

High Performance Electromechanical Actuator for Active Rear Axle Kinematics of a Sports Car

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ABSTRACT

An electromechanical actuator is presented for the active control of a sports car's rear axle kinematics with a high performance-to-size ratio not available on the market up to now. Toe and camber values of both rear wheels are controlled independently to reach the optimum position of the tire contact patch on the ground for each driving situation. The complete system utilizes 4 actuators and is known as Active Kinematics Suspension (AKS).

The actuators substitute the camber and toe links in a multilink suspension, without major modifications, on the prototype of an existing car. The actuators are capable of covering the high axial forces seen in the links and can rapidly alter the wheel position. The heart of the actuator is a hollow-shaft brushless DC motor running efficiently in field control. It drives a recirculating ball-screw which transfers the motor's rotational motion to a translational displacement of a central rod.

Packaging and manufacturing checks are performed in a 3D CAD environment using kinematic and tolerance analysis tools. Sizing of the detail parts are analyzed with the help of non-linear FEM methods to ensure geometric stability and high durability of the complete system.

A first set of four actuators are tested on a 6-DOF bench, incorporating a full rear axle suspension system before installation in the prototype vehicle and final road tests.

CITATION: Ramirez Ruiz, I., "High Performance Electromechanical Actuator for Active Rear Axle Kinematics of a Sports Car," *SAE Int. J. Passeng. Cars - Electron. Electr. Syst.* 5(2):2012, doi:10.4271/2012-01-0974.

INTRODUCTION

Because of the trend of the sports car market towards "smart" electromechanical devices to further improve performance, the actuator in the AKS system was to be a proof of concept of electrical power in highly loaded actuators. However, the design brief was also to produce something very close to production ready, with minimal changes required to the actuator design itself, but also to the vehicle structure to which it interfaces.

Initial simulations and discussion within the company's vehicle dynamics department yielded a set of specifications of an actuator to give acceptable control over the camber and toe angles such that the vehicle dynamics can meet a series of targets. The specifications arrived at were:

- Dynamic camber variation in the range of $+2/-6$ degrees
- Dynamic toe variation of $+/-2$ deg
- Force: Design limits up to 12kN, Misuse up to 20kN without catastrophic failure
- Velocity: up to 300mm/s

- Dynamic response: 80% of maximum speed in 100ms under limit loadcase
- Device shall integrate into an existing multi link suspension without major modification to the kinematic hardpoints
- Must incorporate failsafe hardware to stop the actuator in the case of an emergency or blackout
- Device shall survive the standard durability tests including misuse without failure

ACTIVE KINEMATICS IMPLEMENTATION

The active toe and camber characteristics desired were able to be realised using two actuators per wheel in a sophisticated multilink rear axle which is the test vehicle's current standard rear axle. This particular multilink implementation is perfectly suited to the active kinematics proposition because the axle restraint method. This is realised in such a way that the camber and toe angles can be controlled by replacing simple tension/compression members with a linear actuator to give almost completely independent

control of toe and camber. Figure 1 shows the implementation of the camber and toe actuators into the test vehicle.

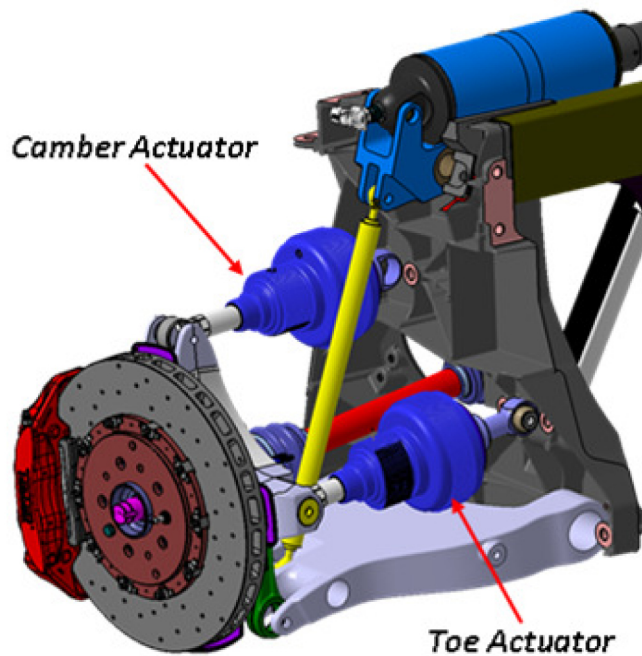


Figure 1. Active Kinematics Suspension implemented in the test vehicle rear axle

The packaging of the actuators into the rear axle posed significant challenges and it became quite clear early on that the power density (power per unit volume) of this device will have to be at the top end of what is possible with current technology because of the small space available.

An additional feature of the prototype is that the toe and the camber actuator comprise of many common parts. All of the main functional systems in both actuators have been unified into a single “AKS-Common” assembly. The common assembly is completely identical for both camber and toe actuators and contains everything except the inboard bearing mounting which is the interface between the actuator and the chassis.

This additional requirement meant that the packaging of the common assembly was very restricted as shown in Figure 2. On the outboard (left) side of the actuator, the wheel and tyre come in very close proximity to the camber actuator in some limit conditions of actuator travel. Similarly, on the inboard (right) side, the packaging limits are dictated by the chassis near the toe actuator. With active kinematics, the wheel has a large “envelope” of operating positions which encroach into the design space of the actuator itself. Careful location and bespoke designs of the main elements, allowed the system to be realised in the space available.

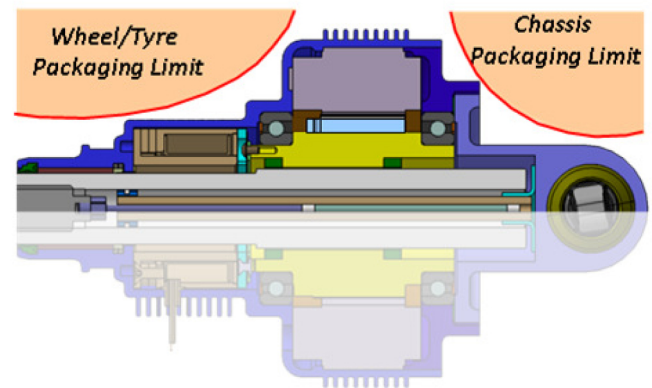


Figure 2. Actuator packaging constraints

MARKET RESEARCH

With the implementation concept set, an initial set of simulations using an ideal actuator were performed to arrive at the force and velocity requirements outlined in the introduction. The high requirements for speed and force and also the dynamic response have placed large demands on the performance of the actuators to be used in the AKS system. Several manufacturers of similar actuator were approached with these requirements but it became clear very quickly that there is nothing with these requirements currently available. Various proposals presented to or otherwise considered for the AKS actuators were far below the performance requirements in terms of force and speed.

Until now, most electrical actuators in the suspension have been for steering which have very low force and stroke requirements. The camber link in the suspension (which is where the camber actuator is to be) has a significant step up in the forces required to actuate the suspension to the required specifications. In particular, the dynamic response requirement meant that the actuator needed a large force reserve to not only react the high forces coming from the contact patch, but also accelerate the complete hub assembly very rapidly in transient situations.

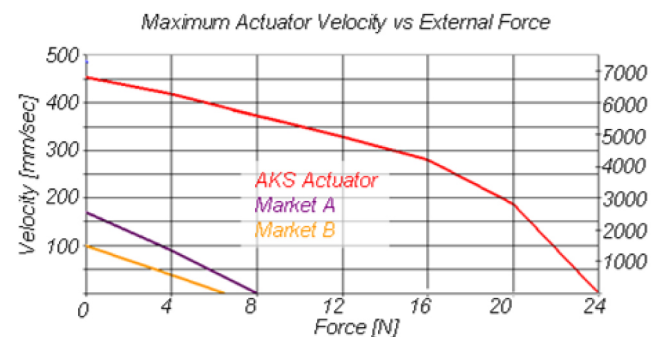


Figure 3. Force/Speed performance of AKS actuators compared to market offerings

Figure 3 shows the simulated force-speed characteristics of the actuators designed for the multilink rear axle against 3 devices proposed by various suppliers. It can be seen that there is a large delta of force and speed capabilities of the AKS suspension as compared to the current market offerings.

DESIGN PROCESS

Drive Method

Using a completely clean sheet, top-down approach to solving the active kinematic problem, the first issue to address was selection of the drive method. Because of the location and high specifications for force and velocity, the drive must be very powerful. Initial simulations indicated a required peak power of 2kW per actuator to satisfy the wheel movement specifications. The drive methods considered were electric and hydraulic. Further reading into the philosophy of this choice has shown that there is a general trend away from hydraulics towards electric power as the technology involved with electric motors and their associated controllers become more efficient, cheaper and more powerful. Additionally, the trend towards fully electric vehicles meant that a high powered electric source will be readily available in production vehicles to power the actuators.

The drive chosen was a brushless DC motor developed specially for the torque-speed requirements of the actuator. Given the packaging constraints, the motor and housing design was largely governed by thermal limitations due to the high power required from a comparatively small volume.

With the motor design starting as a clean sheet, it was decided that the system shall be of a direct-drive nature. That is without the use of a gearbox. The removal of the gearbox greatly reduced the complexity of the system as well as eliminating extra sources of backlash and friction. This means that the only drive ratio in the system is the conversion from rotational to linear movement. The main drive ratio is 4mm/rev. Figure 4 shows the actuator internal layout with the motor and ball screw in the large diameter section of the housing.

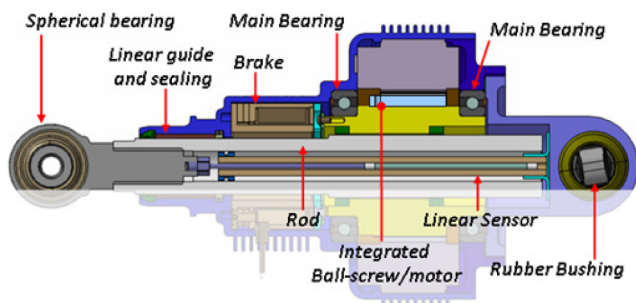


Figure 4. Actuator System Overview

Because of the location of the actuators in the suspension, they are always under load due to the vehicle mass, so the motor required a significant continuous torque rating as it will be operating all of the time. When extra cornering and

misuse forces (up to 20kN) are taken into account, the motor is required to output torques of up to 20Nm in various conditions. With the specification of 200mm/s of operating speed and a ratio of 4mm/rev, the motor is required to be still giving useable torque at up to 3000RPM. With the torque and speed requirements defined, it was quickly realised that there were no commercially available motors which could match the requirements.

The result was a custom designed motor: 16 poles, 100VDC and 1.5kW giving up to 20Nm of peak torque and 3.6Nm constant torque at speeds of 3000RPM. The motor itself is a length of only 50mm and 127mm in diameter.

Actuation Method

With electrical power selected as a power source in the form of a brushless DC motor, the task was then to convert the rotational movement to the required linear movement. Two actuation methods were considered in detail; recirculating ball screws, and planetary roller screws. The decision to use a recirculating ball screw was made because it was seen that the ball screw was able to offer much higher load ratings for a given packaging space.

The ball-screw used in the actuator comprises of a very low profile nut around a 32mm diameter moving rod. The packaging was further improved by working with the motor and ball-screw supplier together to come to a solution requiring the minimum space possible between the bearings. Figure 5 shows the resulting motor/ball screw assembly.

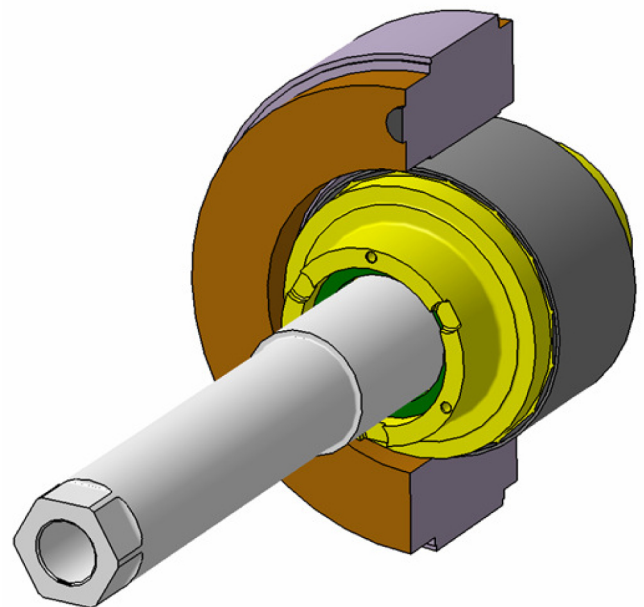


Figure 5. Integrated motor and ball-screw solution

The rotating assembly is completed by two bespoke slimline angular contact ball bearings placed coaxially over the ball screw and sitting directly adjacent to the motor to save axial space. The bearings themselves are a square

section with a modified contact angle to run at the high axial loads and speeds required in the actuator. All of the rotating elements are designed with a slight preload to ensure no free play. Any free play in the axial direction can cause control and stability problems.

Electrical and Electronics Systems

Because of the high power demands of the actuators, considerable design effort has gone into the electrical power supply system. This consists of a power supply system, a motor drive and a controller. The architecture for the complete system is shown in Figure 6.

For this prototype, a unified voltage for the motor and brake of 100VDC has been chosen as the supply voltage. This is obviously higher than a typical vehicle supply voltage of 12V and is implemented using a step up system of a DCDC converter and super-capacitor to ensure sufficient reserve of energy for high current peak demands without affecting the vehicle's 12V supply system.

The hardware for the motor drive consists of a custom inverter and commutation controller which operates with a field-oriented control (FOC). The vehicle dynamics controller then interfaces with the FOC to control the motor based on vehicle dynamics sensors and other internal sensors of the active rear axle.

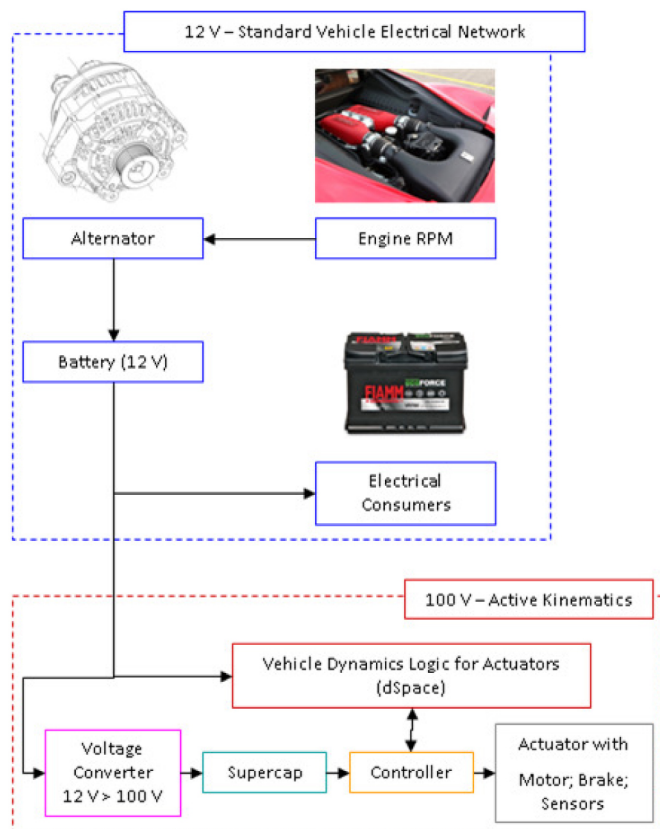


Figure 6. Electrical system for the active rear axle

Safety Systems

With the absence of an overriding mechanical link between the driver and the rear wheels, the failure conditions of the actuators were something closely investigated. The primary safety device in the actuator is a power off permanent magnet brake which is capable of supplying enough torque to stop the motor under emergency power off conditions. The brake can also operate in a power on mode whereby forcing a reverse current through the coils, it can provide even higher torques at a lower response time as compared to its nominal power-off specifications. This strategy is used in the non-black out emergency routines to stop the actuator as fast as possible.

The brake itself required a careful selection of concept, again due to the tight packaging constraints in the area where the brake sits. Concepts considered for the brake included:

- Permanent magnet or spring applied?
- Toothed or sliding interface?

The failsafe requirement of the braking system clearly required a power-off brake to allow operation in the event of a complete blackout during operation. With the operating speeds up to 3000RPM, a toothed interface of the friction surfaces was not acceptable because application at high speeds can see the brake "skipping" teeth and damaging the braking surfaces. The final choice of whether to use a permanent magnet or a spring applied brake was not so obvious. Further investigation into the different technologies showed that permanent magnet type brakes had a higher power density than the spring applied type which meant that more braking power was able to be produced in the small packaging space available.

The prototype actuator contains a semi redundant dual position sensing system to allow a safe return to the static position in the case of a failure within the sensing system. The sensing system comprises of one absolute and one incremental position sensor working together to provide parallel position information under normal working conditions. The sensing system, like the rest of the actuator is subjected to large transverse accelerations from the vertical wheel movement and it was critical to ensure that the sensors do not fail, or provide erroneous readings due to the vibrations.

The absolute position information came from the absolute position sensor in the form of a linear variable differential transformer (LVDT) that is placed coaxially inside the moving rod. Due to the packaging constraints, the sensor is only able to be fixed on one side, so a series of sliding guides are used inside the rod and the sensor itself to ensure resistance to transverse vibrations. The precision of this sensor is increased by combining it a rotary encoder which is synchronised to the absolute position read by the LVDT at power-on. In the event of failure of either sensor, the other