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电动汽车悬架运动学优化方法

Y HUANG[1]，J KÖPLER[2]，A WAGNER[2]，J NEUBECK[2]，B JACOBSON[1]

（1. 查尔默斯理工大学，哥德堡 41296，瑞典；2. 斯图加特大学 汽车工程学院(IFS)，斯图加特 70569，德国）

摘要：电池电动汽车(BEV)的主要挑战之一是续航里程是否足够，因此在任何最近的车辆概念中，最大化的电池体积是合乎需要的。悬架作为最大的电动汽车子系统之一，对这一点有着重要的影响。本研究目的是使用自动化方法进行悬架开发，主要是引入一种新的可转向后悬架概念与电力推进系统，即从配备内燃机(ICE)的传统汽车开始，开发一种悬架，以满足纯电动汽车的新装配要求，同时保持有关驾驶动力学的典型要求。悬架概念是为具有大电池尺寸的高级汽车而优化的，此外还考虑了先进的主动系统，如空气弹簧和大转向角的主动后轮转向。该概念还提出了一种具有良好调整运动学性能的封装解决方案，符合原始设备制造商的调整理念。为解决由此产生的高复杂性，使用了新开发的方法，即运动学优化是用一种创新的方法完成的，该方法根据给定的要求自动提出新的硬点。在设计中，使用简化模型来表示复杂零件的形状，因此可以从包装的角度自动判断运动学概念是否可行。结果表明，新的悬架概念可以处理具有挑战性的装配问题和复杂的运动学要求。

关键词：电动汽车(BEV)；开发过程；运动学；包装

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Optimized Rear-Axle Concept for Battery Electric Vehicles: A Show Case Study for New Suspension Development Methods

Y HUANG[1]，*J KÖPLER*[2]，*A WAGNER*[2]，*J NEUBECK*[2]，*B JACOBSON*[1]

（1. Chalmers University of Technology，41296 Gothenburg，Sweden；2. Institute of Automotive Engineering（IFS），University of Stuttgart，70569 Stuttgart，Germany）

Abstract：One of the main challenges for battery electric vehicles （BEV） is a sufficient range. Therefore，a maximized battery volume is desirable in any recent vehicle concept. Suspension，as one of the largest subsystems，has a significant impact on that. Starting from a conventional car with an internal combustion engine（ICE），a suspension is developed to fulfill new packaging requirements for BEVs，while at the same time maintaining typical requirements concerning driving dynamics. The objective of this study is to use automated methods for suspension development to develop a new steerable suspension concept for an electric propulsion system. The suspension concept was optimized for premium cars with large battery sizes. Moreover，advanced active systems such as air springs and active rear wheel steering with large steering angles were also considered. The concept proposes a packaging solution with a well-tuned kinematic performance which meets typical tuning philosophies. In order to address the resulting high complexity，newly developed methods were used. The kinematic optimization was done with an innovative method，which automatically proposes new hard points，depending on the given requirements. For the design，simplified models were used to represent the shape of sophisticated parts. Therefore，it was possible to automatically judge whether a kinematic concept is feasible from a packaging point of view. The results show，that the new suspension concept can handle the challenge packaging issues and complex kinematic requirements.

Key words： battery electric vehicles (BEV)； development process； automated methods； kinematics； packaging

Suspension systems have a great influence on both vehicle architecture and driving dynamics. With the new requirements for electric vehicles，the suspension systems are required to be more compact and scalable. Furthermore，new functions such as active rear wheel steering and air spring make the packaging for rear axle even more difficult. Therefore，new suspension layouts are invented to fulfill the new functions. For example，Ref. [1]，presented a rear axle suspension which allows steering angles up to 10°. A multi-link rear

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第一作者：Yansong HUANG(1992—)，男，工学硕士，主要研究方向为车辆底盘架构及动力学。E-mail：yansong. huang@chalmers. se

suspension with rear wheel steering system is used to improve the maneuverability. As the range becomes one of the most challenging issues for electric cars, the suspension system aims to improve the battery size without sacrificing the suspension performance. Another new suspension layout is innovated by Ref. [2] to increase the battery size. Other new suspension systems like Ref. [3, 4] are demonstrated to achieve their unique functionalities. In the past years, an increasing number of methods to define and improve the kinematic targets were developed[5-9]. Ref. [10] showed two approaches how the use of carry-over-parts can be addressed during kinematic design. In contrast to that, only few approaches to consider packaging in an early phase were worked on. Those often were mainly focused on the kinematics, but not on specific CAD models, like Ref. [5, 12]. The use of CAD models enables detailed packaging analysis. In other fields, the use of automatic CAD model creation is already in progress, for example, Ref. [11] for crank shaft design.

To meet the complex requirements, the design of a new suspension concept is usually time consuming. The kinematics and packaging of suspensions need to be considered simultaneously, since they influence each other. The goal of the present work is to invent a new suspension system for a battery electric vehicle (BEV) with a maximized battery volume. The underlying algorithms are used to handle the kinematics and packaging automatically. Critical improvements during the development are demonstrated within show cases and subsequently discussed. The tuning process starts from a suspension used in a traditional combustion vehicle, and eventually is adapted to the needs of a BEV. The automatic kinematic tuning method is used to maintain the performance targets for each iteration, while the automatic packaging tool is used to search for a feasible packaging solution. The show cases confirm the efficiency of both kinematics and packaging methods. Eventually, the new suspension layout provides an extra 130 mm for the battery. The results prove the possibility to synchronize the kinematics tuning and packaging processes. Thanks to the automatic methods, the design leading time has been dramatically reduced.

1 Background

In the present work, a new suspension layout for a premium segment BEV is presented. In this section the choices for the reference car, reference components, the focus and limitations as well as the way of the working are discussed.

Besides generating a contribution to the knowledge about BEV concepts, one main purpose of this work is to show the capability of automated methods.

1.1 Focus

The main target of this work is to design a rear axle concept for a BEV. The battery volume shall be maximized while maintaining the driving behaviour from a reference car. In order to achieve more space for the battery, the suspension should take less space in front of the drive shaft. Therefore, the front-end hard points shall be moved as far backwards as possible (see Fig. 1). The wheelbase shall stay unchanged. Therefore, the simple approach of moving the whole axle backwards, is not possible. For the scope of this work, the interdependence between car body and suspension is neglected. For example, the shape of the longitudinal beam is not considered. However, the overall package is kept in a comparable size to the reference. The focus of this work lies in the assessment of kinematic properties and packaging. Elastokinematic effects and structural compliances are not considered. Therefore, mechanical properties of the structure are not taken into account.

1.2 Model selection

In order to prove the capability of the used methods, a complex suspension setup is selected. For example, an active rear wheel steering (AWS) with a large steering angle is integrated. This leads to an increased construction space for all moving parts. In today's largely manual development process, this would lead to a steeply rising development effort. But if the higher complexity is managed by algorithms,

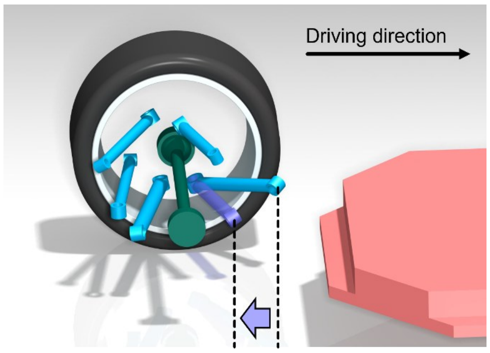

Fig.1　Basic setup of the reference car. (The idea is to move the front-end hardpoint rearwards, to achieve more battery space. Blue and purple marked components: suspension links; green marked component: drive shaft)

the development effort is handled by computers. Therefore, a high complexity in the suspension design provides the chance to prove the benefit of automated methods.

The 5-link axle is made of a comparably large amount of parts —combined with a lot of tunable parameters— and it is widely spread. Thus, the 5-link axle concept is selected for the application in this work. Since it is commonly used in higher class cars, a premium segment car is chosen as a reference (modified version of Audi A8 D5).

An internal combustion engine (ICE) car is used on purpose as a reference, since its suspension concept is not meeting the requirements for a BEV. This also makes the search for an optimal solution more complex. For the propulsion system, an offset motor (design parameters are comparable to the motor of the Mercedes EQC N 293) is chosen. It is mounted in such a way, that the motor is located behind the drive shaft. Therefore, it provides more space in front of the drive shaft, which can be used for a larger battery.

In addition, the reference model is equipped with a full active suspension. Active components increase the complexity of the suspension concept even further. The used components are listed in Tab. 2 in the Appendix.

1.3　Way of working

Each improvement cycle starts with a new idea to modify the kinematic (see Fig. 2). The coordinates of one or more hardpoints are changed. The kinematics algorithm then provides a new set of coordinates for all hardpoints, to regain the desired kinematic behaviour of the reference car. These hard points are then exported automatically. Equivalently, the new hard points are read in by code and checked for feasibility from a packaging point of view. Then, the result is discussed and further improvements for the next iteration step are decided.

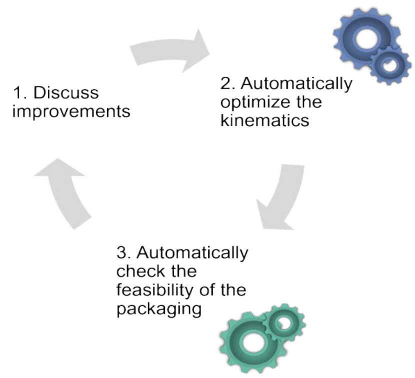

Fig.2　Way of working

2　Methods

2.1　Kinematics

The kinematics behaviours are often described by target characteristic curves. The goal for kinematics tuning is to set up the hardpoints, which gives required target curves. The method used is shown by Huang[13]. For a steerable rear axle, the targets showed at Tab. 1 are considered. Multi-step optimizations are used to search for optimised hardpoints that fit required target characteristic curves. The first step is to transfer the targets into general motions at wheel carrier. The general motions include translational and rotational velocities V, ω and translational and rotational acceleration a, α for jounce motion a translational and rotational of general motions V_s, ω_s, a_s, α_s for steering. For jounce motion, as one of the general motions of velocity are given as input, the general motions of velocity for jounce can be calculated uniquely from five jounce targets. Furthermore, the general motions for acceleration can be calculated from the time derivative of five jounce targets. Following the same principle, the calculation applies for steering motions as well.

The second stage of optimization is designing the hardpoint layout. Instead of optimizing the coordinates of the hardpoints, the algorithms optimize the directions and the length of the links. Therefore, the optimization can provide the intuitive understanding between the kinematics and packaging constraints. Once a point of a link is given, the algorithms search for the correct link direction that goes through the given point and the terminal points along the link direction. The algorithms optimize all the link directions and terminal points simultaneously according to general motions which are calculated from targets. The given point is defined as a point going through the link direction but not necessary inside the link.

For example, the point can be a point at the middle of the link, or a point out side of the link. Indeed, the given points are critical to influence the packaging because the complete hardpoints are optimized according to given points of each link and targets. The automatic kinematics tuning algorithms are built according to mentioned procedures as shown in Fig. 3. In addition, the algorithms interact with automatic packaging method to find a feasible packaging solution with required kinematic behaviours.

Tab.1 Targets considered for kinematics

Static targets	Toe	Camber			
Jounce targets	Bump steer	Bump camber	RCH[1]	Anti lift	Anti squat
Steering targets	Kingpin angle	Caster angle	Scrub radius	Caster trail	WLLA[2]

Note: [1] Roll center height; [2] Wheel load lever arm.

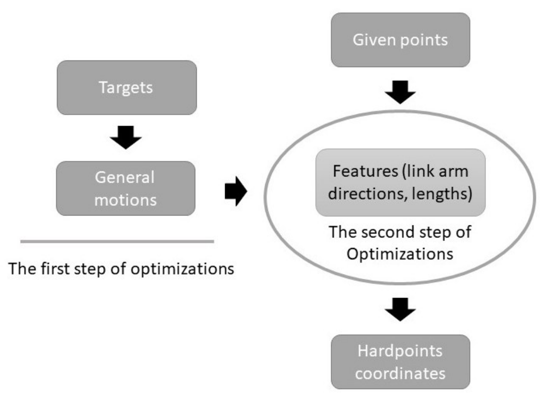

Fig.3 Multi step optimization

2.2 Packaging

Packaging comprises the shape, size, orientation and movement of the used parts. The arrangement must be composed in such a way that no clashes between the parts occur at any given time. Moreover, often minimal allowed safety distances between parts are required. In order to meet those requirements, swept volumes are used. Those are the spaces which are taken by a part during a movement (see Fig. 4). To calculate those swept volumes, a kinematic model is created. This comprises the hard point positions and the types of joints which are used. With that and a motion law (jounce and steering motion), a rigid body simulation can be performed. The results are used to create the swept volumes, which in turn are used to calculate if any clashes are occurring during axle movement. The position and the penetration depth of those clashes are then handed out. This information is the basis to manually find an improved hardpoint setup in the next iteration step.

The geometric models used in the packaging algorithms can be divided into two subcategories: First, carry-over-parts (COPs) are given parts, which cannot be modified. Secondly, parametric models are simplified and adaptable models. If a clash between two parts exists, in which a parametric model is involved, the parametric model can be automatically adjusted. For example, in Fig. 4 a parametric model of a link and a COP of a wheel are depicted. The link gets automatically bent to resolve the clash.

3 Application

During the design process, a lot of different show cases were created. In this section, the most significant improvements are laid out.

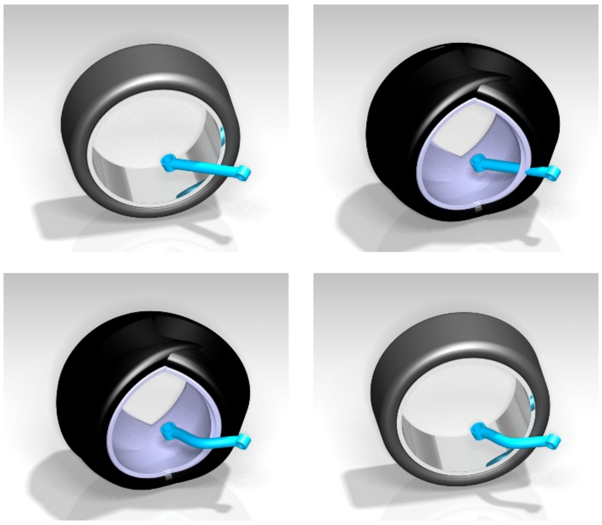

Fig.4　Use of swept volumes to determine the existence of a clash. (Top left: COP for wheel and parametric model for link in design position. Top right: Swept volume of the wheel clashes with the link. Bottom left: The rod gets bent to solve the clash. Bottom right: Wheel and bent rod in design position)

3.1　Show case 1

The target for the first show case is to reproduce the driving behaviour of the reference car. The propulsion system is switched from ICE to BEV. Also, more space for the battery shall be created. In the first show case, it is tried to meet those targets with rather small changes to the reference car.

The axle concept of the reference model and Show case 1 can be seen in Fig. 5. In order to create more space for the battery, the hardpoints in the lower level are moved rearwards. The kinematic behaviours of the reference car are remained for Show case 1, as they are shown in Fig. 15 in Appendix.

One problem with Show case 1 is the positioning of the air spring. For the reference car, the air spring and the damper are mounted concentrically. In order to gain more compartment space, it is decided to mount the air spring and damper next to each other. That leads to a tight packaging situation in the area of the air spring. The toe link (between the hardpoints D and D′, see Fig. 14 in Appendix) has to be bent a lot to make it clash free, see Fig. 6. With this setup, the maximal steering angle is limited to 10°. Otherwise, the toe link is not mountable clash-free. In order to make the setup clash-free, the air spring also has to be positioned far inwards. That comes with the drawback of a lower spring efficiency. To counter this downside, the inner hard-points of the spring link D′ is moved further rearwards. The drawback of this measure is that the length of both links must be increased to fulfill the kinematic targets, so that they both take more space in the area where the subframe has to be designed later on. Moreover, the inner hard-points of those links (C and D) are located in such a close distance to each other that a clash between both parts is unavoidable. It can be concluded, that at least one major change in the axle concept has to be made to resolve these issues.

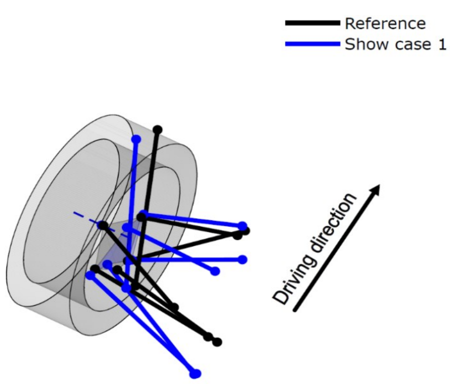

Fig.5　Show case 1

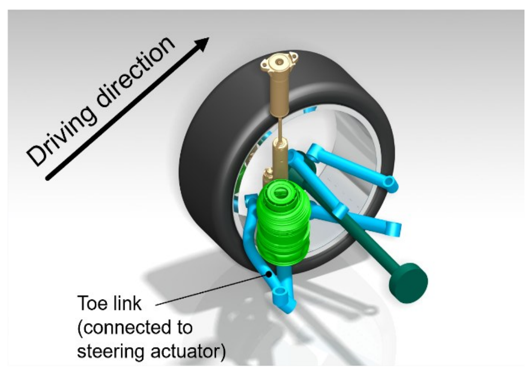

Fig.6　Show case 1 package

3.2　Show case 2

Show case 1 indicates the problem of tight packaging issue at the rear end of the subframe. The toe link and the spring link are difficult to package since they end up in the similar location. In order to solve this issue, a new concept in which the toe link is moved to front is proposed. Additional Show case 2, (see Fig. 7), shows a new layout which the toe link is located at the front part of the subframe. Additional

Show case 2 is able to maintain the similar kinematic performance for jounce motion. However, Fig. 15 indicates the complete different kinematic performance for steering motion comparing with Show case 1.

In order to regain the steering performance, Show case 2 is proposed with compromised toe behaviours for jounce motion. The toe curve that is shown in Fig. 8 is tuned in a more symmetrical way. The toe angle drops approximate to 0. 22° at 100 mm rebound travel. With such compromise, the targets for steering motion remain similar behaviours as shown in Fig. 15 for Show case 2.

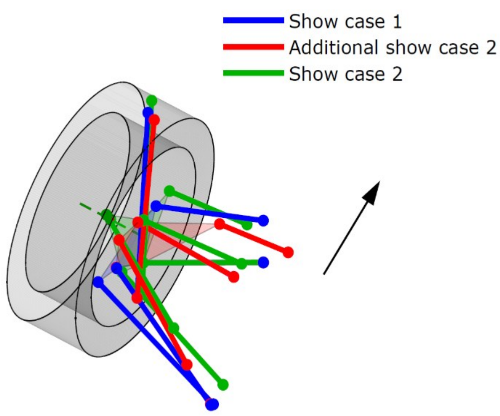

Fig.7 Show case 2

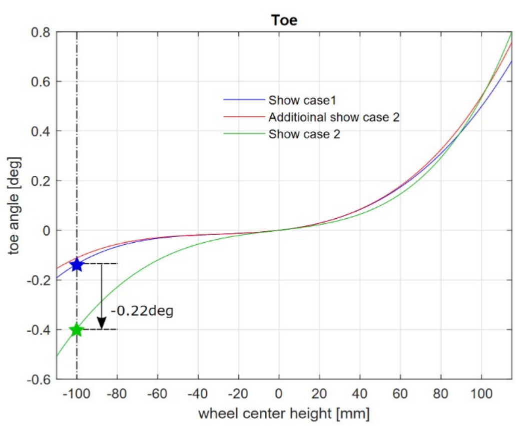

Fig.8 Toe curve of Show case 2

Comparably to Show case 1, the space for the link, which is located most rearward, is critical. But since its position was changed, there are more possibilities to solve the occurring clashes. Fig. 9 shows the swept volumes of the neighbour parts. As long as the link has no interference with any of those swept volumes, it is ensured that no clashes exist during axle movement. It can be seen, that a solution was found, which is clash-free. It allows steering angles of 10°.

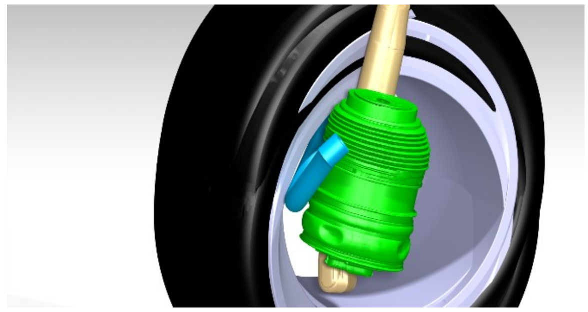

Fig.9 Swept volumes for bent link

3.3 Show case 3

Show case 2 has improved packaging solution for rear end of the subframe. With the compromised toe curve behaviour, the majority of targets remain the same. However, the link arms need to be more compact in order to provide sufficient size for the subframe design. The idea is to shorten all the link arms without dramatically changing the kinematic behaviours. A new modified concept Show case 3 is presented Fig. 10. With the compromised roll center height, Fig. 10 shows the shorted link arms. Fig. 11 highlights the Roll center height increase 36 mm at the 100 mm rebound travel. This adjustment can have impacts on the overall driving properties. For example lateral disturbances at uneven roads might induce increased heave motion. Those effects should be assessed on the full vehicle layer, which is out of the scope of this work. At the same time, Fig. 15 shows that the rest of the targets remain. The improved package can be seen in Fig. 12.

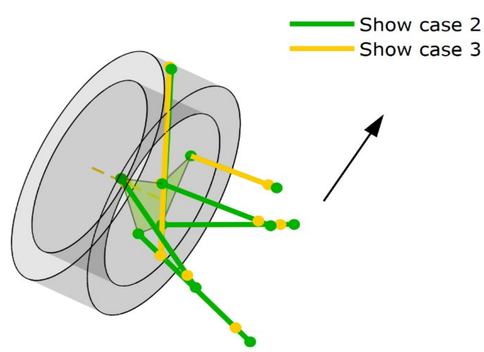

Fig.10 Show case 3

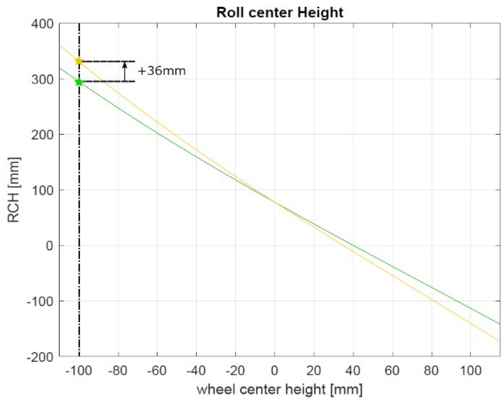

Fig.11　Show case 3 roll center height

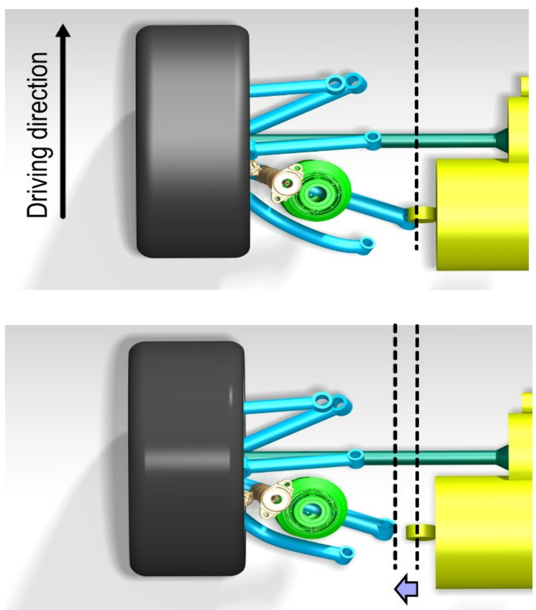

Fig.12　Improved package. (Upper: Show case 2, lower: Show case 3 with more space between the inner bushings and the motor)

With Show case 3, an appropriate solution to the given task is found. With that setup the battery can be located 130mm rearwards compared to the reference car, see Fig. 13. Moreover, it is possible to use an active rear wheel steering with a maximal steering angle of 10°.

4　Conclusion

This paper demonstrates three show case studies to solve the packaging issues. During the transformation from a suspension of an ICE vehicle to a BEV adapted one, the hardpoints are tuned in such a way that the kinematic behaviours are remained as

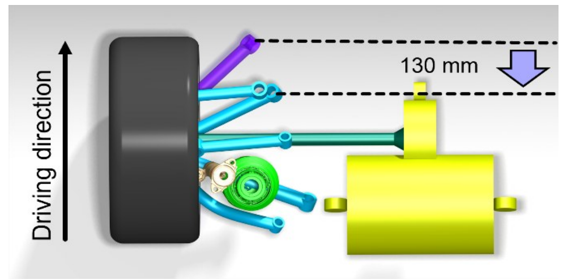

Fig.13　Package of Show case 3 (Violet: critical link from the reference car. It was moved back 130 mm. This space is now available for additional battery volume.

much as possible, and the subframe is being moved backward in order to create space for battery. To balance the overall kinematic performance, certain compromises are done for Case studies 2 and 3.

With the traditional trail-error method, the tuning process is extremely time consuming because the tuning loops for kinematics and packaging have to be considered simultaneously. With the automatic tuning method, the kinematics and packaging work are done automatically. Then engineers only need to synchronise the optimization results and adjust the optimization setups if some conflicts are detected. This new method improves the working efficiency dramatically thanks to the automatic tuning algorithms.

Since this paper has a limited scope, the specific design for the subframe is not considered. However, it is important to have a mutual design especially the connection between the subframe and the body. Furthermore, suspension compliance behaviours are also critical factors to influence the driving dynamic. Therefore the future work should consider these two aspects.

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Appendix 1 Nomenclature (Fig. 14)

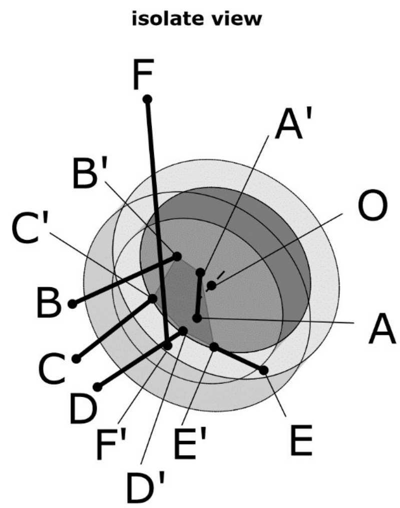

Fig.14 Hard point nomenclature

Appendix 2 Packaging models (Tab. 2)

Tab.2 Models for packaging

Part	Implementation	Derived from
Battery	Parametric Model	Audi Q4 e—tron F4
Motor	COP	Mercedes EQC N 293
Drive Shaft	Parametric Model	Mercedes EQC N 293
Brake System	COP	Mercedes EQC N 293
Active Rear Wheel Steering	COP	Mercedes EQS V 297
Air Spring	COP	Mercedes EQS V 297
Active Damper	COP	Mercedes EQS V 297
Wheel Carrier	COP	AUDI A8 D5
Links	Parametric Model	Own Model
Tie Rod	Parametric Model	Own Model

Appendix 3 Simulation results (Fig. 15)

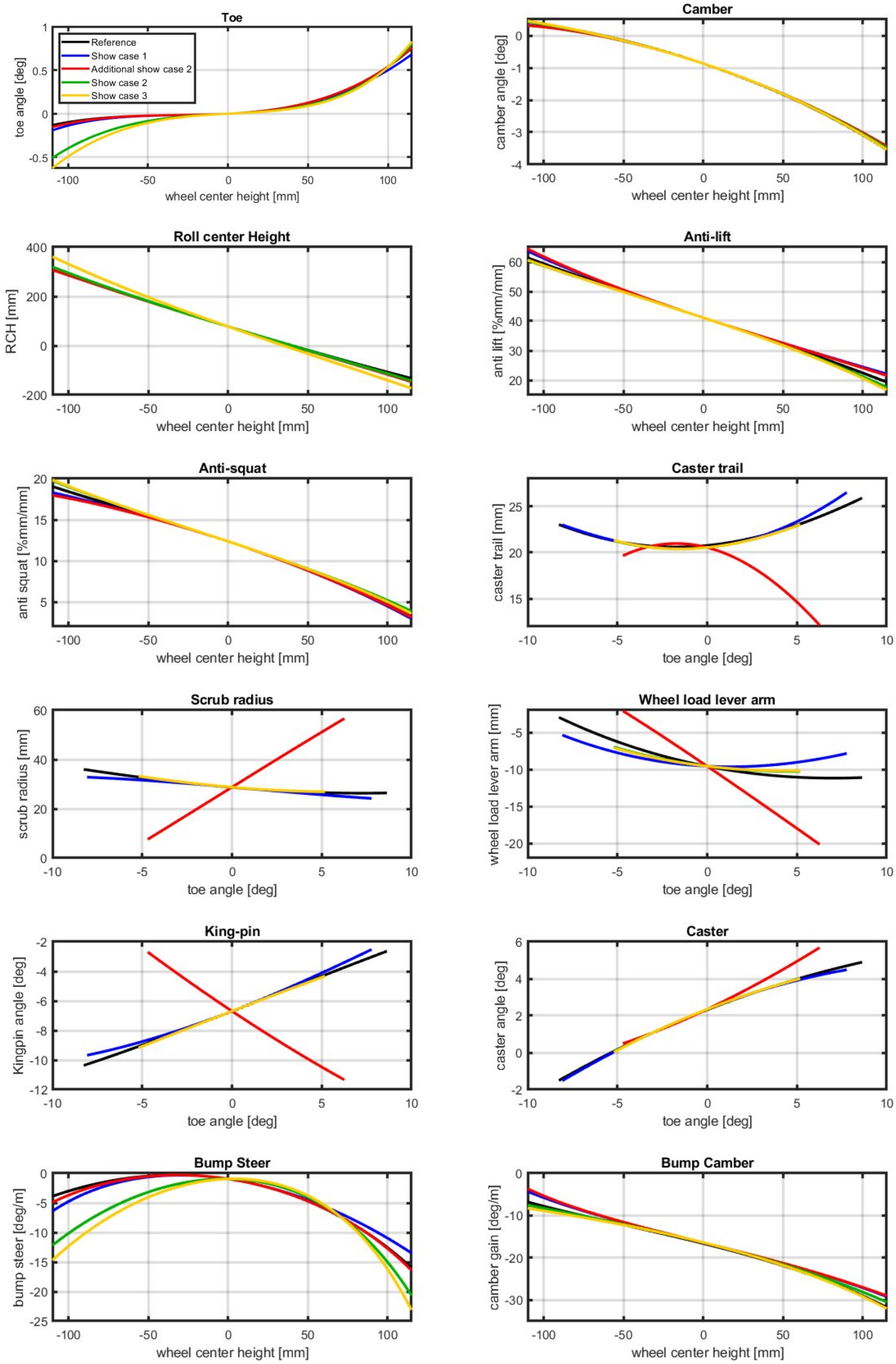

Fig.15　Results from simulation