Paper Number : EAEC2011_A31

STABILITY AND PARKING CROSS DISTANCE ADAPTER MECHANISM FOR ELECTRIC VEHICLE

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Keywords: three-wheeler, cable steering, rollover, stability, track variation

Abstract

Due to the actual vehicle dimensions, low occupancy rates and the increase of world population, a trend to develop new vehicle concepts with smaller dimensions is seen. The objective of this trend is to reduce traffic congestion and park ability related issues in overpopulated areas. Not only footprint reduction is necessary, but also maintaining actual vehicle safety and comfort metrics is a must. Therefore, first of all, a mechanism to reduce wheel track to the third of the space required for a conventional car for low speed and parking situations is designed and developed. Once the track variation mechanism is completely defined, several suspensions and steering mechanism are analysed, concluding with a design of an unconventional front independent suspension and steering systems which are detailed in the present manuscript. By using MBS (Multi-body System) model, loads transmitted from tyre through mechanism joints to the vehicle chassis are computed. Special attention has been paid to the maximum peak loads that mechanism will suffer from road irregularities in extreme conditions. These load values are also used to evaluate the components that compose the track variation mechanism.

Continuing with the actuation of the system, an electronic control strategy is created and virtually tested, with what vehicle track position and vehicle speed and acceleration conditions are instantaneously controlled and related with the wheel track variation system.

Finally, the complete track width variation mechanism is manufactured, assembled and tested in a test rig.

Introduction

Society’s concern in environmental conditions is bringing vehicle manufacturers into electrification. In the same vein, also emission target for 2020 [Dol09] are tightened by European parliament. On the other hand, new vehicle concepts with smaller footprint dimensions need to be studied with the aim of reducing traffic congestion, making streets more quiet places and improving air quality in city centres. Not only footprint reduction is necessary, but also maintaining actual vehicle safety and comfort metrics is a must.

Vehicle manufacturers’ trends are focused on the development of vehicle concepts that combine the potential of both cars and motorbikes. From one side, they are emphasizing motorbike like footprint for urban road use as well as gaining capability to be parked in tight spaces; while on the other hand, they are focused on ensuring comfort and safety values of the current 4-wheeled vehicles. For that purpose, different manufacturers (OEMs) such as Lit Motors, BMW, Renault, Smart and Toyota have launched their personal mobility concepts, prototypes or commercial vehicles.

Regarding 4-wheeler electric vehicles such as the Renault Twizy, Smart for Two or the Nissan Land Glider, are designed and oriented for a continuous urban use. Due to their reduced length, they provide to the driver a better agility to park and drive through urban areas. In the same vein, but with the aim of reducing in a bigger scale the vehicle width, 3-wheeler prototypes such as BMW Clever or Toyota i-Road have been recently presented. These prototypes offer the driver an improved agility compared to the one of 4-wheeled vehicles. In relation to the reduced footprint and agility, a novel 2-wheeler prototype is also presented by Lit motors, where typical motorbike instabilities are solved employing in the lower part of the vehicle two gyroscopes that maintain vehicle equilibrium in any situation. As it is mentioned due to their reduced footprint, the mentioned three wheelers and two wheelers incorporate complex tilting systems to provide vehicle with the required stability [Bar06].

Different tilting proposals for three wheelers are studied. In such cases the goal of tilting systems is to achieve higher cornering lateral accelerations without unloading the vehicle front inner wheel. For instance, an electric three wheeler with a tilting system is proposed by Barker [Bar06, Bar10, Ber15], which is composed by a tilting occupant cabin and a tilting front steered wheel in the front, and a non-tilting rear traction unit where two rear wheels are integrated. Authors analytically demonstrate that the rear inner wheel contact patch load is increased while the tilt angle of the cabin is increased.

Due to the reduced dimensions of these kind of vehicles, it has also been demonstrated that the lateral cornering forces could generate instabilities [Val81, Hus82, Bar06, Ama11, Ber15] and it could cause the rollover of the vehicle, which is mainly associated with the high CG distances in comparison to the reduced vehicle wheel track values. However, it has been demonstrated that the rollover behaviour could be improved by increasing studied vehicle track width [Val81, Hus82].

Therefore, in this paper a design of a mechanism capable of reducing the wheel track for L5 vehicle category for low speed driving and parking situations and maintaining wide track during normal driving situations is developed to avoid installing tilting systems. The main goal of the mentioned mechanism is to reduce vehicle footprint in case of parking situation, while vehicle lateral stability metrics during normal driving situation are maintained due to its wide track position. To achieve packaging requirements, suspension and steering mechanisms have also been studied simultaneously and virtually (by FEM) validated. During the definition of the mechanical design, electronic control logic has also been developed, considering vehicle track position as function of vehicle speed and acceleration.

Vehicle concept and general specifications

The designed mechanism prototype is intended to be assembled in WEEVIL vehicle, which is an L5 vehicle category three wheeled prototype, developed as part of a collaborative research project founded by the European Commission, in which different novel technologies are installed (e.g. parking cross distance adapter mechanism (PINCER), continuously manufactured fiberglass structure and new vehicle power-train technology that integrates a Switched Reluctance Motor (SRM)). This electric vehicle, due to its qualities, it is directly oriented to be used in urban areas, being ideal for cities or mega cities.

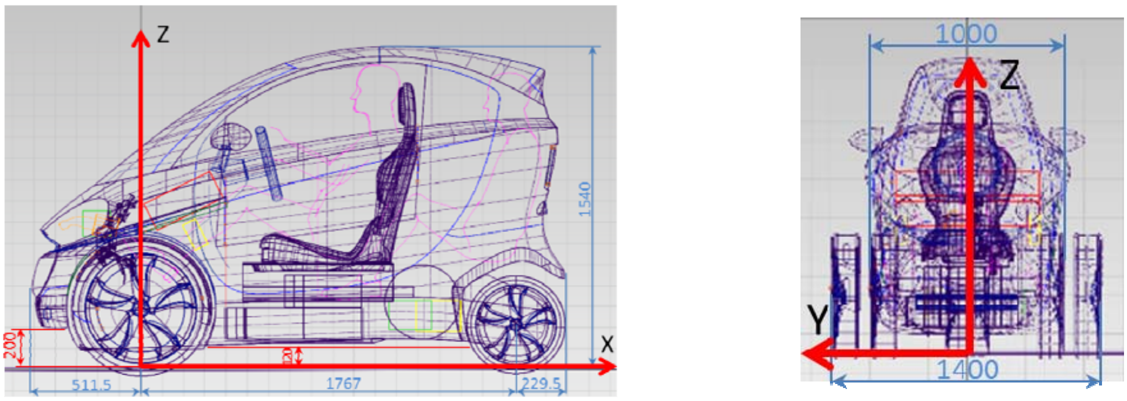
Regarding vehicle characteristics, which are presented in Table 1, this three wheeler is designed in tadpole configuration: 2 wheels are located in the front axle and the other one in the rear axle of the vehicle (2F-1R). The steering system is located in the front end while traction is located in the rear end, providing the vehicle enough power to reach maximum speed of 80km/h.

Feature	Value/description
Wheelbase (L)	1760 mm
Track Open (Wheel Centre) (B_o) / (Outside)	1270 mm / 1400 mm
Track Close (wheel Centre) (B_o) / (Outside)	870 mm / 1000 mm
Overall length	2500 mm
Front tires dimensions	130/70 R18
Rear tire dimension	205/50 R10
Kerb Mass (W)	492 kg
Passenger	75 kg
Kerb+1pass (W_1)	567 kg
Gross Vehicle Weight (GVW) (W_2)	642 kg
Kerb+1pass load condition (CG_X, CG_Y, CG_Z)	(771, 0, 438) mm
Ixx	28844954 kg*mm^2
Iyy	140910033 kg*mm^2
Izz	163913561 kg*mm^2

Table 1: Vehicle general specifications.

Focusing on the presented overall dimensions, this vehicle can be parked in the third of the space required to park a conventional car, which is the equivalent space to park a motorbike. These specifications, brings the user to get the most of a car and most of a motorbike, combining the comfort and safety of a car with the agility and park ability of a motorbike.

As it is intended to be perpendicularly parked in a motorcycle parking, the front and rear ends are designed with enough ground clearance to prevent crash between the car body and kerbs up to 200 mm height. In line with these trim heights, the defined minimum distance between road and vehicle lower part in the wheelbase area is the minimum required to climb a 120 mm kerb without producing any damage in the vehicle lower part.



(a) Lateral view. (b) Frontal view.

Figure 1: General vehicle dimensions

Vehicle mass, CG and inertia parameters shown in Table 1, which are measured taking into account the reference axles presented in Figure 1, are estimated from the CAD data of the vehicle, which was build following all component manufacturer's specifications.

Wheel track variation mechanism design and calculation

The present track variation mechanism is designed to be installed in WEEVIL prototype and extended to any L5-category vehicles, which is composed by an active system that allows increasing or decreasing vehicle's front axle width. The main objective of this mechanism is to reduce the required lateral parking space maintaining satisfactory vehicle dynamic performance.

Mechanism requirements:

This sub-system need to meet the following requirements:

- front wheel track variation from 1400mm to 1000mm while vehicle is being driven in a controlled speed.
- front wheels only displaced laterally due to ingress/egress ergonomic requirements and also in order to maintain the short length of the vehicle.
- supplementary mechanism for wheel toe orientation in opening and closing operations is dismissed because the required toe variation lay into toe alignment tolerances (±1deg) of similar vehicles.
- limited space for the mechanism. Vehicle structure manufacturing and safety target specifications. 200 mm are reserved in the frontal area of the vehicle for fibreglass crash structure.
- Vehicle operation voltages: (12 V or 74 V)
- mechanism track opening and closing time: 8 to 10 seconds and in between driving speed defined by MBS model.
- mechanism need to support loads introduced by road and transmission actuator.

Owing to the novelty of the mechanism and the lack of tyre road friction information, specifically when the car is been driven and tyres are being displaced laterally at low speed, some assumptions are going to be necessary to estate:

- Despite the fact that due to control logic, wheel lateral displacement condition for vehicle static position is not going to be able to happen, the designed mechanism is going to be dimensioned to switch from wide track to narrow track and vice versa considering the loads for such extreme condition. In this situation it is considered that both front tyres are suffering side slip condition in dry road, where a friction coefficient of 0,8 is considered between tyre and road.
- The selected vehicle load condition for such calculations is GVW. This condition is selected because the normal force in front tyres is the maximum permitted by the consortium.

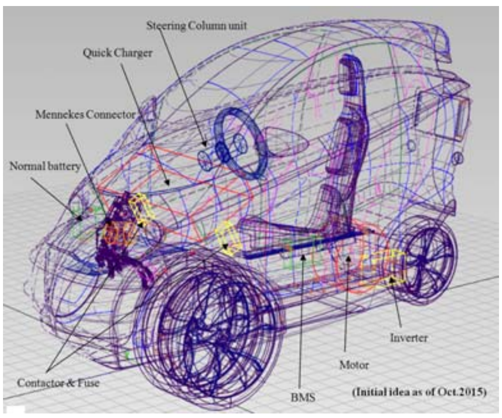
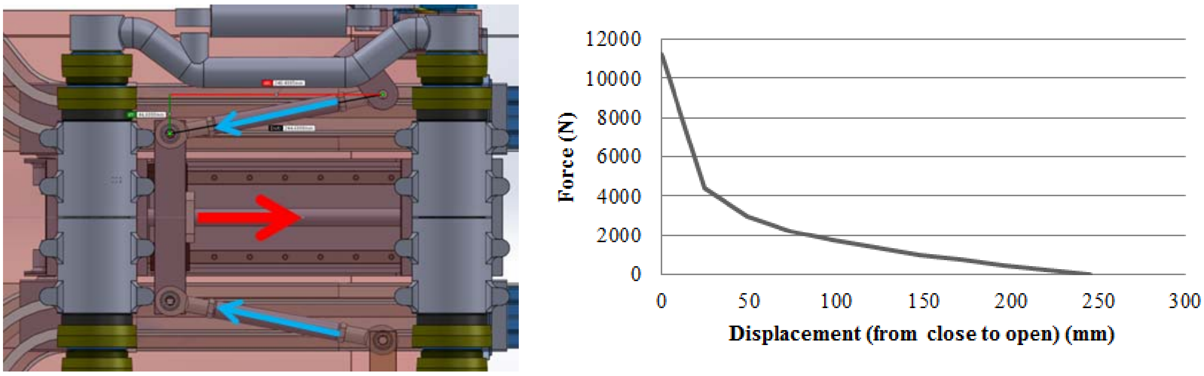


Figure 2: Location of vehicle mandatory components.

Mechanical components selection:

After analysing different conceptual designs and taking into account every technical specification presented before, with the aim of designing a robust and symmetrical component, this mechanism is going to be based on an actuator composed by a servomotor with a ball screw and a telescopic friction track variation guiding system.

In order to size de transmission, tyre lateral forces and push pull bars' dimensions and orientations are defined in Figure 3 (a). Once this geometry is defined, axial load for each displacement is described Figure 3 (b). Screw maximum axial loads are shown in Table 2.



(a) Ball screw disposition. (b) Ball screw load transmission.

Figure 3: Ball screw length and load condition.

Fy (Tire lateral force) (N)	1185
Traction/compression bar load (N)	5751
Ball screw max. axial load CLOSE (N)	11255
Ball screw min. axial load OPEN (N)	0

Table 2: PINCER loads.

Once tyre road load described and screw axial force and displacements calculated, the rest of transmission components are calculated, from ball screw, ball linear guides, belt-pulley transmission, gearbox and servomotor. From those calculations, minimum power of the servomotor is fully defined.

Electronic control strategy:

As the mechanism is embedded in a vehicle and itis intended to be used in parking situations while the vehicle is being driven, a communication between them is mandatory to work in line with vehicle instantaneous conditions. For that, despite the designed control logic is responsible for the correct operation of the developed mechanism, some input and output signal tracking sent in Controller Area Network (CAN) protocol are required between Vehicle Control Unit (VCU) and Pincer Control Unit (PCU). As well as for the mechanical design, this electronic design does not depends on other vehicle components, so it is possible to install it in other L5 category vehicles with minor changes on it.

The complete mechanism logic is integrated in a PCU, which is powered by vehicle auxiliary battery (12Vdc) and it is composed by a Texas Instruments F28335 control card and the necessary signal conditioning electronics. The main input and output signals for this control unit are shown in Table 3.

INPUT	OUTPUT
Vehicle speed request	Servomotor direction
Parking mode switch signal	Servomotor speed
Open limit switch signal	Parking mode
Close limit switch signal	Drive enabled
Front wheel speed sensors	PINCER system information (CAN)

Table 3: PCU input/output data.

The mentioned control card has been programmed in Matlab/Simulink software where two main parts are distinguished: mechanism control and communication. See Figure 4.

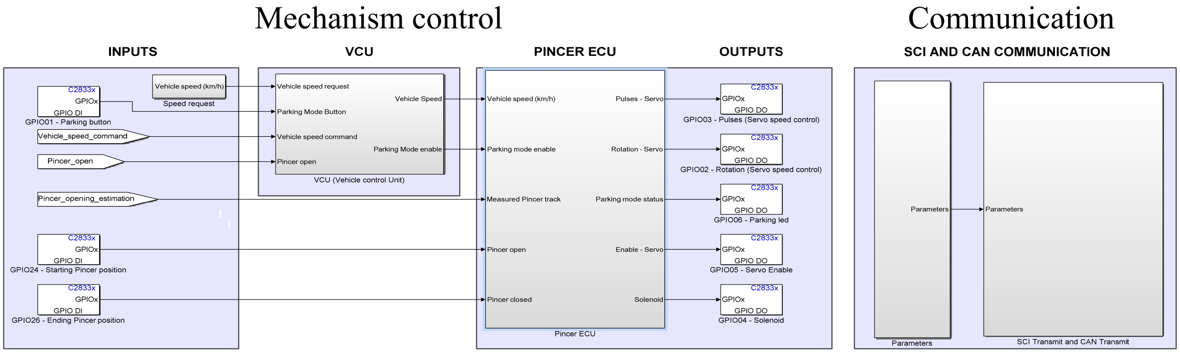


Figure 4: Control algorithm.

As it is mentioned, in relation to vehicle current speed, vehicle speed request, mechanism sensors signals and parking switch signal, the designed control logic is going to define if the mechanism may start its movement or not. Depending on the signals introduced to PCU, mechanism can open wheel track, close wheel track or stop wheel track movement. Once track variation is stopped and considering the received information, PINCER can continue opening/closing condition or revert operation limiting vehicle acceleration and velocity. In addition, in case of being necessary, PCU can send warning messages to VCU.

This control strategy has been virtually tested using Matlab/Simulink, where vehicle inputs are introduced by a CAN protocol simulator. This simulator sends in real time vehicle speed and the mechanism changes its working condition as if it was installed in the real vehicle.

Suspension and steering design and calculation

As well as for the wheel track variation mechanism, for suspension and steering components design, the space available in the frontal end of the vehicle and the track variation conditions of the vehicle are the limiting factors. It is for that reason that none of the commercial suspensions configuration and steering components is adequate for such a design being necessary an in deep study into the matter.

<u>Suspension and steering requirements</u>

Apart from the space available and the mentioned topological constrains, it is necessary to fulfil the following requirements:

- similar comfort and handling metrics of conventional vehicles. 1,8 Hz and 2,5 Hz have been respectively chosen for front and rear end since this vehicle is between conventional 4 wheeler and 2 wheeler. Typical vehicles front ride rates between 1-1,3Hz and 10 to 20% higher for the rear [Rei96], while for two wheelers those values are between 2-3Hz [Cos07].
- from vehicle exterior point of view, vehicle floor height variation is defined to be 15mm from kerb to GVW load case condition.
- a tyre stiffness of 210N/mm is assumed [Hei11].
- vehicle need to be able to turn in a 3,5m radius in low speed condition and close track position.
- steering mechanism need to fulfil Ackerman steering condition and lateral displacement of wheels.

<u>Component sizing and selection</u>

Installed front suspension type is a modified trailing arm, where the spring and damper units are placed in inclined position longitudinal to the vehicle driving direction. This suspension allows the transversal translation of the whole suspension system when track width variation mechanism is activated.

The steering system consists in a mechanical cable that transmits the torque from the steering wheel to the steering mechanism, providing the flexibility needed to vary the front track. This steering mechanism has also a multilink system to satisfy the Ackermann steering geometry.

Spring, damper and jounce bumpers are selected from a damper manufacturer, which are selected due to the custom made production system and due to a Corner Control Valve (CCV) technology available. This technology could detect if the vehicle is driving straight or in turning situation, increasing or decreasing damper force. Due to the wheel track variation mechanism and the reduced space in the front end of the vehicle, it is not feasible to install an anti-roll bar, so this technology could be ideal in order to improve vehicle roll and rollover characteristics of three wheelers.

Based on the vehicle occupancy rate for vehicles that are oriented to be used in urban areas [NTS09, Occ08], actual three wheeler suspension components are optimized for Kerb+1pass. load case. Results shown in Table 3.12 are obtained for such load case:

FRONT	Sprung mass frequency (Hz)	1,8
	Un-sprung mass frequency (Hz)	18
	Motion ratio	0,64
	Wheel compression travel (mm)	70
	Wheel rebound travel (mm)	40
	Spring stiffness (N/mm)	32
	Spring pre-load (kN)	2,14
	Bump stop length (mm)	20
REAR	Sprung mass frequency (Hz)	2,55
	Un-sprung mass frequency (Hz)	20,42
	Motion ratio	0,95
	Wheel compression travel (mm)	70
	Wheel rebound travel (mm)	30
	Spring stiffness (N/mm)	36,55
	Each spring pre-load (kN)	1,32
	Bump stop length (mm)	-

Table 4: Vehicle data.

With regards to design a vehicle with a comfortable ride and good handling values, it is necessary to install dampers with suitable power dissipation properties, which are defined with coefficients such as total average damping, damper asymmetry or progressivity factor [Dix99]. For the actual vehicle calculations the following damper parameters are chosen:

Damping ratio (ξ) (-)	0,35
Compression damping asymmetry (C_{DC}) (%)	30
Extension damping asymmetry (C_{DE}) (%)	70
Progressivity factor (λ) (-)	1
High damper velocity factor (-)	1,5
Low velocity to high velocity compression point (m/s)	0,5

Table 5: Damper data.

A target for vehicle minimum turning radius is defined. In preliminary steering calculations, neglecting Ackerman steering condition, 28 degrees of wheel angle is calculated to satisfy a 3,5m minimum turning radius. Those calculations were performed considering that vehicle

CG is positioned in the centre of it. It is necessary to mention that this angle can vary when Ackerman condition is applied, increasing the angle for the inner wheel and decreasing it for the outer wheel.

In order to satisfy Ackerman steering condition, a four bar link mechanism is designed. This mechanism is optimized to satisfy Ackerman condition when the steering angle is 18 degrees, which is the equivalent of being driving in a 5.5m radius. For this condition, outer an inner front wheels have the following Ackerman reference values: δ_i=20,15deg and δ_o=16,25deg. For the designed steering system, Ackerman error is less than 0.2 degrees for the different steering angles in open track positions.

In order to give the necessary flexibility to the steering system, steering rack mechanism and the steering four bar mechanism are installed near the dashboard of the vehicle. Those components directly actuate in a pulley assembly: from where the rotation of the pulley is transmitted to vehicle front wheels by means of a pulley and cable system, orientating vehicle each side wheel with the desired steering angle.

PINCER, steering and suspension FEM analysis

In parallel to the mechanism design, a three wheeler MBS model has been developed (not detailed here for the seek of clarity) and peak loads are obtained in order to be introduced in FEM calculations. Pothole event is taken as the most critical event and load cases shown in Table 6 are calculated.

	F_X	F_Y	F_Z
Front left wheel centre	+6390N	+2112N	-16074N
Front left contact patch	-	-	+21050N

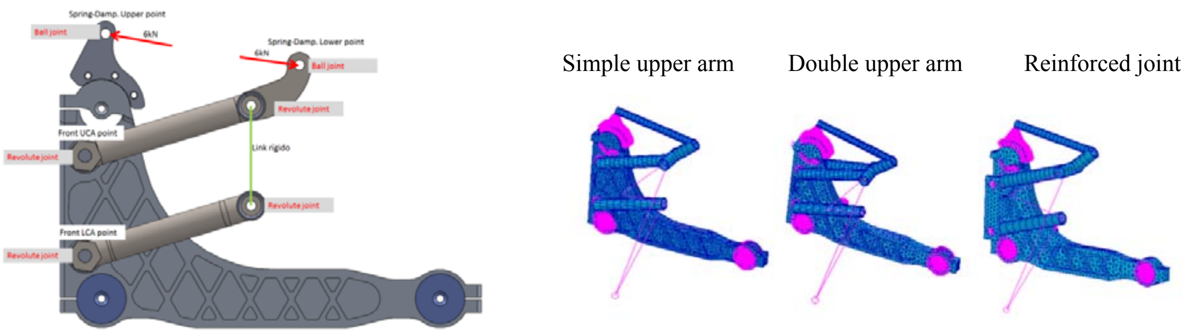
Table 6: MBS pothole load case.

Not only pothole loads are considered but also mechanism opening and closing loads are taken into account.

FEM analysis is divided in different subsystems which are shown below.

Front suspension subsystem:

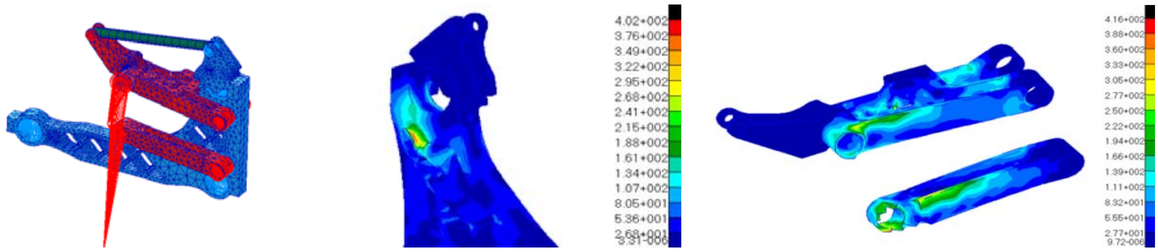
In a pothole event, the most critical condition for the actual assembly is when the suspension is in its full jounce position, where the spring and dampers are fully compressed. In this position, the defined loads are applied, keeping the boundary conditions shown in Figure 5 (a). Also, three different upper suspension arm joints (shown in Figure 5 (b)) are analysed in order to reduce torque and load generated in the upper arm joint.



(a) Front suspension assembly. (b) Upper arm variants.

Figure 5: Analysed front suspensions.

For the present study, the option of a double upper arm is selected, due to the higher stiffness and better load transmission it offers compared to the other considered options. Figure 6 shows the vonMisses stress distribution of this part.



(a) Suspension FEM model. (b) Suspension components max stresses.

Figure 6: Suspension FEM model and results.

As it is shown in Figure 6, the critical pothole load condition is transferring to the designed parts maximum stresses of 415MPa.

<u>Mechanism chassis and guiding supports:</u>

For the track variation mechanism chassis and guiding supports, pothole load case is applied. In such situation, the areas loaded with highest stresses are the attachments between guide supports and mechanism chassis (A area); and the attachments between mechanism chassis and vehicle structure (B area). As it is shown in Figure 7, the maximum vonMisses stresses are identified in those areas, where a value of 390MPa is measured in pothole condition.

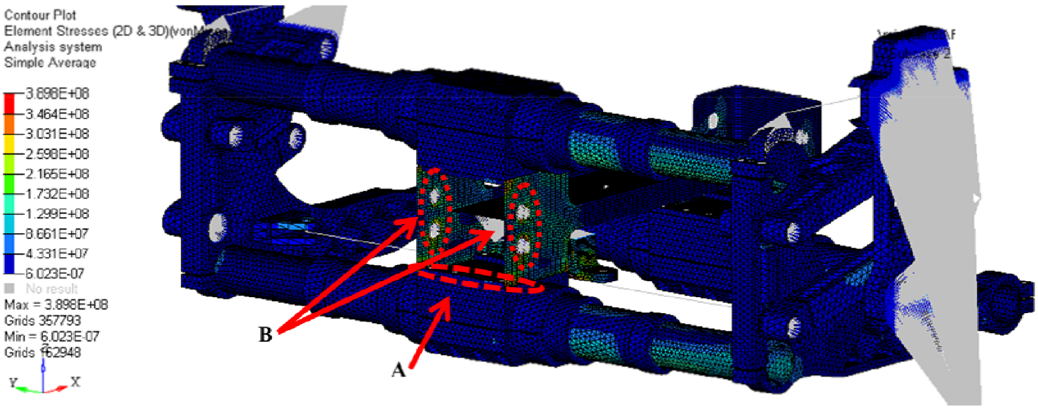
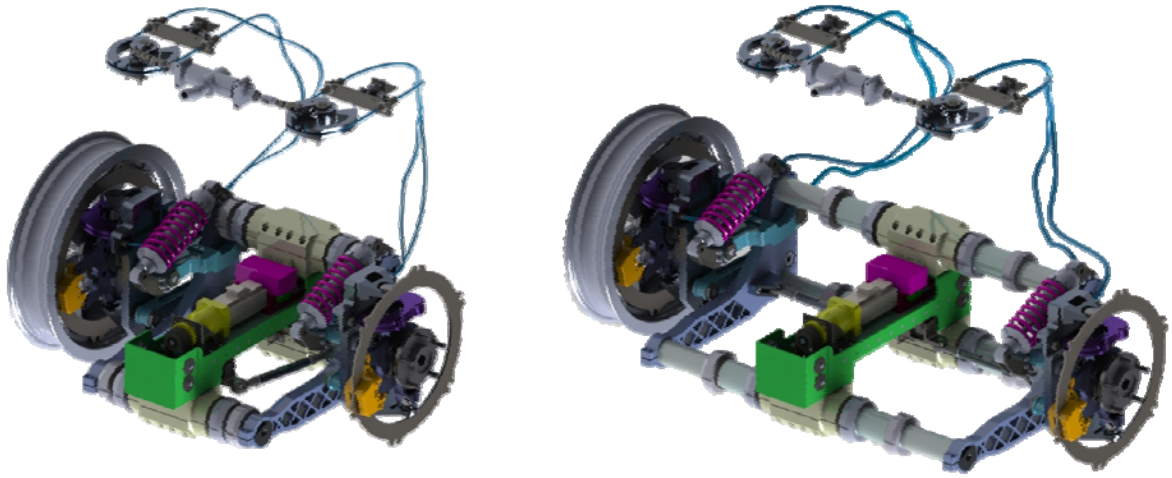


Figure 7: PINCER chassis and guiding support FEM results.

Final mechanism design

In this section the complete front end mechanism is presented. As it can be seen in Figure 8, one of the main advantages of the designed system is that it is completely independent from the vehicle assembly, which can be potentially easily adapted to another L5 category vehicle.



(a) Close PINCER mechanism (b) Open PINCER mechanism

Figure 8: Developed complete PINCER mechanism.

Conclusion

In this work, a modular novel complete track variation mechanism design and development is presented, where mechanism suspensions and steering specifications are fully defined.

It is also presented a modular electronic control unit developed for the control of the designed mechanism, where requested inputs, outputs and the integration in an L5 category vehicle is presented.

The correct function and sizing of the designed components are validated in FEM environment, where the loads introduced by road and track variation mechanism are applied.

After virtually validating the designed mechanism, complete track variation, suspension and steering systems are into manufacturing process. Once assembled all the components and before installing them in WEEVIL vehicle prototype, different laboratory test are going to be performed to it. On one hand, suspension system is going to be tested in a Quarter car test bench developed by Mondragon Unibertsitatea, while on the other hand, complete track variation mechanism is going to be tested in a test rig that introduces the boundary conditions and loads expected in WEEVIL vehicle.

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