26

Chassis systems

Schaeffler is more than just bearings

Manfred Kraus

Introduction

The launch of ABS in the 1970s marked the starting point for the introduction of electronic control systems in chassis. In the following years, driving dynamics control systems were used to an increasing extent in passenger cars, with longitudinal, vertical and lateral dynamic driving characteristics can be improved. Further potential resulted from the ever increasing networking of functions [3]. This further development of the on-board power supply is currently setting prerequisites for further driving dynamics systems with electromechanical actuators.

Some of these systems – such as the superimposed steering system – owe their existence and full functionality to increasing, further development of electromechanical systems and microprocessor technology. Other systems have hydromechanical predecessors such as electric power steering (EPS), rear wheel steering and torque vectoring (distribution of the driving forces to the driven axle using an electromechanical actuator). The expansion of the functions of these systems, however, was only possible with further development of microprocessor technology. For example, the parking assist system of an EPS with hydraulic steering can only be produced at significantly higher cost. In addition, the energy consumption of hydraulic actuators in chassis is much higher than that of electromechanical actuators, which only consume energy when required.

The following objectives can be derived for chassis systems from the above.

The replacement of hydraulic actuators

The objective is to reduce the power consumption and improve existing functions, such as in steering systems. EPS has already achieved high market penetration here and is established on the market.

There has been similar development in the case of hydraulically actuated roll stabilization systems which will soon be replaced by electromechanical systems. It is possible that hydraulic base point displacement of the wheel suspension (so-called active body control ABC) will be replaced by an electromechanical variant.

Rear axle steering, which was offered on the European market as a hydraulic variant in the 1980s and 1990s, is currently experiencing a renaissance. However, the new systems have electromechanical actuators, are more economical, have more favorable fuel consumption and are easier to integrate.

Development of new functions

The superimposed steering system can only be manufactured on an industrial scale using an electromechanical actuator. It made it possible to superimpose the steering angle components calculated by a driving dynamics controller on the driver's steering input, without having to disconnect the mechanical connection between the steering wheel and the front wheels, as would be required by pure, by-wire systems. In the area of vertical dynamics, a mention must be made of controlled damper settings, which not only enable adjustment in accordance with the loads and driving style, but which also contain more or less developed "sky-hook" components in the damper algorithms used.

Linking of systems

The combination of superimposed steering with ESP enables the frequency and level of stabilizing braking interventions to be significantly reduced. The combination of superimposed steering with rear wheel steering, on the other hand, results in a significant increase in the cornering performance at low speeds, while at the same time increasing the agility and stability in the medium and high-speed range. Similar lateral dynamic results must be achieved with the combination of superimposed steering and torque vectoring. Traction can be improved with torque vectoring and is comparable with a controlled locking differential.

If steering and braking systems are linked with environment detection systems with GPS and car-to-car communication, it will open up new possibilities which could lead to adaptive cruise control and a lane-keeping assistant and, in future, collision mitigation and avoidance or even autonomous driving.

Schaeffler – more than a wheel bearing

Schaeffler had already started development of mechanical actuators (ball screw drive, small planetary gear and cylindrical gear units and the bearing support for the entire module) for electromechanical brakes in the 1990s. Mechanical modules for EPS were developed based on these actuators, comprising the ball screw drive including the toothed rack and bearing supports, which have been on the market since 2007 and whose manufacturing quantities continuously increase.

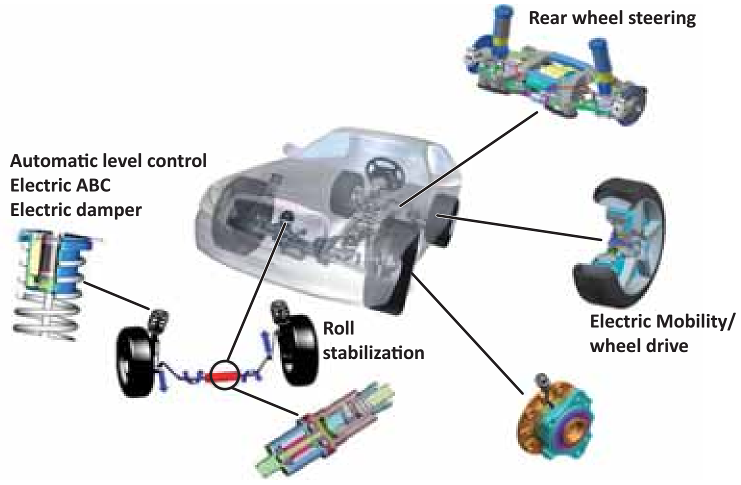

Figure 1 Overview of chassis activities at Schaeffler

Development of electromechanical systems was started based on the mechanical modules and orientated on the strategic direction of Schaeffler. The focus of development is on new chassis functions, for which there has been no market until now. Figure 1 shows an overview of all current chassis development activities.

The development objectives are to increase the power density and reduce the power consumption and costs of the systems, was well as generate new functions. This will increase the attractiveness for customers and widen the lead over the competition. Some chassis system developments are described in more detail below as an example.

Rear axle steering

The earliest rear wheel steering developments showed to some extent the potential in this area [4], [6]. However, current boundary conditions appear to make a reassessment of rear wheel steering or active, toe-in actuation on the rear axle necessary. "Active rear axle kinematics" (ARK) is a useful term to describe this technology.

There are new boundary conditions for actuators due to the current widespread use of electromechanical steering assistance on the front axle and, therefore, the possibility of economically transferring this technology to ARK. In comparison to a hydromechanical solution, the actuation time along with the costs can be reduced, and the application limits and performance, particularly at extreme temperatures, can be increased.

In addition, the increasing networkability of chassis systems and the many state functions available for driving dynamics contribute to better quality chassis control [5]. This means that the possibilities for driving dynamics and the potential of ARK can be nearly fully exploited, as demonstrated by vehicles which have been launched recently with this technology [2].

Requirements and benefits of ARK

The requirements for an ARK system can be summarized in a generalized form:

- Adjustment angle on the wheel: +/- 3.5°

- Adjustment distance on the actuator = 10 to 25 mm (depending on the axle)
- Max. actuation speed up to 100 mm/sec for 50 % of the full stroke
- Positioning/resolution: < 0.1 mm
- Actuation force up to approx. 6 kN, static overload up to 30 kN
- Higher electromechanical efficiency (max. battery current < 40 A with a 12 V on-board power supply)
- Minimal energy consumption with a constant wheel angle
- Fail-safe behavior by holding the position in case of a defect

The features of the actuator can be adapted to the specific requirements of the OEM if required.

The following advantages of the system can be fully used if the named requirements are fulfilled:

- Optimization of the time profile of transverse acceleration and yaw rate for the vehicle under transient driving conditions
- More direct response to steering commands
- Reduced inclination to overshoot (rear of the vehicle breaks away)
- Increased safety and stability, particularly during braking in curves under µ-split conditions
- Potential for improving the directional stability by reducing the structural, kinematic, toe-in change of the rear axle
- Optimized stability during trailer operation
- Reduced turning cycle
- Reduced tire wear and fuel consumption due to the possibility of reducing the initial toe-in

There are many references to tests available which quantify the advantages. The improved response behavior and increased stability are, for example, recorded and measured by means of the 18 m slalom and the ISO lane changing test. ARK enables higher, drive-through speeds during these driving maneuvers, as proved in different tests [1], [9]. These tests can also be used as an argument for improved trailer operation. The reduction in the turning circle can be read from the Ackermann equation by making a simple adjustment. It can be also be shown that the stated pivot range of +/- 3.5° of an ARK system is sufficient to achieve a significant reduction in the turning circle [1].

Actuator design for ARK

The basic actuator configuration comprises the components shown such as the electric motor with the power electronics system, sensor and ball screw drive as the gearing or mechanical actuation element.

The gearing is generally designed to be self-locking so that no retaining forces/currents are required in a static condition and the actuator does not change its position when no current is flowing, as in the case of a possible malfunction. However, because self-locking is coupled with low efficiency, this leads to a significant power consumption during actuation. An actuator with optimized efficiency has a lower power consumption and improved dynamics, but in the presence of static loads, high retaining forces/currents are required to hold a position which has been reached.

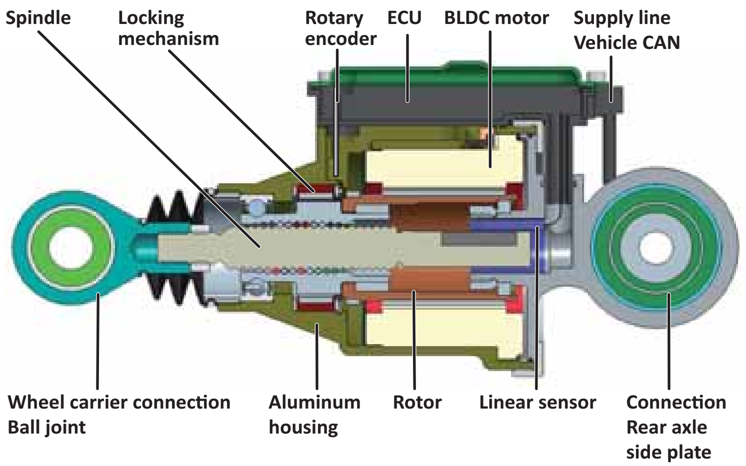

Figure 2 Design of a Schaeffler linear actuator

Figure 3 ARK designs with electric motors which are arranged in parallel (left) and coaxially (right) with the axle for toe-in actuation of individual wheels. Length x diameter of the coaxial actuator is 130 mm x 85 mm without the ECU

Schaeffler has succeeded in improving the energy consumption of the actuators. This improvement has been achieved by combining the efficiency-optimized actuator gearing (ball screw drive) with a locking mechanism, which "locks" the actuator under load in the position it has reached. This solves the problem of the conflict between the objectives, high efficiency and self-locking.

Figure 2 shows the design of such an actuator for linear motion. In addition to the electric motor and a low-friction ball-screw drive for converting rotary motion into translatory motion, a freewheel which can be switched in either direction is integrated in the power flow of the actuator.

An analogy with an electrical diode is helpful in clarifying the function of the locking mechanism, whereby the comparison is only intended as an aid to understanding the concept due to the differences that exist. As a simplification, you can say that the energy which is brought into the actuator from the inside outwards – i.e. induced by the motor – reaches "the outside" where it is available for actuation. However, if external forces are acting on the actuator, then the actuator is locked mechanically by the switchable freewheel. The motor is relieved of stress and is thus protected from high and unnecessary electrical power consumption. At the same time, this mechanical design also meets the fail-safe requirement and holds the actuator in position if power is lost. A separate, electromagnetically operated locking device is not required.

The electric motor can be designed with architecture which is coaxial or parallel to the axle, depending on the design space available or to meet other boundary conditions.

The ARK systems available on the market are equipped primarily on cost grounds with a central actuator, which only allow synchronous actuation of the toe-in. The benefits in terms of driving dynamics offered by wheel-selective ARK compared with synchronous ARK have not been utilized until now because of reasons of cost, weight, and the available design space. Schaeffler has developed an actuator, which can be used for both wheel-selective and central designs, in order to cater for both applications.

The objective was to use the greatest number of identical parts in the wheel-selective actuator and the central actuator.

Only the components marked below in Figure 4 must be adapted to convert an actuator for wheel-selective actuation into a central actuator and this only requires a minimal amount of effort. The spindle is adapted for the increased distance between the pivot points by means of screwed-on extensions. Plain bearings in the fitting parts support the transverse forces. The lugs required on the center section of the actuator are formed by using interchangeable inserts in the pressure die casting

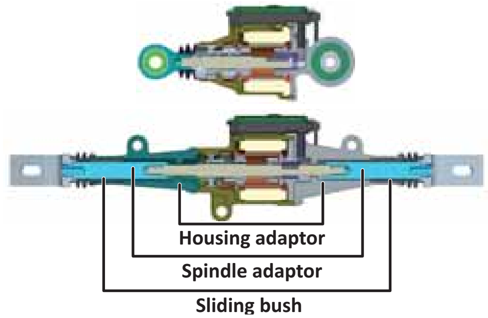

Figure 4 Actuator for ARK with coaxially arranged electric motor and synchronous actuation (below)

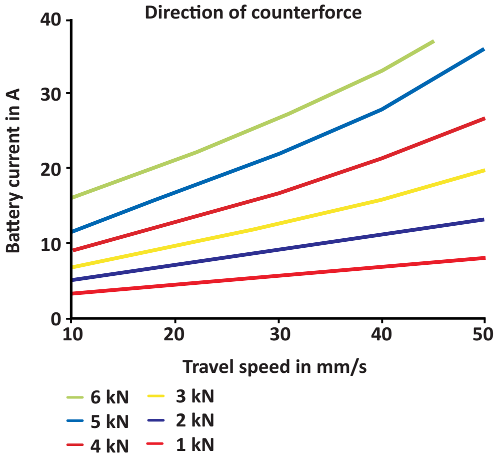

Figure 5 Battery current as a function of the travel speed and axial load of the actuator

mold. This ensures that the wheel-selective actuator can be used without any modifications and any adaption costs for bridging greater distances between the tie rod joints are minimized.

The design selected for the engine and the transmission enables the performance range shown in Figure 5 to be covered.

As a result of the high efficiency, the maximum loading on the on-board power supply of the 12 V design is under 30 A for almost all actuating forces and speeds which are required by OEMs.

The Schaeffler ARK system is offered as a smart actuator with integrated power electronics to reduce the influence of EMC and to meet the requirement that different electromechanical systems are compatible. When deciding on this configuration, great importance was placed on making the actuator as compact as possible and cost-effectively adapting a control unit from an existing EPS application.

The usual sector process chain for driving dynamics systems, consisting of Matlab/Simulink, Target Link and Autobox, is used for developing the functions of the ARK system which are relevant to safety.

System performance and results

Alongside testing on test stands, vehicle tests were conducted with Schaeffler's own demonstration vehicle to verify the performance of the ARK. Figure 7 shows the fitting location of the actuators. The function, strength and reliability of the mechanical components, the electrical and electronic components and the system have been verified in many endurance tests, proof tests and road tests.

Figure 6 Wheel selective actuators for ARK in the Schaeffler demonstration vehicle

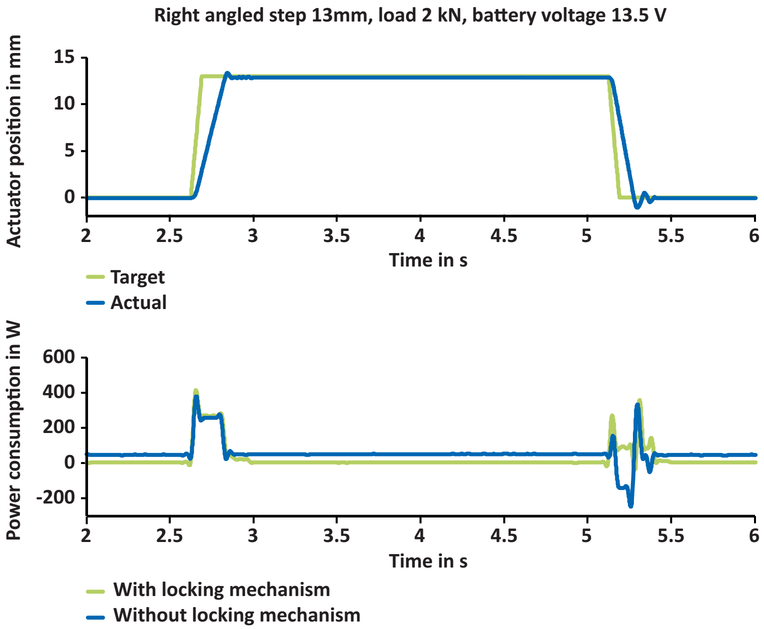

Figure 7 Step response and associated power consumption of ARK

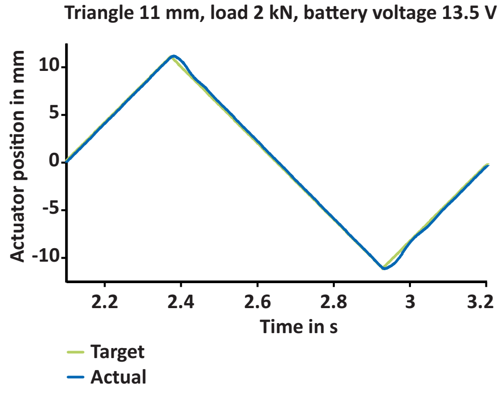

Figure 8 ARK response to a setpoint change

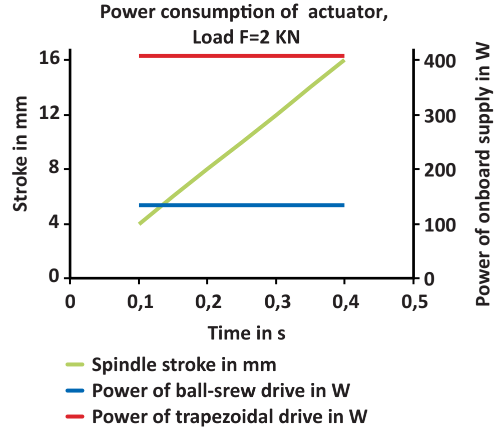

Figure 9 Load placed on the on-board power supply by ARK

Some of the test results are presented below.

A characteristic feature of ARK dynamics is the system step response under a load of 2 kN. The upper part of Figure 7 shows an actuation time of less than 200 ms for a distance of 13 mm. The lower part of Figure 7 shows the power consumption of the actuator with and without a locking mechanism The power required to hold the position without a locking mechanism is 30 W.

Figure 8 demonstrates the response to a setpoint change by comparing a triangular target signal with the corresponding actual position of the actuator. As can be seen in the figure, there is a very good correlation between the actual and target values. The overshoot at the return points is negligible even with the adjustment speed of 40 mm/s shown in the diagram.

Figure 9 compares the simulated power consumption curve for an actuator with a recirculating ball drive and an actuator with a trapezoidal thread under a load of 2 kN. The blue line shows the distance/time curve for both actuators. The line marked in red is the power required for a trapezoidal drive and the green line indicates the power required by a Schaeffler actuator. The loading on the on-board power supply is over three times lower with the ball drive from Schaeffler.

The measurements clearly show the advantages of the Schaeffler ARK system with regard to energy consumption compared with current solutions from competitors. The diagram also indicates the higher level of dynamics available with this system.

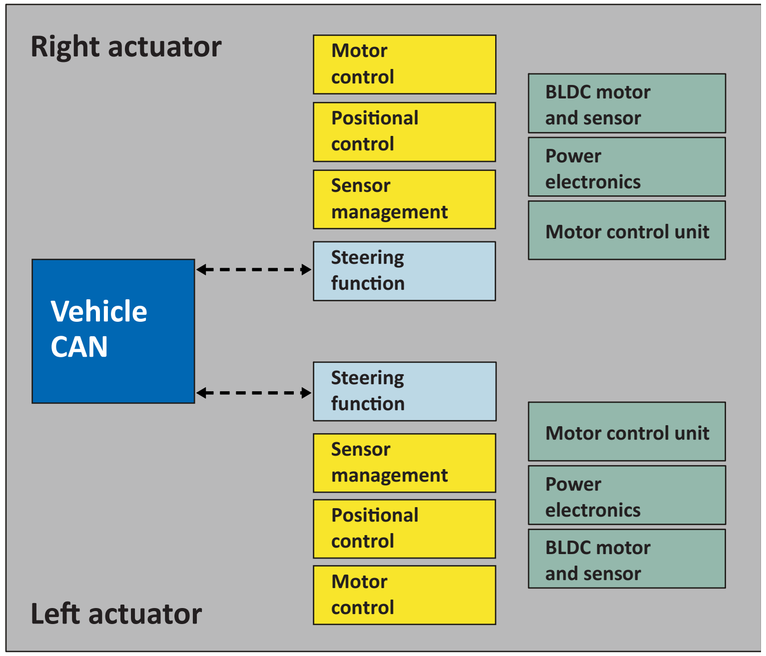

Figure 10 Design of the ARK control unit

26

The development of an add-on control unit for volume production is a further focus of rear axle steering development. Figure 10 shows the concept being pursued by Schaeffler for integrating the control unit arrangement in the OEM vehicle architecture.

The low-level software is applied to the Schaeffler control unit and specially matched to the BLDC motor developed by Schaeffler. For the high-level software of the vehicle controller, the customer can use both his/her own software and the vehicle controller developed by Schaeffler. This can be easily parameterized via a user interface and, therefore, quickly adapted to different customer requirements. The high-level software can be applied to the Schaeffler control unit and also to an existing customer control unit if suitable.

Roll stabilization

Passive stabilizers, usually in the form of torsion bar springs, are used to reduce sideways tilting of the body. These should not be selected with a high torsional rigidity because this will, otherwise, lead to a strong vertical movement of the body, so-called "duplicating", when a vehicle is driving over obstacles (interference) on one side.

To solve this conflict and also further reduce the roll angle, hydraulically actuated pivot actuators are used nowadays which twist a divided torsion bar depending on the transverse acceleration when a vehicle is travelling round bends, and thus noticeably reduce the tilting of the body and significantly equalize the wheel contact forces (Figure 11). When a vehicle is travelling straight ahead in the presence of interference on one side, the stabilizer must have an "open" effect which represents a gain in comfort in comparison with a passive stabilizer. Alternatively, tilting of the body can also be prevented by hydraulically adjustable suspension struts on the individual wheels (active body control ABC), which can eliminate the pitching movement when braking and accelerating in addition to the roll movement. Pneumatic systems are not suitable for this because of the high compressibility of the air.

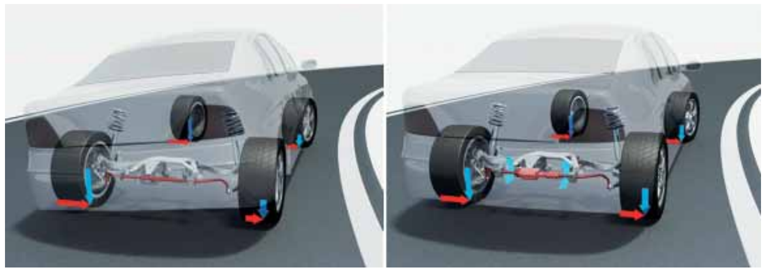

Figure 11 Function of roll stabilization (left without roll stabilization, right without)

Requirements for roll stabilization

The concept of only initiating the actuator in accordance with requirements and at the same time eliminating the hydraulic oil has led to an electric roll stabilizer consisting of an electric motor with ECU and a rotational gear with a high ratio and a compensation module (Figure 12). Without a compensation module to support the electric motor during rotary motion, it is not possible to generate the bracing moment per axle specified in Figure 12 for a mid-range car or SUV with a 12 V system.

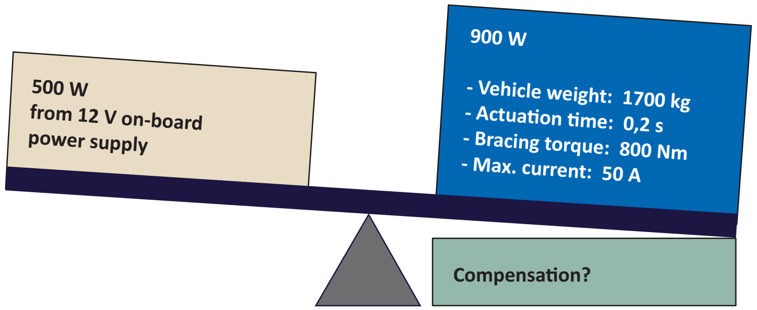

Figure 12 Roll stabilization requirements for an upper, mid-range vehicle

The actuator comprises the electric motor with a flanged planetary gear and the compensation module which is switched in series. The two halves of the stabilizer are fixed to the ends of the pivot actuator by the flow of material. The system-related rigidity that is required for the stabilizer halves necessitates that they are further developed for electromechanical roll stabilization. It is not possible to give a detailed description of this topic due to the complexity.

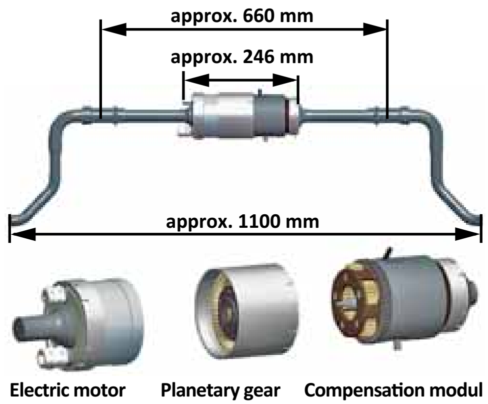

Figure 13 Actuator design with a compensation module for roll stabilization

The function of the compensation module has been thoroughly explained in the conference volume of the Symposium 2006 [11], therefore, the function is only briefly summarized here. The compensation module is a spring-energy store, which as a mechanical actuator supports and relieves the stress on the electric motor, and, therefore, extends the application limits to higher vehicle classes while applying the same load on the on-board power supply.

On the right side of Figure 14, the bracing moment of the stabilizer is entered over the angle of rotation (green line). The area created under it (sum of the green and blue areas) is the work which the actuator must apply. The work performed by the compensation is equal to the area marked in blue. Therefore an electric motor switched in parallel must apply the work represented by the green area which leads to a significant improvement in the energy balance for the actuator.

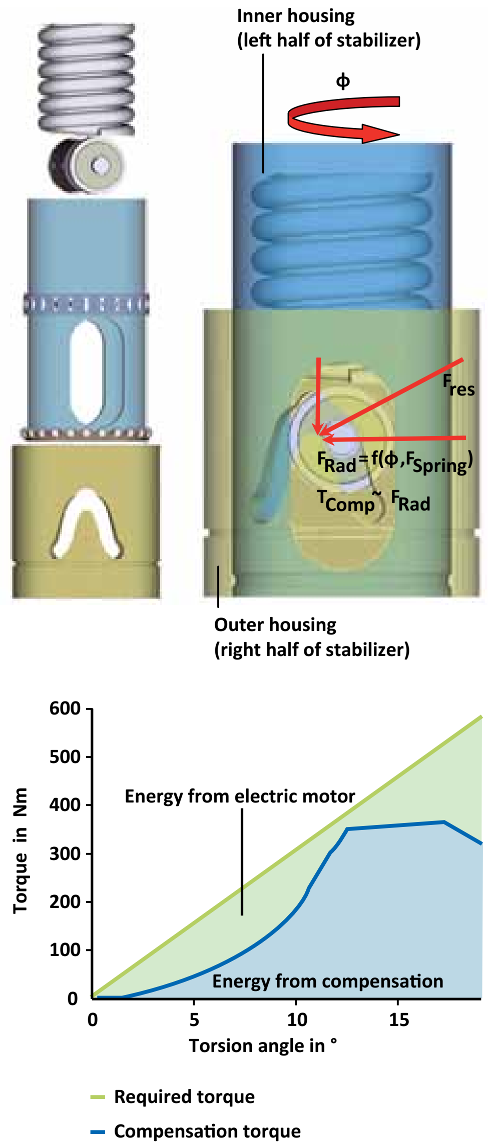

Figure 14 Design of the compensation mechanism and benefits provided by compensation

Results and status of development work

As part of further development, Schaeffler has developed a universal control unit for roll stabilization for phase currents up to 120 A to demonstrate the function of the system from demonstration samples up to A samples in a vehicle (Figure 15).

Figure 15 Schaeffler universal control unit

The control unit is characterized by the features described in Figure 16.

Development of such a control unit was started in collaboration with Continental at the beginning of 2009 because customers are increasingly demanding B samples of control units during development projects. The power features are in accordance with the values of the Schaeffler universal control unit. Development has been focused on the design for an electromechanical steering system (EPS). Experienced gained with the hard and software of Continental control units for EPS has been used during the development. Figure 17 shows the actuator with the add-on control unit. The B sample of the control unit will be available as an add-on control unit for almost all roll stabilization projects from the middle of 2010, and will be successively used in different customer projects.

One objective of further development is improving the comfort. The duplicating behavior is used

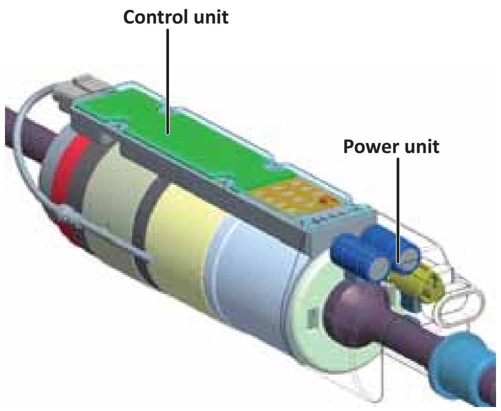

Figure 17 B sample of an add-on control unit

as the comfort parameter and is described as good if the body makes only a small vertical movement as a result of interference on one side. As an effective measure for improving/reducing the interference transmission behavior, the moment of inertia of the actuator was successively reduced by 40 % from generation 1 (Gen1) to generation 3 (Gen3) by means of design measures (Figure 18).

The controller was also optimized in different optimization loops. This means the actuator reacts more sensitively and dynamically to interferences when a vehicle is travelling straight ahead and can, therefore, avoid the interference more easily.

Parameter	12 V	24 V	
Temperature	- 40 ... 95		°C
Phase current (RMS)	75	50	A
Phase current (t < 100ms)	120	80	A
Voltage	9 - 18	16 - 40	V
Min. controller voltage	6 - 9	9 - 16	V
PWM frequency	5, 10, 20		kHz
Current measurement	0 ... 150	0 ... 100	A
Standby power consumption	< 50		µA
Dimensions (l x w x h)	338 x 132 x 56		mm

Figure 16 Features of the Schaeffler universal control unit

At the same time, the power density was increased by 25 % by means of different measures. This was achieved by making improvements to the gearing, the compensation module and the electric motor.

Furthermore, it was possible to decouple interference from the vehicle body by designing elasticity into the pivot bearing support and the actuator bearing support. The pivot bearing connec-

tion can be made more flexible by using a special rubber-metal bearing.

Figure 19 compares the interference transmission behavior of an actuator with a reduced moment of inertia for the generations 1 to generation 3. The interference transmission behavior is quantified using the dynamic rigidity plotted on the vertical axis in Figure 19. The value F_{actual}/z specifies the force acting on the vehicle in a vertical direction in relation to the excitation in this direction. The greater the value, the stiffer the chassis will be for a constant level of excitation.

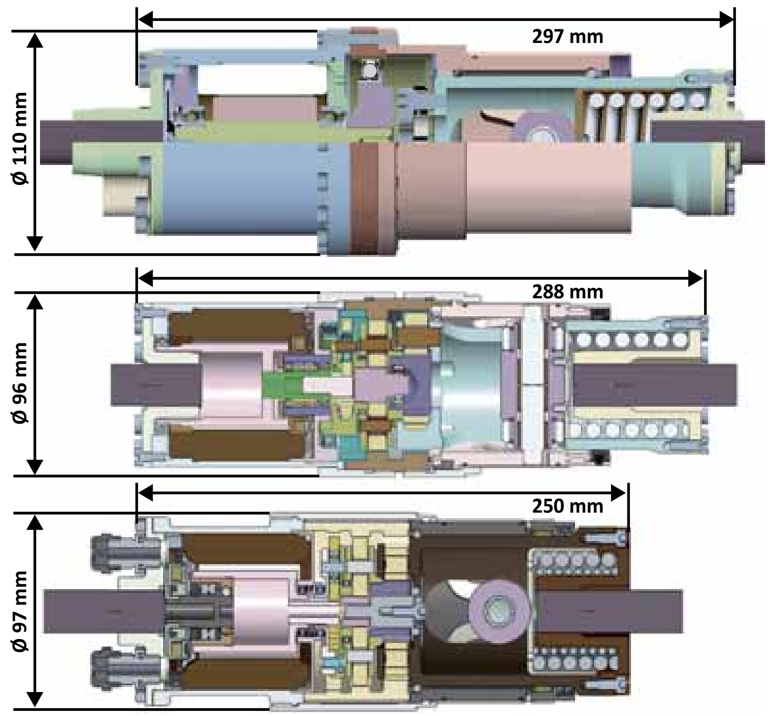

Figure 18 Comparison of different designs showing how the power density has been increased by 25 % while the moment of inertia has been reduced by 40 % (generations 1 to 3 from top to bottom)

There is significantly less interference transmitted to the body and, therefore, the body acceleration is reduced in the range up to approximately 6 Hz which is relevant for vehicle comfort.The influence of the compensation can be seen by the quick response of the actuator compared to an actuator without compensation.

The current customer projects require a range of bracing moments from 800 Nm to 2000 Nm. This requires a modular design and the objective of concentrating on two sizes. Figure 20 shows both sizes of the design.

The required power is matched to the series using the length of the electric motor, the width of the gear unit and the length of the compensation module (this is dependent on the energy to be stored). The B sample of the control unit previously described remains unchanged for all series. This ensures that adaption is based on customer requirements, while maintaining the greatest number of identical parts. This is the basis for a cost-optimized design.

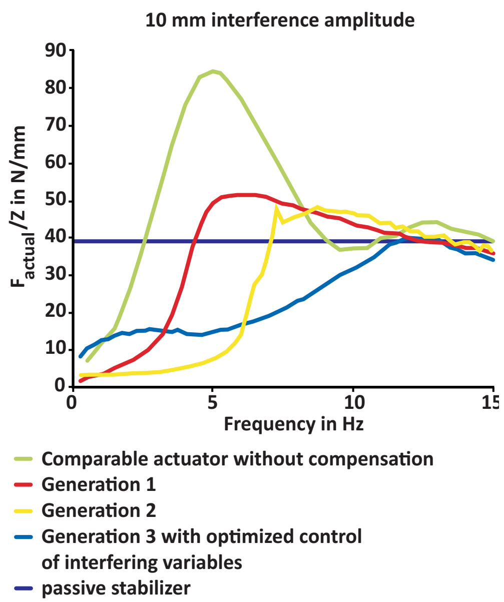

Figure 19 Measurement of the interference transmission for different development statuses of roll stabilization on a test stand

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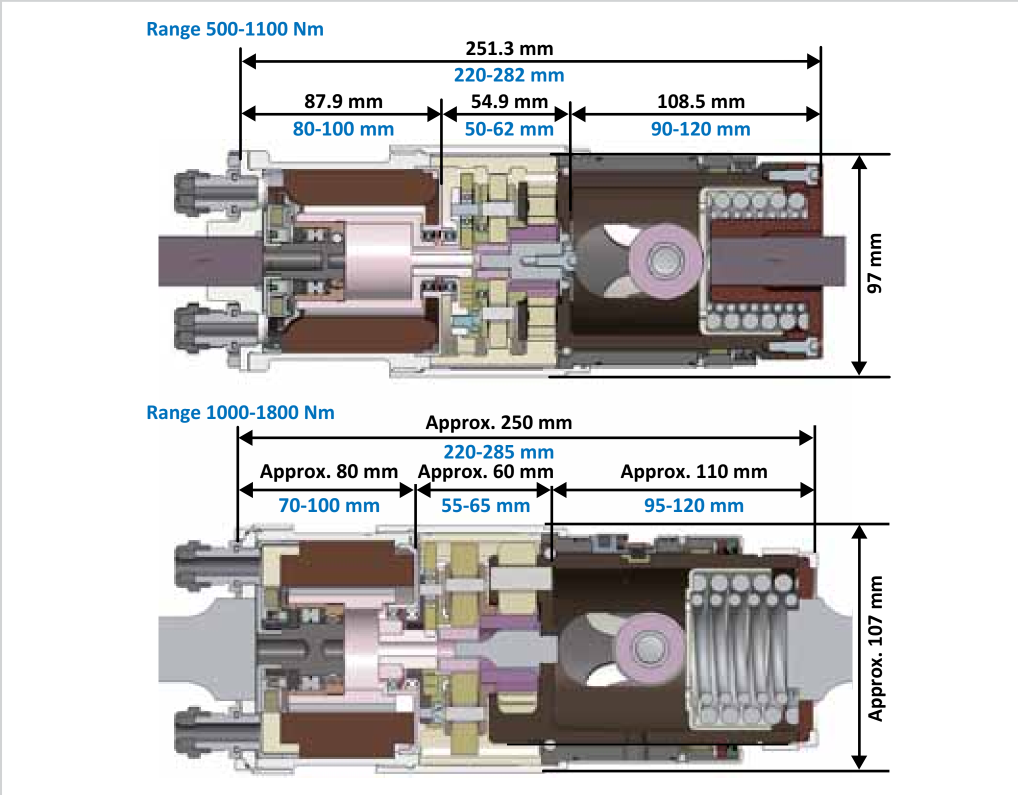


Figure 20 Modular system for roll stabilization

Electromechanical damper (electric damper)

At the beginning of the 1990s, the idea of using the energy lost when damping a vehicle lead to the concept of operating an electric motor (BLDC) as a generator using a ball screw drive to convert the vertical movement of the wheel into a rotational movement of the rotor and, thus, to the regeneration of the damper energy [12].

The electric damper also offers the prerequisites for optimizing damper characteristics that are beyond the scope of the hydraulic system and provides the basis for realizing a semi-active suspension system. It can make a contribution both to improving driving comfort and increasing the performance of driving dynamics. Electric damping has been blighted in the past by the failure of developers who have tried to use linear motors. The intrinsic energy requirements of these motors exceed the damping energy many times and the design space requirements are critical.

The system proposals by Michelin und Bose, which have been presented in the trade press, are certainly well known. But they are complex and/or cannot be integrated in the available design space. Alongside the unfavorable cost-benefit ratio of well-known systems, other requirements such as the overload capability and the response to low excitation has prevented further development in this area. The unfavorable results have lead to electric damping being generally regarded as negative.

Schaeffler is developing a damper which will fit in the available design space of a hydraulic damper, and has a better cost-benefit ratio than existing active dampers, and an improved overload capability. This damper is described in more detail below.

Design and function of the Schaeffler electric damper

The basic configuration of the damper comprises a BLDC motor, a ball screw drive with bearing support and a damper tube (Figure 21). The forces introduced into the chassis follow the flow of force as shown in Figure 21.

The wheel carrier and suspension strut are excited in a vertical direction by the road. This translatory motion is converted into rotary motion in the damper and damped by the electric motor which is operated as a generator. The centrifugal brake is used to brake the rotary motion of the electric motor rotor if large impulses occur.

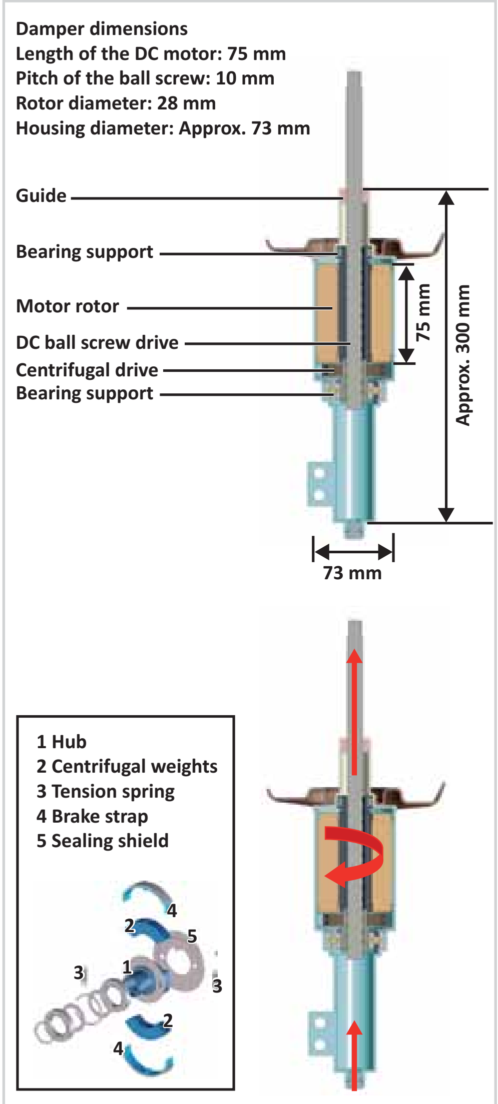


Figure 21 Design of the Schaeffler electric damper

Dimensioning of the electric damper

The electric damper is dimensioned according to the characteristic curve during compression and rebound of a hydraulic damper and the physical limits of the electric motor during generator operation. Compression is indicate by the green line and rebound by the blue line.

The performance range marked in brown in Figure 22 can be covered by an electric damper. If the travel speed of the damper is approximately 1000 mm/s, the electric motor goes into saturation mode and the maximum damper force is approximately 2500 N. Any overloading can be reduced by means of a centrifugal brake developed by Schaeffler.

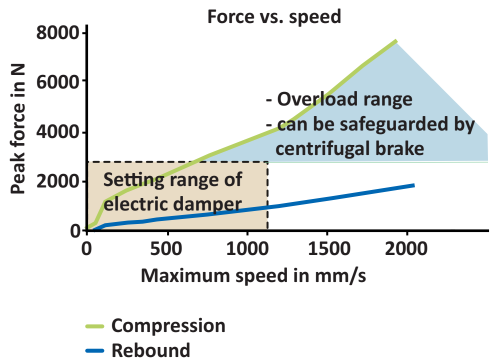
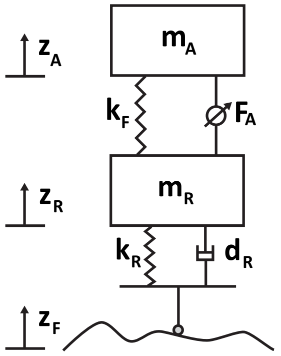


Figure 22 Characteristic curve of a hydraulic damper and the application range of an electric damper

The functional equations for the damper are derived using the quarter-scale model of the vehicle. The damper element is removed from the model and replaced by an electromechanically generated force F_A.

Figure 23 shows the quarter-scale model of the vehicle with the relevant coordinates and symbols for compiling motion equations. The actuator force is due to the behavior and activation of the electromechanical system during generator operation and is a function of the motor voltage.



m_A Quarter of vehicle mass/body mass
m_R Wheel mass

k_F Spring rigidity of the chassis
k_R Wheel stiffness
d_R Wheel damping
F_A Axial force on the thread/actuator force

z_A Deflection of body
z_R Deflection of wheel
z_F Deflection of road

$$m_A \cdot \ddot{z}_A = F_A + k_F \cdot (z_R - z_A)$$

$$m_R \cdot \ddot{z}_R = k_R \cdot (z_F - z_R) + d_R \cdot (\dot{z}_F - \dot{z}_R) - F_A - k_F \cdot (z_R - z_A)$$

$$F_{A(U)} = \frac{2 \cdot \pi}{s} \cdot (M_{(U)} - M_R - J_G \cdot \ddot{\varphi})$$

Figure 23 Forces acting on the damper on the basis of the quarter-scale model of the vehicle

Comparison of the electric damper and a hydraulic damper

The effectiveness of the electric damper must be equal to that of a hydraulic damper. Figure 24 compares the body acceleration and the dynamic wheel load as a function of the wheel frequency. This comparison shows a good correlation between both damping designs. The electric damper with compensation has an even better correlation with the hydraulic damper. The compensation eliminates the slightly higher moment of inertia of the electric damper by means of a pilot control developed by Schaeffler. This means that the same damping behavior can be achieved as with a hydraulic damper [10].

Energy benefits of an electric damper

Figure 25 shows the power generated by the electric motor as a function of the damper speed and the load. The information in this diagram can be summarized very simply: The poorer the road, the greater the energy gains. The damper force and speed increase as the road gets worse.

The generated performance is limited to 3500 N due to the design and dimensions of the motor. Damper speeds of over 0.15 m/s occur infrequently and in conjunction with very large unauthorized loads. According to information provided by the Federal Motor Transport Authority, 80 % of German roads are in a condition which results in the damper forces and speeds marked in blue and light blue in Figure 25. This means that a maximum energy gain per wheel of 50 to 100 W can be assumed in Germany. The evaluation of this saving in relation to the price per piece must be conducted on the basis of future CO_2 taxes and increasing fuel prices. A comparison with roads outside of Germany, which have worse road conditions in some cases, results in an improved basis for electric damper technology.

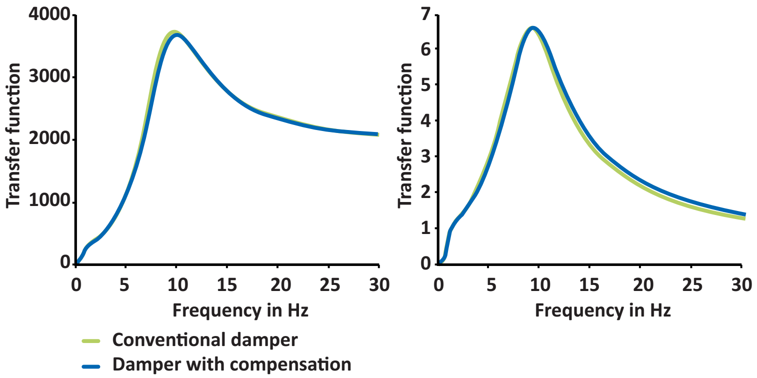


Figure 24 Comparison between the electric damper and a hydraulic damper

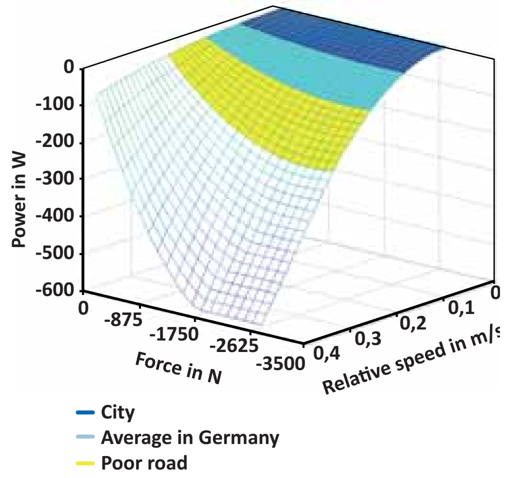


Figure 25 Generated power as a function of damper force and speed

Conclusion

To ensure a distinction compared with competitor products, Schaeffler is developing different actuator principles which go beyond the approach comprising an electric motor, sensor and self-locking gear unit. This also includes the development of advanced solutions with an improved energy balance. By using the breadth of knowledge and experience available within the Schaeffler Group, it is possible to develop solutions which are advanced with regard to functionality, energy balance, design space and power density compared with the current state of the art. The development and test status of rear axle steering, roll stabilization and automatic level control corresponds to an A sample.

The Schaeffler Group can use its well-developed expertise in manufacturing technology, especially the experience gained in the volume production of ball screw drives for EPS, for successfully and quickly implementing volume production. Volume production of the ARK system will be implemented initially within the next few years, followed later by volume production of the roll stabilization system.

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