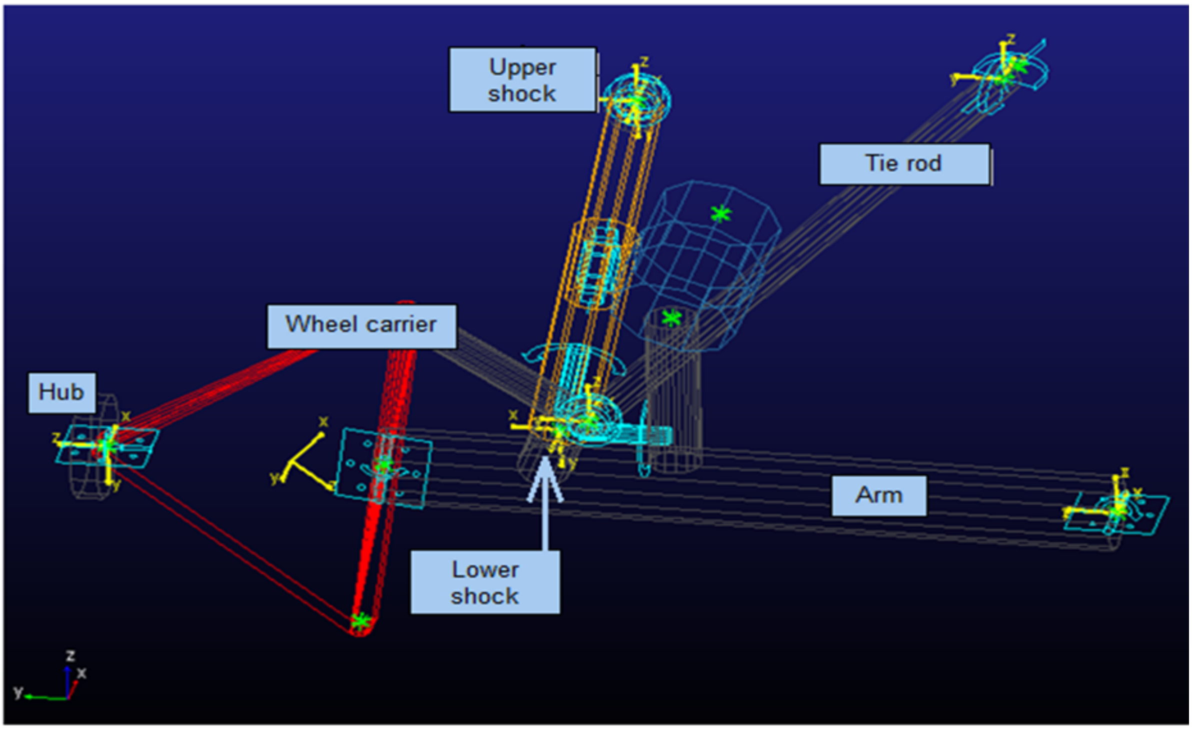

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Fig. 3 - One half of front suspension model

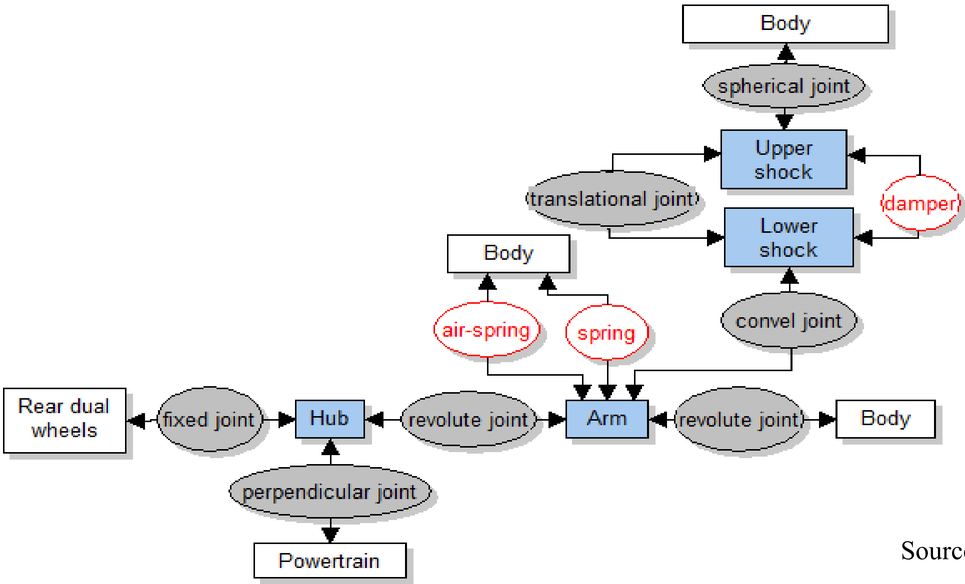

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Fig. 4 - One half of rear suspension model topology

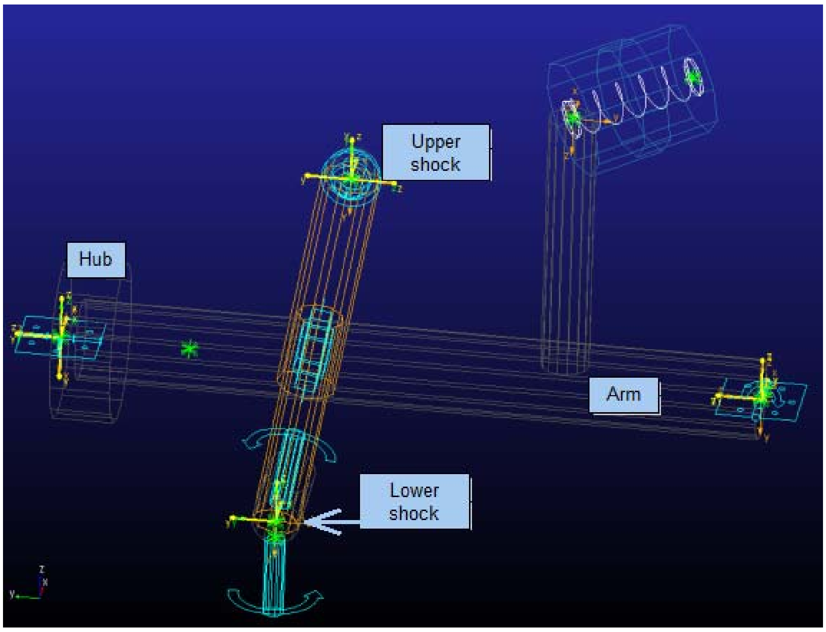

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Fig. 5 - One half of rear suspension model

1.3 Body

The body template is composed of two general parts which are connected by a fixed joint. One part represents the mass properties and geometry of the rigid “backbone” tube with frame, cabin, tipper body and accessories. The other part represents cargo.

1.4 Powertrain

The powertrain template is based on the *msc truck powertrain* (5). The main modifications are change of powertrain type from two-axle drive to only one-axle drive and connecting the axle drive-shafts to swinging half-axles. In Fig. 6 the model of rear wheel drive powertrain topology is shown. Drive torque acts on driveshaft which by a revolute joint is connected to the body subsystem. This revolute joint is coupled with the rear driveshaft revolute joint by a reduction gear with reduction ratio equal to 1. Next the torque is distributed via the differential gear between rear left and rear right drive-shafts. On these drive-shafts, point torque actuators are symmetrically added, which simulate the torque vectoring differential. Rear left and rear right drive-shafts are connected by perpendicular joints with rear suspension hubs.

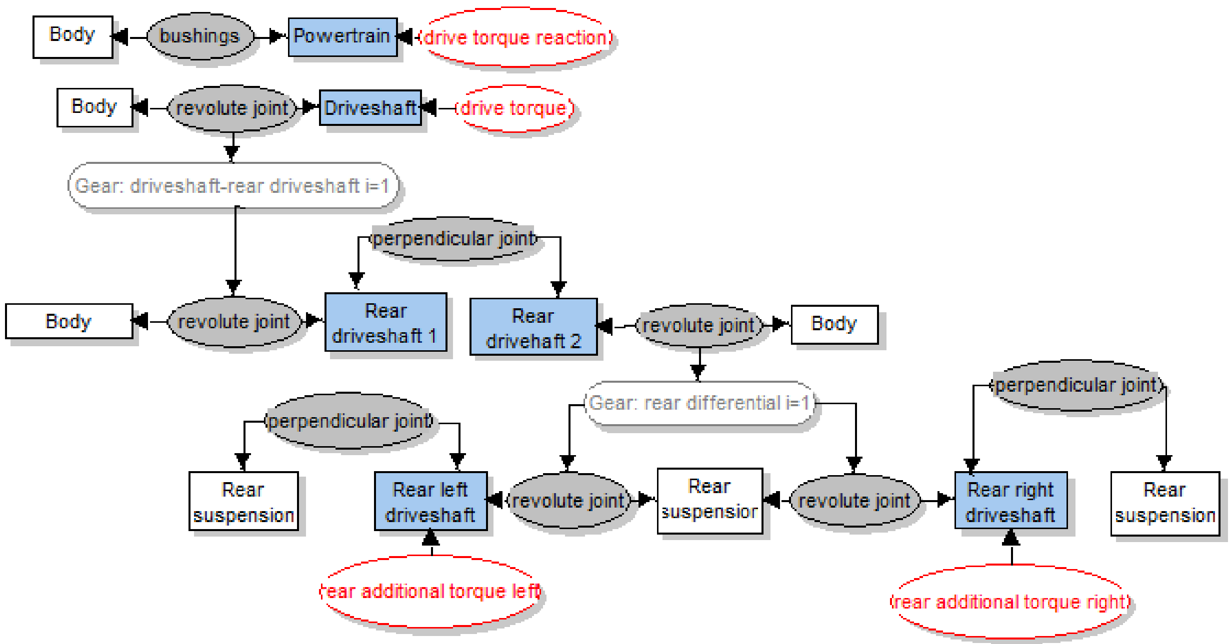

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Fig. 6 - Topology of the rear drive powertrain model with torque vectoring

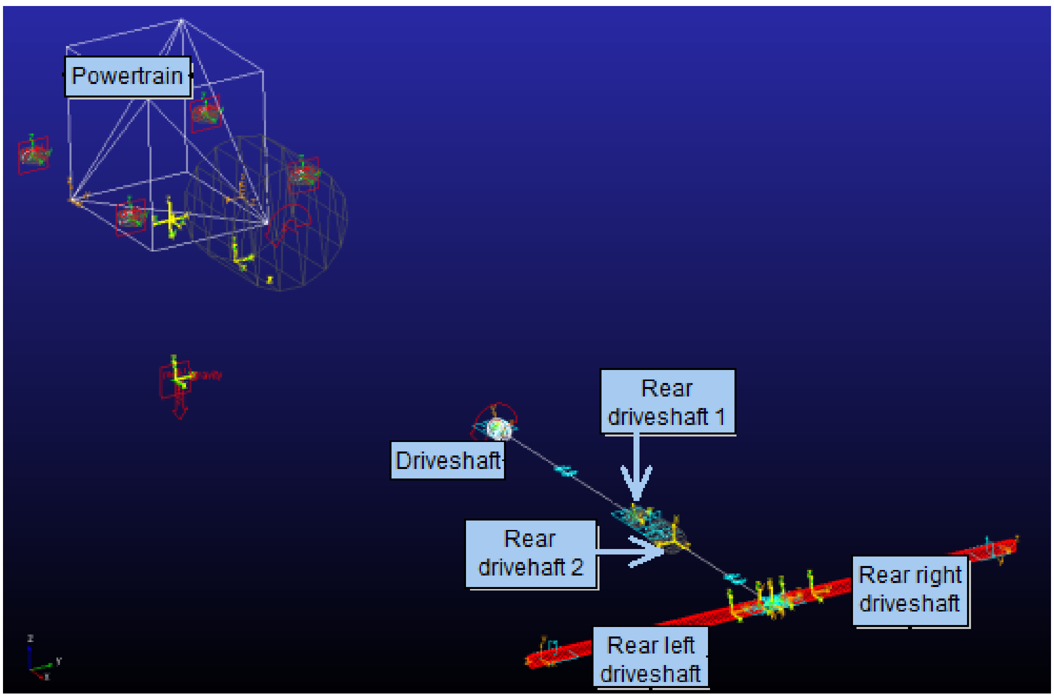

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Fig. 7 - Powertrain model

In this template the control of additional torque has to be defined. Optimal torque vectoring control is serious problem whose solution is not the subject of this article so a simplified control through 3D-spline is used (5). The values of the additional torque actuators on one axle have to be additive inverse to each other as action and reaction forces. At the same time these actuators depend on the required turning radius of the vehicle. There are more parameters which impact the optimal torque vectoring value, but for the simplified control, only the steering wheel angle and lateral acceleration were chosen. The left and right torque values for the rear wheel drive are defined as follows:

$$T_{arl} = \frac{T_r}{2} * AKISPL(\varphi_{sw}, a_y, i_{tv}) \tag{1}$$

$$T_{arr} = -\left[\frac{T_r}{2} * AKISPL(\varphi_{sw}, a_y, i_{tv})\right] \tag{2}$$

where T_{arl} and T_{arr} are additional torques on the rear left and rear right drive-shafts, T_r is total drive-torque (input) acting on the rear axle, φ_{sw} is steering wheel angle, a_y is lateral acceleration, and i_{tv} is vectoring intensity from the interval <-1;1>.

This definition is based on the fact, that the classic differential splits the input torque in ratio 50:50.

1.5 Wheels

The vehicle model includes front wheels and rear dual wheel templates. In the wheel templates PAC2002 Magic-Formula tire model (developed by MSC Software according to Tyre and Vehicle Dynamics by Pacejka) is used (6). This model is applicable to cars and trucks. It is possible to simulate all vehicle handling and stability maneuvers at smooth roads with this tire model (5).

1.6 Steering

For the appropriate control of the vehicle, own pitman arm based steering system is used with translational rack. The rotational input motion from the steering wheel is transmitted to the translational motion of the rack and then by steering gear to the rotational motion of the pitman arm. The pitman arm pulls and pushes the steering link which causes the steering input arm to steer the front wheels. The steering model is shown in Fig. 8.

1.7 Brakes

In order to make the basic model of the vehicle complete, it was also necessary to make the brakes template. In this paper, the air-drum-brake model is used. These brakes are simply the point torque actuators that act between the wheels and the suspension templates. The torque value depends on driver's demand and the friction coefficient.

1.8 Test-Rig

The whole model of the vehicle was assembled together with MDI SDI TESTRIG that allows the model to perform a wide range of the full vehicle analysis using a driving machine. In this paper single lane-change open-loop steering event is used. In this kind of analyses the steering input is defined as a function of time. Specifically, during the single lane-change maneuver the steering wheel input goes through a complete sinusoidal cycle with the amplitude of 200 degrees and 6 seconds time length (Fig. 9).

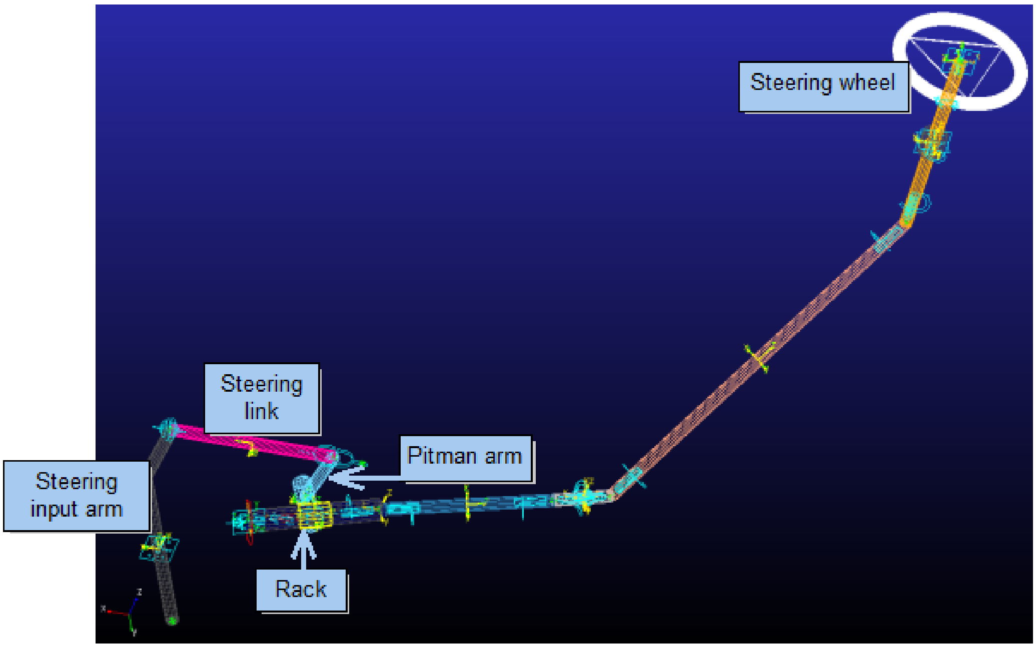

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Fig. 8 - Steering model

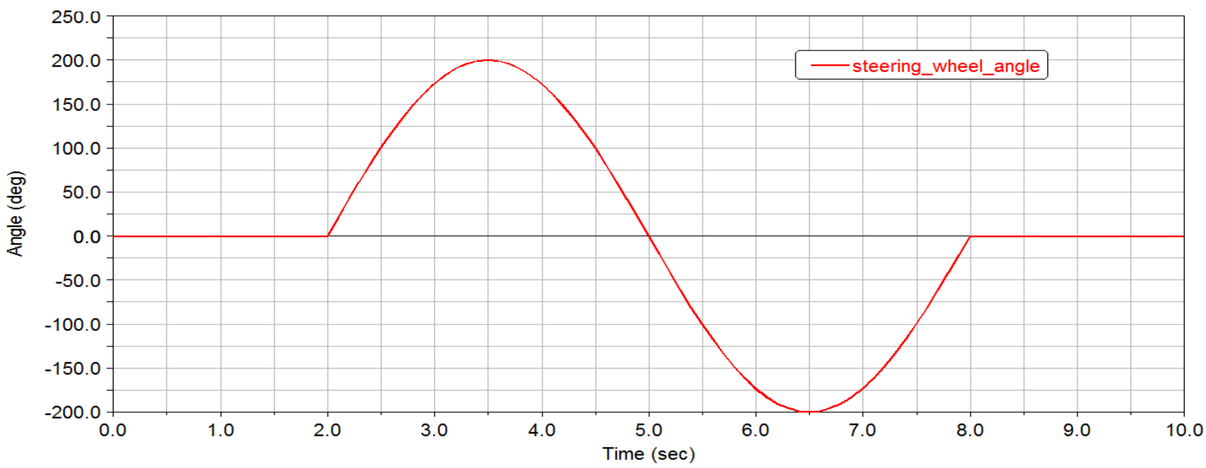

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Fig. 9 - Steering wheel input in single lane-change (positive angle means turning to the left side)

2. RESULTS

Two simulations were performed with the rear wheel drive. The first simulation is a single lane-change without torque vectoring and the second simulation is the same maneuver with torque vectoring. For the analyses 2D-flat road is used. Between the road and wheels is used default one point follower contact and the friction coefficient in the road property file is set to 0.5. The initial velocity is set to 50km/h (5). Fig. 10 shows the comparison of the total torque acting on driving wheels during maneuver with and without torque vectoring. It is obvious that during the single lane-change test with torque vectoring when the steering angle reached its maximum, nearly the whole driving torque is transmitted by outer (more weighted) wheel. While without torque vectoring is the driving torque during whole maneuver divided in ratio 1:1. So the main benefit of the torque vectoring is the transferring of the longitudinal (driving) force from the less weighted (inner) driving tire to the more weighted (outer) driving tire while the total longitudinal driving force of both wheels is maintained. This means that the torque vectoring system is able to keep the forces acting on the tires under their friction force limits and thereby increase the whole vehicle dynamic limits, as it is shown in Fig. 11. In this figure the vehicle side slip angles comparison during the single lane-change maneuver is shown.

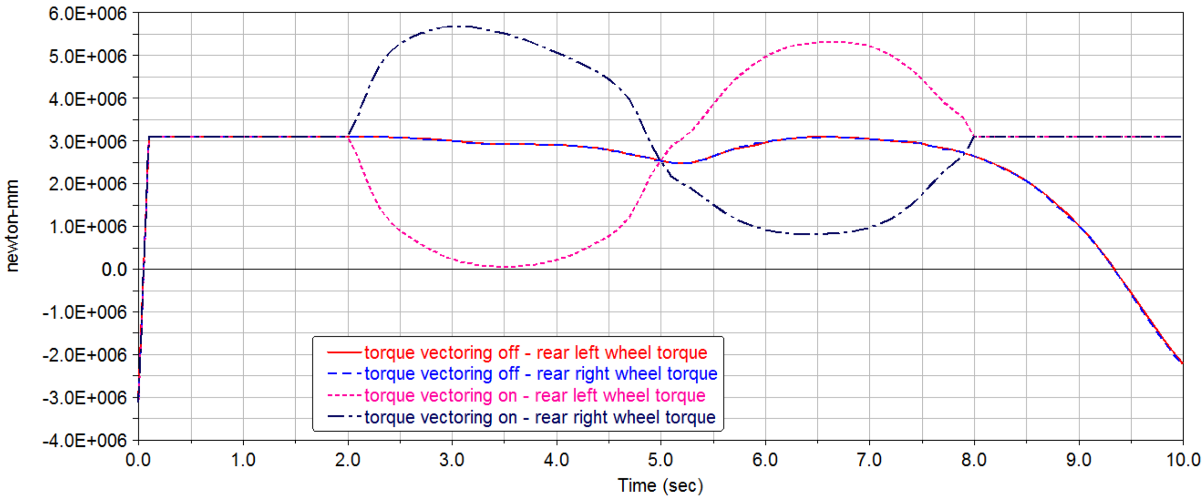

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Fig. 10 - Comparison of rear wheel drive torques during simulations with and without torque vectoring

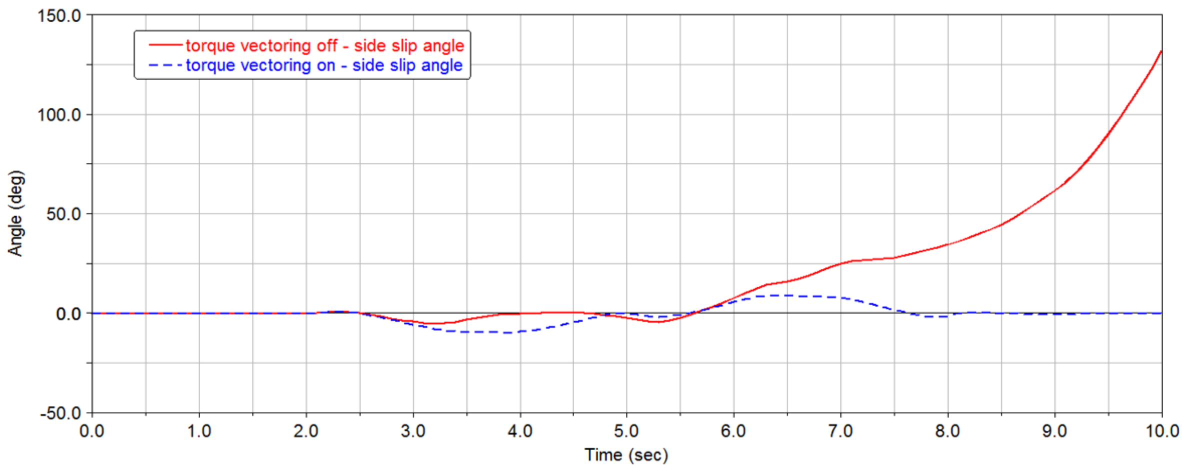

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Fig. 11 - Comparison of the side slip angles during simulations with and without torque vectoring

CONCLUSION

Simulation results show that the multi-body model is useful for demonstrating the torque vectoring basic effects. This can be very helpful for real lateral torque vectoring system design and development. However, it will be necessary to validate the results of the simulations with the real vehicle parameter measurement. Then the model will be able to provide a powerful tool to analyze vehicle behavior with various torque vectoring control strategies. It is possible to use not only the 3D-spline control but also to test real control algorithms by the MATLAB-ADAMS Co-Simulation that can be very helpful for control algorithm development and testing. This paper shows the basics of a heavy truck model created in ADAMS Software as well as the simulation of the vehicle behavior during a single-lane-change maneuver with and without the torque vectoring system. Simulation results show the potential of the torque vectoring technology that is in direct reducing and increasing of the longitudinal wheel forces which means an improvement of the vehicle dynamic limits. Systems like this are very useful for vehicle safety and operational efficiency. These facts are directly related with the main targets in modern vehicle development.

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