7 Chassis and Suspension Concepts

Chassis and suspension concepts for ULSAB-AVC feature a range of high strength steels and new technologies such as TWBs for wishbones and tailor tube hydroforming.

7.1 BACKGROUND

The principal goal for the development of front and rear suspensions of the ULSAB-AVC concept vehicles (C-Class and PNGV-Class) was the design of steel intensive lightweight suspension. Conventional and new steel materials should be applied together with their related manufacturing processes and assembly technologies. The goal was addressed within the overall approach of designing the lowest overall mass of the complete vehicles rather than the lightest possible suspension concepts. An alternative, lighter weight suspension system with low mass may not contribute to achieving the lowest overall vehicle mass because they could cause mass increases in other components (e.g. body structure).

7.2. Approach

Although the ULSAB-AVC Program involves the development of two vehicle concepts, with either a diesel or gasoline engine variation, it was necessary to develop similar suspensions for both C-Class and PNGV-Class vehicles. Of course tuning components (e.g. springs, dampers) would have to be specific to each vehicle to accommodate the range of wheel loads in conjunction with vehicle mass. The final set-up would have to be tuned on the test tracks. Components were calculated, with the resulting forces of the load cases described in Section 7.8.2. in terms of stress, and designed accordingly. Specified carry-over parts were adapted respecting the different mass and loads of the ULSAB-AVC vehicles.

The emphasis on safety was, in some respects, a principle driver of the concept design of the body structure. This decision required that a significant deformable distance be achieved between the vehicle skin and the shock tower location under the hood and influenced the shock and spring location. Therefore, the selection of the front suspension type took into account anticipated pedestrian safety.

In addition, the front end module had to be designed as a system including engine, transmission and front suspension attached to a front subframe in order to enable assembly and disassembly as one unit without the requirement to realign the suspension after assembly.

A critical assessment of emerging non-steel chassis technologies (e.g. ceramic brake discs) was undertaken and their use was considered in relation to overall program objectives whenever a significant benefit would result from their use. The impact on manufacturing cost and specified mass production (> 225,000 units/year) was also considered.

7.3. Scope of Work

The scope of work for the development of the suspension concepts included the following tasks:

- Assessment of benchmarking results of chassis components of the Ford Focus and Peugeot 206 from the Benchmarking Report (found in the ULSAB-AVC Consortium Document Technical Transfer Dispatch #3) and publicly available data for overall target setting
- Evaluation of suspension mass
- Calculation of chassis components, mass, curb weight, gross weight and axle load
- Mass estimation of chassis components
- Preliminary consideration of front and rear suspension concepts
- Concept design of front and rear suspensions with regard to mass reduction of suspension components and subframe components
- Definition of suspension kinematics
- Selection of steering system concept
- Selection of engine mounting concept (without evaluation of bearing stiffness)
- Selection of carry-over brake system
- Calculation of axle loads and calculation of the forces and torque on the components of the suspension
- Calculation of component load and forces
- Selection of component manufacturing concept (e.g. stamping, hydro-forming)
- Design Layout of suspension components

- FEM-calculation to analyze the suspension component design layout (e.g. dimensions, materials)
- Incorporation of FEM-calculation results into suspension designs
- Definition of drive shafts
- Assessment of manufacturing and assembly costs

7.4. Benchmarking Data and Target Setting

Prior to the start of the concept design phase, benchmarking data for mass of various suspension systems was gathered to set the targets for both C-Class and PNGV-Class vehicle suspensions. These were developed based on experience and engineering judgement and are displayed as total mass targets in Table 7.4-1 along with the benchmarking data of a Ford Focus and a Peugeot 206, which was reviewed.

Table 7.4-1 Suspension benchmarking data and ULSAB-AVC suspension targets

Component Name	Ford Focus (kg)	Peugeot 206 (kg)	ULSAB-AVC Targets** C-Class and PNGV-Class (kg)
Front Suspension incl. subframe	55.05	53.79	50.0
Rear Suspension incl. subframe	55.04	42.13	42.0
Pedal system	7.51	7.82	5.7
Wheels with tires	84.85	67.81	46.2
Brake system	39.59	45.55	38.5
Steering system	19.39	16.86	16.0
Others	2.84	2.87	n/a
Total Mass*	264.26	236.82	198.5

** Total mass of suspension does not include drive shafts*
*** For cost reduction and as a result of the small difference in total vehicle ma the decision was made that both vehicles will share the same suspension wit exception of tuning parts*

Using these component mass targets, the curb and gross weight of the vehicles were calculated.

7.5. First Assumptions for FEM-Calculation

The following assumptions were used as input parameters for the initial load calculation:

7.5.1. Suspension Load Input Parameters and Assumptions

Table 7.5.1-1 Axle load input parameters

	C-Class			PNGV-Class		
	Front suspension	Rear suspension	Weight	Front suspension	Rear suspension	Weight
Curb weight (kg)	n/a	n/a	998	592	485	1077
Design load (kg)	n/a	n/a	1253	n/a	n/a	1302
Gross weight* (kg)	730	713	1448	741	836	1577
Maximum payload (kg)	450			500		

n/a = not applicable
** at wheel center*

7.5.2. Kinematics Layout Assumptions

For the first kinematics layout, the following assumptions were used.

Table 7.5.2-1 Kinematics layout assumptions

Dimension	Front Suspension	Rear Suspension
Wheel base C-Class/PNGV-Class	2950 mm / 3035 mm	
Track	1537 mm	1540 mm
Height of center of gravity at curb weight	550 mm	
Wheel (tires/rims)*	175/65 R14 5J x 14	
Wheel travel	+ 90 / -105 mm	+/- 105 mm
Steering rack travel	+/- 79 mm	

** optional 185/60 R15 5.5Jx15*

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7.6. Front Suspension

7.6.1. Concepts Considered

7.6.1.1. McPherson Principle

The first front suspension considered and assessed for its suitability in the ULSAB-AVC vehicle concepts was a McPherson front suspension. It was considered because of its good balance of drive-ability (handling and comfort), mass and cost. A principle layout of a McPherson's front suspension is illustrated in Figure 7.6.1.1-1.

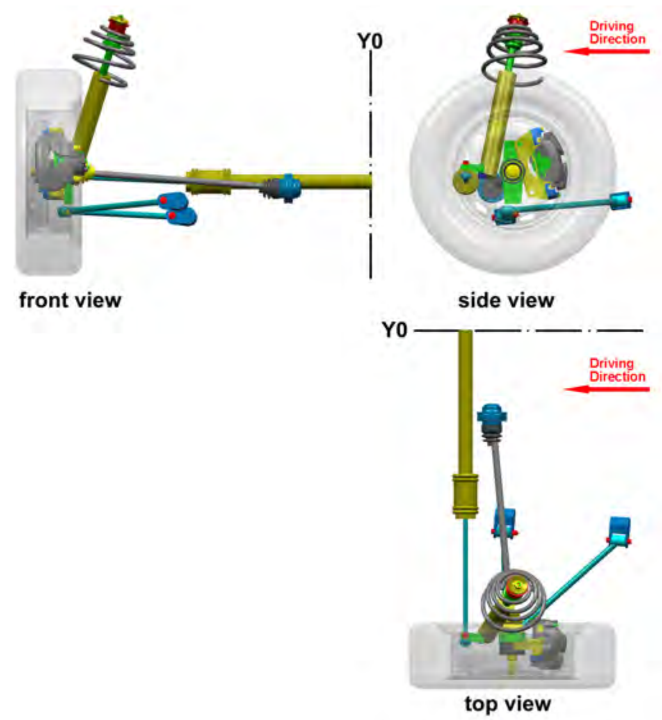

Figure 7.6.1.1-1 McPherson principle layout

The main reasons the McPherson concept was not selected for the ULSAB-AVC vehicle were:

- Strut attachment points lead to a high shock tower, which is located too close to the outer vehicle skin (e.g. hood) and will not provide sufficient space in the event of pedestrian impact (see Figure 7.6.1.1-2 on next page)
- Concept requirement of a powertrain/suspension module with advantages in assembly and disassembly, cannot be met. Suspension has to be aligned after disassembly and cannot preset prior to assembly in final assembly line.

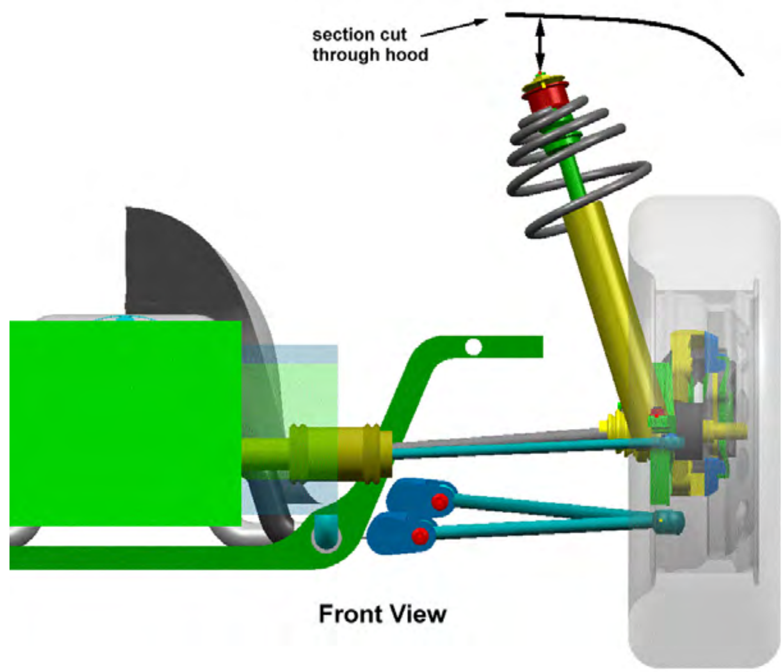

Figure 7.6.1.1-2 McPherson suspension with section cuts through the body structure

7.6.1.2. Double Wishbone Principle

The next system, which was considered for implementation into the ULSAB-AVC Program was a double wishbone front suspension concept in various configurations.

7.6.1.2.1. Double Wishbone with Coil Spring

Two solutions for the double wishbone with coil spring were considered, but rejected– first, a double wishbone suspension with a spring strut and next, with a separated damper and coil spring. In the first case using a spring strut, a package investigation showed insufficient clearance between the other suspension components. The only feasible position was to package the coil spring above the wheel, which resulted in a similar position (at the upper shock absorber point) to the McPherson principle; close to the outer skin of the hood. In the second case, placing the coil spring beside the wheel and packaging the damper in a different position to lower the upper attachment point of the shock absorber was not practical due to insufficient clearance.

7.6.1.2.2. Double Wishbone with Torsion Bars

Another possibility to lower the upper mounting point of the damper was investigated using torsion bars. In Figure 7.6.1.2.2-1, both possible positions of the torsion bar are shown (upper wishbone and lower wishbone). Either torsion bar locations could cause passenger compartment intrusion in frontal crash accidents, increasing the risk of leg injury to passengers and therefore, this suspension concept was not selected.

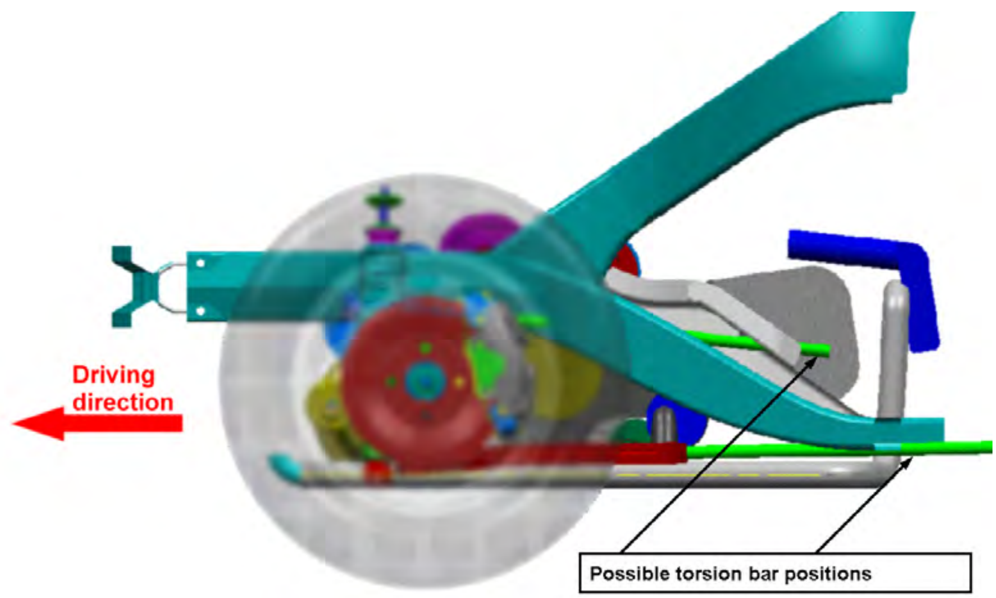

Figure 7.6.1.2.2-1 Possible locations of torsion bar

7.6.1.2.3. Double Wishbone with Transverse Leaf Spring Principle

Since packaging of the coil spring and torsion bar were not possible, a transverse leaf spring was investigated. The double wishbone concept (see Figure 7.6.1.2.3-1) with transverse leaf spring showed the most potential for packaging, as well as achieving mass targets. Although this concept does not represent the lowest suspension mass, at lowest cost, it was selected as the most promising front suspension concept for the overall vehicle concept.

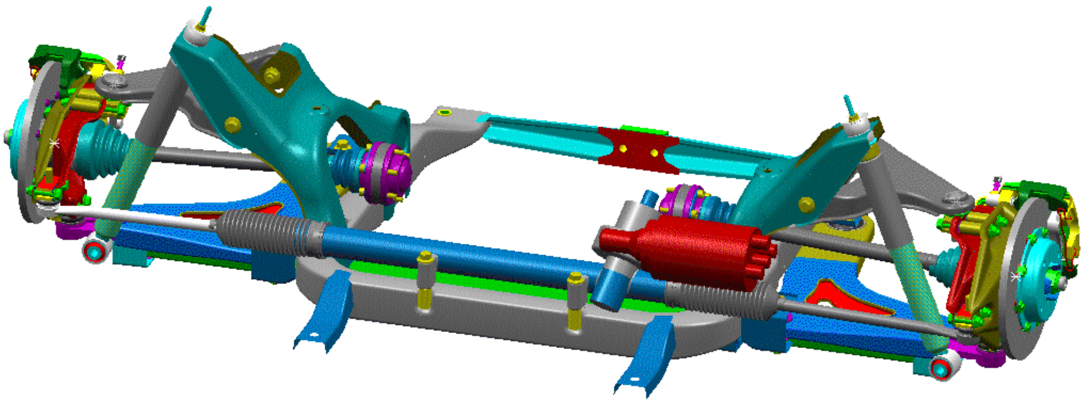

Figure 7.6.1.2.3-1 Double wishbone with transverse leaf spring

7.7. Front End Module

The package studies highlighted the benefits of a front end module made of a subframe with engine, wishbones, steering knuckle and damper, steering gearbox and tie rods, transverse leaf spring and the cooling system (radiator and intercooler). This front end module would allow ease of assembly during vehicle manufacture, as well as disassembly for maintenance or repair over the vehicle life.

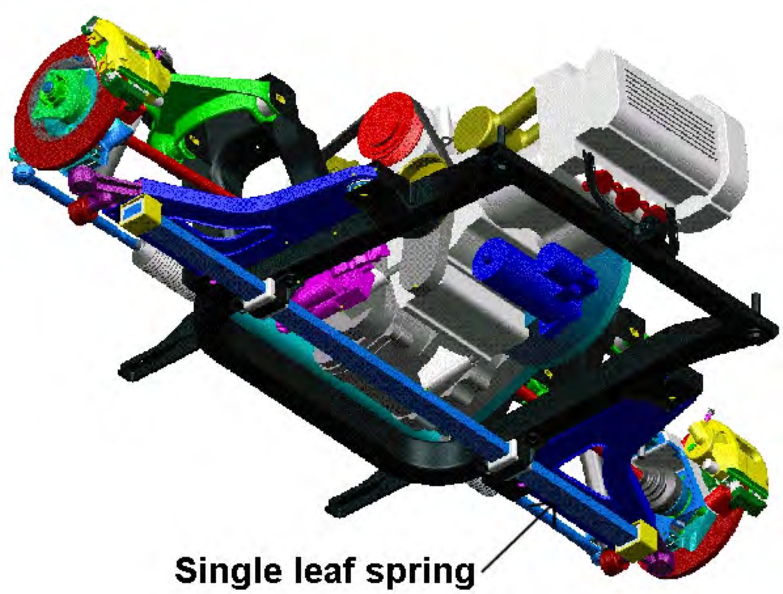

Figure 7.7-1 Front end module bottom view

7.7.1. Engine Mount Concepts

For the selection of the engine mount system, manufacturing feasibility, function and comfort had to be considered. Three point and four point engine mount concepts were considered

A four-point engine mount has the advantage that only small support brackets are necessary to attach the engine mount bearings. Additionally, these brackets can be placed close to the subframe tube as shown in Figure 7.7.1-1. Another advantage of this system is the reduced movement of the powertrain.

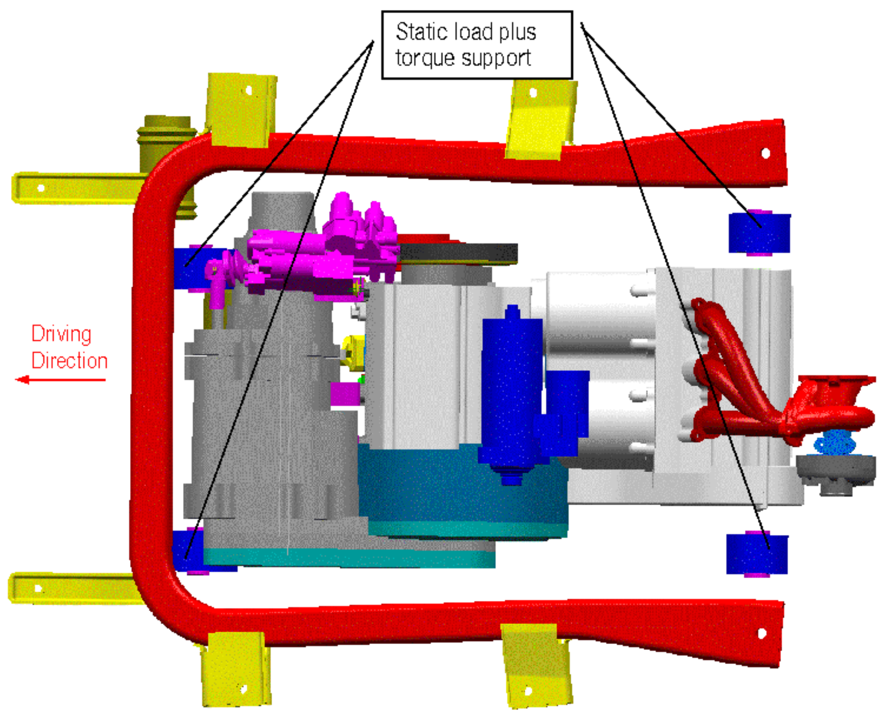

Figure 7.7.1-1 Four-point engine mount concept bottom view

7.7.1.1. Selected Engine Mount Concept

As an alternative to the four-point engine mount concept, a three-point engine mount system as shown in Figure 7.7.1.1-1 was investigated for potential of implementation into the subframe. Compared to the four-point engine mount system, a three-point engine mount system shows superior acoustic behavior when the bearings are carefully placed. In the three-point engine mount concept, two bearings absorb the static loads of the powertrain and are therefore, located close to its center of gravity of powertrain. The third bearing functions as a torque support. With a three-point engine mount concept, tuning of the bearings becomes easier. A cross beam has to be provided for the rear mount to increase lateral stiffness and increases the subframe mass.

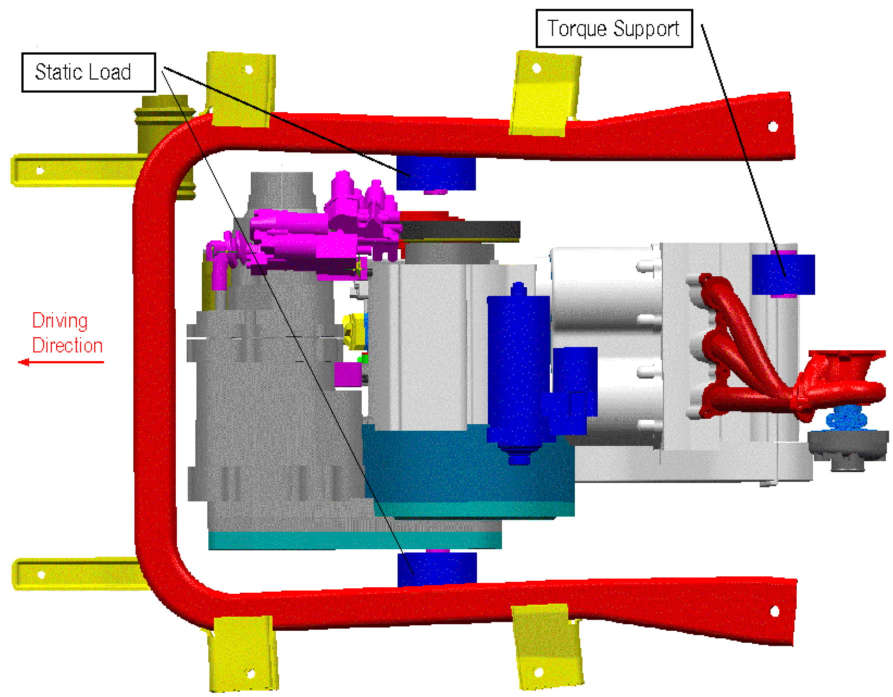

Figure 7.7.1.1-1 Three-point engine mount concept bottom view

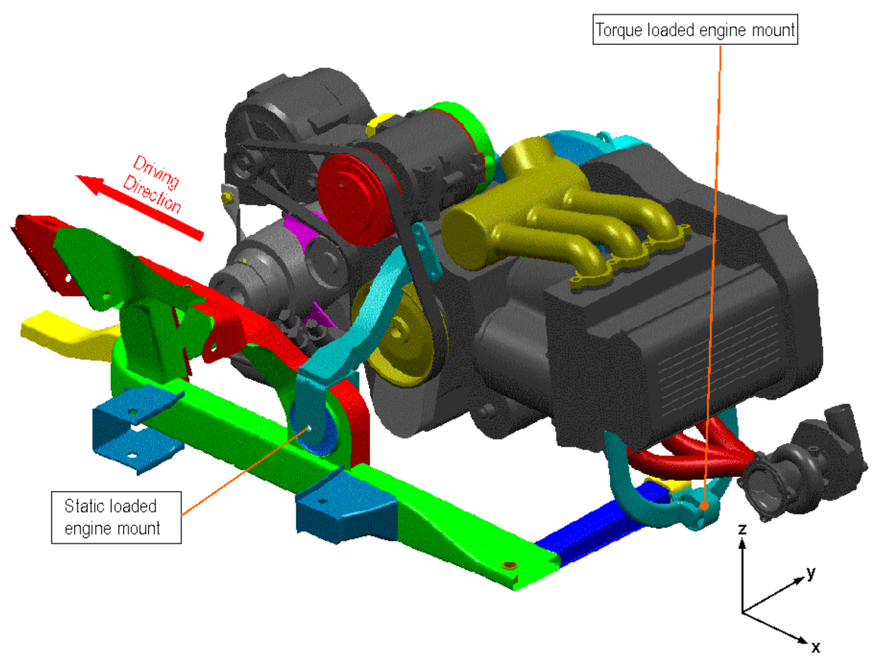

Figure 7.7.1.1-2 Selected three-point engine mount system

7.7.1.2. Engine Mount Brackets

The front engine mount brackets are assemblies made from square sections with a material thickness of 2.5 mm and two brackets, one for the attachment to the engine and one for the attachment to the subframe. DP 350/600 material was selected for the brackets. The rear bracket is produced out of bent sheet steel.

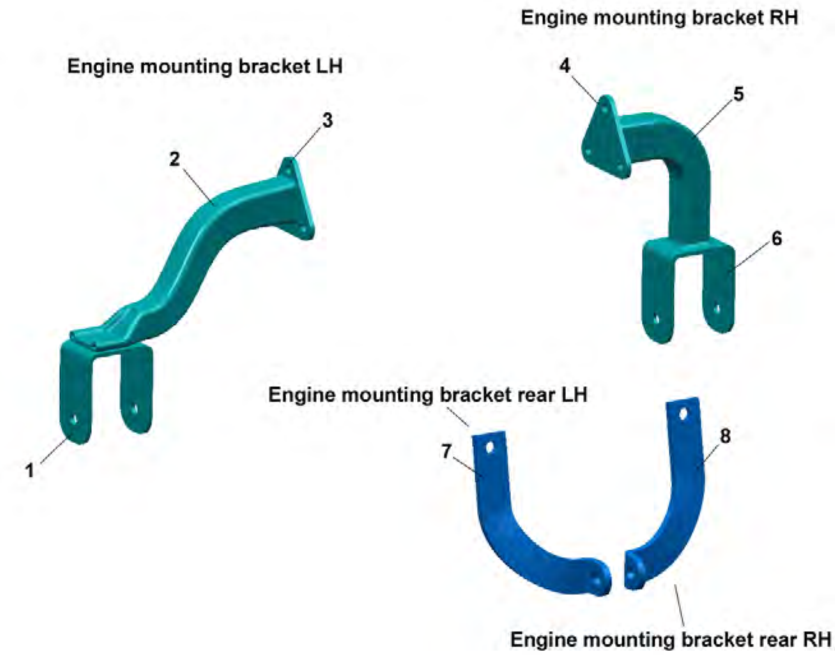

Figure 7.7.1.2-1 Engine mount brackets

Table 7.7.1.2-1 Engine mount bracket parts list

	Part Number	Material Thickness [mm]	Material Type	Mass [kg]
Engine mounting bracket LH	1	4.0	DP 350/600	1.047
	2	2.5	DP 350/600	
	3	5.0	DP 350/600	
Engine mounting bracket RH	4	7.0	DP 350/600	0.809
	5	2.0	DP 350/600	
	6	4.0	DP 350/600	
Engine mounting bracket rear LH	7	7.0	DP 350/600	0.37
Engine mounting bracket rear RH	8	7.0	DP 350/600	0.37
Total mass engine mounting brackets [kg]				**2.596**

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The forces on the brackets were calculated considering the following load cases:

Table 7.7.1.2-2 Load case

Acceleration	g
Vertical acceleration	3.4
Lateral acceleration	1.0
Longitudinal acceleration	0.45
Acceleration rearward	0.4
Static	0

In the FEM-calculations, the maximum force calculated on the two mounts located on the side of the engine was 5500 N in z-direction (vertical). For the rear engine mount, maximum forces of 2636 N in z-direction and -985 N in x-direction (longitudinal) were calculated with insignificant forces in y-direction.

These forces were calculated using the mass of the diesel engine. The same parts are used for the gasoline engine, which would generate lower forces due to its lower mass.

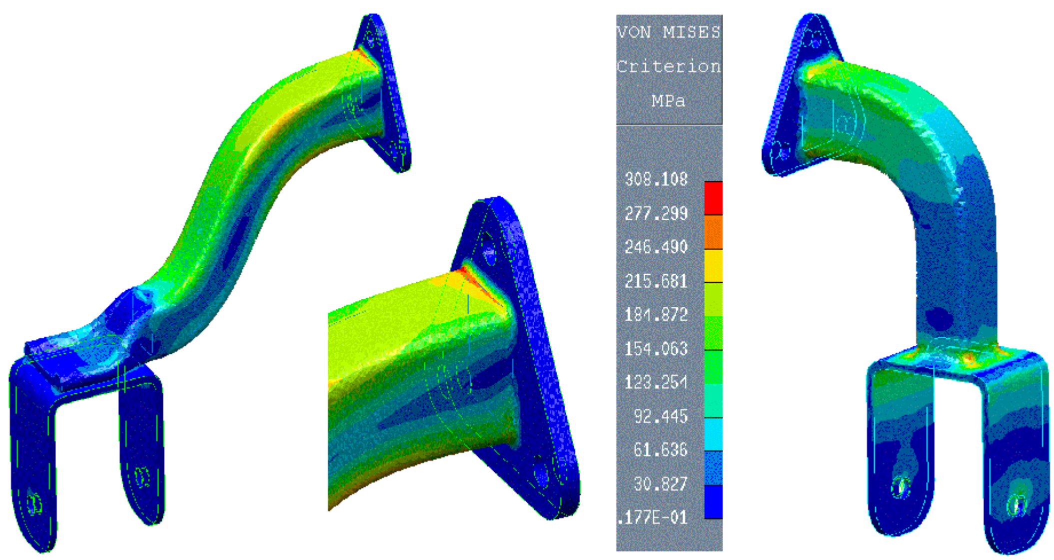


Figure 7.7.1.3-2 FEM calculation results engine mount brackets

The FEM-calculations shown in Figure 7.7.1.3-2 predict a stress level of about 300 MPa in the area of the welding seam. These areas would be optimized in a detailed design phase.

7.8. Double Wishbone Front Suspension

As previously mentioned, a double wishbone front suspension was selected for implementation in the ULSAB-AVC vehicle concept (see Section 7.6.1.2.3). Under the given boundary conditions, including the modular assembly approach, the task was to create a front suspension layout, which was compatible with the other package requirements.

7.8.1. Subassembly Front Suspension

The subassembly front suspension includes the subframe, upper and lower wishbone, steering knuckle, damper, steering gearbox with tie rod, and transverse leaf spring and is shown in Figure 7.8.1-1.

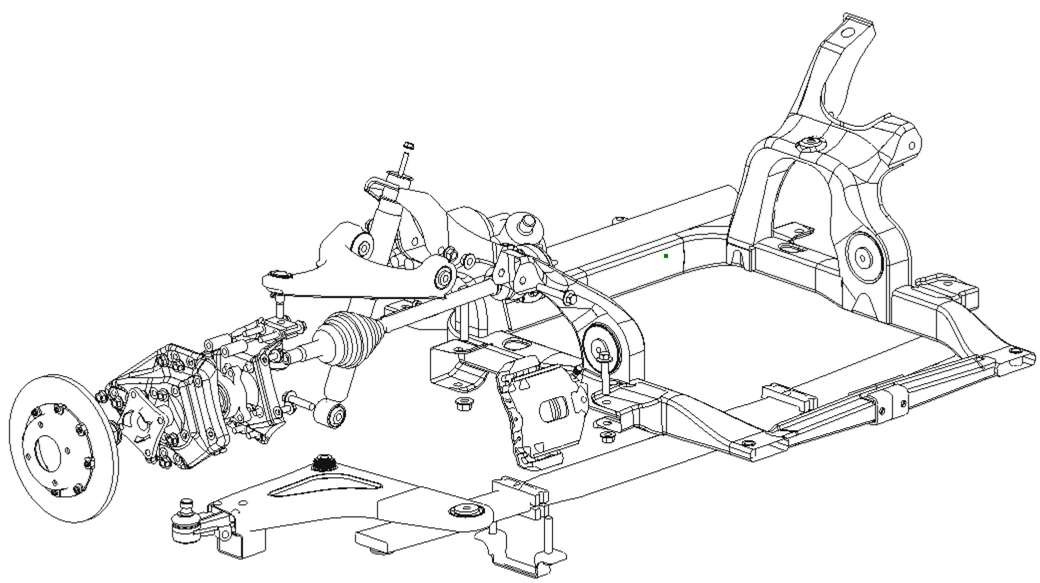


Figure 7.8.1-1 Subassembly front suspension (LH)

During assembly, the upper and lower wishbone are attached to the subframe using two nuts and bolts for each. An alignment jig is necessary for the assembly of the front suspension because the upper wishbone must be assembled to the subframe in design position in order to get symmetrical torsion angles of the rubber bushings for jounce and rebound. An unsymmetrical torsion angle of the rubber bushings could cause damage to the bushings themselves.

The upper wishbone controls the wheel camber and castor angle and also supports the damping and spring forces. The wishbones are attached to the steering knuckle at their outer edges.

The steering knuckle is made of two steel stampings, a 2nd generation wheel bearing, a brake caliper, two spacer sleeves and attachment brackets for the upper and lower wishbone and steering rod. The wheel hub is pressed into the wheel bearing and attached to the outer joint of the drive shaft. The brake disc completes the wheel hub assembly.

The steering gearbox, including the steering rods, is attached to the front of the subframe and the outer ball joints on the steering rods are attached to the steering knuckle. A transverse leaf spring is attached from below, with two brackets to the underside of the subframe. The leaf spring is loaded by attachment to the lower wishbone near the outer ball joint. The damper's lower attachment point is the outer ball joint housing of the lower wishbone and its upper attachment point is the subframe.

7.8.1.1. Kinematics Layout

The concept for the kinematics layout was made to reach the following characteristics, which are shown in Table 7.8.1.1-1:

Table 7.8.1.1-1 Kinematics results

Track	1542 mm	**Track change**	1.19 mm/cm
Toe in	10'	**Toe in change**	3.4'/cm
Camber	-0.50 deg	**Camber change**	-0.1deg/cm
Kingpin offset	-7.7 mm	**Kingpin inclination**	15 deg
Castor angle	2 deg		
Castor offset	23 mm	**Castor offset in wheel center**	14 mm
Roll center height	44 mm	**Spring rate rel. to wheel center**	19 N/mm
Spring rate	30 N/mm	**Damper ratio**	0.74

The kinematics characteristics are based on existing vehicles currently on the market. To achieve a safe and stable driving behavior, the kinematics characteristics were tuned towards an understeering behavior. This kinematics layout should be validated in tests using ULSAB-AVC prototype vehicles and will likely be changed during the development process. The elastokinematic behavior is also important for the vehicle response during maneuvers and must be tuned. The ULSAB-AVC design concept layout allows for such variations of hardpoints and mounting stiffness, which were considered in the package development for future tuning as described above.

In general, two of the most significant kinematics characteristics are the change of toe-in and camber in relation to wheel travel and are shown in Figure 7.8.1.1-1.

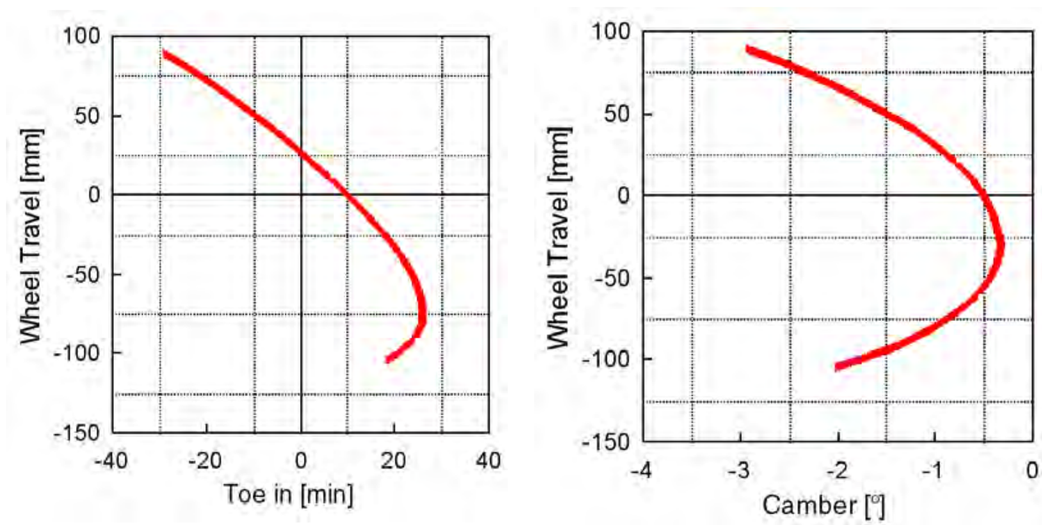


Figure 7.8.1.1-1 Toe in and camber characteristics of the front suspension

The elastokinematic behavior (e.g. longitudinal compliance) of the suspension is not specified here since it is dependent on results of vehicle dynamic tests. There is enough package space in the area of the rubber bushing locations for optimization of the elastokinematic behavior by changing the shape of the rubber bushings, if necessary.

7.8.2. Front Suspension Components Load Distribution

The load distribution of the front suspension components had to be taken into account for the design of the front suspension. The load distribution is a product of vehicle mass and the resulting wheel loads (see Table 7.5.1-1), as well as the different forces resulting from steady state driving maneuvers. The maneuvers and resulting accelerations considered are shown in Table 7.8.2-1. In addition, a factor of 1.5 for safety was considered. The maneuver "curbstone push" is a misuse load case for the front suspension. This load case describes a vehicle standing with two wheels directly besides a curbstone and the vehicle will be pushed from the curbstone by turning the steering wheel slowly.

These maneuvers create the tire forces acting on tire contact area. The forces are transformed into forces and torques in the wheel center and distributed to each component of the suspension by using an elastokinematic program and also taking the wheel travel into account. The resulting values of reaction forces and torque for each component of the suspension are used as load input in the FE-models for part stress analysis and can be found in Appendix - Section 4.3.

Table 7.8.2-1 Maneuvers and accelerations taken into account

Manuever	Acceleration (g)	
Vertical push	Vertical acceleration	3.4
Steady state cornering	Lateral acceleration	1.0
Braking forward	Longitudinal deceleration	-1.0
Braking rearward	Longitudinal deceleration	0.9
Acceleration	Longitudinal acceleration	0.45
Curbstone push (front axle	Static	0

Combined load cases can be analyzed by superposing the results of the individual FEM-calculations. If the stress levels of the components are in an uncritical range, in most cases, the stiffness of the suspension components are also in an uncritical range. The design is mostly determined by the misuse loads.

7.8.3. Subframe Design Concept

The subframe (see Figure 7.8.3-1) is designed to transmit the forces from the suspension into the body structure while suppressing the acoustic excitation of the body structure by the suspension. With the selection of rigid attachment of the subframe, the subframe stiffness is an important contributor to the rigidity of front suspension and body structure system. The square section of the tube was chosen to get the best stiffness for the support of the vertical and lateral forces from the front bearing of the lower wishbones and the leaf spring and the rear bearing of the lower wishbones and the engine mounts. These spring forces are supported in the tube and therefore, the console, which is designed as a closed section supports the forces of the shock absorber to provide high stiffness and transfers the loads into the body structure. The subframe design was developed in an iterative process with crash analysis.

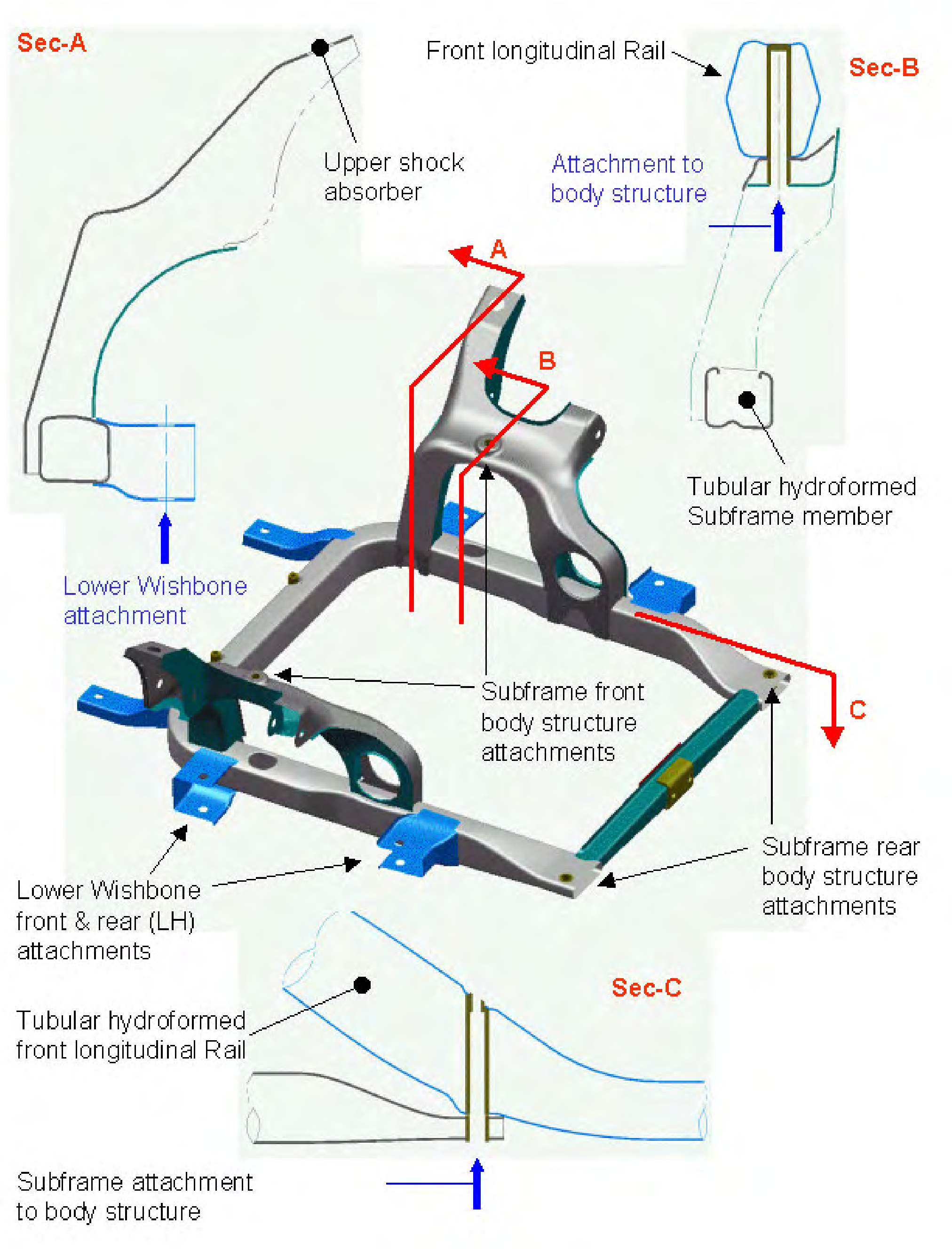
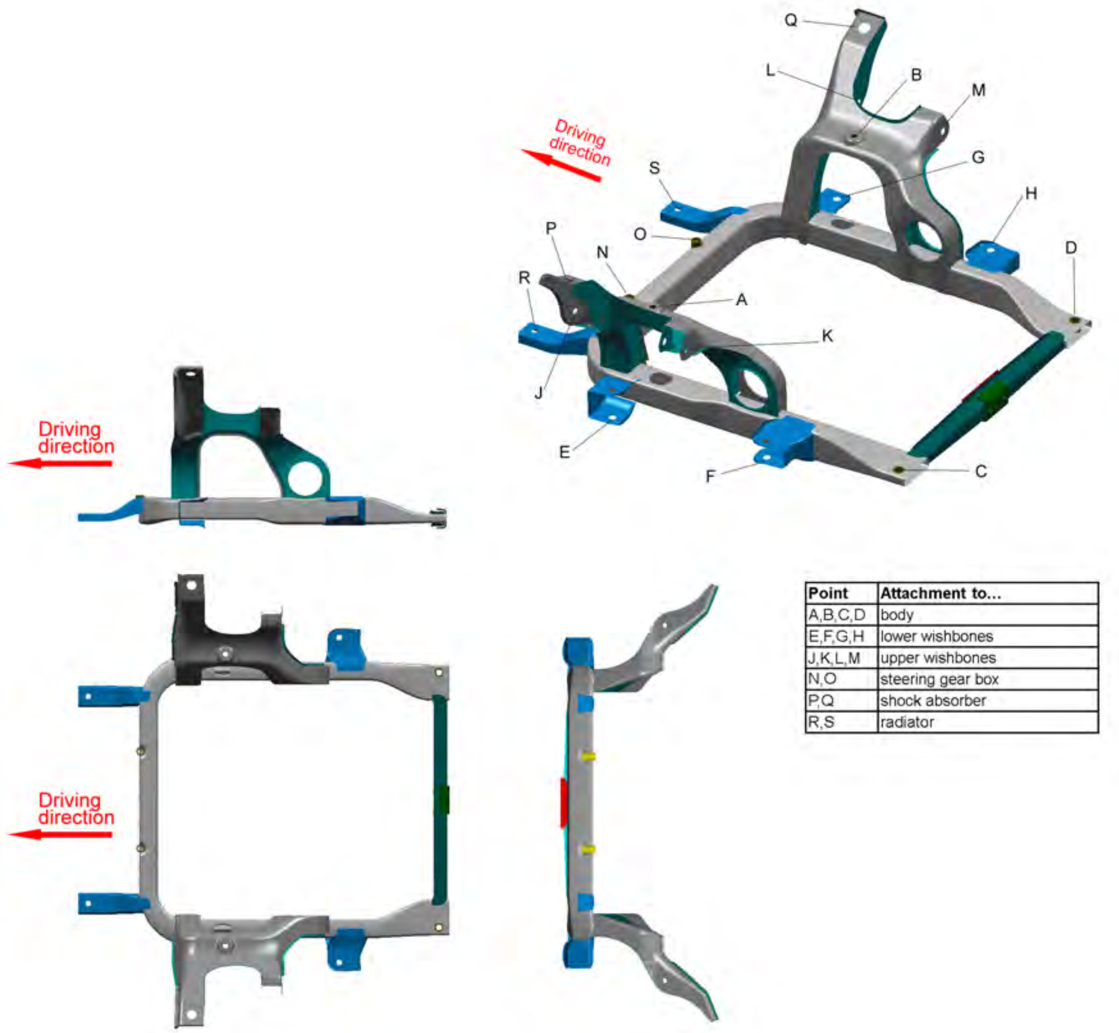


Figure 7.8.3-1 Subframe design layout

The subframe is manufactured from a hydroformed tube with a stamped steel cross member welded at the rear. This cross member functions as an attachment for the rear torque support of the engine mount. Two brackets for attachment of the cooling unit and two sleeves for attachment of the steering gearbox are welded to the front of the subframe. On each side, the support for the lower wishbone is made of two deep drawn steel brackets, which are welded to the tube. The transverse leaf spring clamp is attached by projection welding nuts inside of the tube through access holes in the upper side of the tube. To attach the upper wishbone, shock absorber and engine mount, a console made of two steel stampings is welded onto the tube.

To attach the complete subframe to the body structure, four (4) attachment points (A,B,C,D), two (2) located in the console and two (2) located in the rear end of the hydroformed tube, are defined. The attachment points of the subframe to the body structure and the attachment points of the suspension components are shown in Figure 7.8.3-2.



Point	Attachment to...
A,B,C,D	body
E,F,G,H	lower wishbones
J,K,L,M	upper wishbones
N,O	steering gear box
P,Q	shock absorber
R,S	radiator

Figure 7.8.3-2 Subframe attachment points

7.8.3.1. FEM-Calculation of the Subframe

The FEM-calculations of the subframe were made by superposing the suspension loads, as well as the engine loads. The results for the most significant load cases are shown in Figure 7.8.3.1-1 through Figure 7.8.3.1-5 and predict only small areas with higher stress levels. One of these areas is the front attachment point of the console to the body and the front attachment of the console to the tube for the load case "vertical push." The other area is the rear cross member for the load case "acceleration." The stress level for the load case "vertical push," shown in A, is caused by the forces to the attachment of the shock absorber. It can be reduced by designing a smoother transition from the attachment point to the structure of the console. The stress level in B would also be reduced by redesigning this area to the structure of the console. For the load case "acceleration," the stress level can be reduced by optimizing the transition from the rear engine mount to the crossmember. These design changes could be introduced in a further development phase.

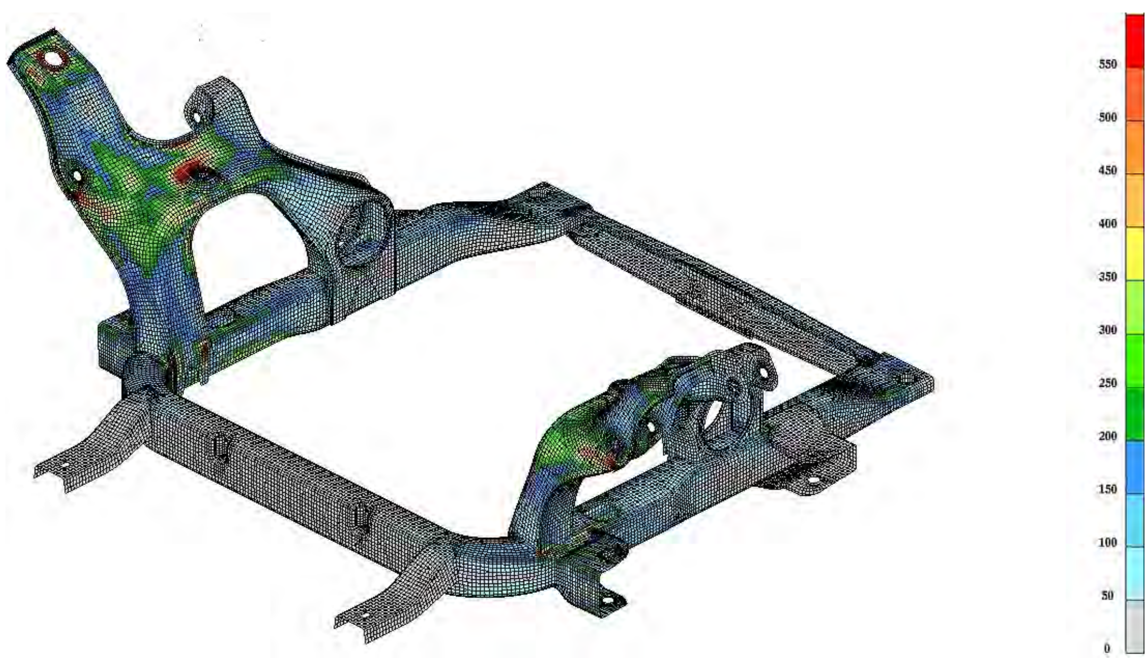


Figure 7.8.3.1-1 FEM-calculation subframe load case vertical push

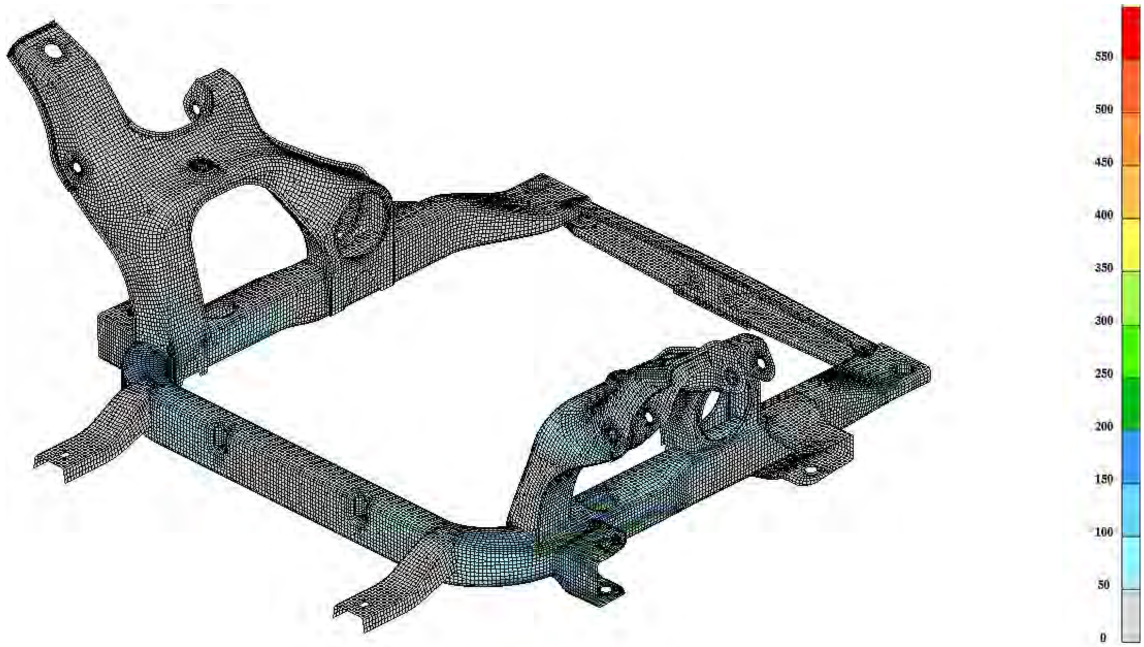


Figure 7.8.3.1-2 FEM-calculation subframe steady state cornering

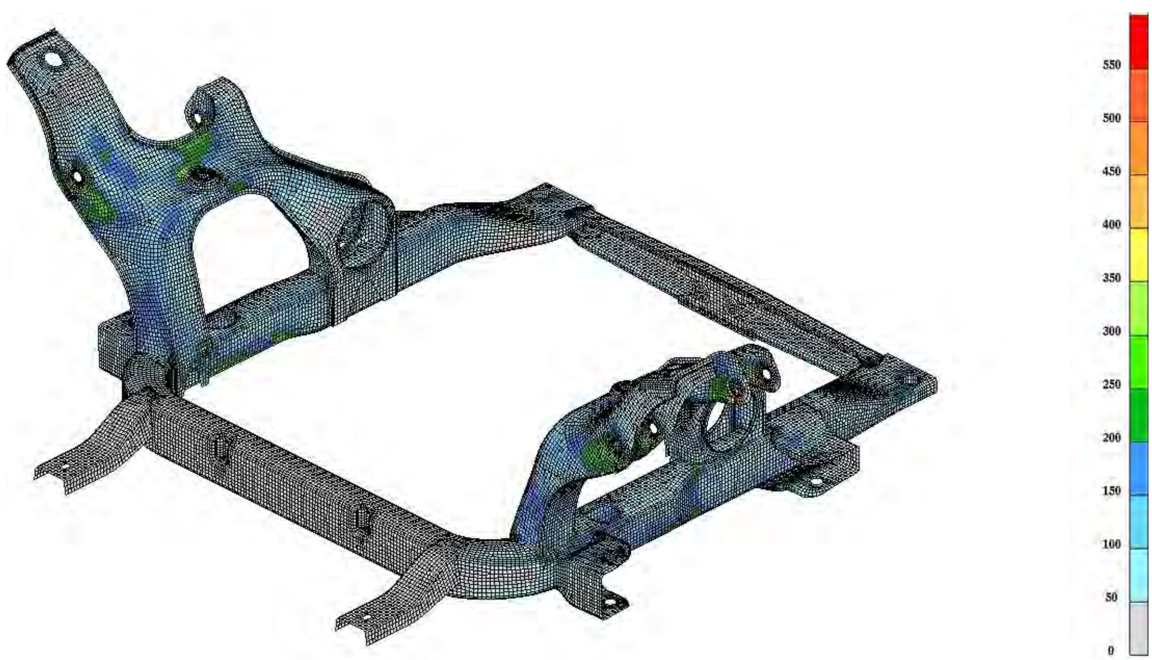


Figure 7.8.3.1-3 FEM-calculation subframe braking forward

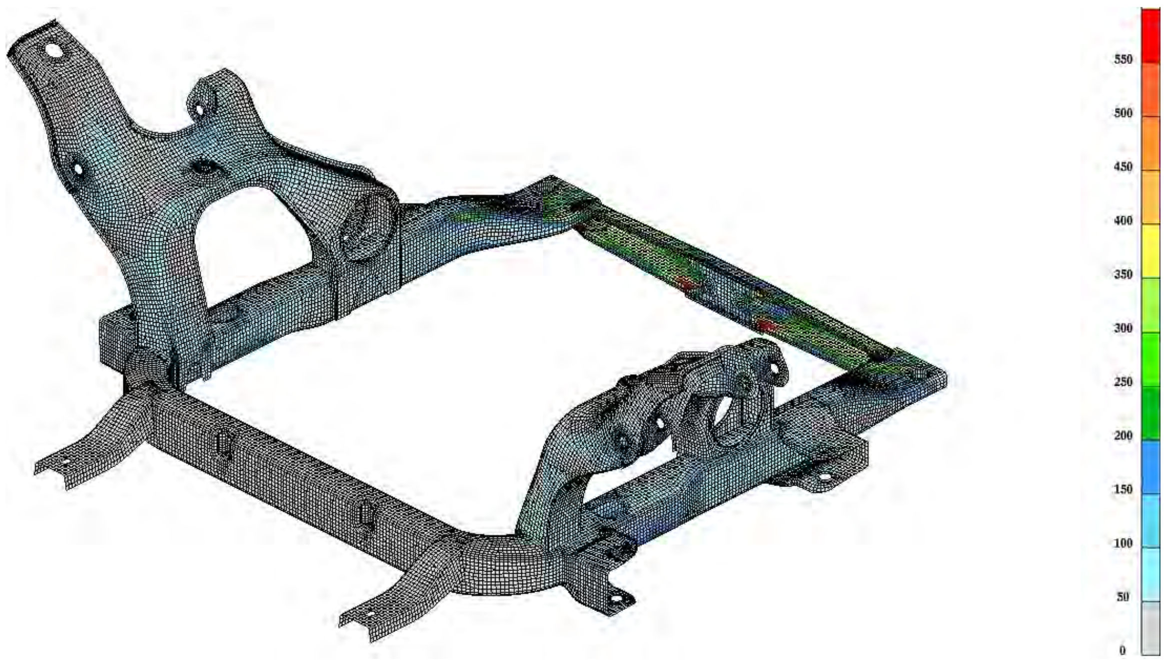


Figure 7.8.3.1-4 FEM-calculation subframe results maximum acceleration

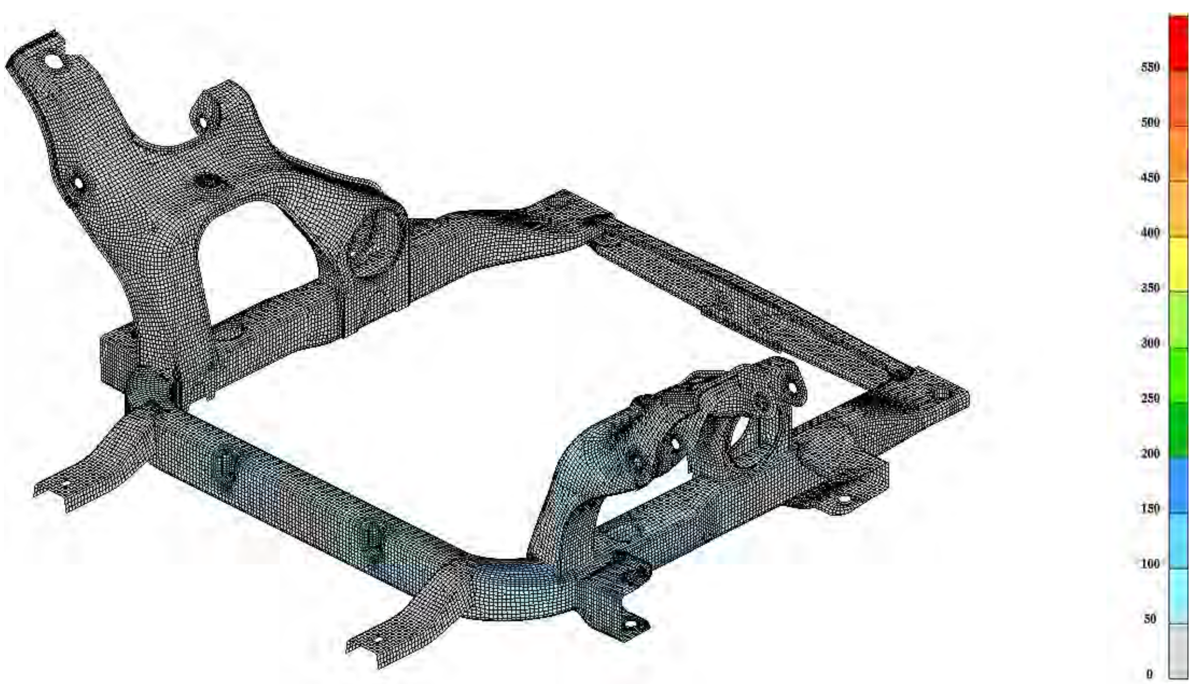


Figure 7.8.3.1-5 FEM-calculation subframe results curbstone push

7.8.4. Steering Knuckle Module

A steering knuckle transmits forces from the wheel to the wishbones. The steering knuckle is required to provide sufficient strength and stiffness to minimize deviation as much as possible from the kinematics characteristics in regards to toe-in change and camber change.

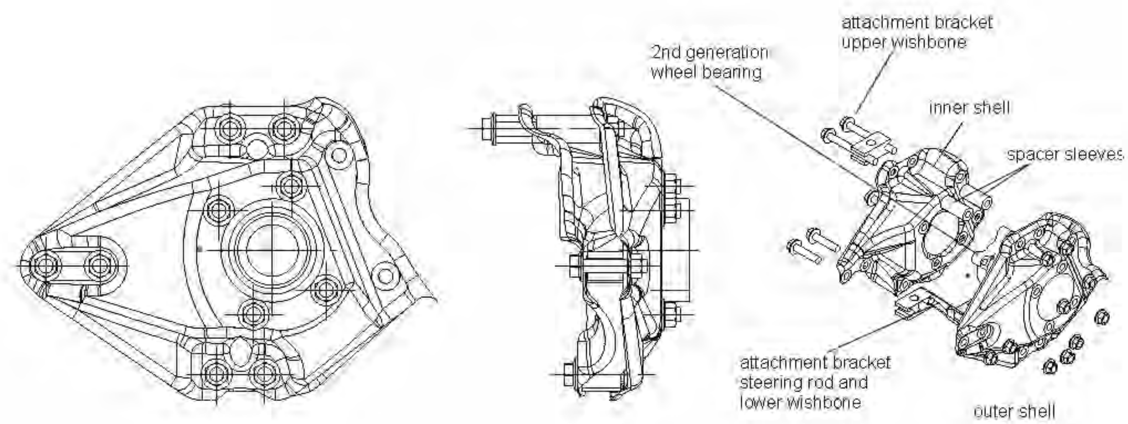


Figure 7.8.4-1 Steering knuckle module

A decision was made to design the steering knuckle from two (2) shells made of DP 350/600 with a material thickness of 3.0 mm for lighweighting potential. The steering knuckle module as shown in Figure 7.8.4-1 is assembled in the following sequence:

- The wheel bearing is assembled with 4 nuts and bolts at the outer ring between the two steering knuckle shell parts, therefore, for assembly reasons a 2nd generation wheel bearing is needed (see Figure 7.8.5-1).
- The brake caliper is attached between two steering knuckle parts with two nuts and bolts.
- The spacer sleeves in the upper area of the steering knuckle are attached between the two steering knuckle parts. The mounting bracket for attachment of the upper wishbone is bolted through the spacer to the outer shell of the steering knuckle.

- The mounting bracket for the attachment of the lower wishbone and the steering rod is manufactured as a forged steel part. This part, with its relatively high mass, is needed for strength in the load "curbstone push".

Several components, including the wheel bearing, spacer sleeves, mounting attachment for the upper wishbone, wheel hub and brake disc, are identical for both left and right hand sides of the steering knuckle module. The steel steering knuckle parts, the brake caliper and the attachment bracket for the steering rod and the lower wishbone, are mirror-imaged parts from left to right.

7.8.4.1. FEM-Calculation Steering Knuckle Version 1

The initial FEM-calculation steering knuckle outer and inner shell (variation 1) using the resulting forces from the considered load cases (see Section 7.8.2 and Appendix - Section 4.3) are shown in Figure 7.8.4.1-1 for the outer steel shell and in Figure 7.8.4.1-2 for the inner steel shell parts. It is important to recognize that the red colored areas around the points of constraint (location of connecting bolts) do not represent real stresses, they result from the representation of these constraint points in the FE-model. They are not considered to be problem areas from a durability perspective.

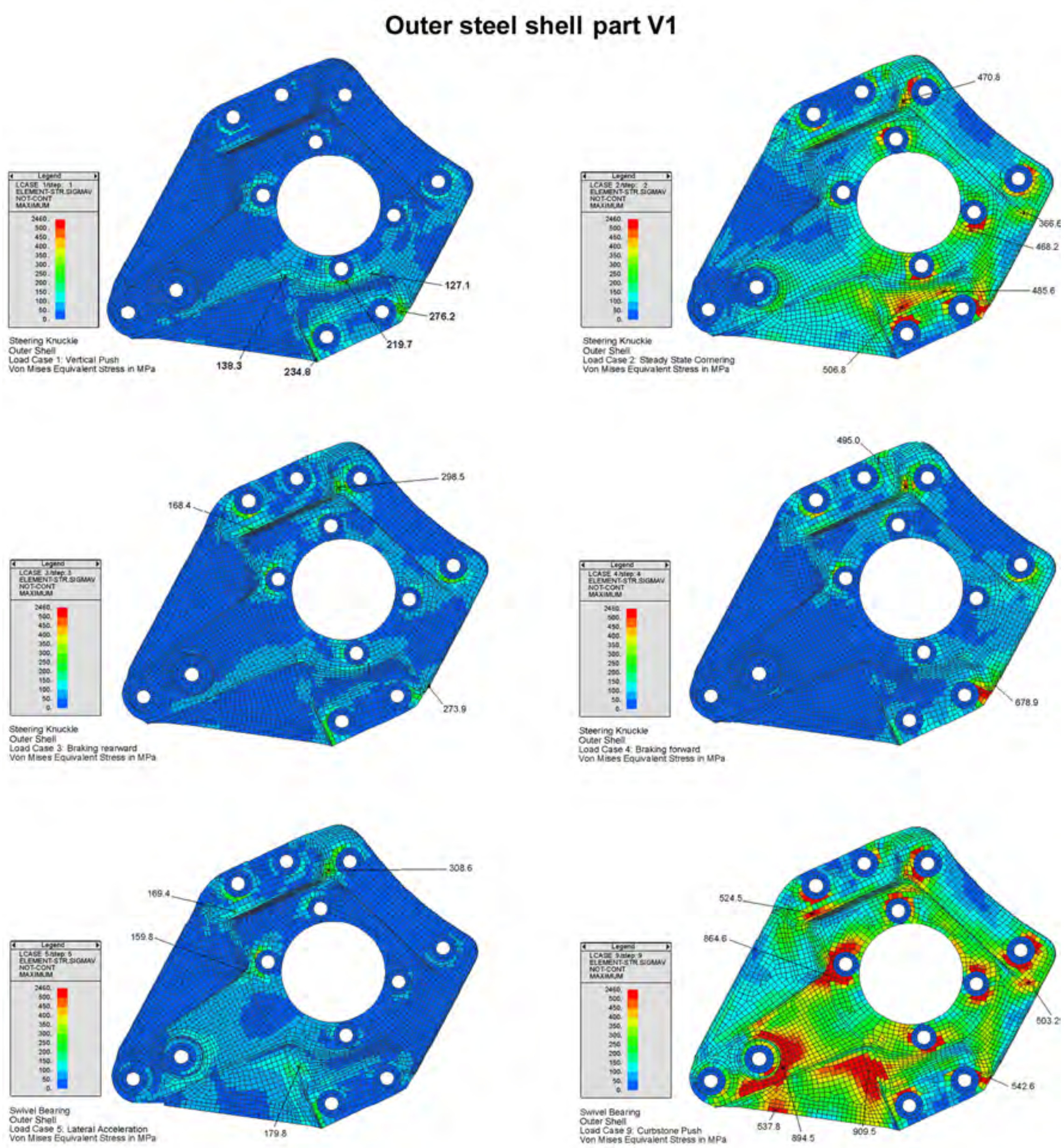


Figure 7.8.4.1-1 Steering knuckle outer sheet FEM-calculation version 1

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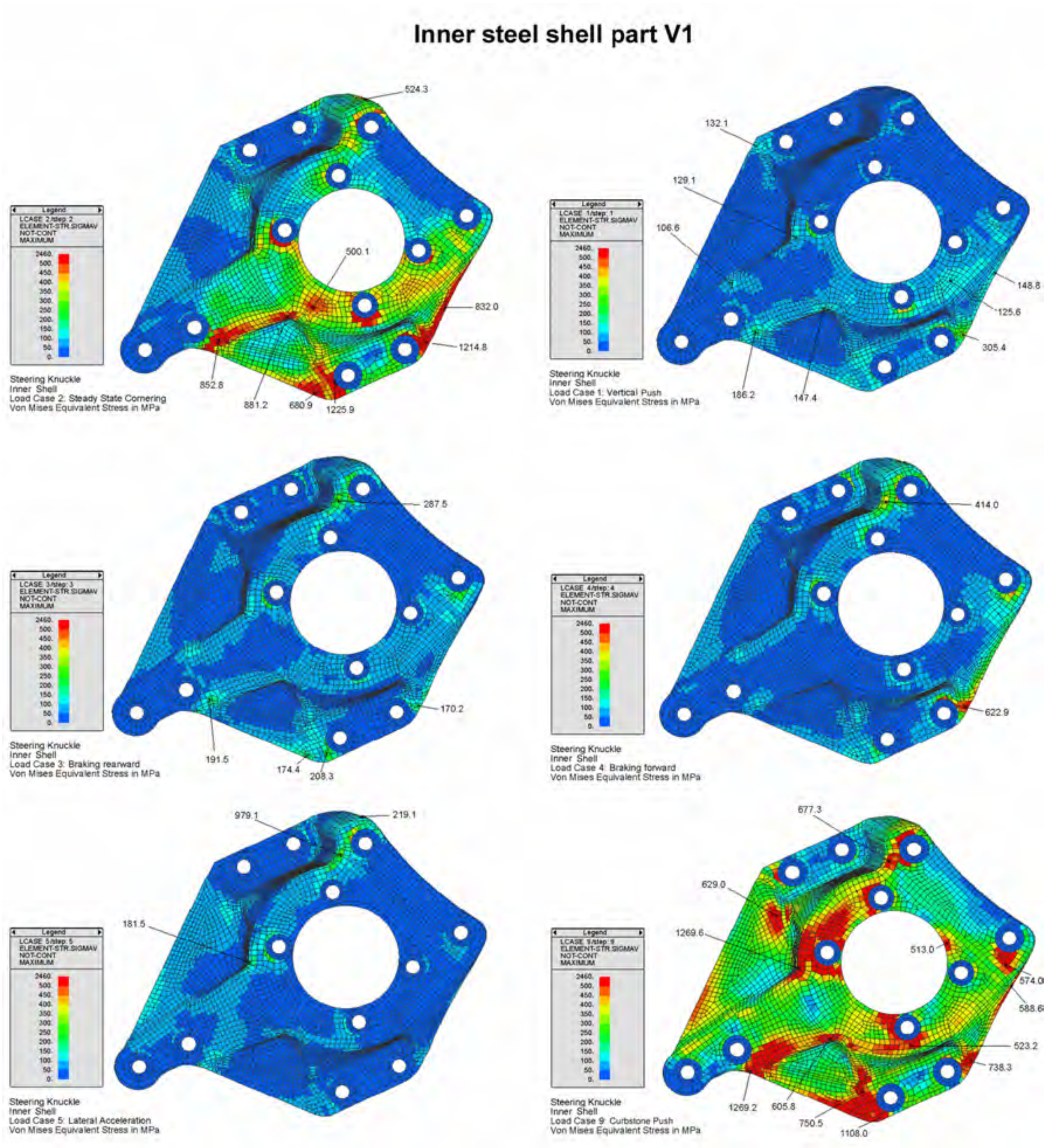

Figure 7.8.4.1-2 Steering knuckle inner sheet FEM-calculation version 1

Figure 7.8.4.1-3 shows a comparison of the steering knuckle inner shell of version 1 and a modification, version 2, which has been updated with design improvements in the area between the steering rod attachment mounting and the lower wishbone attachment mounting without any change of shape. This measure reduced stress in this area significantly.

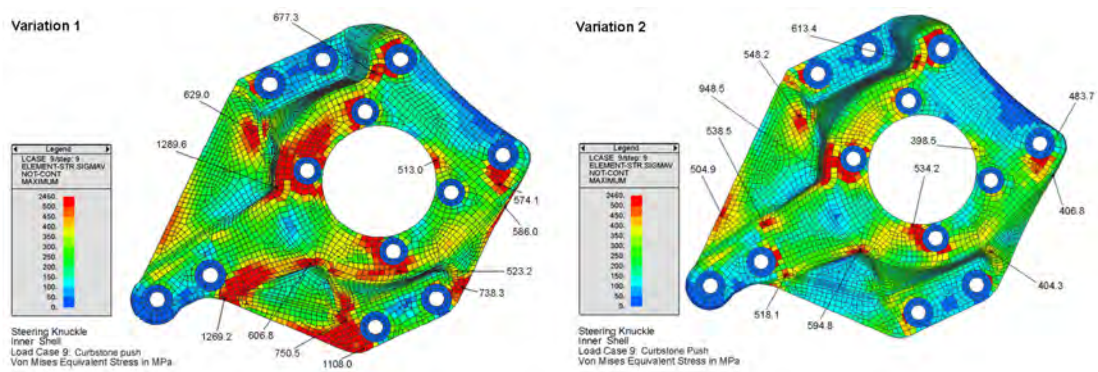

Figure 7.8.4.1-3 Comparison steering knuckle inner sheet FEM-calculation version 1 and 2

7.8.4.2. FEM-Calculation Steering Knuckle Final Design

For the design optimization of the steering knuckle in the various design stages, forming simulations (one-step and incremental) were performed by ULSAB-AVC Consortium member companies. As a result of these forming simulations , several changes were made to the steering knuckle inner and outer shell designs. The FEM results of the final design are shown in Figure 7.8.4.3-1 for the two critical load cases, "steady state cornering" and "curbstone push" for version 4 of both parts.

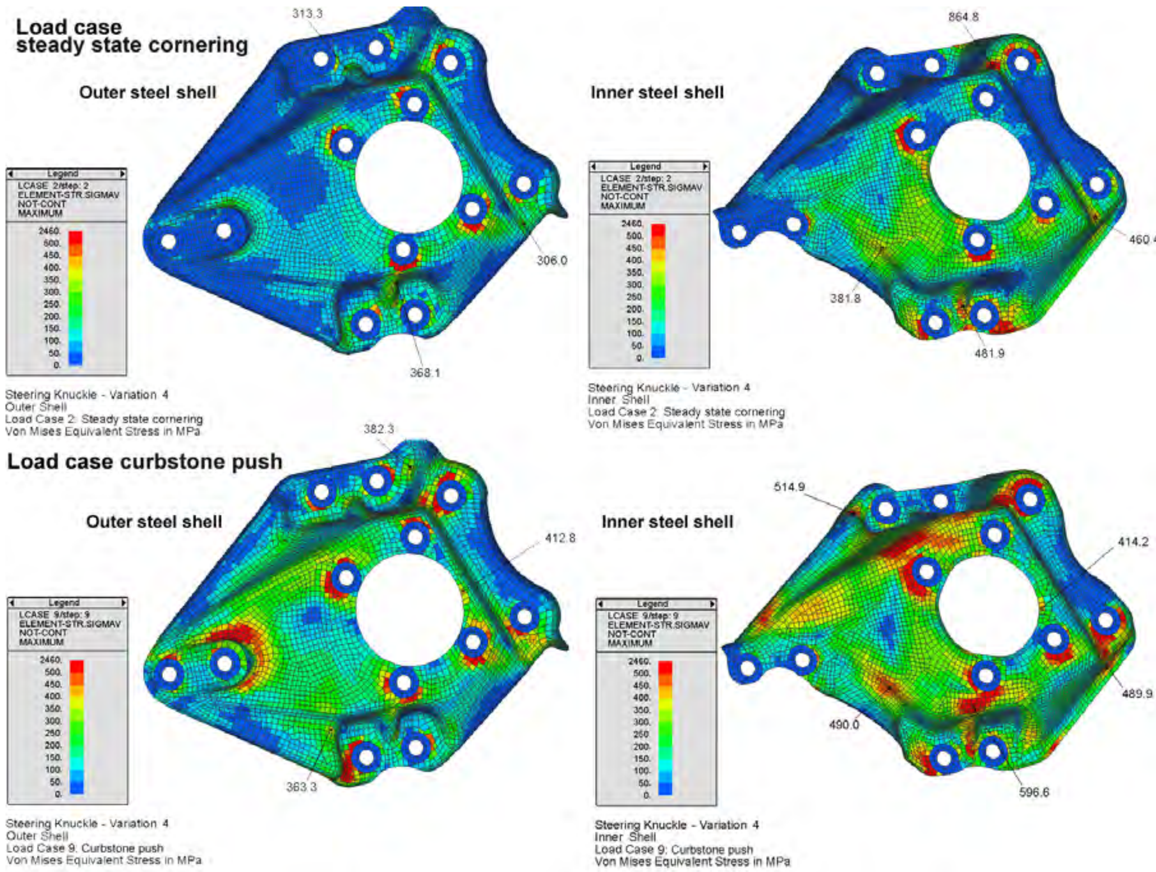

Figure 7.8.4.2 -1 FEM-calculation steering knuckle outer shell version 4

Figure 7.8.4.2-2 shows a proposal for additional beads, which can be added on the steering knuckle inner and outer sheets in a detail design phase and would strengthen the parts and lower the stress level in the high loaded areas, if necessary.

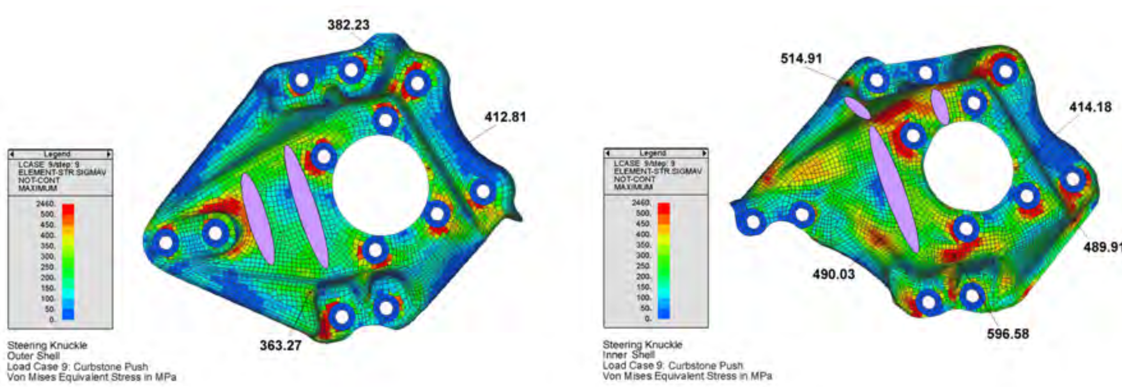

Figure 7.8.4.2-2 Measures for stress optimization

7.8.5. Wheel Bearing and Hub

The wheel bearing as shown in Figure 7.8.5-1 encompasses a double row annular ball bearing with the following dimensions:

Outer diameter	72 mm
Inner diameter	36 mm
Width	39 mm

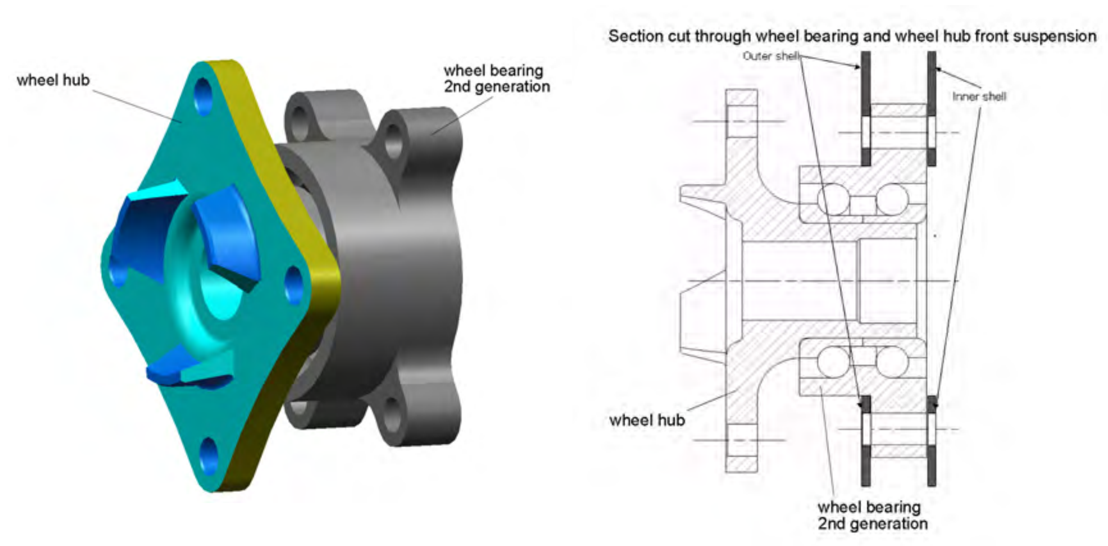

Figure 7.8.5-1 Wheel Bearing and Hub

The bearing has multiple sealing with lifetime greasing. At the outer ring of the bearing are four eyes distributed on the outer circumference. The final shape has to be committed after some tests regarding the camber stiffness of the axle. The wheel bearing is bolted between the steering knuckle outer and inner sheets at these locations.

The wheel hub (Figure 7.8.5-2) is manufactured out of 42 Cr Mo 4 with a tensile strength of 900 MPa. It is pressed into the inner ring of the wheel bearing and will be attached with a nut to the drive shaft. A very important point for durability of these parts is their temperature resistance, which influences the sealing of the wheel bearing, as well as the dimensional stability of the hub, because of the potential problems caused by the brake system such as squeezing and wobbling. The standard wheel hub features a round surface for the wheel attachment. For mass reduction, this shape has been reduced to a minimum and has to be confirmed or optimized (adjusted) in conjunction with tests of the axle regarding camber stiffness. The manufacturing of the wheel hub needs to be investigated in a further development phase.

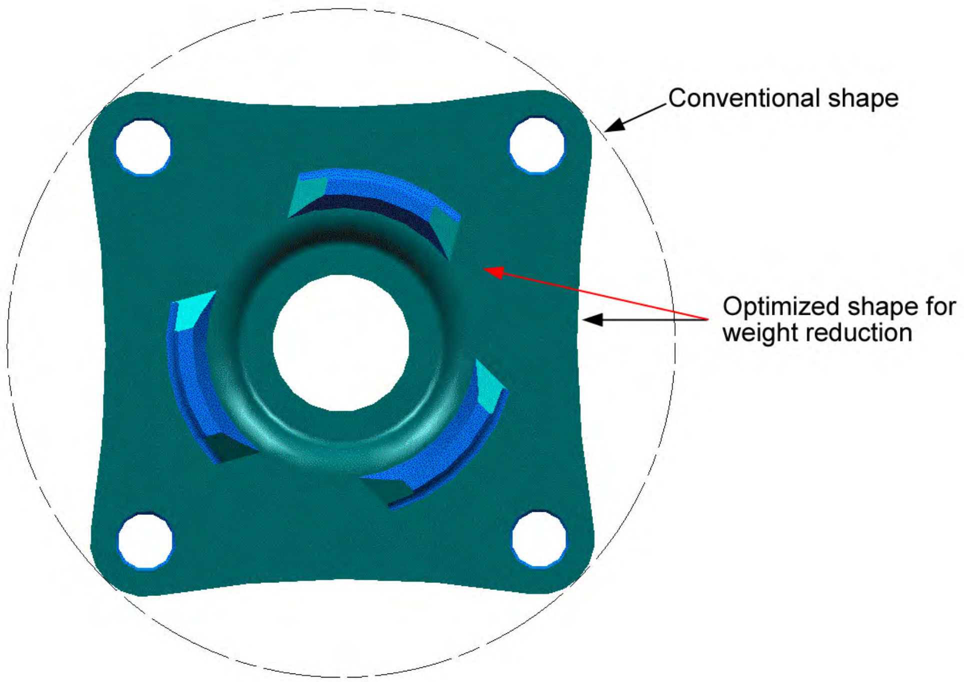

Figure 7.8.5-2 Wheel Hub

7.8.6. Lower Wishbone

7.8.6.1. Lower Wishbone Assembly

The lower wishbone assembly is shown in Figure 7.8.6.1-1. It's made of two tailor welded blank steel sheet stampings, with both halves, upper and lower, being mirror images of each other. The proposed joining process for the two halves is butt-welding, respectively plasma welding. The welded wishbone is identical for both left and right-hand side. The destination for the left and right-hand side is completed in the assembly with the rubber bushing and the inner and outer ball joints and the bracket for the leaf spring.

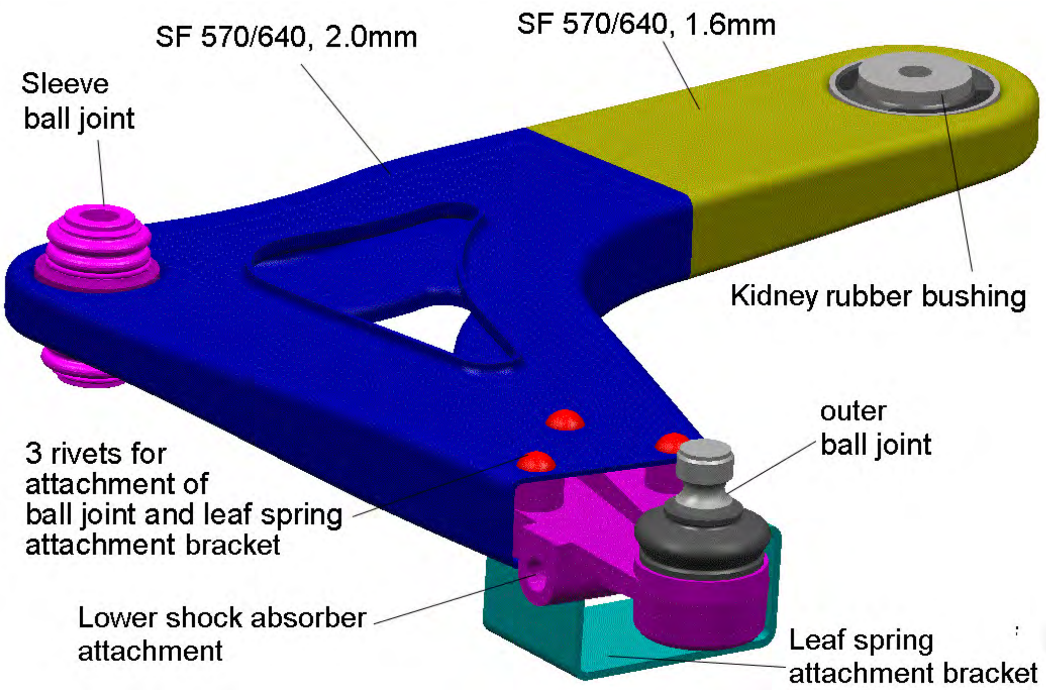

Figure 7.8.6.1-1 Lower wishbone assembly

The same inner ball joint, the same kidney rubber bushing and the same bracket for the leaf spring are used for both the left and right hand lower wishbone assemblies. The outer ball joint is also designed for use on both sides of the vehicle. With this approach, the parts costs and tooling investment costs of the lower wishbone could be reduced.

7.8.6.2. Lower Wishbone Design

A FEM calculation of the lower wishbone was performed using material thickness of 2.0 mm for both the upper and lower half of the wishbone. Figure 7.8.6.2-1 shows the results of the FEM-calculation with higher stress levels evident on the front side of wishbone from the outer and inner ball joint toward the rear kidney rubber bushing, mainly in the outside radii area. The cut out of the sheet has the shape of a collar. With this shape the sheet gets a good stiffness and bulging can be avoided

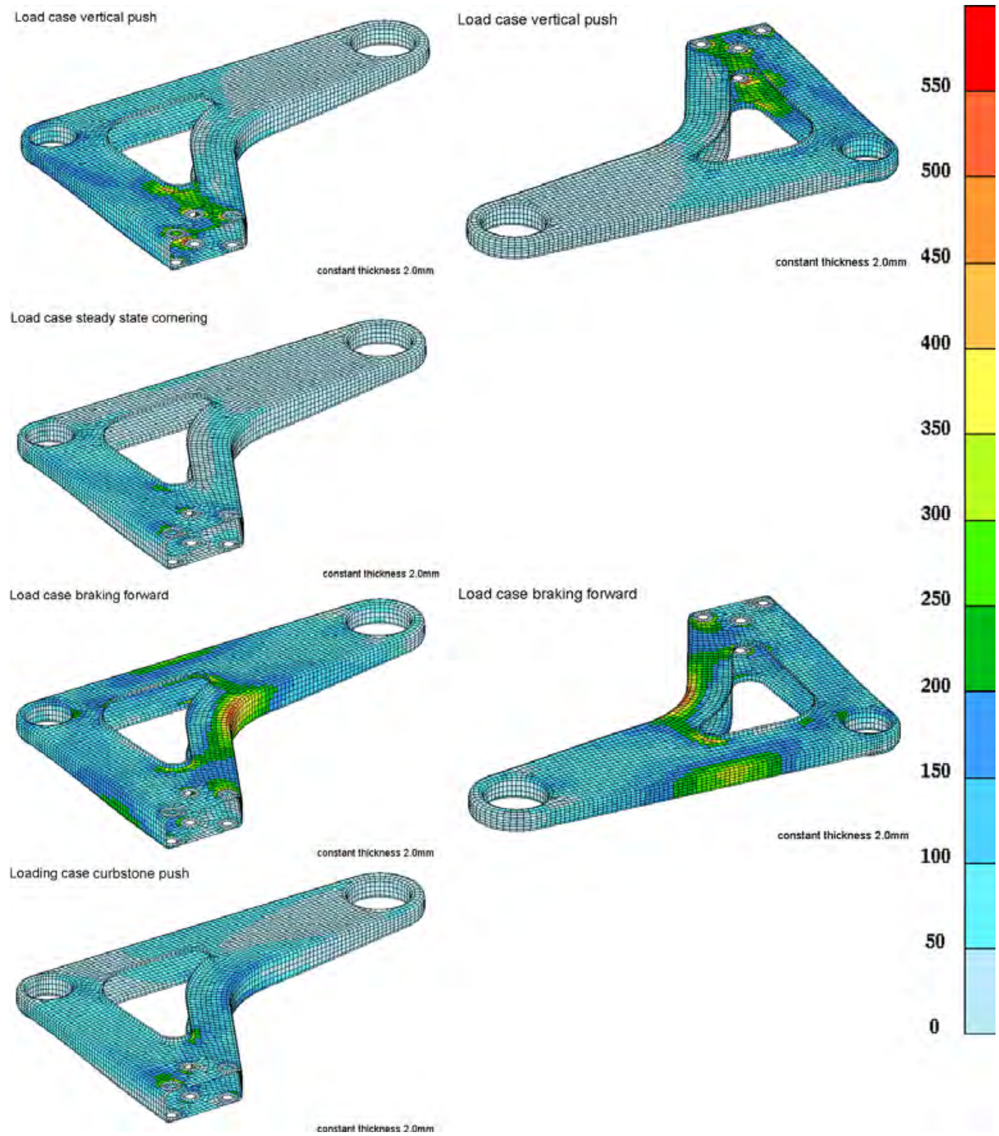

Figure 7.8.6.2-1 FEM-calculation of lower wishbone

As a result of the FEM-calculation, it was decided to manufacture the lower wishbone, upper and lower parts, as tailor welded blanks utilizing a material thickness of 2.0 mm for the higher stress areas and 1.6 mm for the lower stress areas.

Initial forming simulations performed by ULSAB-AVC Consortium member companies showed failure on the flanges of the holes of the inner ball joint and kidney rubber bushing locations with the material DP 350/600. Additional forming simulation with Stretch Flangeable material (SF 570/640) for enhanced stretch flanging performance predicted part manufacturing feasibility.

7.8.6.3. Outer Ball Joint

The outer ball joint is designed to be symmetrical so that it can be used for the left or the right side of the vehicle. The ball has a diameter of 30 mm. For mass reduction considerations, the ball pin is designed as a hollow part and requires a non-cutting manufacturing process, such as the spin forming manufacturing process. The location of the outer ball joint in the steering knuckle to lower wishbone assembly is shown in Figure 7.8.6.3-1. A section cut of the ball pin is shown in Figure 7.8.6.3-2.

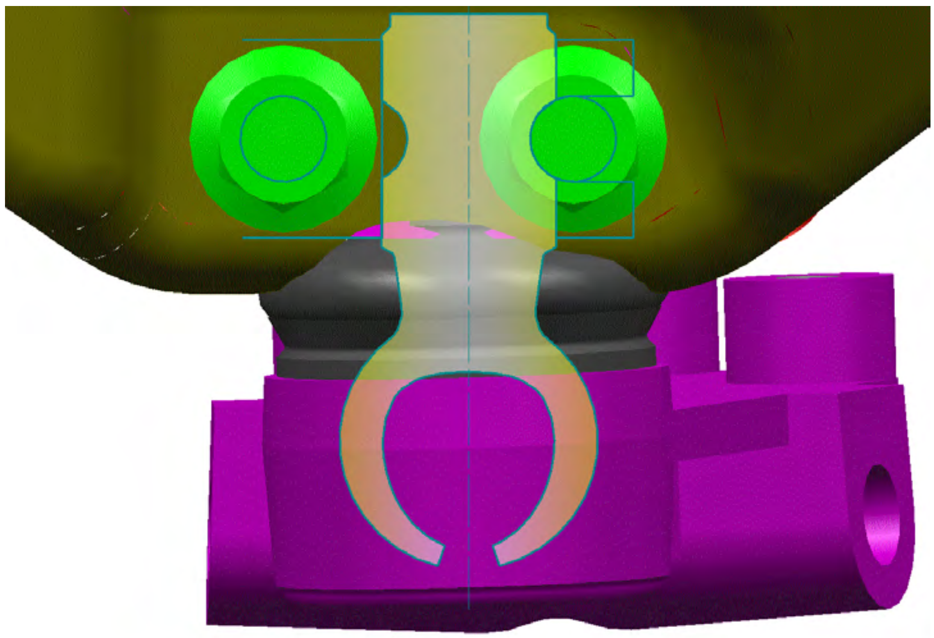

Figure 7.8.6.3-1 Location of outer ball joint in knuckle to lower wishbone assembly

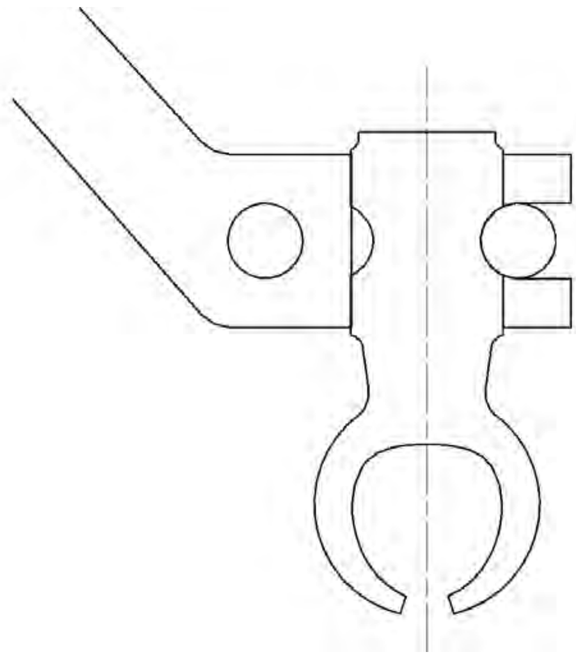

Figure 7.8.6.3-2 Section cut - hollow ball pin outer ball joint

The outer ball joint has an annular range of about ±21° and lifetime grease lubrication. The outer ball joint has to prove its water-tightness in a salt spray test. The force to pull out the ball pin of the housing must be greater than 25kN, which is the force for testing including a safety factor of 2. The maximum force occurring at the ball pin is about 12 kN in z-direction. The ball pin is fixed in the mounting location by friction generated by the bolt force and by positive engagement. The friction of tilting and rotating must be minimized in a way without affecting the service lifetime.

7.8.6.4. Sleeve Ball Joint

The sleeve ball joint connecting the lower wishbone with the subframe, has an annular range of ±22° and lifetime grease lubrication. The sleeve ball joint has to prove its water-tightness in a salt spray test. The force to pull the sleeve ball out of the housing must be greater than 15 kN. The friction during tipping, tilting and rotating the sleeve must be minimized in a way that does not affect the service lifetime.

7.8.6.5. Kidney Rubber Bushing

The kidney rubber bushing has a compliance in y-direction that influences mainly longitudinal compliance of the suspension. For compliance optimization, the diameter, Shore hardness, loss angle and the clearance of the rubber bushings may be varied to a certain degree. For durability reasons, the loss angle, which is a unit of measurement for the damping of the rubber bushings, of the kidney rubber bushing must be as great as possible.

7.8.7. Upper Wishbone

7.8.7.1. Upper Wishbone Assembly

The upper wishbone assembly (see Figure 7.8.7.1-1) is made of the upper wishbone stamping, the front and rear rubber bushing and the ball joint. The front and rear bushings and the ball joints are identical for left and right hand side of the upper wishbone assembly.

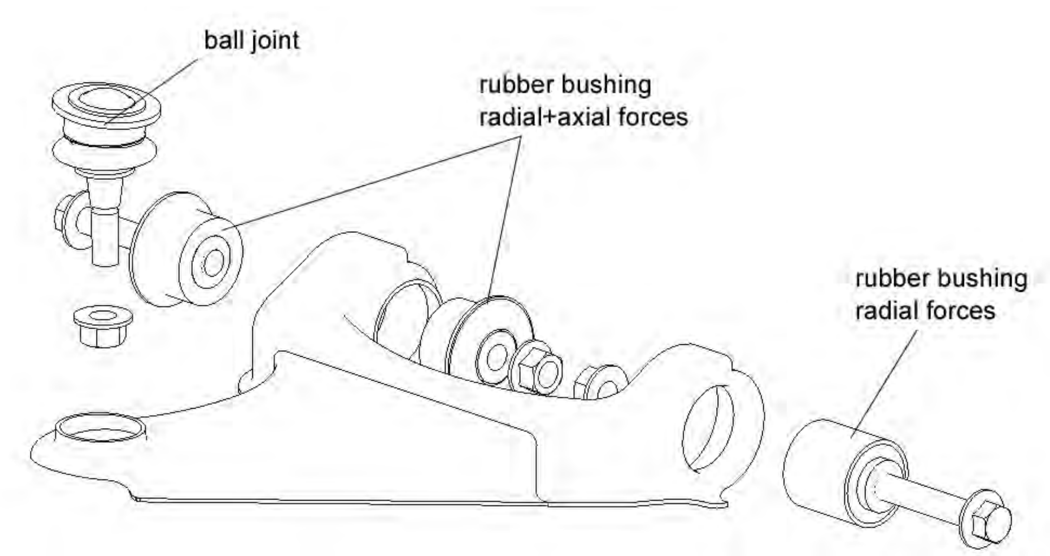

Figure 7.8.7.1-1 Upper wishbone assembly

7.8.7.2. Upper Wishbone Design

The upper wishbone is designed as a single stamped tailor welded blank sheet steel part. The material thicknesses of the blanks are 1.6 mm and 2.0 mm.

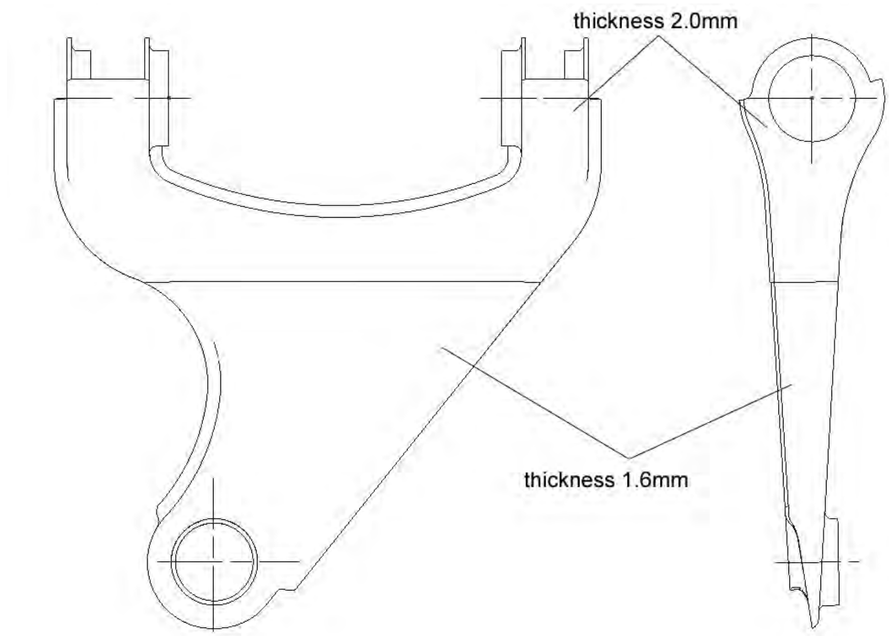

Figure 7.8.7.2-1 Upper wishbone

The results of a FEM-calculation are shown in figure 7.8.7.2-2, with only one critical area with a stress level of approximately 600 MPa, especially in load case braking forward.

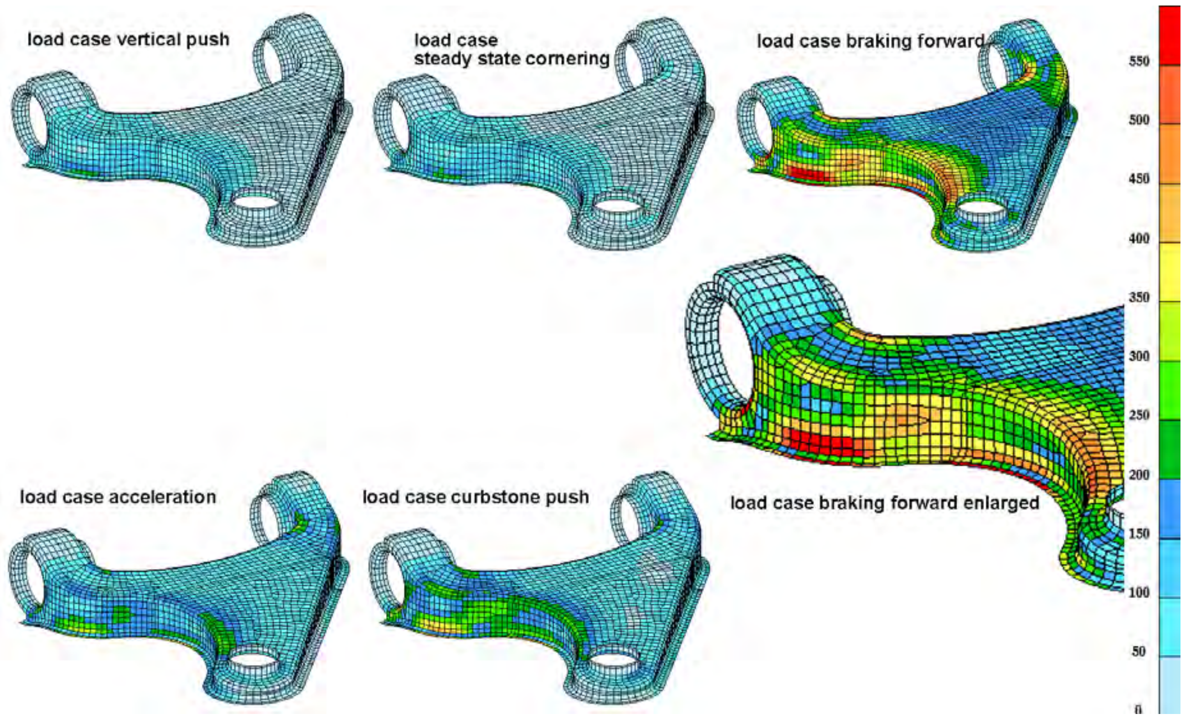

Figure 7.8.7.2-2 FEM-calculation of upper wishbone

Forming simulations were performed by ULSAB-AVC Consortium member companies. The first forming simulation with the material DP 350/600, the forming simulations show critical areas of fracture therefore, Stretch Flangeable material SF 570/640 was used to overcome this problem. In a detail design phase, the design should be optimized in the critical areas as shown in the FEM-calculations, which were identical to the critical areas as predicted in the forming simulation results.

7.8.7.3. Rubber Bushings

The front and rear rubber bushings are identical for the left and right side wishbones. The front rubber bushing is made of two identical parts and supports the longitudinal and lateral forces. The rear rubber bushing supports the lateral forces of the upper wishbone.

7.8.7.4. Outer Ball Joint

The outer ball joint has a ball pin with a diameter of 23 mm. For mass reduction reasons, the ball pin is designed as a hollow part. For the manufacturing process of the ball pin, the same considerations apply as previously mentioned in Section 7.8.6.3.

7.8.8. Leaf Spring

For ULSAB-AVC, the transverse leaf spring (see Figure 7.8.8-1) has two functions. First, it functions as a spring and second, it functions as a stabilizer. Using this approach allows reduction in cost and mass. Two alternatives were considered in the beginning of the ULSAB-AVC program.

- Leaf spring made of steel
- Leaf spring made of fiberglass reinforced plastic

Although forged steel leaf springs are used in commercial vehicles and trucks, normally in longitudinal direction, the parts designed for ULSAB-AVC are small in comparison and this, together with the specific compliance characteristics and the transverse mounting position, presented a major challenge for the steel based design. Therefore, for the final selection, a fiberglass reinforced plastic leaf spring was preferred.

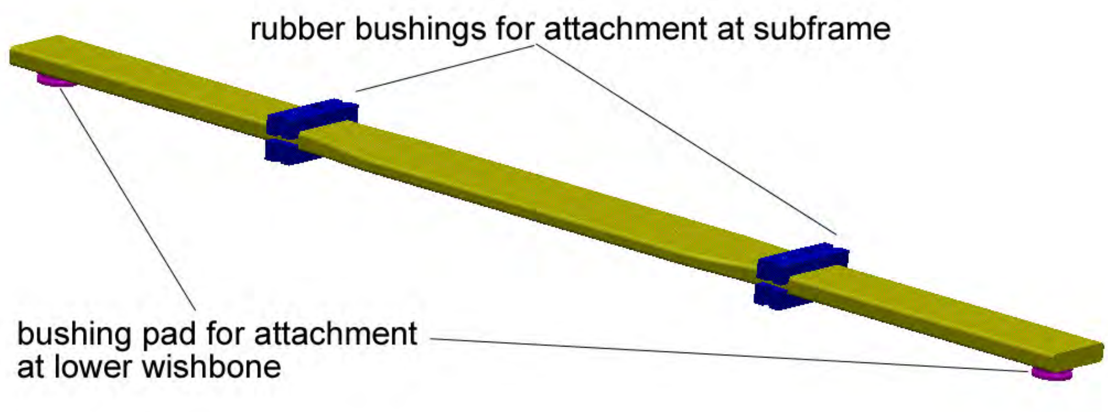

Figure 7.8.8-1 Transverse leaf spring

Because of the difference in vehicle mass for the ULSAB-AVC C-Class and PNGV-Class vehicles, two different leaf springs with different spring rates are needed.

The leaf spring is attached to the subframe with rubber bushings allowing the compensation of spring movement in transverse direction, with two brackets riveted to the lower wishbone. The leaf spring is put into the leaf spring bracket of the lower wishbone, where forces, resulting from movement of the lower wishbone caused by wheel travel, are transferred into the leaf spring. At the two ends of the leaf spring, rubber pads are bonded to the leaf spring. Relative movement, in transverse and in longitudinal direction, between the wishbone bracket and the leaf spring will be compensated by sliding the rubber pad relative to the bracket.

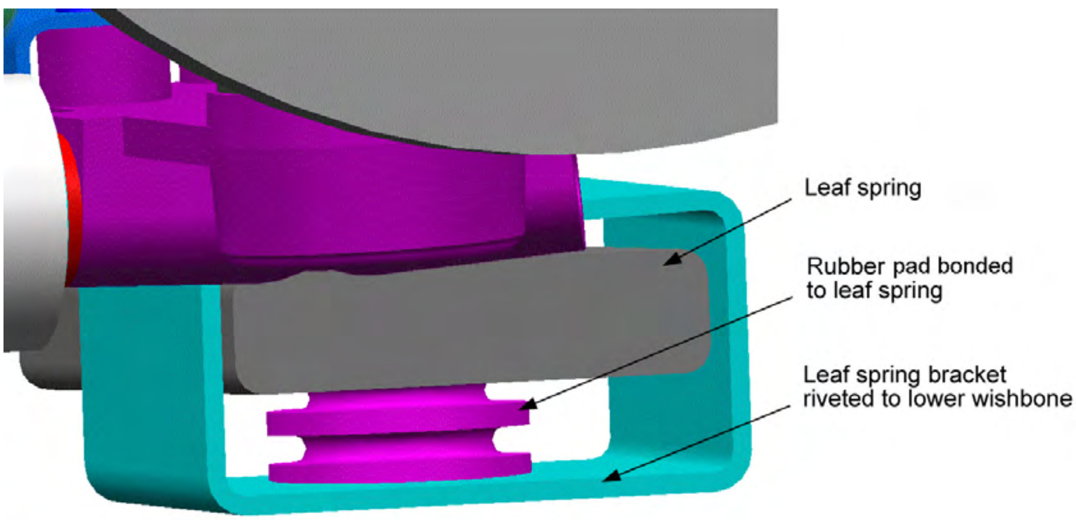

Figure 7.8.8-2 Leaf spring rubber pad in bracket

7.8.9. Shock Absorber Front Suspension

For each C-Class and PNGV-Class vehicle, with either diesel or gasoline engine variants, a different shock absorber with special characteristics is needed. The shock absorber is designed as a single-tube shock absorber (see Figure 7.8.9-1).

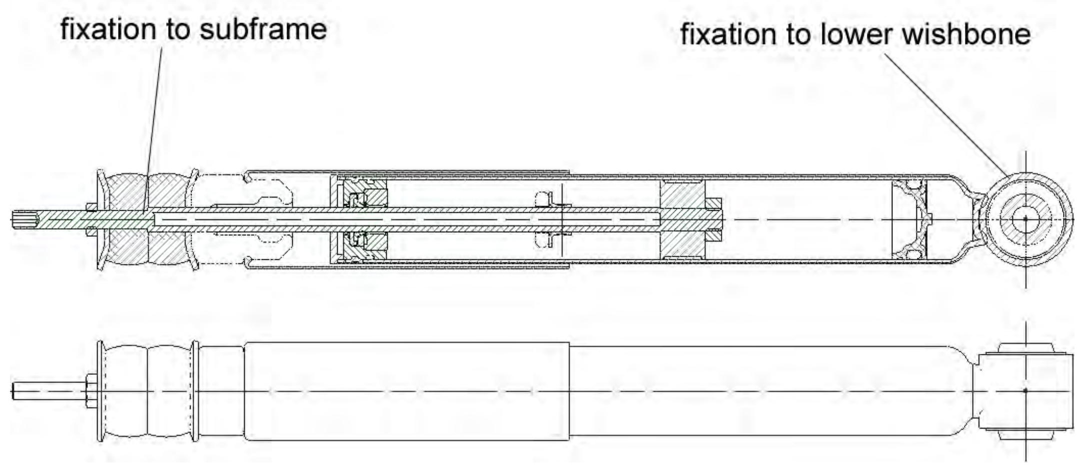

Figure 7.8.9-1 Shock absorber front

The wheel movement bump stop is integrated in the shock absorber. The shock absorber is attached to the subframe with a nut, washer and two rubber bushings at the threaded pin located at the end of the hollow piston rod. At the lower end shock absorber is bolted to the wishbone through a rubber bushing, which is pressed into a sleeve. The wall thickness of the shock absorber tube is minimized and a plastic coating is applied to protect the tube against damages caused by stone chipping. The shock absorber must function in a temperature range from -40°C to 160°C.

7.9. Steering System

7.9.1. Steering Gear

In the early stages of development, the concept selected for the steering gear was a conventional hydraulic power assisted rack and pinion steering system because such a system has the largest potential for mass reduction. However, this requires a hydraulic pump, which would only be necessary in the vehicle for the steering system. To eliminate the need for a hydraulic pump, it was decided to utilize a power steering system with electrical power assistance. The advantage of an electrical power steering system is mainly the reduction of fuel consumption. Additionally, power assistance can easily be made speed sensitive compared to a hydraulic system. The tie rods are part of the steering gearbox module (see Figure 7.9.1-1) with a length of 305.7 mm for both tie rods. For the connections of the tie rods to the steering gearbox, an axial ball joint with a diameter of 25 mm is used. A 22 mm diameter axial ball joint is used to attach the steering rods to the steering knuckle. For mass reduction, both ball joint pins are designed hollow.

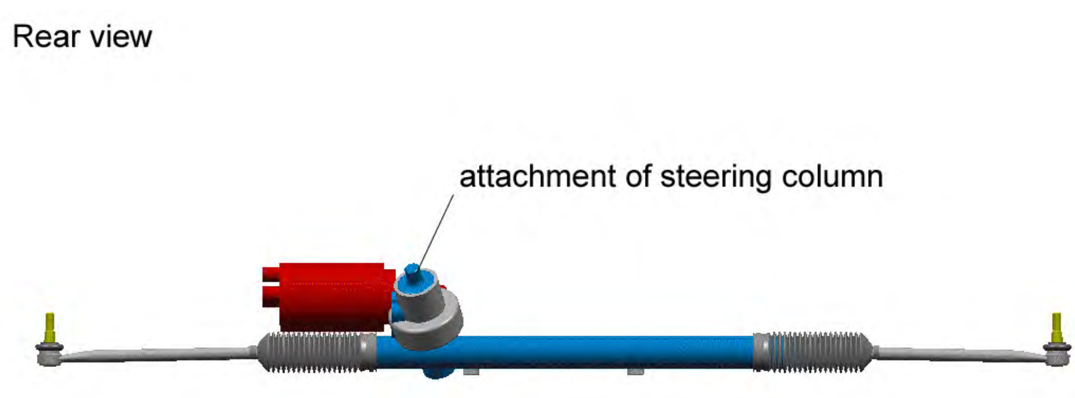

Figure 7.9-1 Steering gear module with electrical power steering

7.9.2. Steering Column Concept

The vehicle concept features fixed front seats for improved vehicle side impact crashworthiness utilizing an adjustable pedal system. Based on the package studies, the steering column had to be designed to have a longitudinal adjustment of 120 mm and a height adjustment of 58 mm (see Figure 7.9.2-1). For mass reasons, it was decided to utilize a mechanical adjustment with a friction locking mechanism instead of an electrical adjustment with a positive engagement mechanism. Therefore, special attention has to be made for the synchronization of the air bag in a further development phase.

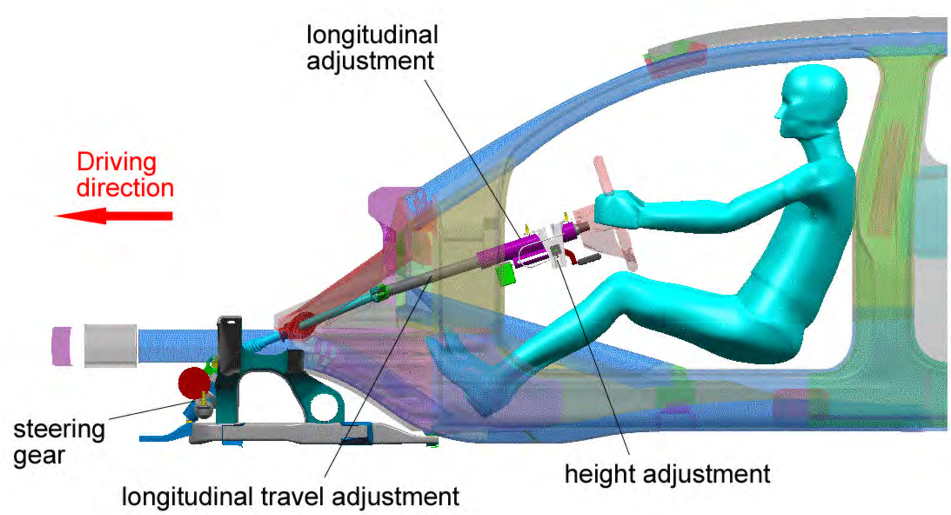

Figure 7.9.2-1 Steering column concept

The steering column with the friction locking mechanism and electrical steering column lock, as well as the spring for weight compensation is shown in Figure 7.9.2-2. The spring for weight compensation for height adjustment prevents the steering wheel from raising or lowering when the locking lever is released.

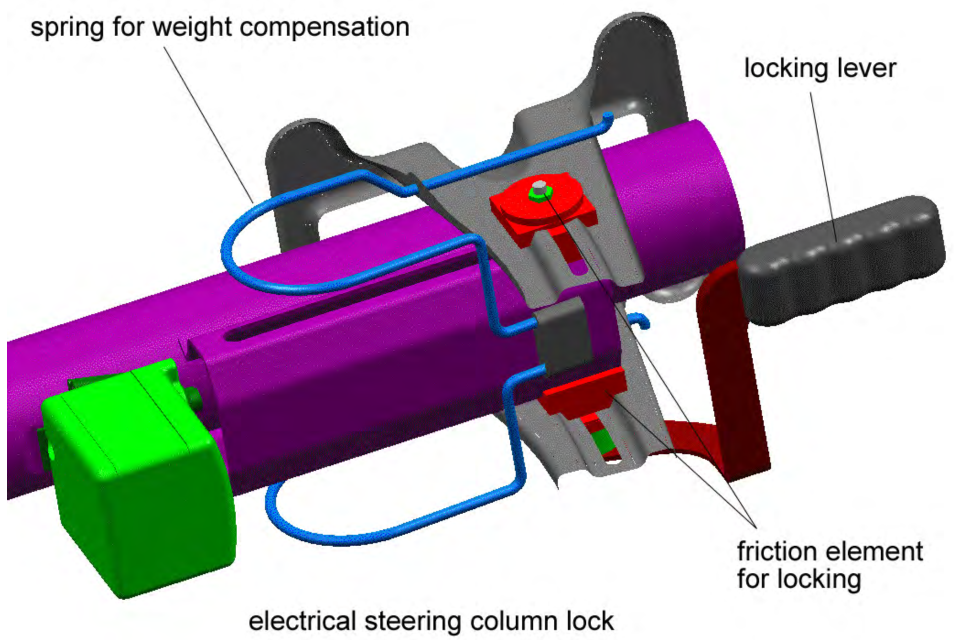

Figure 7.9.2-2 Steering column with the friction locking mechanism and electrical steering column lock

The assembly concept of the steering column is shown in Figure 7.9.2-3. The upper steering column is assembled from the passenger compartment with the lower steering column being attached to the body structure on the upper end and to the steering gear at its lower end.

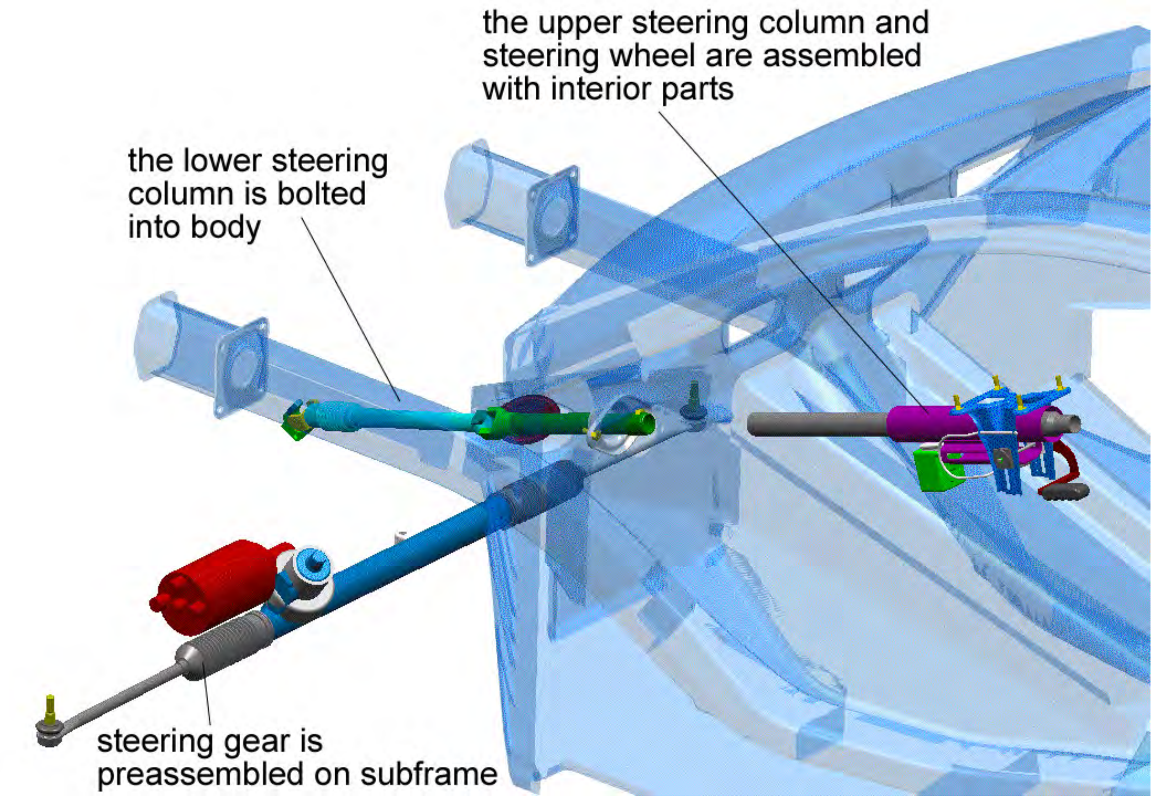

Figure 7.9.2-3 Assembly concept of steering column into vehicle

7.9.3. Steering Wheel Concept

The ULSAB-AVC vehicle concepts features a multi-functional steering wheel concept (see Figure 7.9.3-1) with buttons for the following controls:

- Turn signals
- Horn
- Wipers
- Light
- Cruise control
- Gear shifting

A highly reliable short distance telemetry system will transfer the data from the steering wheel to the corresponding receiver.

The advantage of combining all these functions in the steering wheel are:

- Elimination of gear shift on the tunnel or dash panel
  - Reduction of parts, mass, parts costs, tooling investment costs, assembly costs in trim line
- Elimination of levers for wiper, lights (high beams), signals, cruise control
  - Reduction of parts, mass, parts costs, tooling investment costs, assembly costs in trim line
- Elimination of ergonomic disadvantages for different sized drivers
  - Constant distance to function controls

The ergonomic aspects of such a steering wheel have not been analyzed in full in the framework of this study. Vehicles with multi-function steering wheels including push button shifting can already be found on the high-end sports car market segment with the visible trend of moving into the lower market segment. In addition to the ergonomic aspects, the crash behavior and the adaptation of the air bag module are not developed in this concept phase and would need to be further analyzed and the final design of the steering wheel may deviate from the styling shown here.

Figure 7.9.3-1 Steering wheel concept

7.10. Drive Shaft

7.10.1. Drive Shaft Concept

The drive shaft concept shown in Figure 7.10.1-1 is a conventional designed with the outer joint bolted to the wheel bearing/wheel hub and the inner joint bolted to the differential flange. The connection of the outer joint to the wheel hub depends very much on the philosophy of the car manufacturer.

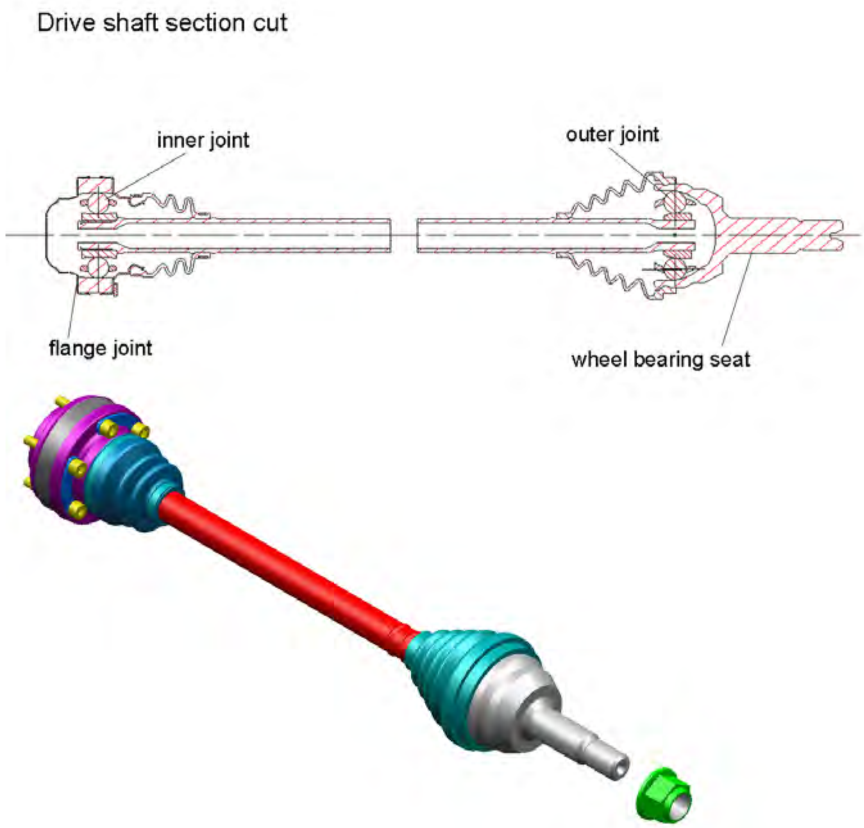

Figure 7.10.1-1 Drive shaft concept

For the ULSAB-AVC diesel and gasoline engine variants, two drive shaft designs with different joint size and tube diameters are needed as a result of different power and torque characteristics of both engine types.

7.11. Rear Suspension

7.11.1. Twist Beam Rear Suspension

Gathered benchmarking data showed the twist beam rear suspension concept (see Figure 7.11.1-1 and 7.11.1-2) to be one of the lightest among the possible alternative systems such as multi-link, double wishbone, de Dion and twist beam. The twist beam suspension was selected for the ULSAB-AVC vehicle concepts. Additionally, the twist beam rear suspension concept has proven its capabilities in respect to cornering handling and comfort in many classes of vehicles and is very popular in vehicles with similar total vehicle mass as the ULSAB-AVC vehicle concepts.

The twist beam rear suspension has to transfer the forces and torque from the wheel to the body. An important point is to suppress the acoustic stimulation of the body by inputs from the axle. The stiffness of the rubber bushing at the support to the body has to be adapted to the kinematics and elastokinematic behavior of the axle. That means that the over-steering by side forces must be counteracted with special rubber bushings as used in existing production twist beam rear suspensions.

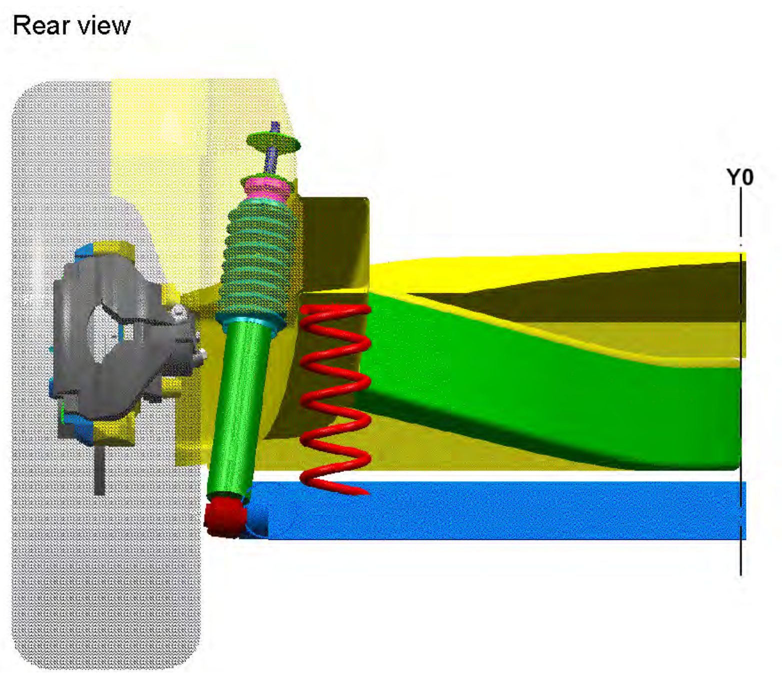

Figure 7.11.1-1 Twist beam rear suspension

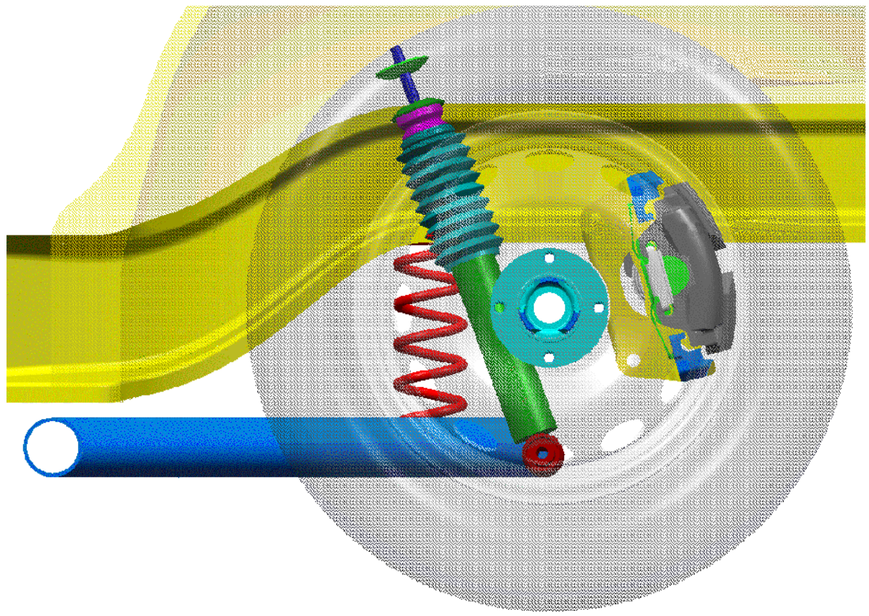

Figure 7.11.1-2 Twist beam rear suspension side view

The advantages of the twist beam rear suspension concept are reduced number of parts compared to an alternative multi-link concept (see Figure 7.11.1-3). Additionally, this design does not add mass to the body structure for brackets, cross-members and reinforcements, which are needed for other suspension concepts. Compared to double wishbone suspension concept, no additional sub-frame is needed for the attachment of the lower wishbones. Other advantages of the twist beam rear suspension are the facts that it can be pre-assembled off line as one (1) unit and the final assembly to the vehicle (body) requires only four (4) attachment points (two (2) for the shock absorbers and two (2) on the trailing arms).

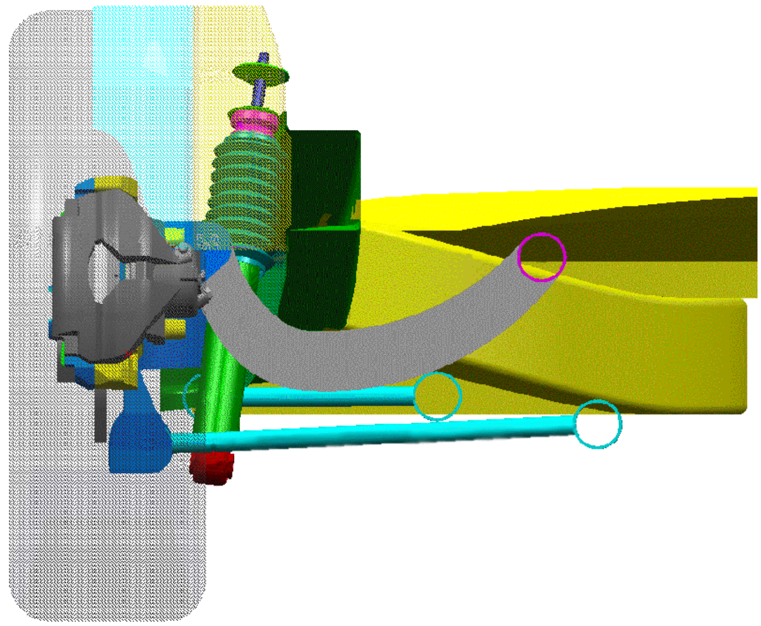

Figure 7.11.1-3 Multi-link rear suspension system

7.11.2. Rear Suspension Module

The pre-assembled rear suspension module, as delivered to the final assembly line, is shown in Figure 7.11.2-1, including the electrical parking brake module and the bushings for the attachment to the body. In the final assembly, the springs are placed between the longitudinal rail of the body structure and the spring cup.

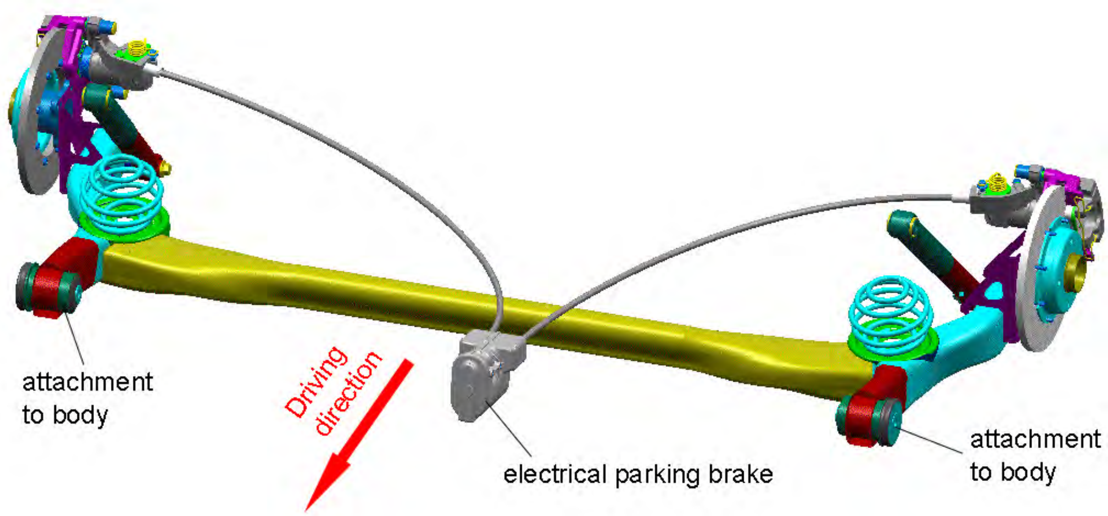

Figure 7.11.2-1 Rear suspension module

The assembly unit as shown in Figure 7.11.2-2 comprises of the welded twist beam assembly, shock absorbers, springs, rubber bushings, wheel bearings, brakes discs, and brake calipers.

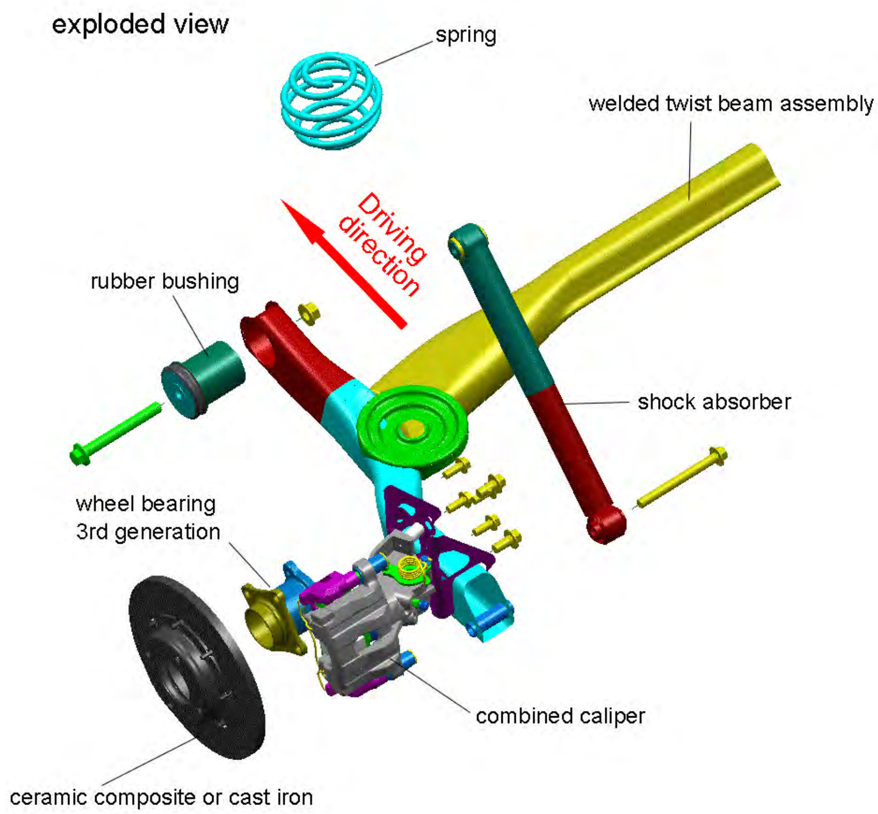

Figure 7.11.2-2 Rear suspension exploded view

The twist beam assembly is made of five (5) parts as shown in Figure 7.11.2-3, the twist beam, two (2) trailing arm subassemblies and two (2) spring cups.

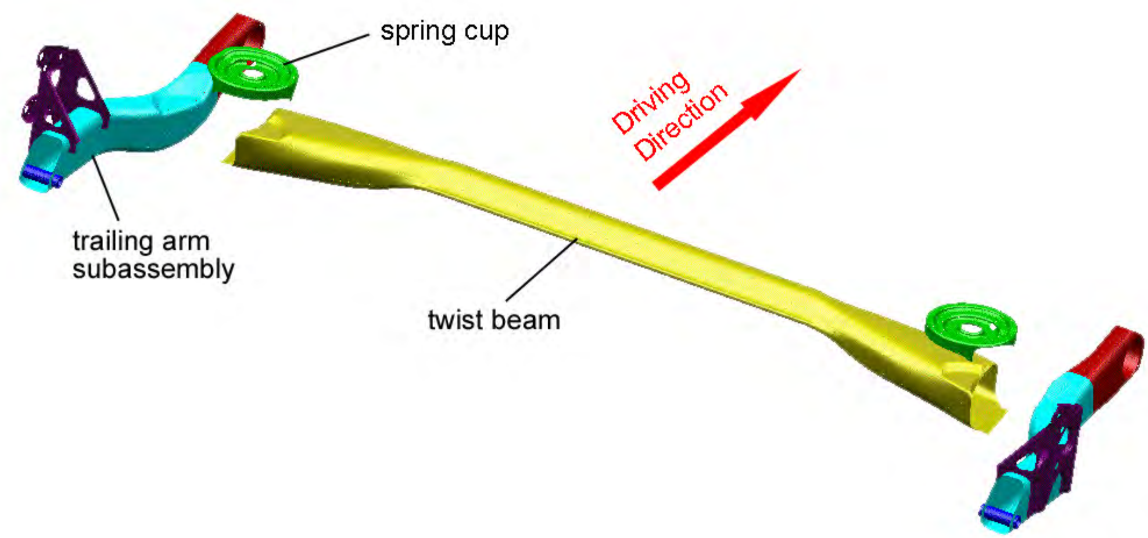

Figure 7.11.2-3 Twist beam rear suspension welding assembly

7.11.3. Kinematics Hardpoints and Layout

For the first kinematics layout, the following results were achieved as shown in Table 7.11.3-1:

Table 7.11.3-1 Kinematics results

Track	1540 mm	Track change	0 mm/cm
Toe in	0'	Toe in change	0/cm
Camber	0 deg	Camber change	0/cm
Spring rate	54 N/mm	Spring rate rel. to wheel center	31.6 N/mm
Spring ratio	0.66	Damper ratio	1.14

This kinematics layout has to be verified with prototype vehicle tests and can change during the vehicle development. The elastokinematic behavior is important for the vehicle response during maneuvers and must be tuned. The ULSAB-AVC design concept layout allows for such variations of hardpoints and mounting stiffness for future tuning as described above.

The most important characteristics are the change of toe in and camber in relation to wheel travel as shown in Figure 7.11.3-1. Because this suspension does not have toe-in and camber change during parallel jounce and rebound because there is no offset in design position, only the results for reciprocal deflection are of interest. The reason, therefore, is the vehicle behavior for the driving maneuver cornering.

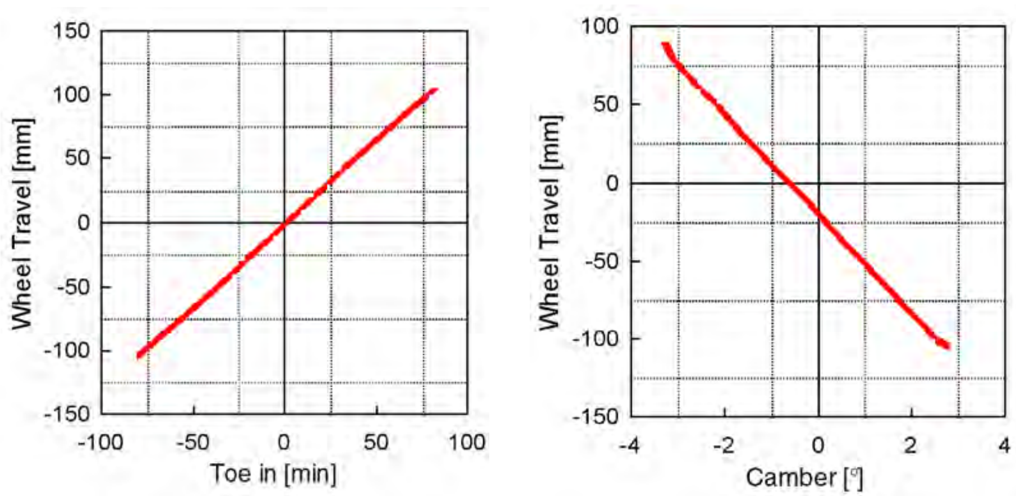

Figure 7.11.3-1 Toe in and camber characteristics for the rear suspension

The elastokinematic behavior (compliance in longitudinal and transversal direction) of the axle cannot be specified here since it is dependent on results of vehicle dynamic tests. The design provides the possibility to tune the stiffness of the bushing without changing the design of the main parts.

7.11.4. Rear Suspension Components Load Distribution

The load distribution of the rear suspension components had to be taken into account for the design of the suspension components. The load distribution is a product of vehicle mass and the resulting wheel loads (see Table 7.5.4-1) as well as the different forces resulting from steady driving maneuvers. The maneuvers and the resulting accelerations considered are shown in Table 7.11.4-1. In addition, a factor of 1.5 for safety was considered. The forces distributed to each component of the suspension are calculated with an elastokinematic program, also taking the wheel travel into account. The resulting reaction forces for each suspension part are used as load input in FE-Models for the analysis of part stress.

The resulting values of reaction forces and torque for each component of the suspension are used as load input in the FE-models for part stress analysis and can be found in Appendix - Section 4.3.

Table 7.11.4-1 Driving Maneuvers and accelerations

Manuever	Acceleration	(g)
Vertical push	vertical acceleration	3.4
Steady state cornering	lateral acceleration	1.0
Braking forward	longitudinal deceleration	1.0
Braking rearward	longitudinal deceleration	0.9

7.11.5. Rear Suspension Trailing Arm

The trailing arm weld assembly as shown in Figure 7.11.5-1 is made by joining the trailing arm, the wheel carrier and the damper attachment sleeve. The trailing arm assembly will be completed by pressing the two rubber bushings for the body mounting into the rubber bushing housing. The trailing arms are made from hydroformed tailored tubes with both tube sections made of DP 350/600. The tailored tubes have two (2) material thicknesses 2.2 mm and 3.0 mm. In the hydroforming process, the rubber bushing housing is formed and the holes for the sleeve of the shock absorber are punched. Special attention has to be given to the longitudinal location of the weld seam of the tailored tube for bending process feasibility. The wheel carrier has to be machined at the wheel bearing and brake caliper attachment locations.

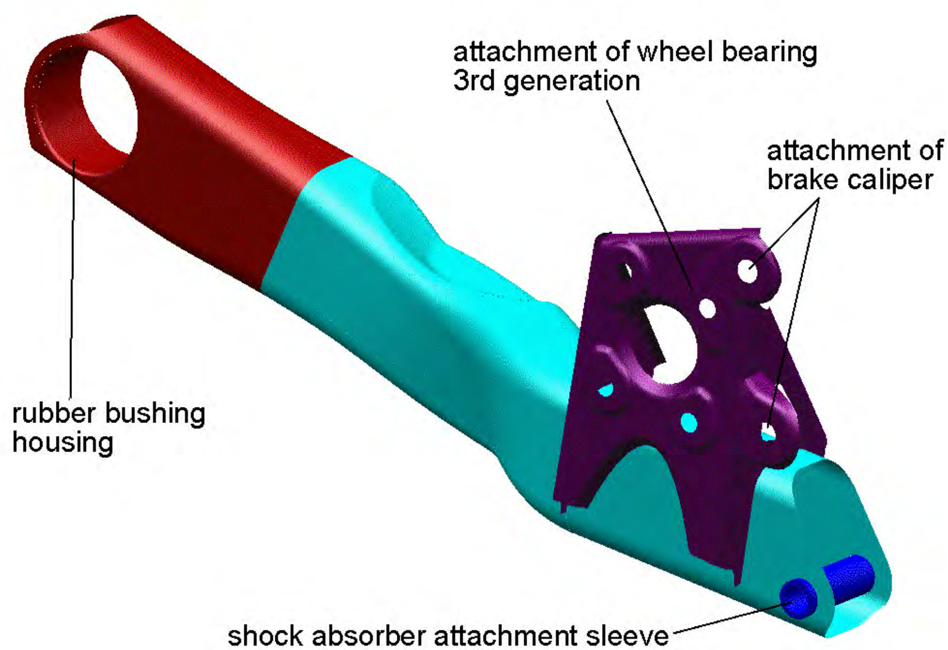

Figure 7.11.5-1 Rear suspension trailing arm weld assembly

7.11.5.1. Trailing Arm FEM-Calculation and Design Optimization

FEM-calculation and forming simulations were used to optimize the design of the trailing arm assembly. Several iterations were conducted to finalize the design.

7.11.5.1.1. FEM-Calculation Design Iteration Version 1

Figure 7.11.5.1.1-1 shows for the load case "steady state cornering," a maximum stress level of 814 MPa in the area where the twist beam is welded to the trailing arm and a stress level of approximately 1115 MPa in the wheel carrier, where it is connected to the trailing arm.

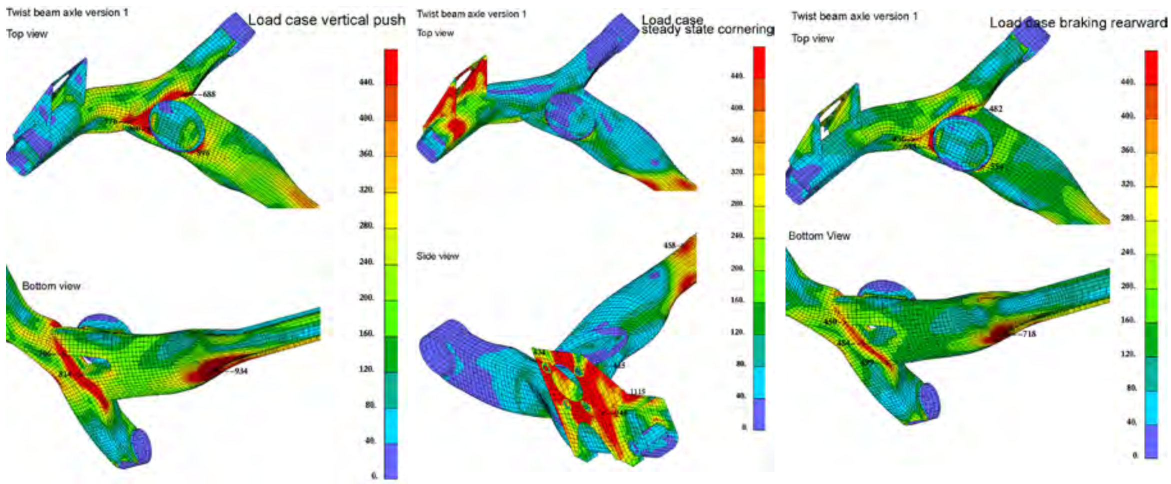

Figure 7.11.5.1.1-1 FEM-calculation results version 1

7.11.5.1.2. FEM-Calculation Design Iteration Version 2

For the second FEM-calculation (see Figure 7.11.5.1.2-1), the design was updated with two (2) changes:

- Shape of the twist beam
- Shape of the wheel carrier

With these design changes, the stress levels at the rear connection of the twist beam to the trailing arm was lowered from more than 700 MPa to approximately 400 MPa. The stress levels at the front connection of the twist beam to the trailing arm is still at a high value.

In the load case "braking rearward," the stress level at the connection of the twist beam to the trailing arm was increased from 500 MPa to approximately 850 MPa.

The stress levels at the connection of the wheel carrier to the trailing arm was lowered from more than 1100 MPa to approximately 400 MPa for the load case "steady state cornering," while showing high stress levels on the wheel carrier itself.

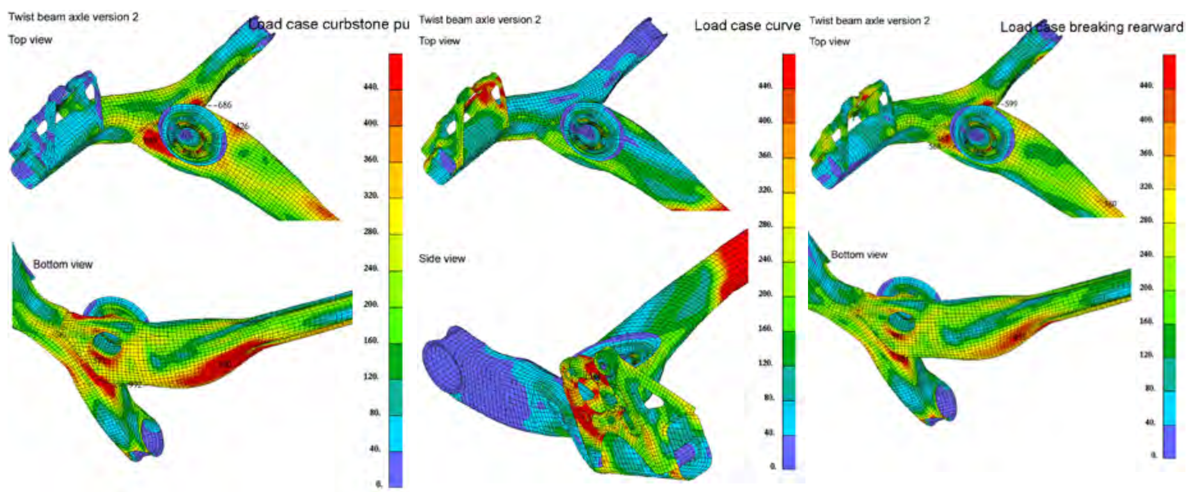

Figure 7.11.5.1.2-1 FEM-calculation results version 2

7.11.5.1.3. FEM-Calculation Design Iteration Version 3

For the third FEM-calculation (see Figure 7.11.5.1.3-1), the design was updated with four (4) changes:

- Position of the spring is located more towards the trailing arm to reduce the bending on the connection of the trailing arm to the twist beam.
- Shape of the wheel carrier
- Shape of the trailing arm
- Welding transition of the twist beam to the trailing arm

In load case vertical bump, no stress peaks remain in the trailing arm. For the load case "steady state cornering", the stress level at the rear connection of the wheel carrier to the trailing arm has increased and the stress level in the wheel carrier is still high. With minor changes as shown Figure 7.11.5.1.3-2, the stress levels can be lowered to a safe level. The stress level in the connection of the twist beam to the trailing arm has decreased and is in a safe range.

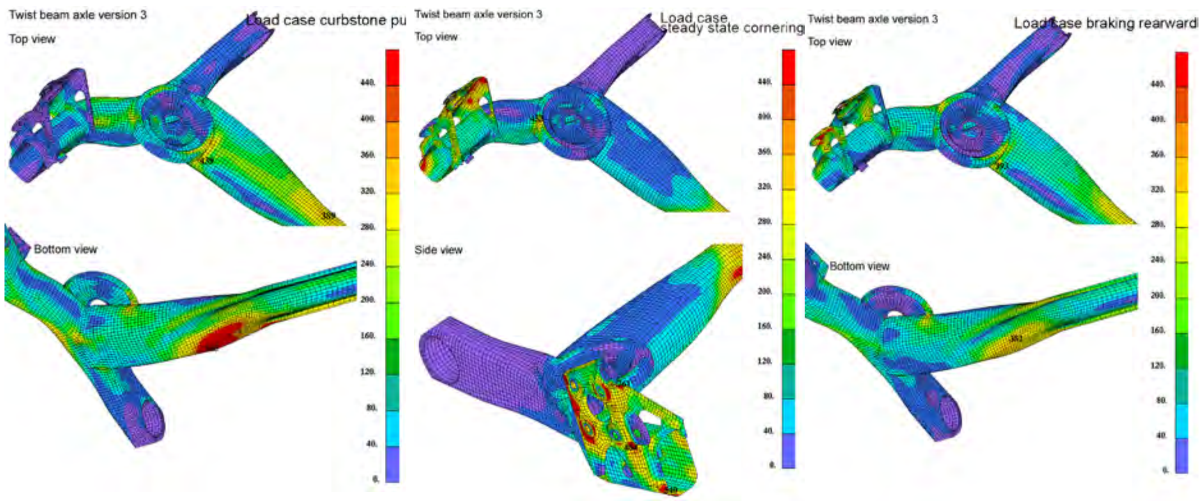

Figure 7.11.5.1.3-1 FEM-calculation results version 3

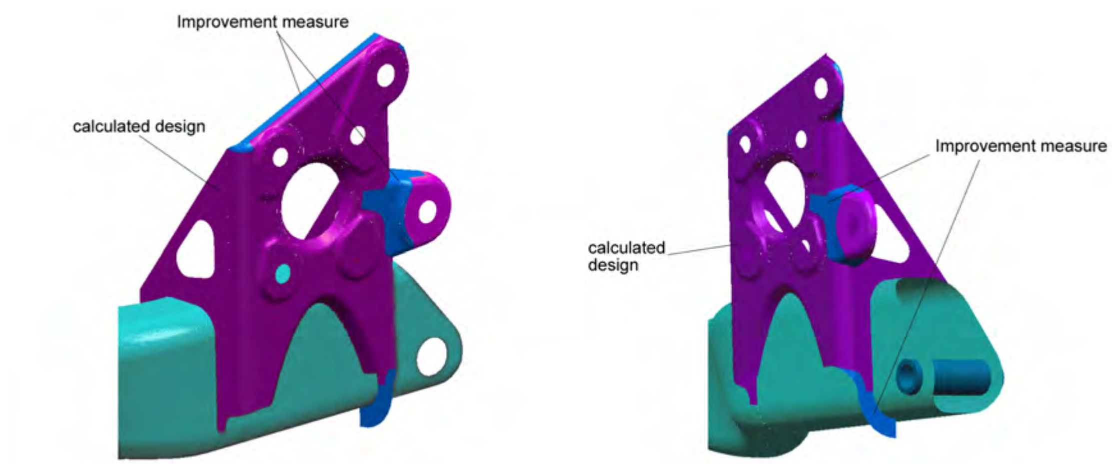

Figure 7.11.5.1.3-2 Modifications of the wheel carrier

To assess the manufacturing feasibility of the trailing arm, ULSAB-AVC Consortium Member Company conducted a forming simulation for the tubular trailing arm, analyzing each manufacturing step. The forming simulation results showed that design optimization of the trailing arm would need to be done in a detail design phase.

7.11.6. Twist Beam Profile

The material used for the twist beam profile is MnB 1200/1600 in a material thickness of 2.5 mm. The flat material properties in delivery condition (before heat treatment) are 280/450.

The twist beam profile (see Figure 7.11.6-1) has a complicated shape. Similar shapes are used in existing high volume production twist beam production rear suspensions. The main distinction is that the torsion profile for the ULSAB-AVC rear suspension is flared at the outer ends of the welded seam to the trailing arms. With this measure, the length of the welding seam is increased and the support base for side forces is enlarged with the effect that the toe out behavior of the rear suspension is minimized.

The parts manufacturing is performed in a 2-stage process. The first stage is a deep drawing of the tube to obtain the spring rate of the desired torsion bar. In the subsequent hydroforming process (second stage), the ends of the tube are widened. Axial force-feeding is used to achieve a nearly constant wall thickness at the widened tube ends.

After the forming process, the twist beam must be heat treated to achieve the required strength.

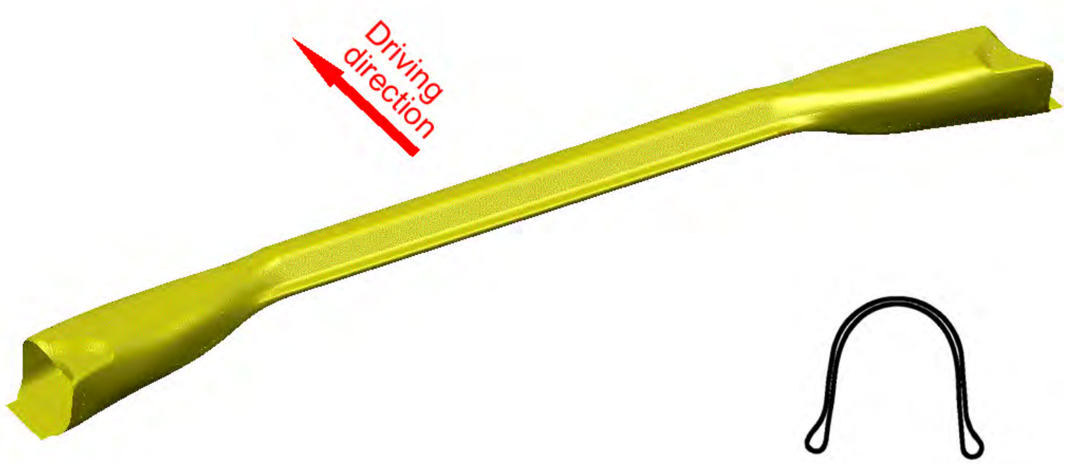

Figure 7.11.6-1 Twist beam profile

7.11.7. Wheel Bearing and Hubs

The wheel bearing as shown in Figure 7.11.7-1 encompasses a double row annular ball bearing with the following dimensions:

Outer diameter	72 mm
Inner diameter	36 mm
Width	39 mm

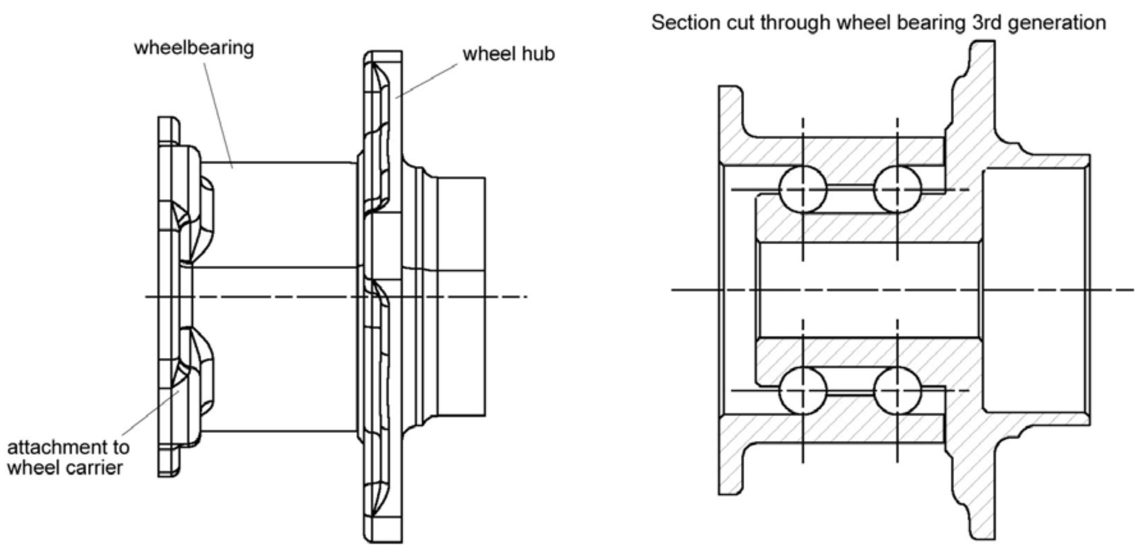

Figure 7.11.7-1 Wheel bearing and hub

The bearing has a multiple sealing with lifetime greasing. A very important point for durability of these parts is their temperature resistance, which has an influence to the sealing of the wheel bearing, as well as to the dimensional stability of the hub, because of the potential problems caused by the brake system, such as squeezing and wobbling.

For mass reduction, the shape of the surface for the wheel attachment has been reduced to a minimum and has to be confirmed or optimized (adjusted) in conjunction with tests of the rear suspension in regards to camber stiffness.

7.11.8. Shock Absorber Rear Suspension

For each C-Class and PNGV-Class vehicle, with either diesel or gasoline engine, a different damper with a special characteristic is needed and is designed as a single-tube shock absorber with an integrated bump stop for limiting wheel movement. (see Figure 7.11.8-1).

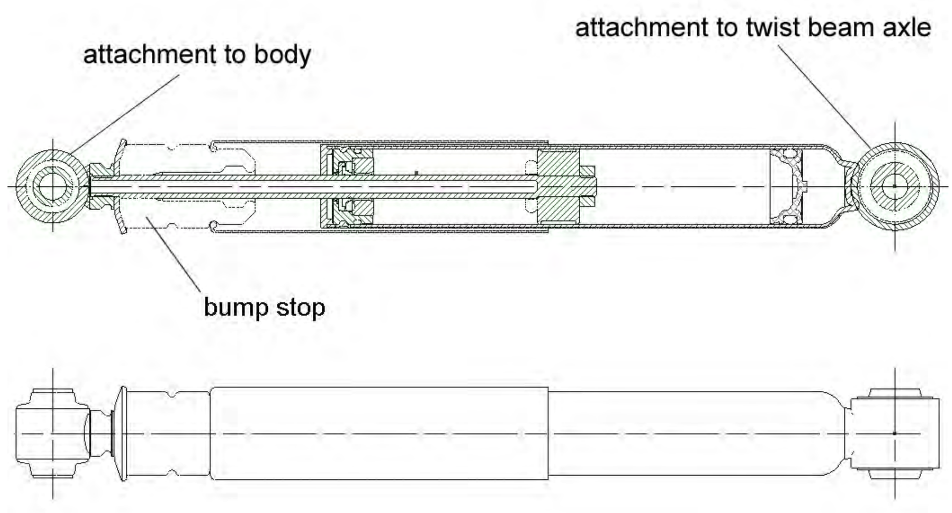

Figure 7.11.8-1 Shock absorber rear suspension

The shock absorber is bolted to the body structure and to the trailing arm through rubber bushings at its upper and lower mounting points. The wall thickness of the shock absorber tube is minimized and a plastic coating is applied to protect the tube against damages caused by stone chipping. For mass reduction, the piston rod is designed as a hollow component. The shock absorber must function in a temperature range from 40°C to 160°C.

7.12. Brake System

The brake system of the ULSAB-AVC concepts (see Figure 7.12-1) consists of the following components:

- Electro-hydraulic brake unit (EHB)
- Electrical parking brake
- Brake disc at both axles (ceramic composite or cast iron)
- Steel brake caliper at front axle
- Combined brake caliper rear axle

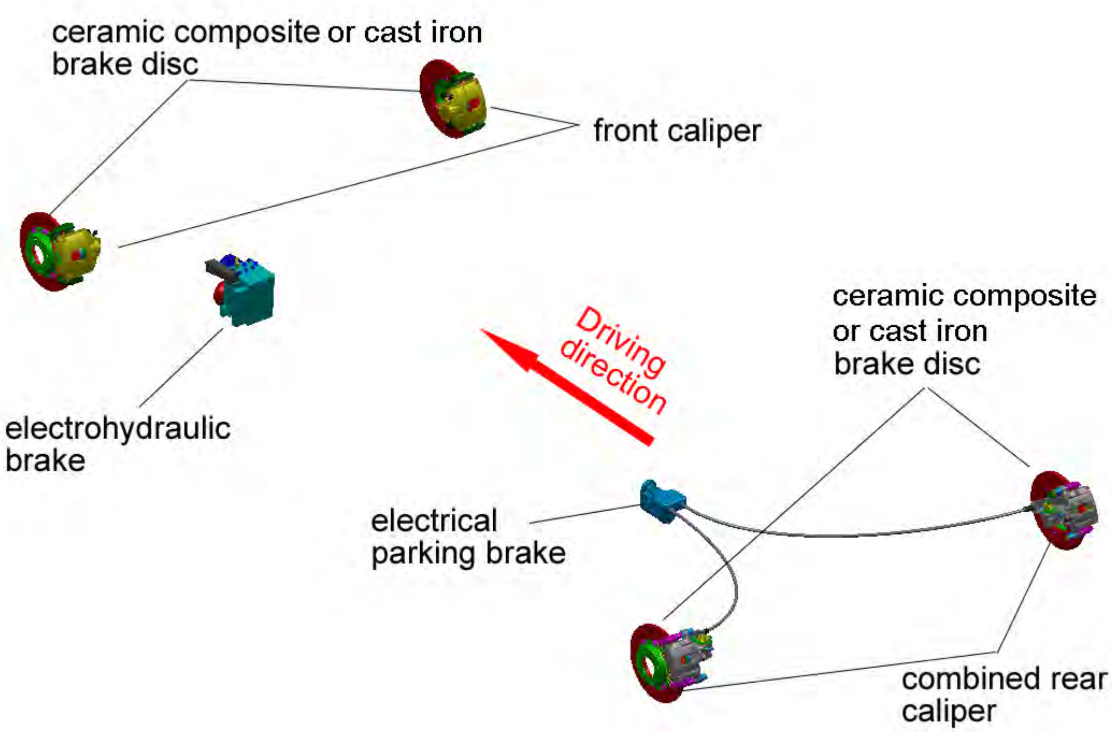

Figure 7.12-1 Brake system with parking brake

7.12.1. Brake Discs

For the ULSAB-AVC vehicle concept, the utilization of ceramic brake discs for mass reduction was investigated. The design allows the alternative use of conventional cast iron brake discs. This design requirement was implied because, in the beginning of the design, the cost of ceramic discs was not available and the possibility to switch to the less expensive conventional system was needed.

The parts cost assessment showed that the costs for ceramic brake discs would drive the vehicle manufacturing and assembly cost, therefore it was decided to utilize conventional cast iron brake discs.

7.12.2. Brake Calipers

The brake calipers used for ULSAB-AVC are carry-over parts from vehicles currently in production. The dimensions are typical for vehicles with similar mass and performance.

At the front axle, compared with similar vehicles, about two-third of the brake performance is applied. For the concept, a sliding one-piston caliper made of forged steel was used for stiffness and braking sensibility reasons. Brake pads as well as the accessory parts such as pins and springs are also carry-over parts from other vehicles.

The brake performance at the rear axle of a front drive vehicle with front-mounted engine is much lower than at the front axle. The caliper of the rear axle is a sliding one-piston caliper combined with a parking brake. The lower stiffness of the rear caliper does not affect the brake sensibility, due to the lower force level. The accessory parts used are carry-over parts from other vehicles and the same as used for the front axle.

For the front and rear brakes, a good acoustic and comfort behavior of the system is essential. Squeaking noise and juddering is not acceptable. Therefore, very complex vibration calculations and prototype testing would be necessary in a detail design phase.

7.12.3. Electro-Hydraulic Brake (EHB)

The electro-hydraulic brake unit (EHB) includes an electrical pump, a reservoir, valve block, pedal force simulator and control unit. The EHB unit is a carry-over part from a Tier One Supplier. The adaptation of the system to different vehicles is achieved by modifying the software of the control unit. To avoid too many different variations of hardware and software for the vehicle variants, different software solutions must be stored on the control unit.

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7.12.4. Electrical Parking Brake

The electrical parking brake system as shown in Figure 7.12.4-1 includes an electrical actuator and two (2) bowden cables. The actuator is attached to the body structure. The advantage of the electrical parking brake system is that the bowden cables from the rear axle to the hand brake lever in the cockpit as well as the hand brake lever itself are eliminated. The actuator is controlled from a switch in the cockpit and connected with a wire. Other advantages are in the final assembly, where no bowden cables have to be routed through the body structure.

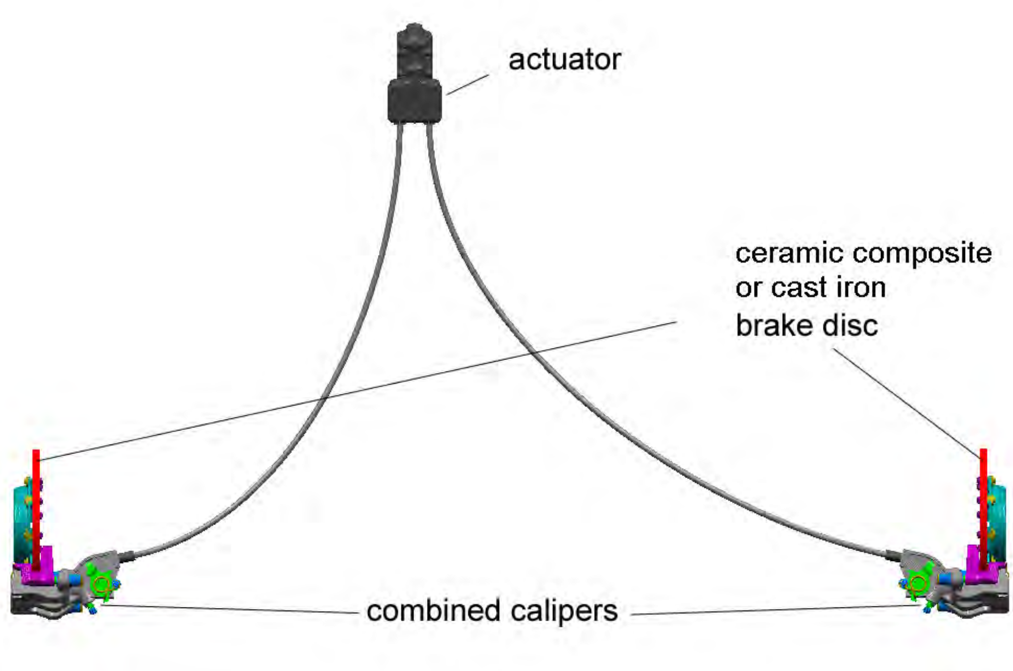

Figure 7.12.4-1 Electrical parking brake system

7.13. Wheels and Tires

The wheels selected for the ULSAB-AVC vehicles are made of steel, utilizing High Strength Steels (HSS) and tailor welded blanks. More information can be obtained in the ULSAB-AVC Wheel Technology Report.

The wheel size is 5J x 14 and the tire size selected is 175/65 R14.

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7.14. Parts/Mass Lists and Drawings

For a detailed part/mass/material specification lists and part drawings – see Appendix - Section 7.1.

7.15. Suspension Concepts Mass Summary

Table 7.15-1 shows the mass achieved for the ULSAB-AVC C-Class and PNGV-Class Suspension Concepts.

Table 7.15-1 Suspension concepts mass summary

Component Name	C-Class and PNGV-Class Target (kg)	C-Class and PNGV-Class - Mass (kg)	Remarks
Front Suspension incl.. Subframe	50.0	44.236	
Rear Suspension incl. Subframe	42.0	25.761	
Pedals Main Brake Cylinder Parking Brake Gear Shift	5.7	6.720 NA 2.113 NA	Adjustable pedal system
Wheels (4)	20.0	16.208	Wheel size 5x14 including bolts (12)
Tires (4)	26.2	26.000	175/65 R14
Steering incl. Power System	16.0	23.150	
Brake System hydraulic/ABS	8.5	36.865	electrohydraulic brake system w/ steel disc brakes
Front Brake System	15.5		
Rear Brake System	14.5		
Tire fit kit		1.000	Replacement for spare wheel tire
Total Chassis Mass *	**198.5**	**182.053**	

* Total mass does not include drive shafts

7.16. Vehicle Load Distribution

The vehicle load distribution relative to front and rear suspension at vehicle curb mass is shown in Table 7.16-1.

Table 7.16-1 Front/Rear axle distribution

Front/Rear Axle Distribution	Front Suspension (%)	Rear Suspension (%)	Curb Mass (kg)
C-Class Gasoline	55	45	935
C-Class Diesel	56	44	965
PNGV-Class Gasoline	54	46	1000
PNGV-Class Diesel	55	45	1030

7.17. Summary

The ULSAB-AVC suspension concepts show the target has been surpassed using steel as the material of choice. In combination with a range of high strength steels, new technologies such as tailor welded blanks for wishbones and tailor tube hydroforming were implemented.

Suspension components such as lightweight steel wheels, electrical parking brake, electro-hydraulic brake system, tailor welded blanks and wishbones utilizing high strength steel have contributed to surpass the mass reduction target. However, further mass reduction is possible with the utilization of ceramic brake discs.

The front end module concept could be implemented into the front suspension design, thus providing the desired advantages in final assembly.