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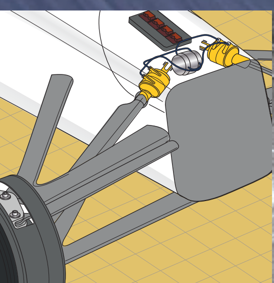
Return of the snake

Viper gets ESC but stays true to its V10, rear-drive roots



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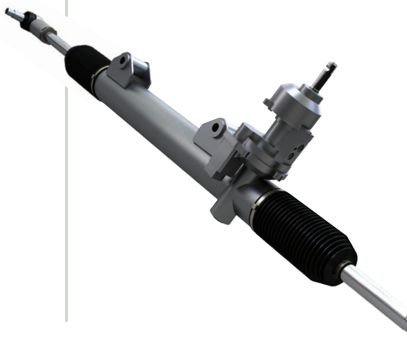

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A NOTE FROM THE EDITOR

Forgive us if we sound like a broken record, but does anybody really care anymore whether vehicles are comfortable to drive? Apparently all that car buyers are interested in – in Europe, at least – is better fuel economy/lower CO_2 emissions (hello, feel-free steering and low-rolling-resistance rubber), concept car-like styling (oversized wheels), 'sporty' handling (a 'Sport' steering setting on a Kia Cee'd?), and endless forms of personalization (stripey cars handle better, naturally).

It strikes me that there are two factors at work here. Legislation aimed at reducing tailpipe emissions is, politics aside, broadly well intentioned. I'm not sure that anyone outside of a James Bond film is keen on destroying the planet, so I guess we'll have to sit tight until EPS systems recover the tactile feel of the best hydraulic setups (the signs are promising) and tire engineers find a way to add more grip to low-CO_2-friendly rubber (likewise).

The second factor is less altruistic. In our experience, larger wheels, 'Sport' buttons, and contrasting roof colors generally add little to everyday cars beyond an unsettled ride and marketing ammunition – and higher profit margins. The latter 'benefit' is surely the most telling in a tough European market and therefore, in its own way, also hard to dispute. Perhaps when sales pick up we can all go back to proper steering tuning and sensible sidewall heights...

Graham Heeps

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RACHEL EVANS
VDI's assistant editor took a trip to Padua, Italy, for a look behind the scenes at the new Bhai Tech Advanced Vehicle Science Centre (p42). The company has drawn on years of race engineering experience to develop new vehicle dynamics software


THOMAS MIRWALDT
Porsche engineer Thomas has written for us on his PhD subject: controller optimization of active anti-roll systems (p24). His co-author here is also his supervisor at the University of Stuttgart, Professor Peter Eberhard


JOE WALTER
A highly respected consultant in the tire development community, Bridgestone's former head of European R&D is also a regular contributor to *VDI*'s sister title, *Tire Technology International*. Here he remembers the late, great Bill Milliken (p58)


MATT YOUSON
Matt follows the Formula 1 circus around the globe, which gives him great access to the sport's top technical minds. In this issue he uses Lotus's stillborn reactive ride-height system as the springboard for a look at banned suspension technology (p48)


PAUL ZANDBERGEN
Together with Wolfgang David and Ed Knoy, Paul has written a fascinating account of the development of the new Ford Mondeo's 'integral link' rear suspension (p18). The team has sought to deliver improved ride comfort over an already class-leading setup

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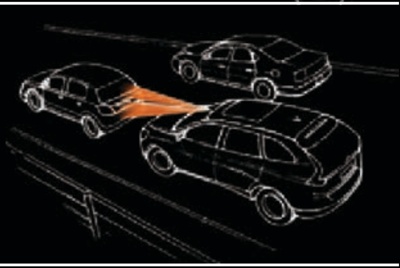

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Vipeout!

MARC NOORDELOOS GOES UNDER THE SKIN OF THE BORN-AGAIN SRT VIPER AND TRIES A PRE-PRODUCTION PROTOTYPE ON TRACK AT GINGERMAN RACEWAY

THE OVERALL LENGTH OF THE 2013MY VIPER IS 4,463MM; WIDTH IS 1,941MM. WHEELBASE MEASURES 2,509MM AND WEIGHT DISTRIBUTION IS NEAR 50/50

"I told the team to make a 640-horsepower Miata (Mazda MX-5)." Ralph Gilles – president and CEO for SRT Brand and Motorsports, as well as senior vice president of product design for Chrysler – answered us with that unique line when we asked him what the goal was with the new Viper. Chrysler's halo sports car has never been even remotely lithe like the Japanese roadster. In fact, we're quite sure no one has ever put the V10-powered Viper and the MX-5 in the same sentence before those words came out of Mr Gilles' mouth.

The original Viper concept arrived in 1989 as a simple, powerful, and extroverted halo car for Chrysler. It has never featured an overly friendly chassis beneath the wild exterior design. The basic details of the car haven't changed dramatically as this fifth-generation Viper comes to market in early 2013. It's still left-hand drive only, with a giant V10 that sends its power – now 631bhp from 8.4-liters – to gigantic rear tires solely through a manual gearbox.

The space frame continues on with a boxed steel structure, albeit heavily revised. SRT engineers told us that Viper owners track-drive their cars and drive them hard. Exotic technology makes repair far too expensive after the occasional shunt. A new magnesium bulkhead and aluminum engine x-brace contribute to a 50% stiffer structure. Carbon fiber for the hood, roof, and decklid, combined with aluminum door skins, help drop the Viper's curb weight by 45kg compared with the last version.

Bolted to this revised structure is a suspension system with aluminum A arms front and rear. The setup is similar to the fourth-generation car, so much so that German-sourced KW threaded coil-over dampers from the old track-focused Viper ACR bolt right up to the new car. Not that you necessarily need that modification given the improvements in this area of the Viper. SRT worked with Bilstein for the two damper options. The base Viper features non-adjustable aluminum-body shocks. The top-spec GTS model uses gas-charged monotube DampTronic Select dampers with 46mm pistons. These units feature two specific modes via a button on the center console. The road setting is softer than the base car's tuning and the track setting is firmer. The GTS combines this with 10% stiffer springs. The 27mm sway bars on the Viper are hollow and the rear suspension has been fettled for improved rear-toe compliance.

The steering system continues to be a fully hydraulic setup. It utilizes a 16.7:1 ratio, with 2.4 turns lock-to-lock. The increased structural rigidity of the space frame and the retuned steering combine to give excellent feel and accuracy, something lacking in previous-generation Vipers.

Strong brakes are always important on a 631bhp sports car that can break 200mph. Viper engineers worked with Brembo as well as StopTech in California. The four-piston – 44 and 40mm – Brembo calipers are forged aluminum. All Vipers feature 355 x 32mm rotors front and rear, but the optional SRT Track Pack adds StopTech two-piece slotted rotors that offer increased performance and lighter weight. Our time in a pair of near production-ready development Vipers at Gingerman Raceway in South Haven, Michigan, USA, revealed an overall impressive braking system, but pedal feel and performance did degrade a touch with heavy use – even with the SRT Track Pack rotors.

Putting all this power to the ground and allowing the newly refined chassis to work to its full potential meant that engineers had to revisit the wheel and tire package. The newest Viper utilizes Pirelli tires for the first time. The fourth-generation Viper had bespoke Michelin tires – either Pilot Sport PS2 or Pilot Sport Cup. The new standard tire is the Pirelli P Zero. It's a non-runflat design, with 275/35ZR-18 installed up front and a mammoth 355/30ZR-19 in the back. The SRT Track Pack adds ultra lightweight

"The basic details of the car haven't changed dramatically. The Viper is still left-hand drive only, with a giant V10 that sends its power – now 631bhp from 8.4-liters – to gigantic rear tires"

NEWS-IN-BRIEF

Bridgestone has announced a partnership with Mazda to supply the Mazda 6/Atenza models with its latest Turanza T001 tire. Bridgestone claims that the Turanza T001 has been designed for improved fuel efficiency as well as providing increased traction in both wet and dry. Bridgestone claims that these are elements well suited to the sporty sedan.

Hankook continues to expand its European R&D program with the recent inauguration of new facilities in Hanover, Germany. The European Technical Center (ETC) has been continually expanded since its inception in 1997, and this latest development "offers new possibilities and clearly signals a long-term investment" according to Stefan Fischer, head of ETC at Hankook.

The extensive overhaul of MIRA proving ground's UK facilities has taken a step forward with the completion of the first building of the new site. The 43,000ft² building represents a £6 million investment and includes workshop space and a new control center.

track wheels fitted with the circuit-oriented P Zero Corsa tire. Both tires performed flawlessly during extended lapping sessions at Gingerman, but the Corsa tire added a welcome degree of steering precision and front-end grip. Power is managed through a GKN ViscoLok speed-sensing limited-slip differential and it does an excellent job, giving the Viper impressive traction.

Possibly the biggest advancement in the new Viper is the fitment of stability control for the first time. Viper engineers could have gone down the easy route and added a simple system, just to keep the regulatory bodies happy. Instead, they developed a system that allows owners to have fun with their cars, even with it fully turned on.

The base car's stability control system features two modes – fully on or fully off. A simple button on the steering enables the driver to quickly toggle between the two. The GTS adds two additional settings, a sport mode with a higher threshold before intervention, and a track mode that disables the yaw control but keeps wheel spin in check. We found all three of the 'on' settings to be excellent and even the most stringent default mode was very unobtrusive. It's a very impressive system, especially considering it's a first for Viper.

VDI SAYS

As you can see, Chrysler's newest Viper offers little groundbreaking advancement. It's a perfect example of taking a product with known technology and massaging it into something much better. The balance and purity of the newest Viper is a breath of fresh air in an automotive world that produces more and more cars each year that lack a basic, visceral dynamic personality in the chassis. Ralph Gilles was thankfully onto something when he talked about the MX-5 and the Viper in the same sentence.

SNAKES ON TRACK

In a move that will very likely lead the Viper team back to the Le Mans 24 Hours in 2014, SRT commissioned a pair of GTS-R race cars for the 2012 American Le Mans Series (ALMS), *writes Jim McCraw*. The silver machines were built by Riley Technologies and are run by Riley Motorsports, with heavily restricted Roush engines.

ACO and IMSA granted the Viper a couple of significant waivers, one being a new Xtrac transmission located on the rear axle, which necessitated some slight modifications to the standard Viper frame. The other was the use of an 8-liter V10 engine in a series where the engine displacement is restricted to 5.5 liters.

The 1,294kg Riley race cars are built to the new ACO/ALMS wide-body rules with a maximum vehicle width of 2,050mm, and a maximum body extension of 150mm, so the big, wide Viper GTS-R body was redesigned for the rules with an eye toward aero performance. All of the suspension control arms of the Viper's short/long arm suspension are longer than stock, located on or near the standard mounting points, within 20mm of original locations, according to the new rules. Brake calipers are enlarged Brembo 380mm six-piston fronts with 330mm four-piston rears, on Riley hubs and uprights mounting Michelin 30/68-18 front and 31/71-18 rear tires on TWS forged alloy wheels.

Lead engineer Matt Bejnarowicz says the Riley team has been running both cars on Penske aluminum remote-reservoir shock absorbers, because they are proven and widely adjustable. However JRI through-rod dampers have been trialed in private testing.

TOP: **OPTIONS FOR TRACK USE INCLUDE KW SPRING-DAMPER UNITS**

RIGHT: **VIPER GTS-R RACE CARS WILL COMPLETE A FULL ALMS SEASON IN 2013, WITH A RETURN TO LE MANS PLANNED FOR THE FOLLOWING YEAR**

Stability packed

HONDA'S CR-V IS THE USA'S BEST-SELLING SUV. THE NEW, FOURTH-GENERATION CAR PUTS CLEAR WATER BETWEEN EU AND US VERSIONS. BY **GRAHAM HEEPS**

When the first Honda CR-V appeared back in 1995 it was a soft-roader pioneer, helping define a segment that has since moved from niche to mainstream. With pretty much every major manufacturer now competing in the compact SUV sector, Honda faces its toughest challenge yet in building on the five million CR-V sales it has notched up to date.

The CR-V is sold in more than 160 countries and built in seven of them, but in development terms the latest, fourth-generation car came down to two key versions. With the USA as the car's biggest market, the North American variant – also sold everywhere but Europe – took the lead. Around 80% of that car was carried across into a dedicated European version, where market conditions dictate that the SUV is sold with more equipment, different chassis settings, and a higher price. The move to increased differentiation represents a return to the philosophy behind the second-generation CR-V, as deputy development leader and dynamics leader for the car, Akihiko Mori, explains.

"When we developed the third-generation car, we tried to unify all the good parts and offer one solution to the world," he says. "But for the fourth generation we acknowledged that it's not really feasible, especially when you think about Europe, where driving speeds are much higher and you have very different road situations to the USA. Therefore we have again separated into one specification for Europe, and one for the rest of the world.

"That decision required careful consideration, but once it was done, we looked at competitors [to benchmark] in the same category in Europe. Certainly a competitor we continued to look at during development, including for dynamics, was the Volkswagen Tiguan," says Mori.

Putting more distance between the US and European chassis setups naturally had cost implications.

"The feedback [from the outgoing car] was that the Americans wanted a softer, gentler CR-V, whereas the Europeans wanted a car that was stiffer and more stable," Mori explains. "Of course, it costs more money to have individual settings, but I'm very proud of achieving the best settings for both regions. Hopefully the customers will be satisfied that we haven't compromised between the two."

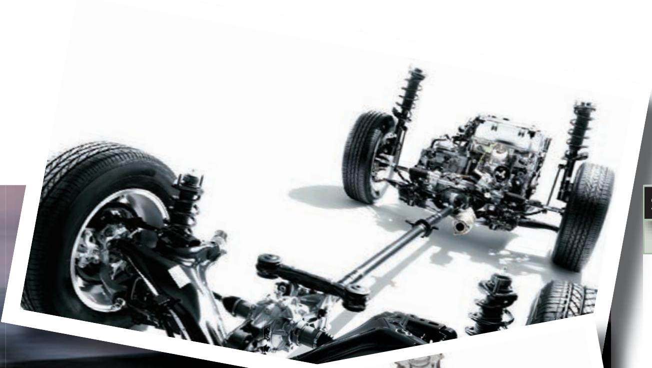
In particular, Mori speaks of wanting to increase the CR-V's high-speed stability and "flat feel",

NEWS-IN-BRIEF

Maplesoft has introduced a new release of MapleSim. With tighter Modelica integration, as well as more simulation, analysis, and connectivity capabilities, MapleSim 6 offers even more ways for engineers to meet and exceed their system-level requirements.

The Scania Group has selected HEEDS MDO, the Multidisciplinary Design Optimization Software from Red Cedar Technology, for vehicle dynamics optimization. Anders Ahlström, senior analyst, Scania, said, "HEEDS helps us solve problems faster, better, and easier than we were able to do alone. This software surpasses anything on the market in its ability to help us drive innovation."

Nissan's steer-by-wire technology, to debut on selected Infiniti models within a year, allows independent control of a vehicle's tire angle. The system reads the driver's intentions from steering inputs and controls the tire movements via electronic signals. A regular mechanical steering connection is only used in extremis. At all other times, multiple ECUs ensure that, in the event of one ECU malfunctioning, another will instantly take control.



HONDA'S CR-V HAS A MORE CAR-LIKE DRIVING POSITION AND 30% BETTER STEERING-WHEEL REACH/ RAKE ADJUSTMENT

to improve comfort and driver confidence. Rear dampers with 10% more volume gave the chassis team increased tuning freedom, while for the European car, the coil springs were stiffened by around 10% over the US version (see Table 1). He adds that, when combined with a softening of the rear anti-roll bar, this improves camber control and therefore rear-axle stability, without majorly impacting on ride comfort.

The base chassis settings were done in Japan, then brought to Europe, where local R&D teams in the UK, and in Germany, evaluated and confirmed them under real European driving conditions, and fed back to Tochigi. Note that the largest possible wheel size for each market – 17in for the USA, 18in for Europe – was selected for the base tuning, to ensure that the suspension could handle the worst-case load scenarios.

The CR-V retains a similar MacPherson strut/multilink rear suspension to the outgoing model but is "completely different to the Civic", according to Mori. At the front, the suspension's dimensions and the geometry have been slightly modified to accommodate the new EPS motor.

At the rear, he maintains that the suspension design was not influenced by the cargo floor height, even though it's 30mm lower on the new car. The main change here is the lower attachment point to accommodate the aforementioned greater damper volume.

Other minor chassis hardware changes were dictated by the 18in summer tire option on the European version. Aside from a slightly stiffer front spring rate, the bigger wheel demanded that the front brake caliper size be increased by 1in.

"The combination of the 18in wheel with the diesel engine makes the car quite front-heavy," Mori explains. "It wasn't necessarily to reduce the braking distance, but it improved the pedal feel."

The CR-V is also fitted with numerous ADAS systems. Mori's team did the base settings on the proving ground next to the R&D center in Tochigi; fine-tuning was carried out by Mori himself under real road conditions in Europe. US cars only get ACC; cost and market reasons preclude further fitments.

SPECIFICATIONS

2012MY Honda CR-V 2.2-liter i-DTEC

Dimensions: 4,570mm (L) x 1,820mm (W) x 1,685mm (H, inc. aerial)
Wheelbase: 2,630mm. Track: 1,570mm (F), 1,580mm (R)

Dry weight: 1,653kg

Suspension: MacPherson strut front, multilink rear. Showa dampers

Brakes: Nissin, Bosch, 315mm front discs (ventilated); rears 302mm (solid)

Steering: NSK EPS. Ratio 16.8:1 (10% slower than outgoing car). Turning circle at body 11.8m; 3.16 turns lock-to-lock

Tires: 225/65 R17 or 225/60 R18

Table 1: CR-V spring rates

	EU		US model
	Diesel	Petrol	Petrol
Front	37N/mm	35N/mm	33N/mm
Rear	73N/mm	73N/mm	71N/mm

VDI SAYS

Driving a top-spec CR-V on German roads flattered its secondary ride, but this 18in-wheeled SUV was still stiffer than we'd have hoped. The diesel is more dynamic than the 2-liter gasoline-engined car; time for a downsized turbo, Honda!

NEWS-IN-BRIEF

TRW Automotive's combined electric park brake (EPB) and electronic stability control (ESC) system will first launch in Europe and Asia Pacific with major manufacturers in 2014. The integration of the two technologies eliminates the need for a separate ECU in the system network, as the EPB control functions are performed within the upgraded ESC electrohydraulic control unit.

Renault and Caterham Group have teamed up to design and develop future sports cars. The vehicles "will carry the respective DNA of Alpine and Caterham Cars", the automotive division of Caterham Group. They will be built at the Alpine plant in Dieppe, Normandy, in France. The Caterham Group will own a 50% stake in the Automobiles Alpine Renault company, currently held 100% by Renault SAS.

The GoPro video cameras have been integrated into Race Technology's data logging systems. This provides the capability to analyze data and video side by side and create HD videos with data overlays. The DL1 data logger controls the GoPro camera via an interface cable, so both the video and data are recorded at the same time.

Café culture

THE VAUXHALL MOKKA IS A GLOBALLY DESIGNED COMPACT SUV. **JOHN O'BRIEN** DISCOVERS WHERE IT DIFFERS FROM ITS BUICK, OPEL, AND CHEVROLET COUSINS

The UK is a key market for compact SUVs, with Nissan's Qashqai and Juke models proving unprecedented successes. The Mokka is GM Europe's first foray into this segment and its first car on the new global small vehicle platform, Gamma II. The car will be sold in a variety of guises, from Buick through Opel, but it is Vauxhall, GM's UK subsidiary, that was instrumental in the UK and European chassis specifications.

The development of the Vauxhall model started right at the beginning of the Mokka's gestation. "We didn't start with one generic model and develop the European spec from it," explains Gerry Baker, vehicle dynamics manager at Vauxhall. "We started right at the beginning of the program, with the very first mule vehicles, and developed a European car in isolation from there."

The bulk of this work was not done at Opel's International Technical Development Center (ITDC) in Rüsselsheim, but instead at the Millbrook proving ground in the UK. "We approached this by recognizing that different European markets have different preferences, and in some segments those preferences diverge more than in others," explains Baker. "As we continued through the program and continued to review the requirements and differences, we recognized that this is one of the segments where there isn't a great deal of difference."

As a result of this, the European and UK specifications grew ever closer, a move accelerated by the fact that Mokka is an SUV. "When you're off road," explains Baker, "the things that we would normally tune differently – steering calibration, damper settings, bushes, and so on – none of that really matters, it's all the same and the expectations are all the same."

The extensive off-road testing program at Millbrook (see sidebar, opposite page) also resulted in a common standard for the Mokka's electrical driving aids being defined, with the ABS, stability, and traction controls all sharing a pan-European state of tune. "There's a great advantage to that," explains Baker. "We can tune one system exceptionally well rather than trying to rush two."

This common state of tune also extends to other subcomponents and systems within the car. Extensive testing in the UK was also backed up with programs in Germany and Spain, resulting in the European-spec cars all sharing damper settings, while the two steering setups ended up converging far more than expected.

"Quite late in the program," says Baker, "we came to the conclusion that the typical European calibration, where the steering is light on center and the effort and steering wheel torque builds up gradually as you steer, didn't actually benefit this car very much.

"The perception that this is the best thing to do on the Autobahn didn't pan out and the use of the car in town said that we didn't need a

NEWS-IN-BRIEF

GM Europe has opened its completely renewed high-speed circuit at the Dudenhofen proving ground in Germany – part of an on-going US$37.5-million modernization program at the test center, which first opened in 1966. Performing the official opening, engineering vice-president, Mike Ableson, said: "This is a clear sign of our commitment to the long-term future of our development facilities in Germany."

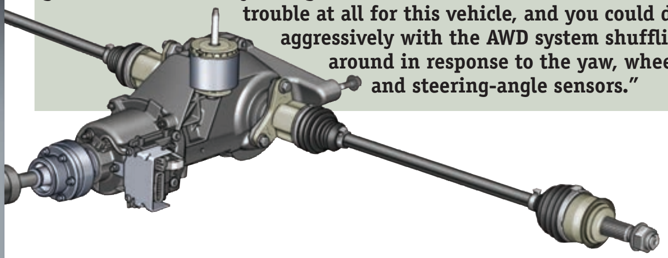
More than a third of the investment has been spent on the revamped high-speed track. With its increased angle of banked corners – from 37.5° to 40° – cars can now be driven at almost 160mph without lateral forces. The remaining US$24 million will be spent on new tracks for testing pass-by noise and calibrating new engines, an all-new traffic control system for the entire proving ground, new tracks for testing transmissions and driving dynamics at Formula 1 levels, a near-identical copy of a public street and a city-driving route.

LET'S OFF-ROAD

The Mokka's AWD system was honed at Millbrook proving ground. The electronically controlled system is built around a two-piece driveshaft that is driven from a power-take-off unit located in the final drive; an electronically controlled multiplate clutch, just ahead of the rear-axle differential; and then driveshafts out to the rear wheels. This allows the Mokka to transfer anything between zero and 50% torque to the rear end, depending on grip availability.

The same system also works in collaboration with the Mokka's anti-lock brakes and stability control systems. By applying individual brakes and transferring torque to the wheel with the most grip, the system mimics the qualities of a limited-slip differential.

Millbrook's off-road trails were used to hone the ABS system, hill descent control, and the stability control systems. "There are some very severe off-road trails, up to 37% gradient loose-surface trails, but some moderate ones too," explains Gerry Baker. "We quickly found, though, that the moderate gravel trails, such as you might find on a hillside somewhere, were no trouble at all for this vehicle, and you could drive very aggressively with the AWD system shuffling torque around in response to the yaw, wheel-speed, and steering-angle sensors."



SPECIFICATIONS

2013MY Vauxhall Mokka 1.7 CDTI

Dimensions: 4,278mm (L) x 1,777mm (W) x 1,658mm (H, exc. aerial). Wheelbase 2,555mm. Track: 1,540mm (F), 1540mm (R)
Dry weight: 1,354kg
Suspension: MacPherson strut front, compound crank rear
Brakes: Ventilated 300mm front discs, ventilated 268mm rears
Steering: EPS. Turning circle at body 10.9m
Tires: 215/55 R18

heavy steering setup. We don't need that wide center window; instead we need the responsive feel you get from having a slightly steeper torque build-up curve."

Meanwhile, the UK steering setup was criticized for its lack of feel. "The British settings came from a heavy, meaty, firm steering and we found that it made the car feel quite dead, dull, and unresponsive."

The result for the UK car is a compromise between the two, with some of the 'effort' taken from the steering so as to lose any of the dampening, hysteresis, or 'connected' feel for the driver.

The Mokka's column-mounted EPS has also been refined to reduce the 'typical springy feel' associated with that type of assistance. "Every time we develop a car," acknowledges Baker, "there are more parameters to play with and it gets more complicated, but you can end up with a much finer finish. We ended up with two settings for the steering that were very, very similar. And even though the calibrations are different, they are so similar that we recognize that for this segment in future, we may not even try to do a separate UK steer, although it's something we'd keep in mind."

The front suspension uses MacPherson struts paired with offset springs. "The car has been designed for a wide range of intended uses," explains Baker, "from an urban family carrier, to some capabilities for some fairly tough off-road expedition. So we wanted to reduce friction caused by the wheel position on the strut." By allowing the strut to rotate through steering, friction is reduced and sensitivity is heightened in the steering, ultimately resulting in a "much smoother ride".

The rear suspension is by way of a compound crank. It comprises a crushed tubular crossbeam, indented into a V section, the orientation of which dictates the roll-center height and position of the rear suspension, while also setting up the toe and camber combination for the axle.

"The length of the V-section indentation is critical to roll stiffness," explains Baker. "With those parameters, we've got very fine control over roll-stiffness for handling balance and roll-center position for the front/rear roll force distribution."

The dual nature of the car made the suspension settings critical. "We had to tune this vehicle for a much wider range of capabilities than a normal car, which was a challenge in itself," says Baker. "First and foremost, it's going to be carrying people around town, on motorways, and on relatively smooth roads. So a smooth ride was paramount."

The Mokka's relatively wide track enabled Vauxhall engineers to give it good ground clearance without it becoming top-heavy and having a tendency to roll, or needing such a high roll-stiffness that the ride became uncomfortable. "The vehicle does exceptionally well off-road and we had a huge amount of fun defining those parameters," states Baker. "But given that the basic architecture is moderate ground clearance and a wide track, none of the things we did for off-road capability impacted on the on-road refinement."

As a result, the Mokka's dampers have been tuned for on-road prowess, while the spring rates have been kept deliberately low to aid ride comfort. Because Vauxhall was able to tune the roll center heights, roll-stiffness has been kept to a minimum, reducing 'head toss' across moderately rough country roads.

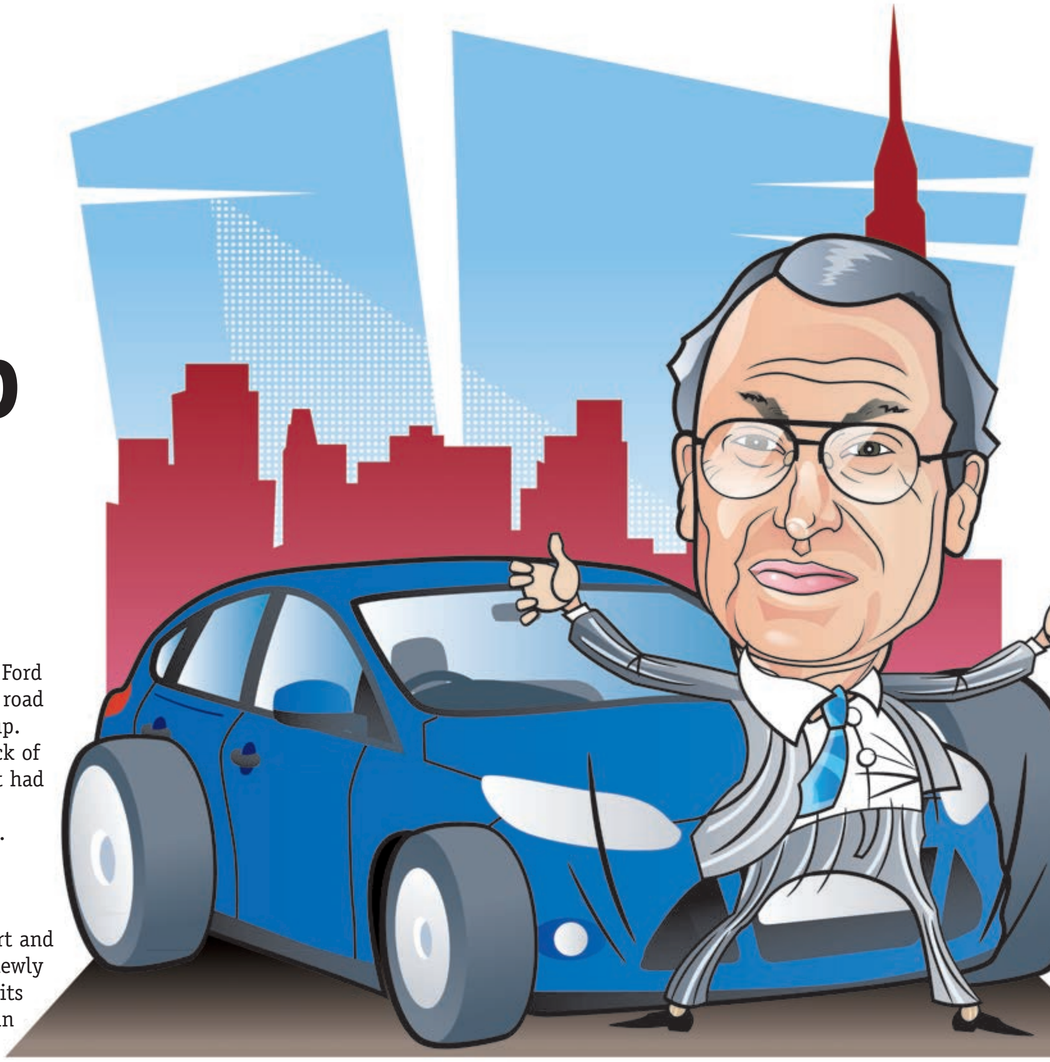
"We were able to get the on-road comfort through damper tuning, springs, and ARBs. This means that the off-road capability comes from the ground clearance and the cleverness of the AWD system."

NEWS-IN-BRIEF

Fowlerville proving ground, Michigan, is investing US$12 million to create new test tracks that emulate US highway surfaces. The 870-acre facility has four test tracks, including 48,000ft² of low- and middle-μ tiles and a 20-acre dynamic pad.

Arctic Falls has begun a partnership with BD Testing, a subsidiary of the tire design and development company Black Donuts, to provide a full tire-development service. The partnership means customers can now send their tires for winter testing without having to provide a team of engineers to undertake the work.

Test World has opened an indoor winter test facility in Ivalo, Finland. The building offers natural snow and adjustable ground and air temperatures all year round, removing the vagaries of the winter climate from the testing equation for more predictable and repeatable results.



On the job

A wheel shame

FOCUS-BUYING **JOHN MILES** IS LEFT FRUSTRATED BY OVERSIZED ALLOYS

To be honest, I'd had enough of my Mk2 Ford Focus 1.8TDCi. I despaired of its howling road noise and restless ride – especially two-up. A new set of dampers helped, but the lack of refinement had become too much to bear – just as it had been on my Mk1.

Of course Ford was aware of the refinement issues. The new model really had to be hugely better to compete with the extremely refined Mk6 Golf. Well, it is – but for many, it will not be.

Inspired by the truly transformational ride comfort and Golf-like road noise isolation of a work colleague's newly acquired Mk3 Focus 1.6 petrol Ecoboost wagon, and its lack of any perceptible powertrain or steering column shake, I hot-footed it down to my local Ford dealer.

Oh dear. I expect to be able to specify the wheels and tires on my everyday car. I did not expect to be forced to suffer the scourge of ultra-low-profile tires on such a mass-market car. It seems I'm completely out of step with the rest of the car-buying public, because no attention has been paid to what the customer might want in the way of tires (thus disregarding considerations such as replacement cost, rim-damage potential, ride comfort, wear rates, aquaplaning resistance, and spare wheel inconvenience). Whether you like it or not, the top-spec Titanium Focus comes as standard with 235/40-18 rubber, while the mid-spec Zetec (powered here by the mechanically interesting 1.0 Ecoboost petrol engine), has 215/50-17 boots.

From a road noise standpoint, the Zetec was the worst – worse even than my raw-edged Focus Mk2 on the UK's rough aggregate surfaces. Steering column ride on my friend's Estate is truly excellent, but was mediocre here, with the wheel responding to any kind of transverse ride, cat's-eye, or concrete road surface. The body interior had a resonant quality that I disliked a lot. The engine buzzed away smoothly, but seemed a bit breathless in such a big body. Perhaps the lack of apparent powertrain mass at certain frequencies, and therefore a differentially variable (compared with 1.6TDCi-equipped cars) front axle mass, had something to do with the resonant road-surface response of the car, because the even more up-spec 1.6TDCi Titanium felt a more 'solid' package.

But it was both cars' busy low-speed ride and aggressive off-center response (plus the road noise) that really depressed me. Ford's first go at all-electric steering in a Focus is not as good as the earlier hydraulic systems, in having a somewhat 'compliant' response quality, but I was to find out (yet again) that steering linearity and on-center aligning torque are far more satisfactory on 215/55-16 tires fitted to the basic 1.6TDCi Edge model.

> "It was both cars' busy low-speed ride and aggressive off-center response (plus the road noise) that really depressed me"

Here were the genuinely much better cars I had read about – quieter, less road-surface-sensitive, and with a ride that combined a supple undertone with good primary body control. So it's a pity one has to accept a basic, 1.6-liter turbodiesel engine (or a 1.6 petrol Ecoboost wagon) to get the gains. The 215/55-16 Michelin Primacy HP-shod example had just the better steering linearity, and was a bit less noise sensitive on different road surfaces, while the Goodyear Efficient Grip-shod machine demonstrated somewhat superior comfort on the larger sharp inputs. Both cars were much more settled at high speeds – more grown-up one might say – and produced a more flowing drive experience than the Mk2 or current Titanium/Zetec cars.

So it seems there is no discussion. Dealers and Ford PR confirm that you get what you are given and like it – except I don't like it at all. The conclusion is that if you want to ruin your Focus's (or Golf Mk6 for that matter) steering, ride, and handling, then specify a version that comes with the 17in or 18in rubber.

The inability to specify the most important element of chassis componentry is extremely frustrating. It depends on the ignorance of the customer who is tricked by the styling- and marketing-led nonsense. For an RS version of course, but it should be obvious that we don't need yet further gains in grip for this kind of bread and butter car. We need further gains in wet grip, lower replacement cost, wider availability, and above all, a quieter, less stressful, and more refined driving environment.

So once again I am a Focus owner – a one-year-old Edge 1.6TDCi. It's a bit gutless, but it's a relief not to get beaten up on a long journey. The Focus Mk3 may still not quite have the ability and quality of a Mk6, 1.4 turbo Golf, but it is a very significant step forward, and a car with more 'athletic' handling than the VW.

Made in Italy

Modular thinking

VOLKSWAGEN HAS ANOTHER WINNER ON ITS HANDS WITH MQB, SAYS **MATT DAVIS**

Volkswagen Group's MQB architecture, a big-time strategy gestated directly by the VW brand, is supposed to change our lives. That is, at least in so far as our relationship with cars ranging from the next Polo up to the next-generation Passat, or from the next A1 up to the next A5's smaller engine trims with front-wheel drive. That's a bunch of cars. More modular than this you're probably never going to find, unless something else sprouts forth from Wolfsburg, that is.

The basic idea physically is to leave the front axle and engine mounting points in place while moving the rear axle back various distances to accommodate the various class segments. So far VW Group has let me at their latest (third generation) Audi A3 and the all-new Golf Mk7, these being the first two models to be assembled using MQB. The cars are larger, lighter, and nimbler, plus the new premium wiring harness possibilities with MQB mean that the dynamics tech and options available to the compact and mid-size segments have reached a previously unthinkable level. All other manufacturers – at least those not yet owned by Dr. Piech – have been put on notice and they're no doubt wondering how on earth they'll be able to compete on either the product or the cost savings sides. As VW Group tech boss Ulrich Hackenberg tells it, MQB is more of a killer business strategy rather than one that will drastically alter current default driving dynamics as seen on outgoing architectures PQ25, PQ35, and PQ46.

The last instances of this thinking, as I recall, were in the early part of the last decade when Ford worldwide made a biggish deal regarding its Generic Architecture Process beneath the Mk2 Focus and other front-wheel-drive corporate cousins. Then a few years later Fiat and Ford launched the 500 and Ka with Fiat's small car architecture used first under the then-new Panda. But neither of these has gone anywhere near as far or quite as global as MQB.

In long hand, MQB stands for Modularer Querbaukasten or 'modular transverse matrix'. Prior to the wonders of MQB, VW was bleeding money like everyone else, with a wide range of track widths across several similar cars in various regions, as well as several engine placements and bolt patterns. Nowadays, however, every single front transverse engine on MQB places the engine at a 12° angle leaning rearward with the exhaust side facing the rear. Immediately all of this greatly improves leg and knee room for everyone and is also said to improve pedestrian impact protection.

But I was there for dynamics, of course. I'd driven the new A3 a while ago and I was pleased with the MacPherson front/multilink rear setup, together with the more elaborate Dynamic Chassis Control and now standard XDS brake-steer-style electronic virtual differential working the front axle. The higher rigidity and lighter steels put to use, together with judicious usage of aluminum for this less pricey segment, create a much tauter set of responses, while making all the bits and pieces less noisy. Throw in previously unavailable Progressive Steering and these smaller architectures start feeling like bigger premium German cars. Yet the larger A3 is built at 165 lb less than the preceding car, while the Golf Mk7 loses up to 220 lb. It doesn't seem possible, but lightness is the keyword going forward.

"Prior to MQB, VW was bleeding money like everyone else, with a range of track widths across several similar cars"

I went so far as to ask a couple of the more whiney German and British journalist colleagues what they thought regarding this new generation of cars from VW riding upon MQB. They said that these are damned near perfect cars and all the more so for actually being entirely new despite the only evolutionary look on the outside. They added the thought that these could be the most boring driving cars to date. I mostly agree with this assessment, but I won't use the word 'boring' until cars are driving themselves and I'm essentially out of a job worth doing.

"As usual I met a lot of interesting people and spent the whole day talking to people – some new faces, as well as people I knew from before"

Koen Reybrouck, director, Reybrouck Consulting & Innovation

CD changer

DR PAUL ZANDBERGEN AND **DR WOLFGANG DAVID** (FORD AACHEN) AND **ED KNOY** (FORD COLOGNE) DETAIL A NEW REAR SUSPENSION FOR THE GLOBAL CD PLATFORM

Ford is introducing a completely new rear suspension on the new 2013MY Mondeo. This so-called integral link suspension is one of the most technically advanced suspension concepts and falls into the category of suspensions used in the upper premium segment. As well as the new Mondeo, all Ford and Lincoln vehicles built on Ford's new global CD platform will benefit from the new suspension architecture.

Control blade SLA rear suspension was first adopted on the Mondeo wagon in 2000 and on the sedan from 2007. This axle still performs very well for steering and handling, but to achieve premium level comfort the system has some limitations. As indicated in Figure 1, the height of the control blade pivot is actually slightly below the height of the wheel center position, resulting in so-called kinematic precession of the wheel during jounce motion. However, for optimized impact harshness and noise the wheel should move rearward in jounce, which is called kinematic recession. Moreover, for optimized noise on smooth road surfaces the vibrations transferred to the body must be minimized. The flexibility of the control blade and the bushing mounting the blade to the body structure result in torsional and bending modes of the blade. These modes are transferred to the vehicle body via different paths, the most important of which are:

- Wheel carrier -> transverse links -> subframe -> body;
- Wheel carrier -> shock absorber -> shock absorber top mount -> body.

Integral link suspension

The key components of the integral link suspension system are indicated in Figure 2 and the relevant suspension hardpoints are indicated in Figure 3. The suspension comprises a lower control arm, which connects to a subframe at two pivot points. The rear pivot is a ball joint and the front pivot is a bushing. At the outer side, the lower control arm connects to a wheel carrier by means of a ball joint. This pivot point is positioned at a certain distance behind the wheel spin axis. The so-called integral link is positioned in front of the wheel spin axis. A lower pivot point connects the lower control arm to the integral link and a higher pivot point connects the integral link to the wheel carrier. In addition to the lower control arm, two other links connect the wheel carrier to the subframe. The upper link is called the camber link and the lower link is called the toe link. The inner and outer pivot points of these links are all realized with bushings.

The suspension further comprises a coil spring, a shock absorber and a stabilizer bar. The coil spring is supported by the lower control arm. The top of the spring connects to the

KEY SUSPENSION REQUIREMENTS

A number of requirements were defined for the new suspension. First, sufficient wheel travel has to be provided. Next, the underfloor of the vehicle should be as low and wide as possible to provide a low and wide load floor. The stringent package requirements for the underfloor imply that the side rail structure of the vehicles is very low. This results in a small section between the side rail and the ground clearance plane, which explains the tight curvature of the camber link.

The majority of the suspension parts used for the FWD application must also be used for the AWD applications to minimize additional cost. The AWD system comprises a differential, driveshafts and a propshaft. The differential needs to be mounted to the subframe by means of four elastic mounts. The integral link suspension is very compact in the lateral direction, so there is sufficient space for a standard AWD as well as a torque vectoring AWD differential (this compactness also enables the exhaust to be easily routed below the suspension). The camber link can move in parallel to the driveshaft and therefore the packaging of a driveshaft does not compromise the suspension travel.

The wheel and tire sizes that need to be fitted to the new suspension cover a large range. For the sedan applications on the CD platform, the wheel size ranges from 16in to 19in and for the crossover vehicles they range from 17in to 21in.

Finally, when it comes to attribute requirements, the goal for the new suspension architecture is to build on the Mondeo's high level of steering and handling performance and also further improve the ride comfort, in particular impact harshness and road noise.

vehicle body. The shock absorber is also supported by the lower control arm. It is located behind the wheel spin axis and outside the spring. On the top of the shock absorber an assist spring is mounted, which provides a progressive suspension rate.

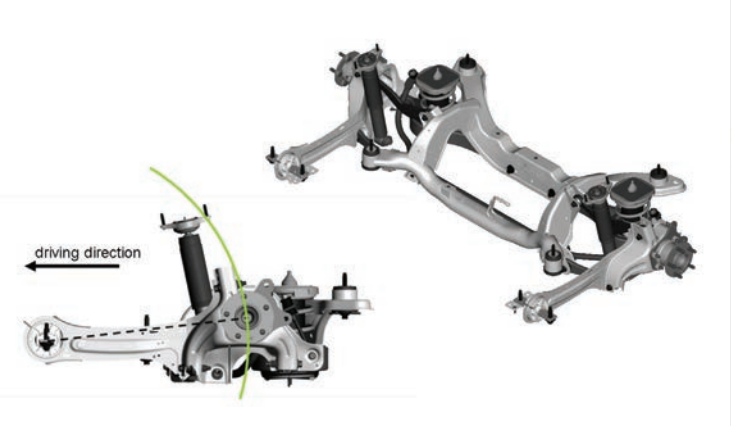
Impact harshness targets

An important aspect of a vehicle's ride comfort is impact harshness. Impacts due to road imperfections, such as expansion joints or tar strips, should be absorbed smoothly. The behavior of the vehicle to such discrete events is tested on the Ford Proving Grounds by driving over metal strips of specified height. Related to impact harshness is the so-called aftershake, which is the remaining oscillation of the vehicle after the impact. This oscillation must be suppressed as quickly as possible via damping.

During the course of this project, multibody CAE analysis was performed to simulate impacts. Accelerations in both the X and Z directions were triggered at the seat rail. Differentiating these signals results in the so-called jerk in X and Z (m/sec^3). Accelerations and jerk as a metric for impact harshness and aftershake were used through the course of this project. Figure 4 displays a typical result for a simulation of a vehicle driving over an impact strip. The first peak (A) relates to the front suspension impact and the next (B) relates to the rear suspension impact. The most important design parameters influencing the suspension impact harshness and aftershake were identified via parameter studies.

The longitudinal mode frequency is determined by the dynamic stiffness of the suspension in the longitudinal direction and the mass of components involved in the longitudinal motion. The dynamic stiffness is calculated in CAE at the wheel center position and is mainly determined by the dynamic rates of the suspension and subframe bushings. CAE studies have demonstrated that the impact harshness (jerk) in the X direction is proportional to the longitudinal mode frequency: a lower longitudinal mode frequency results in better impact harshness. The lower limit for the mode frequency is driven by longitudinal mode frequencies of other relevant systems, for example the powertrain.

Increasing the longitudinal damping in the suspension via the dynamic stiffness of the bushings would increase the longitudinal mode frequency, which, as discussed above, deteriorates the impact harshness. For a correct assessment of the effect of damping, the longitudinal frequency will have to stay constant via constant dynamic



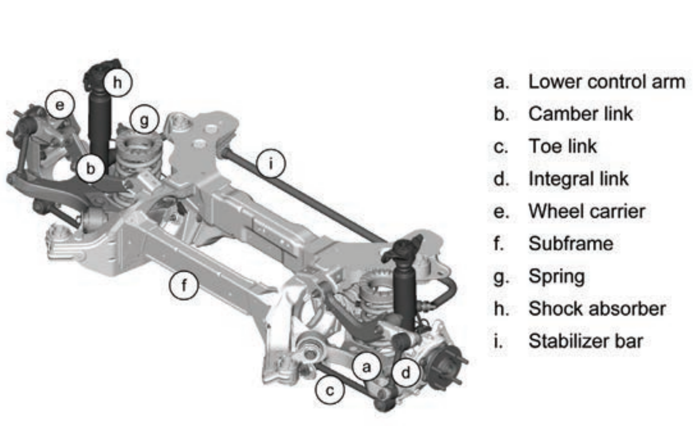


FIGURE 1 (ABOVE): **2007-2011 FORD MONDEO CONTROL BLADE SLA REAR SUSPENSION**

FIGURE 2 (ABOVE RIGHT): **THE FORD INTEGRAL LINK SUSPENSION – COMPONENTS**

FIGURE 4 (BELOW): **ACCELERATIONS AND JERK CALCULATED DURING DRIVING OVER IMPACT STRIP**

FIGURE 5 (BELOW RIGHT): **EFFECT OF INCREASED BUSHING DAMPING ON JERK X**

stiffness of the bushings. This can be achieved via a lower static stiffness of the bushings and increased damping. Increased longitudinal damping at constant longitudinal mode frequency demonstrates an improvement in impact harshness (Figure 5). The impact aftershake and the overall ride comfort on different road surfaces also improve when longitudinal damping is increased.

The wheel trajectory angle is defined as the slope of the curve that plots the wheel center vertical displacement versus the wheel center longitudinal displacement. The angle is defined as positive when the wheel moves rearward during upward movement (Figure 6). The influence of impact harshness to wheel trajectory angle was studied for different suspension systems. It was found that the impact harshness decreases rapidly when the trajectory angle falls below zero. A positive trajectory angle is beneficial for impact harshness up to a maximum of 5-6°. In this case, the wheel moves rearward and upward when it hits a bump.

The parameters listed above primarily influence the impact harshness and aftershake in the X direction. The characteristics of the shock absorbers, shock absorber mounts, springs and spring aids primarily determine the impact behavior and aftershake in the Z direction. These parameters are considered as suspension-tuning parameters rather than design parameters. One design parameter influencing the vertical dynamics is the unsprung mass. In general, a lower unsprung mass is beneficial for ride comfort.

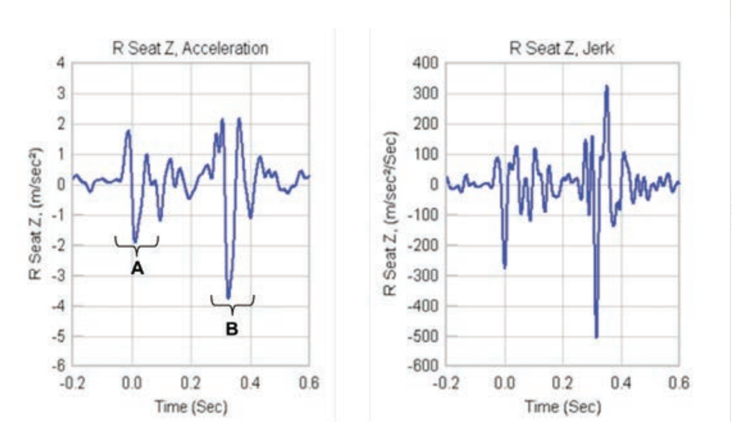
Steering and handling targets

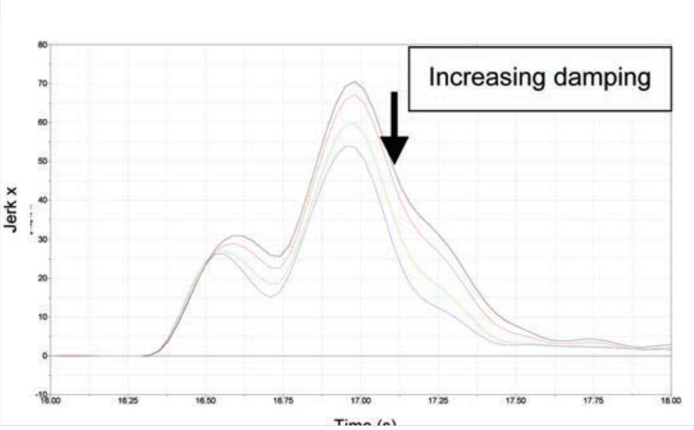
The most important metrics related to steering and handling can be divided into metrics related to the kinematics of the suspension and metrics related to the compliances of the suspension. Important kinematics related to metrics are roll steer, roll camber and roll center height; important compliance-related metrics are lateral compliance, lateral compliance steer, camber compliance, castor compliance, longitudinal compliance, longitudinal compliance steer and aligning torque compliance steer.

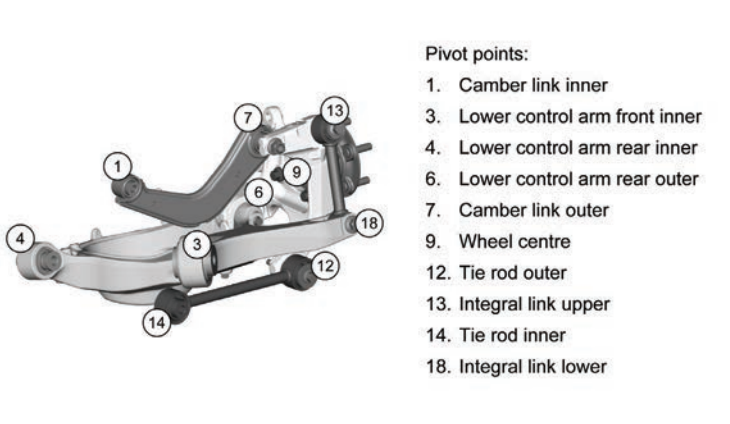
Suspension kinematics define how the wheels translate or rotate during suspension vertical travel or suspension roll. For steering and handling, the metrics related to roll are of highest relevance. During cornering the vehicle body rolls and the resulting suspension travels should result in some level of understeer (roll steer). On a rear suspension, this is achieved via a certain amount of toe-in on at least the wheel on the outer side of the bend. On an independent suspension, the wheel on the outside of the bend usually loses camber angle with respect to the road. This camber loss results in reduced grip and it is desirable to minimize it via maximized suspension roll camber.

Another important aspect of vehicle roll behavior is the roll axis, which is defined by the line connecting the suspension roll centers. The vehicle rolls with respect to the ground about this instantaneous axis. The rear suspension roll center is targeted higher than the front suspension roll axis, creating an inclined roll axis in side view. This supports a linear steering response and high steering precision at corner entry.

In general, lower suspension compliances are better for steering response and agility. A good steering system that makes the driver feel connected to the road requires very low compliance in lateral direction. Low camber compliance is crucial for low lateral compliance. For handling stability, the suspension ideally delivers a certain amount of compliance understeer during cornering and braking. This is







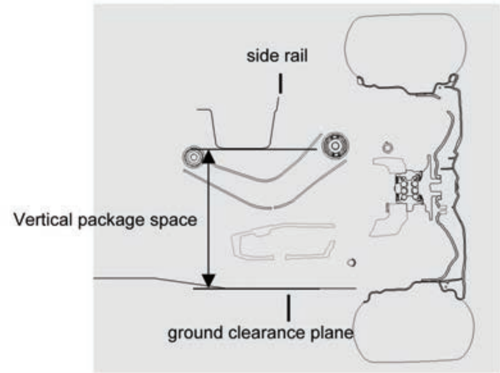


FIGURE 3 (ABOVE LEFT): **THE FORD INTEGRAL LINK SUSPENSION – DEFINITION OF PIVOT POINTS**

FIGURE 6 (BELOW LEFT): **WHEEL TRAJECTORY ANGLE (α) BENEFICIAL FOR IMPACT HARSHNESS**

FIGURE 7 (BELOW): **INTEGRAL LINK ROLL POLE AND ROLL CENTER HEIGHT**

defined by the lateral compliance steer and longitudinal compliance steer characteristics. During braking or other longitudinal wheel loading, such as impacts, the castor angle change should be minimal to prevent variations in bump steer, roll steer and mechanical trail. This is controlled via the castor compliance characteristic.

Suspension design principles

An important requirement for a suspension used for a global platform with many different vehicle applications is the ability to easily tune the suspension to the required characteristics.

For a double-wishbone suspension, the roll center can be constructed via a simplified, two-dimensional approach, as detailed by. The same construction can be applied to the new integral link suspension (Figure 7). The roll pole is the intersection of lines drawn through the lower control arm and the camber link as projected on a vertical plane through the wheel centers. This is only a rough approximation, as in reality the toe link front view inclination will also affect the roll pole to some extent.

A line connecting the center of the tire contact and the roll pole intersects the vehicle center plane in the roll center. As well as determining the roll center, the roll pole also determines the roll camber. Moving the roll pole inboard by a more parallel front view inclination of the lower control arm and camber link lowers both the roll center and the roll camber.

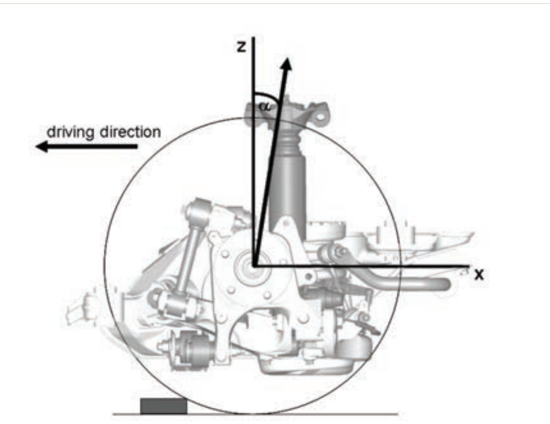
A sensitivity study was performed in ADAMS/Insight to see the effect of the suspension geometry to both roll center height and roll camber to confirm what would be the best strategy to adapt roll center or roll camber (Figure 8). The parameters with highest sensitivity to camber gain are the heights of points 1 and 7. The roll center height is affected by several parameters, the heights of points 1, 4, 6 and 7 being the most sensitive. As expected, the toe link front view inclination also plays a role in the roll center height.

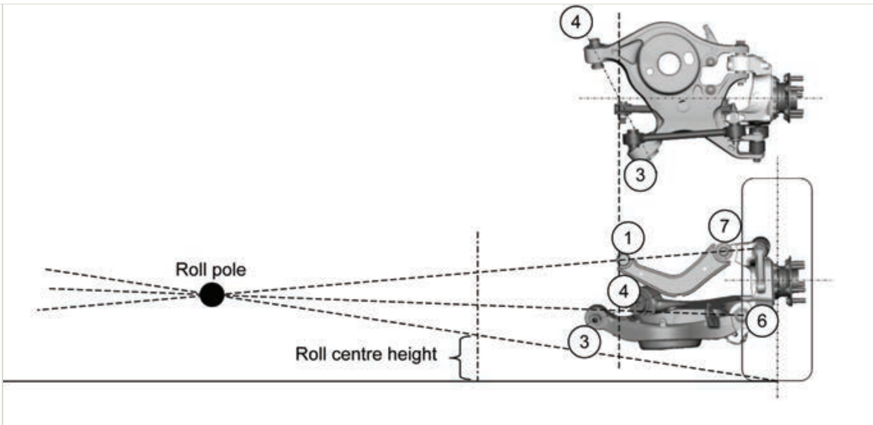
In the selection of the heights of points 1, 4, 6 and 7, a few restrictions play a role:

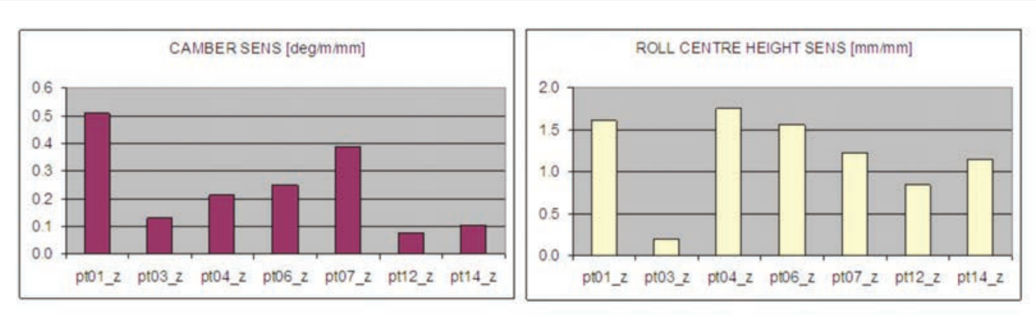
- Point 4 – lower limit for height is given by the ground clearance;
- Point 6 – lower limit and point 7 upper limit for height is given by smallest rim diameter;
- Point 1 – upper limit is constrained by vehicle floor height;
- The vertical distances between the wheel center and points 6 and 7 need to meet a minimum in order to achieve low camber compliance.

Based on the sensitivities and restriction described above, the following guideline is used to set roll center height and roll camber. Points 6 and 7 are positioned as close to the smallest rim as possible. Points 1 and 4 are set to meet the target roll center height. If the resulting roll camber is too high, then point 7 can be lowered to match the target roll camber. The roll center height would then slightly drop, which could be corrected by raising point 4. If the resulting roll camber is too low, then point 1 can be lowered. This would raise the roll center, which could be corrected by lowering point 4 and/or raising point 6.

The roll steer of the integral link suspension is mainly dependent on the inclination of the toe link in the front view (Figure 9 left). When the outer connection to the wheel carrier is raised, the roll steer gradient increases, which is a modification toward understeer. The wheel carrier steers about a virtual steer axis, which passes through point 7 and a point between points 6 and 12 (Figure 9 right). The location of this point depends on the stiffness of the pivot points 4 and 6 versus the stiffness of the pivot points 12 and 14. It is more biased toward point 6 as pivot points 4 and 6 are ball joints







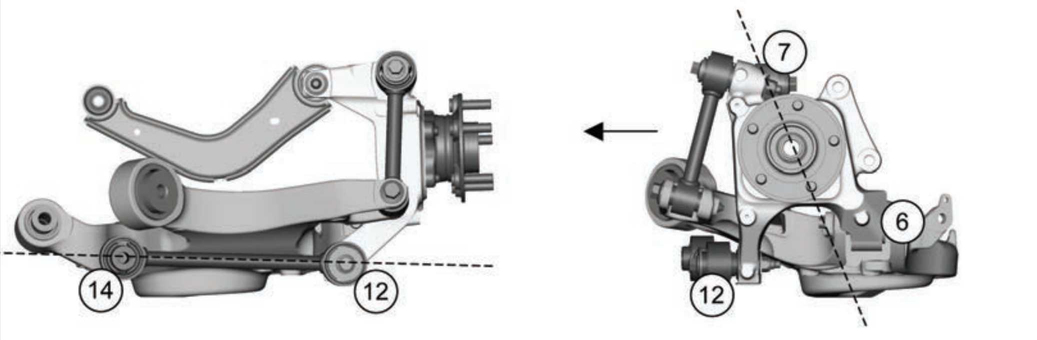


FIGURE 8 (TOP): **SENSITIVITY OF SUSPENSION GEOMETRY TO ROLL CAMBER AND ROLL CENTER HEIGHT FOR INTEGRAL LINK SUSPENSION**

FIGURE 9 (ABOVE): **INTEGRAL LINK FRONT VIEW (TOE LINK INCLINATION) AND SIDE VIEW (APPROXIMATE STEER AXIS)**

FIGURE 10 (BELOW): **INTEGRAL LINK INSTANTANEOUS AXIS AND PITCH POLE**

FIGURE 11 (BELOW RIGHT): **KINEMATIC RECESSION, ANTI-LIFT AND ANTI-SQUAT**

and the toe link pivot points 12 and 14 are bushings with some level of compliance.

Figure 10 shows the bottom view of the integral link suspension. The lines connecting points 3 and 4 and points 6 and 18 are extended and the intersection point in this view is connected to the roll pole to form the suspension's instantaneous axis. This is the axis about which the wheel rotates during suspension travel. The intersection point of the instantaneous axis and the plane through the wheel center is named the pitch pole.

As shown in the side view of the suspension (Figure 11), the pitch pole is above the wheel center. This provides kinematic recession during suspension jounce travel, anti-lift and anti-squat. The pitch center is located far in front of the wheel center, which results in a relatively low bump castor. The pitch pole can easily be modified via the location of the lower control arm pivot points.

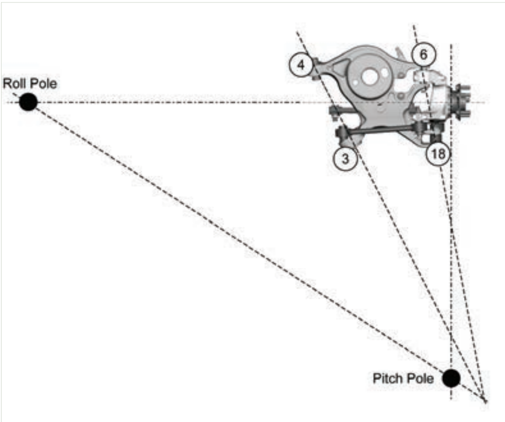
As explained, the integral link has a great amount of longitudinal compliance in order to absorb road impacts smoothly. This compliance is mainly provided by the inner bushing of the lower control arm (Figure 12, point 3). When the wheel hits a bump, vertical and longitudinal forces are induced at the wheel center. As a result of the longitudinal force, the lower control arm rotates around the rear pivot (Figure 12, point 4), which is a ball joint. The resulting translation in the bushing at the front of the lower arm is controlled by the stiffness of this bushing.

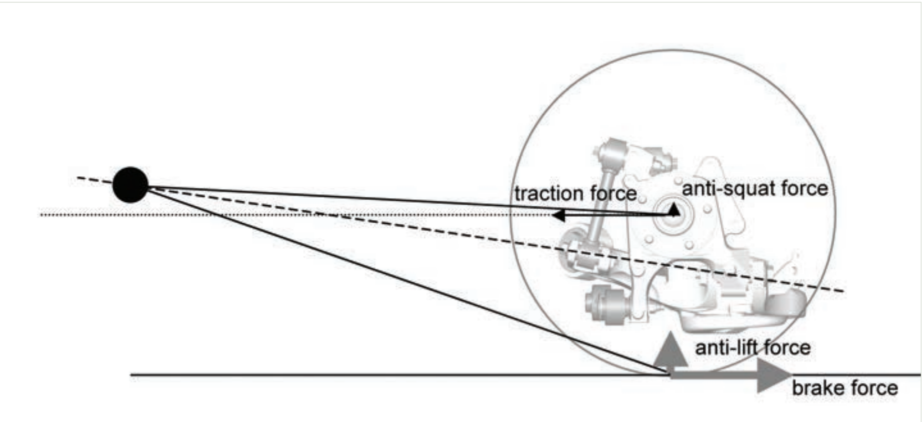
The integral link connecting wheel carrier and lower control arm prevents rotation of the wheel carrier during impact or, for example, braking load. This reduces vibrations introduced to the suspension and prevents castor changes for wheel impacts or braking while driving around a corner. Such castor changes could cause instability due to varying roll steer and castor trail. When more longitudinal compliance is desired, the stiffness of the bushing in point 3 should be reduced. For a classic double-wishbone suspension, such a change would greatly reduce the castor stiffness. This is not the case for the integral link suspension because the castor load path is decoupled from the longitudinal load path via the integral link mechanism.

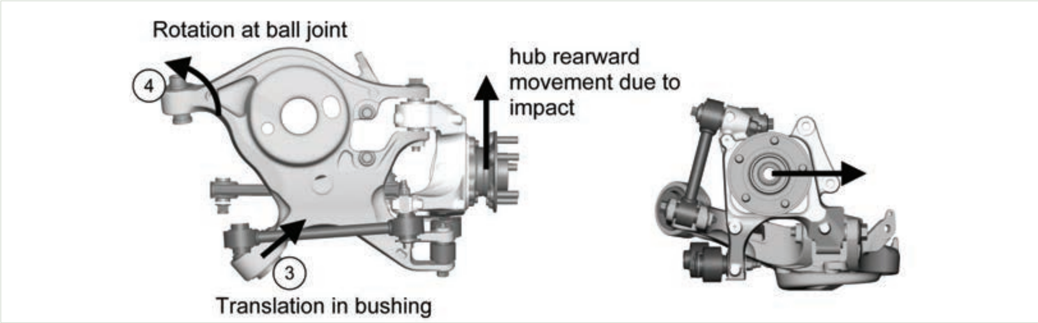
During braking, the lower control arm rotates as described for the longitudinal compliance mechanism. Should the wheel follow this arm rotation, the wheel would rotate to toe-out during braking, which is not desired. However, this rotation is restricted by the toe link. Depending on the bottom view angle of the toe link, the wheel could even deflect to toe-in during braking.

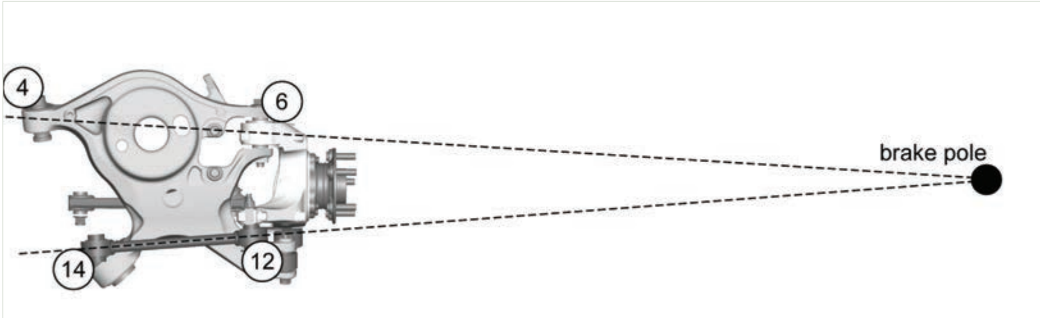
This is achieved via the mechanism displayed in Figure 13. The lines connecting points 4 and 6 and points 12 and 14 are extended and the projected intersection in the bottom view is named the brake pole. This is the point about which the wheel rotates when a longitudinal load is applied. When the lines do not intersect, the pole is in infinity, resulting in zero longitudinal compliance steer (neutral brake steer). When the pole is outboard of the wheel center line (as in Figure 13), this gives toe-in under-braking. Moving the pole toward the wheel center line would increase toe-in. When the pole is inboard of the wheel center line, the wheel would turn to toe-out under braking. The lateral position of the pole can easily be altered by the bottom view angle of the toe link.

Brake forces are applied at the wheel contact patch and introduce a moment on the wheel carrier.









This moment is transferred via the integral link to the lower control arm and the inner pivot points connecting the arm to the subframe. The rear pivot point (4) is a ball joint and therefore very stiff. The front pivot point (3) is a bushing responsible for longitudinal compliance and therefore it is soft in the direction in plane with the lower control arm. To resist the moment induced by braking, the bushing must be stiff in the direction perpendicular to the lower control arm. To meet both requirements, the bushing is voided in the soft direction.

Results

The integral link suspension was assessed at the Ford proving grounds in a number of prototype vehicles. After initial subjective assessment of the built hardware, a tuning phase started in which springs, shock absorbers, link bushings, top mount bearings, etc., were varied according to a systematic approach.

Subjective assessment is supported with measurements according to Ford standard DNA procedures. The results are then processed and visualized in so-called 'fingerprints' for ride, steering and handling. The achieved improvements in impact harshness, aftershake and noise are illustrated in Figure 14. Despite clear improvements for comfort and noise, the steering and handling characteristics have not degraded compared with the outgoing Ford Mondeo. This was verified during several management drives and resulted in implementation of the integral link suspension for all applications built on the global CD platform.

Figure 14 shows the accelerations in X and Z, measured at the front seat rail versus time, while driving over a 30mm high strip at a speed of 30km/h. The highest peaks are found at the point where the strip is passed first by the front wheels and then by the rear wheels. The first cycle of body acceleration is the part of the signal reflecting the impact harshness (A). The remaining vehicle acceleration after the impact is the aftershake (B). The vertical accelerations show very small differences between integral link and control blade. This meets expectations as the driven vehicle setups have very similar spring and shock-absorber characteristics.

In the longitudinal direction, the difference between the two suspensions becomes very clear. The integral link shows much lower accelerations during the rear suspension impact and the aftershake accelerations are also much lower. During the subjective evaluation, it was clearly observed that for the integral link vehicle the rear impact was on a similar level as the front impact. For the control blade SLA vehicle the rear impact was much more pronounced than the front impact. The observed difference between the two suspensions became even clearer for the rear passengers.

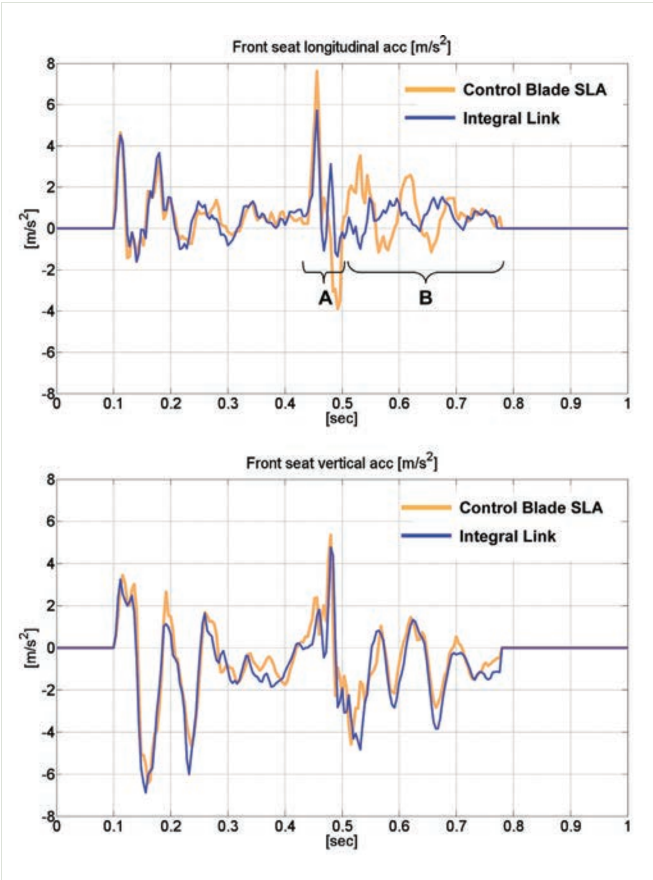
As discussed above, the control blade SLA suspension has some known sources of vibrations that are transferred to the vehicle structure. The key contributor to noise is the blade between the wheel carrier and the pivot attachment point to the vehicle body. Taking away this blade and introducing the new integral link removes several vibration modes contributing to noise in the rumble range (80-200Hz). The resulting cruising noise was measured in the interior. On the front seat, the integral link shows around 2dB lower noise than the control blade SLA. On the rear seat, the difference is an even greater 4dB. During subjective evaluation, this substantial reduction of noise was clearly experienced.

The steering and handling was assessed during extensive testing at the Ford proving grounds. In general, the steering and handling performance of the integral link is at a similarly high level as for the control blade suspension. The measured kinematics and compliance characteristics back up this result. Although the longitudinal compliance has greatly increased, the lateral compliance, camber compliance and castor compliance are still very low. Tuning of the longitudinal compliance to a different level hardly affects the steering and handling performance.

FIGURE 12 (TOP): **MOTION OF LOWER CONTROL ARM AND HUB DURING IMPACT**

FIGURE 13 (ABOVE): **INTEGRAL LINK BRAKE POLE**

FIGURE 14 (BELOW): **FRONT SEAT LONGITUDINAL AND VERTICAL ACCELERATIONS MEASURED FOR CONTROL BLADE SLA AND INTEGRAL LINK SUSPENSION DURING IMPACT**



Ready for roll

PORSCHE'S **THOMAS MIRWALDT** AND **PETER EBERHARD** (UNIVERSITY OF STUTTGART) PRESENT AN ONLINE-OPTIMIZATION ROUTINE FOR AN ACTIVE ANTI-ROLL SYSTEM

Active anti-roll systems – as part of vehicle chassis – are able to adapt their system behavior to the current driving situation. It is possible to vary between comfortable and sporty setups. The forces applied to the chassis are determined by ECU-controlled actuators. This ECU includes a global chassis control that generates set points for an internal actuator feedback control.

In the development process the final system application always takes place in the real car. Typically it consists of a manual parameter adjustment based on drivers' subjective criteria. An objective and efficient way to set up system parameters is given by numerical optimization routines, which enable automated calibration of the anti-roll system. The optimization objectives consist of drive-dynamic criteria that describe the vehicle's sporty driving behavior. The current states are calculated online from vehicle measurement variables.

Active anti-roll system

An anti-roll bar reduces the body roll angle of a vehicle and consists of a torsion spring connected to the opposite wheels of an axle. During cornering the relative deflection between the opposite wheels applies a torsion moment to the rod, which leads to the chassis being pushed toward the horizontal. In contrast an active anti-roll system applies active forces to the chassis, for example through hydraulic swivel motors integrated into the anti-roll bar.

The major drive-dynamic features of the active anti-roll system are separated into features that influence the vertical and horizontal vehicle dynamics.

With regard to the vertical vehicle dynamics, the active anti-roll system is able to apply any desired anti-roll moment M_{act} on the chassis – as long as the actuator limits are not exceeded. Thus the vertical anti-roll bar control strategy determines the static and dynamic roll angle behavior. In contrast to the passive system – which is comparable to a spring element – an active anti-roll system allows full body roll-angle compensation. The control strategy's main reference variable is based on the measured lateral acceleration a_y which is fed to the ECU to calculate the required set anti-roll moment M_{act}.

With regard to the vehicle's horizontal dynamics, the influence of the active anti-roll moment is caused by the axle-specific change in the relative wheel load:

$$\Delta F_{z,i} = \Delta F_{z,i,\ dyn} + \Delta F_{z,i,\ act}$$

which increases with a higher set anti-roll moment M_{act}. According to Zomotor, a higher relative wheel load $\Delta F_{z,i}$ leads to lower side force F_y being transmitted. The decline is caused by the gradual decline in the side force's characteristic curve, which in turn is influenced by the tire characteristics (Figure 2). Thus a variable roll-moment distribution *WMV* between the front and rear axle enables a specific change of the vehicle's self-steering behavior.

Strict separation between the influences on the vertical and horizontal vehicle dynamics is only possible for the general system effects. Due to kinematic relationships there are also (less

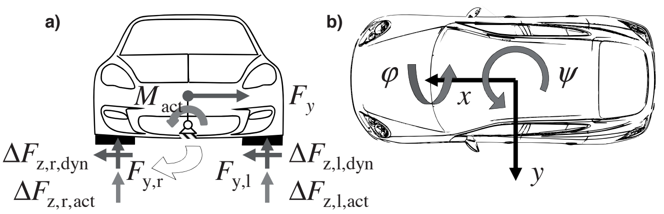
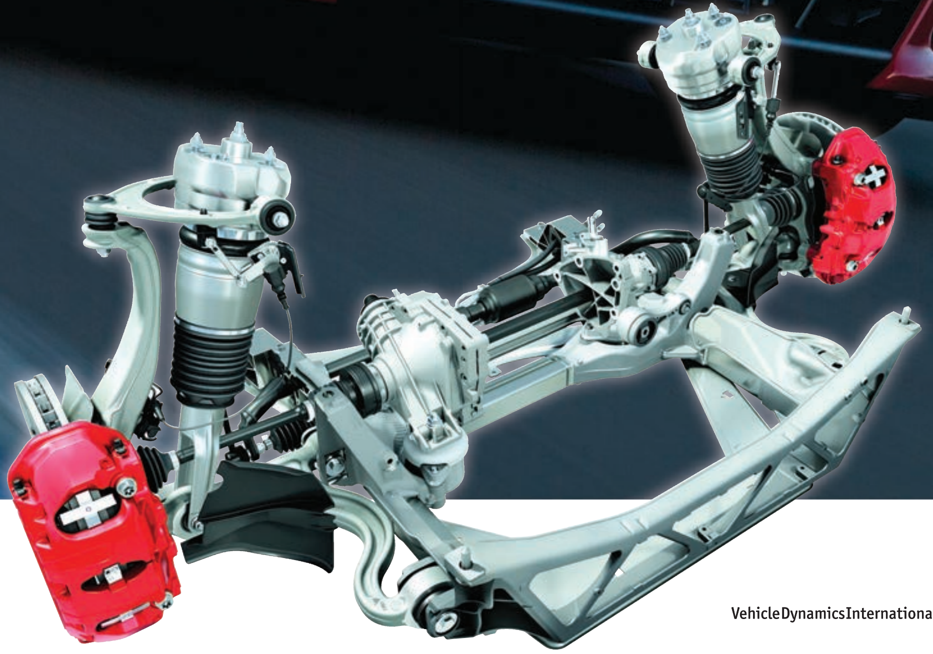


FIGURE 1: **a) CHASSIS' DYNAMIC LOADS DURING CORNERING (RIGHT TURN); b) VEHICLE'S COORDINATE SYSTEM**



PORSCHE PANAMERA'S DOUBLE TRACK ARM SETUP WITH ADAPTIVE AIR SUSPENSION AND ACTIVE ANTI-ROLL

Table 1: Extracted control sub-functions and parameters of the active anti-roll system		
sub-function/phenomenon	variable	optimization parameter p_i
static roll-angle behavior	φ_{stat}	$p_{stat,j}$
dynamic roll-angle behavior	φ_{dyn}	p_{lead}
anti-roll moment distribution	WMV	$p_{WMV,k}$

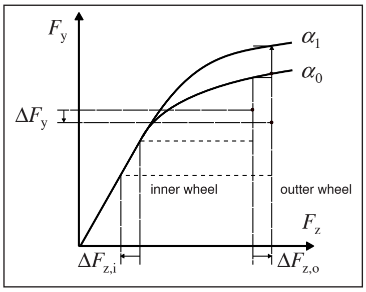


FIGURE 2 (TOP RIGHT): **DECREASE IN MEAN SIDE FORCE ΔF_y, CAUSED BY A HIGHER RELATIVE WHEEL LOAD $\Delta F_{z,i}$ AS A RESULT OF A HIGHER ANTI-ROLL MOMENT M_{ACT}**

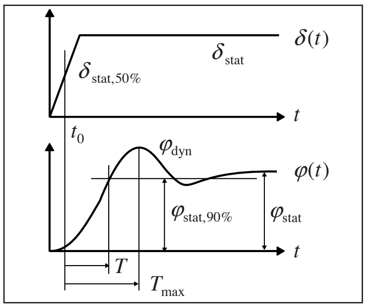


FIGURE 3 (MIDDLE RIGHT): **STEERING STEP MANEUVER**

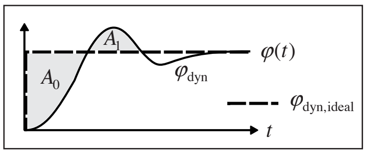


FIGURE 4 (BOTTOM RIGHT): **A_0, A_1 AND THE IDEAL ROLL-ANGLE BEHAVIOR $\varphi_{dyn,ideal}$**

FIGURE 5 (RIGHT): **RESULTS OF SENSITIVITY ANALYSIS OF THE STEERING STEP MANEUVER**

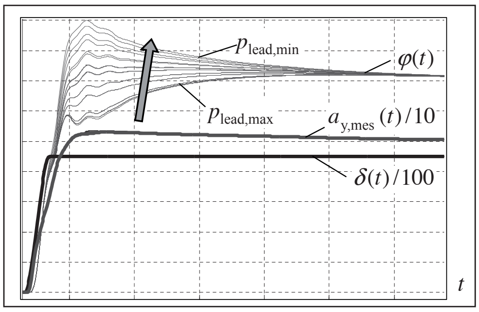
FIGURE 6 (BELOW RIGHT): **RESULTS OF $B_{\varphi dyn}$ (p_{lead})**

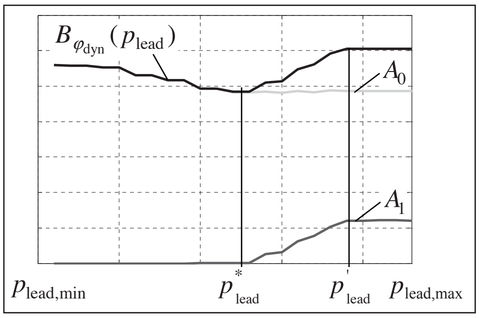
significant) interactions between the behavior of the chassis and the self-steering effect.

Objectives

In the definition of an optimization problem, the chosen optimization objectives, optimization parameters and design variables largely categorise the most suitable optimization method. This in turn determines the convergence of the optimization as well as the required computing time per optimization step and the total number of iterations.

Concerning real-time online optimization, there are rigid criteria for the computing time and a small number of iterations – in this case, of the test maneuver – are required in practical use. This can be achieved by a fast converging optimization. However, in order to obtain high-quality optimization the pre-analysis of the system is of special importance. Aiming at a simple design of the objective functions, the goal is to reduce the given control structure on primary influence parameters and basic effects – referred to as 'sub-function' or 'phenomenon' in the following – without a loss of control features.





Derivation of objectives

Based on customer and performance specifications, the technical design process – using classical product design methods – begins with the definition of a global product function. Subsequently this global function is subdivided into single functions with a higher degree of detail. In this process the system objectives correlate with the performance specifications. They evaluate the system quantitatively using specific values that are calculated by measurable states.

The optimization of the active anti-roll system's control structure represents a parameter optimization of a given system. Therefore an inverse system analysis is more useful. Instead of asking "What should the control be able to do?" we ask "What is the control capable of?" Therefore the given control application process of the active anti-roll system is analyzed by a parameter variation. In this way the dependency between control parameter and control features as well as the application order is examined.

This bottom-up approach – usually used in technical construction processes and software design – delivers the required functional subsection of the given active anti-roll system control. If possible the sub-functions are assigned to an independent parameter. Any small number of 'phenomena' can be treated separately during the optimization. The whole of these subsections represents the total system function.

Applying this analysis method to the present control structure – without presenting the application itself – leads to the primary sub-functions of the given active anti-roll system control, listed in Table 1. The sub-functions are affected by the functionally combined optimization parameter p_i.

It is clear that these extracted sub-functions match the desired basic features of the active anti-roll system. The first two criteria represent the influence of the active anti-roll system on the vertical chassis dynamics. The relation of the values can be illustrated in a steering step maneuver (Figure 3). It represents a standard maneuver to analyze a vehicle's dynamic behavior.

The dynamic roll angle behavior φ_{dyn} describes the chassis's transient roll behavior up to the moment when motion decays to a steady state with the static roll angle φ_{stat}.

As stated above, in contrast to the passive anti-roll system, the active system has no rigid relationship – predetermined by the material properties – between the applied lateral acceleration a_y and the built-up body roll angle $\varphi(t)$. From a drive comfort perspective different approaches for the design of the roll angle behavior exist. Usually a reduced body roll angle correlates with the perception of a sporty driving character. Concerning drive dynamics, a reduced body roll angle permits an increase in the maximum lateral acceleration due to larger tire contact areas.

The general requirement for the dynamic roll angle behavior φ_{dyn} is a fast reference response to reaching the determined static roll angle φ_{stat}. A fast reference roll angle response in combination with a small requested static roll angle amplifies the sporty driving sensation due to minimal chassis motion. In comparison, a slowly responding anti-roll system – which equals low roll damping – in combination with a high requested static roll angle leads to a transient overshoot of the body roll angle during the steering step maneuver (Figure 5). In addition to the control strategy, the dynamic roll angle behavior φ_{dyn} also depends on the dynamic properties of the system actuators.

The third sub-function, the anti-roll moment distribution *WMV*, influences the vehicle's horizontal self-steering behavior. The desired independence between the active anti-roll system's objectives for increased efficiency of the optimization process is only possible for the basic effects. From an axle-specific point of view, an applied body-roll angle φ correlates with an opposite deflection Δz_{rel} of the wheel carriers toward the vehicle axle. A rotational displacement of the wheel carrier is applied during deflection. A decrease in the symmetrical angle δ_{V0} between the wheel carrier and the longitudinal vehicle axle X is called 'toe-in'; an increasing angle is 'toe-out'. Typically a vehicle's front axle toes-in for a compressive deflection in contrast to the rear axle, which toes-out. Thus the wheel-specific steering angle δ_W decreases during cornering and in turn increases the self-steering effect.

This toe-in behavior caused by the body-roll angle is called 'roll steering'. Hence a changed body roll angle also affects the self-steering behavior. Further relationships between the roll angle behavior and the horizontal vehicle behavior exist, for example due to the change in the kinematic camber and the steering stiffness.

Considering those sub-functions while deriving optimization objectives provides a basis for the drive-dynamic setup of an active anti-roll system. Drive comfort criteria and the recognition functions of special drive situations are not included at this point.

Objective $B\varphi_{dyn}$ (p_{lead})

In the following, the description of the implementation of the online optimization routines considers only the sub-function of the dynamic roll angle behavior φ_{dyn}.

Despite the interaction between the sub-functions, an isolated view is permitted by the application order of the phenomena. In the application process of the active anti-roll system the parameter setup that influences the dynamic roll angle behavior φ_{dyn} ranks first. The application of the further sub-functions – static roll angle φ_{stat} and roll moment distribution *WMV* – are executed afterward.

Subsequently the sub-function needs to be expressed in an objective value that is usable for an optimization routine. The general objective evaluation of a vehicle's roll behavior has been described by Botev, Kraft, and Riedel and Arbiger with appropriate test maneuvers. To demonstrate the implementation of the optimization routine, the steering step maneuver is used to calculate the objective variable $B\varphi_{dyn}$ (p_{lead}). It is calculated from the sum of the root mean square of A_0 and A_1 such that:

$$B\varphi_{dyn} = f(RMS(A_0) + RMS(A_1))$$

The desired response of the body-roll angle during a steering step maneuver is a fast – delay-free – rise of phi to φ_{stat} without overshoot (Figure 4). Thus A_0 represents the error area that correlates with an overcompensation or overly fast responding system behavior respectively. On the other hand, A_1 represents the overshoot error area correlating to a system's slow response time.

As indicated earlier, the measured lateral acceleration $a_{y,\,mes}$ is used as major reference variable to calculate the set anti-roll moment applied on the chassis. The control strategy includes a calculated, model-based acceleration $a_{y,\,mod}$ that is used in combination with the measured lateral acceleration to increase the system's response time. The parameters that influence this calculation are summarized to the parameter p_{lead}, as shown in Table 1. p_{lead} sets the lead rate of the lateral acceleration a_y,lead that is used to determine the set anti-roll moment M_{act}.

Optimization

Existing literature presents numerous methods and algorithms for parameter optimization problems. The task consists of a sufficient analysis for choosing an appropriate algorithm, so a sensitivity analysis determines the basic characteristics of $B\varphi_{dyn}$ (p_{lead}).

The aim of the sensitivity analysis is to explore the dependency of an objective function on its optimization parameter. Due to $B\varphi_{dyn}$ (p_{lead}), which is only dependent on the single optimization parameter p_{lead}, the sensitivity analysis is straightforward.

p_{lead} influences the lead rate of the control's lateral acceleration, which is a combination of the measured lateral acceleration $a_{y,mes}$ and the model-based calculated acceleration $a_{y,\,mod}$. Figure 5 shows the results of the simulated step steering maneuver for a parameter variation of:

$p_{lead,max\ 0} = 20 p_{lead,ref} > p_{lead,ref} > p_{lead,min\ 0} = 0$

For high values of p_{lead} the behavior converges on $B\varphi_{dyn}$ (p_{lead}). This is caused by internal control saturations. Thus the analysis shows that the objective function has the basic form shown in Figure 6.

The convergent behavior shows the requirement for a small number of optimization iterations. Due to the simple implementation – and without considering the computational costs – a basic grid search method is used for this scalar optimization problem. It represents a parameter

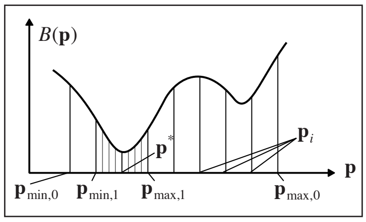


FIGURE 7 (LEFT): **EXEMPLARY GRID SEARCH**

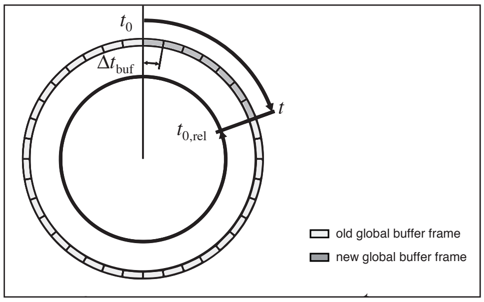


FIGURE 8 (BELOW LEFT): **CIRCULAR BUFFER WITH CONTINUOUS REORDERING TOWARD $t_{0,rel}$**

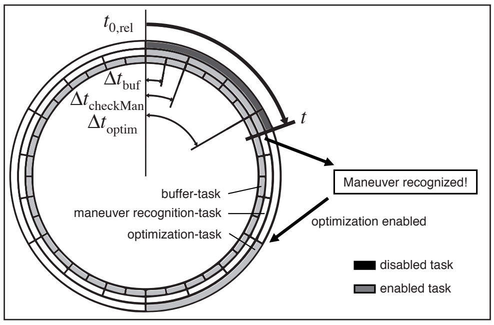


FIGURE 9 (BOTTOM LEFT): **PARALLEL PROCESS WITH REORDERED BUFFER DATA AND DIFFERENT SAMPLE TIMES**

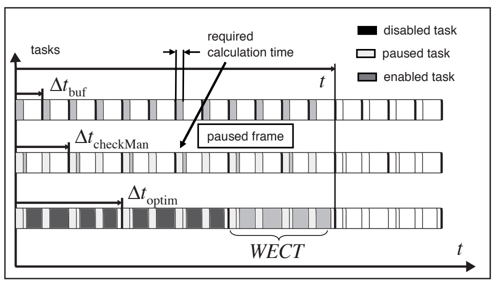
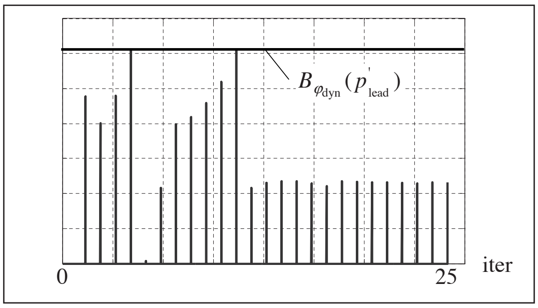


FIGURE 10: **STATIC SCHEDULE OF THE PARALLEL PROCESS WITH PAUSED, DISABLED AND ENABLED TASKS AND THE *WCET***



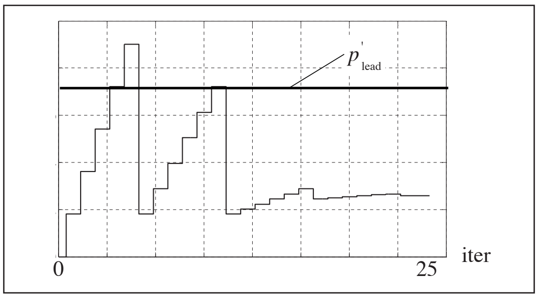


FIGURE 11 (TOP): **SIMULATION RESULTS OF** $B_{\varphi dyn}$ (p_{lead})

FIGURE 12 (ABOVE): **SIMULATION RESULTS OF** p_{lead}

optimization method of 0^{th} order, which only evaluates the function values $B\varphi_{dyn}$ (p_{lead}). $p_{lead,i}$ are determined symmetrically by separating the range between the upper and lower parameter boundary $p_{lead,max\ 0}$ and $p_{lead,min\ 0}$ in multiple equidistant spaces. After evaluating the function values, the grid is again spanned around the focus area of the current minimal value (Figure 7). The abort criteria consist of a minimum improvement in the function value *TolFun* and parameter *TolX* for a maximum number of iterations *maxFunEvals*.

Real-time implementation

The optimization implementation as a real-time process requires handling of the optimization task parallel to the active anti-roll system control. Hence, due to the hardware boundary condition of a used rapid prototyping single processor unit, it is a multitasking system. In addition the optimization needs to be available during the driving operation. Therefore a continuous data buffer as well as automated driving maneuver recognition are necessary.

The calculation of the objective function $B\varphi_{dyn}$ (p_{lead}) requires the entire data set of a driven steering step maneuver (Figure 4). Hence the measured vehicle u(t) is continuously stored over a period of time $t_{buf} = n\Delta t_{buf}$ in a buffer on the rapid prototyping ECU. The data storage implementation as a circular buffer provides a simply way of data post processing (Figure 8). The data record initially starts with the commissioning of the optimization environment t_0. The circular buffer's sample time equals the control sample time $\Delta t_{buf} = \Delta t_{ECU}$. In addition the stored data is reordered toward a relative start time $t_{0,rel}$ at each time step t. Thus the currently recorded data is always placed last in the buffer's data set, enabling continuous driving maneuver recognition.

The driving maneuver recognition routine calculates the error areas of the objective function $B\varphi_{dyn}$ (p_{lead}). Therefore the buffer-delivered steering wheel angle $\delta(t)$ is analyzed for the following criteria:

1. $\delta \geq \delta_r$ for $t_r \geq t_{r,0}$
2. Maintaining the static steering wheel angle $|\delta(t)| = \delta_{stat} \pm \varepsilon$ for $t_s \geq t_{s,0}$
3. Scaling A_0, A_1 in regard to $\delta_{stat}(t)$

As mentioned, parallel handling of the buffer, driving maneuver recognition, and optimization tasks are required for control of the active anti-roll system. The implementation hardware consists of a real-time rapid prototyping (RP) ECU. The RP-ECU provides a flexible control structure during the application of the active anti-roll system. Due to the RP-ECU's single core processor, the implementation requires a scheduling of the multiple tasks. Another boundary condition is the processor's fixed-base sample time Δt_{ECU}, which is generally required for real-time-capability of the ECU.

A real-time computer consists – in addition to the correctly calculated result – of a fulfilled physical time for each step. The tasks are further classified according to what occurs when the processing time for the calculation exceeds the deadline given for that step. The first task – the control of the active anti-roll system – is assigned as hard real-time task. Its compliance is essential, otherwise an overrun may influence the vehicle's driving behavior.

The parallel buffer, driving maneuver recognition, and optimization tasks do not directly influence the vehicle driving behavior if their permitted sample time is exceeded. In the event of an overrun their calculation result simply can not be used. Hence the number of timeouts only influences the efficiency of the optimization process. These tasks are classified as firm real-time tasks.

The classification of the tasks according to the result of missing the permitted deadline, correlates with a task's handling priority. The execution of the tasks is time-triggered and organized in a schedule. Within the schedule a priority is assigned to a task by defining its sample time. The base sample time Δt_{ECU} is assigned to the task with the highest priority – the control of the active anti-roll system. Parallel tasks with a smaller priority are associated with the higher sample times Δt_{buf}, $\Delta t_{checkMan}$ and Δt_{optim}, which are a multiple of Δt_{ECU}.

The parallel process is illustrated in Figure 9 according to task content.

The circular buffer's sample Δt_{buf} correlates with Δt_{ECU} to avoid a loss of sensor data:

$\Delta t_{buf} = \Delta t_{ECU}$

The driving maneuver recognition can be retrieved at each base time step, because of the reordering buffer. Its minimum sample time $\Delta t_{checkMan}$ is determined by the requested calculation time for one maneuver recognition iteration. Hence: $\Delta t_{checkMan} \geq \Delta t_{buf}$

The sample time Δt_{optim} defines the duration available for the optimization task. This event-triggered task is only executed if a recognized maneuver occurs, otherwise the task is disabled. Due to the task's fixed step sizes, which are determined periodically, the schedule of this multitasking system is called a 'static schedule'. The major task is executed preemptively. Thus the optimization tasks are interrupted if the major control task requests service. The static schedule of the multitasking system is shown in Figure 10.

As mentioned, a subordinated task is paused if a higher prioritized task is requested. Hence the sample time of a minor task has to recognize the termination time and the pausing time. In a static schedule system the feasible schedule of a task set is calculated off-line by assigning sufficiently high sample times. A measure for a feasible schedule represents the worst-case execution time *WCET*. This is the guaranteed upper boundary of the required calculation time for the multitasking process. Applied to the optimization, *WCET* equals the task time when the optimization is running.

Results

Figure 11 shows the exemplary online optimization process for a simulated steering step maneuver. $B\varphi_{dyn}$ (p_{lead}) converges as desired.

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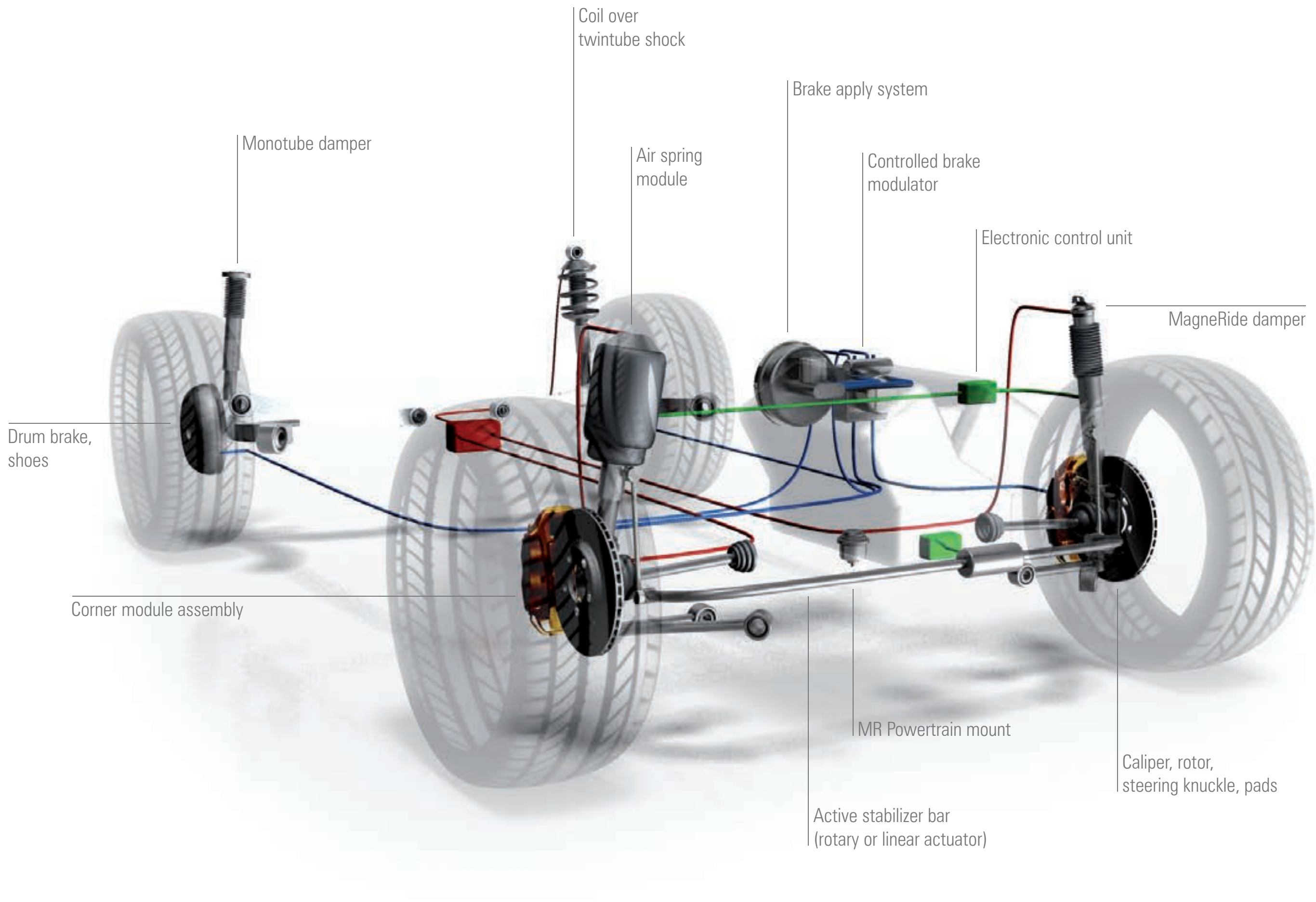

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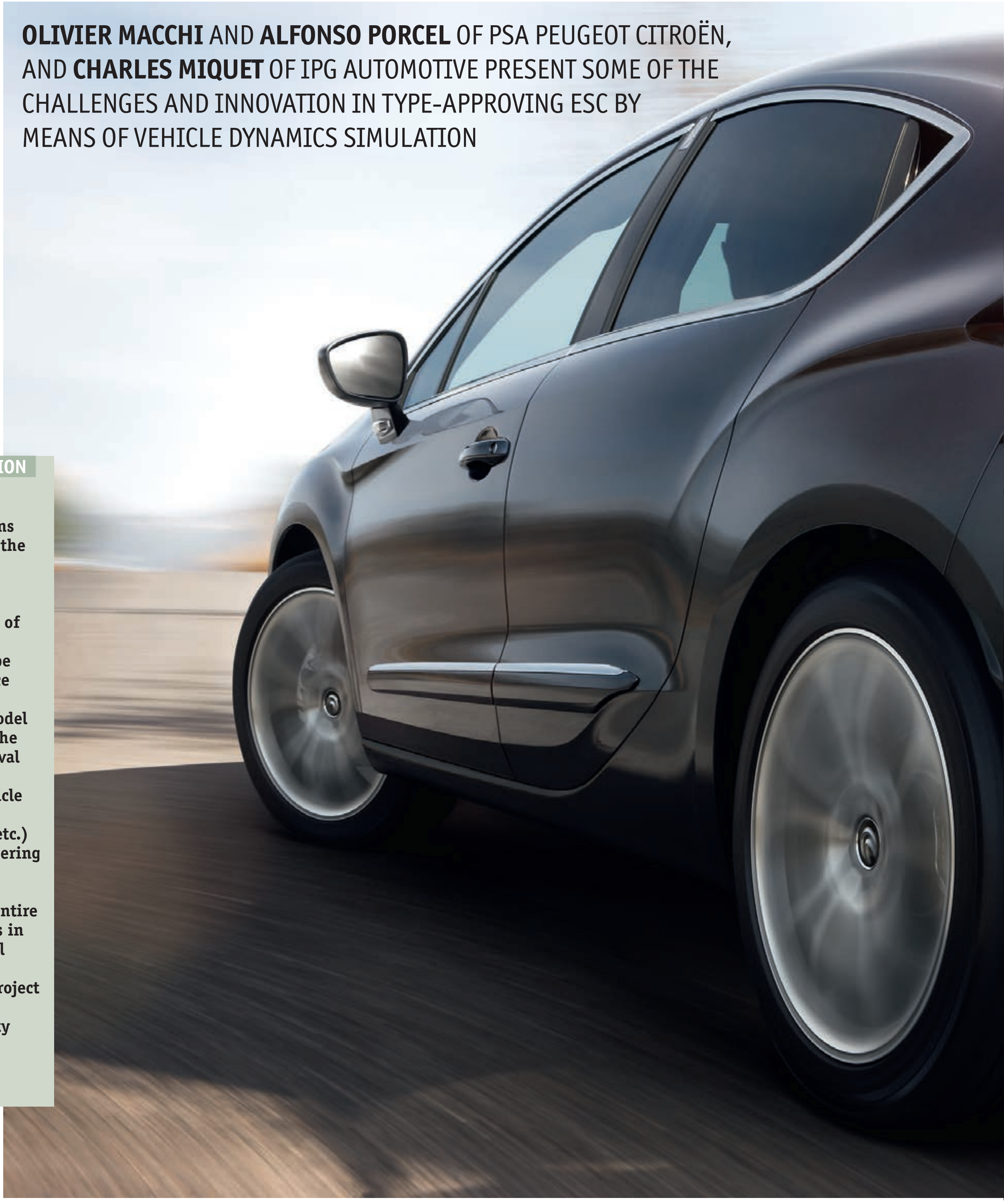
Sine of the times

OLIVIER MACCHI AND **ALFONSO PORCEL** OF PSA PEUGEOT CITROËN, AND **CHARLES MIQUET** OF IPG AUTOMOTIVE PRESENT SOME OF THE CHALLENGES AND INNOVATION IN TYPE-APPROVING ESC BY MEANS OF VEHICLE DYNAMICS SIMULATION



PROGRAM FOR TOOL VALIDATION

- Presentation of the PSA HIL process and the results of the numerical/physical correlations of the reference vehicle from the project under consideration
- Prior to the physical type approval test, construction of additional numerical variants of the reference vehicle
- Physical Sine with Dwell type approval tests on the reference vehicle
- Validation of HIL vehicle model performance compared with the results of physical type approval tests:
  – Detailed verification of vehicle behavior (yaw rate, lateral acceleration, side slip angle, etc.) by incorporating different steering wheel angles into the model measured on the test vehicle
  – Validation throughout the entire series of Sine with Dwell tests in accordance with type-approval criteria
- HIL simulation for all the project variants to be simulated
- Vehicle parameter sensitivity study to demonstrate the parameters affecting the type approval criteria

Electronic stability control (ESC) is the first active safety system to have produced a quantifiable effect on road accident statistics. In fact all the studies investigating the impact of this system (Swedish, American or by the manufacturers themselves) have shown that, under the same conditions, a population of vehicles equipped with this system, compared with an identical but unequipped population, will see its risk of accidents reduced by between 10% and more than 60%, depending on the type of accident (loss of control, vehicle-vehicle impact, etc.), grip and type of vehicle (SUV, sedan, etc.).

In 2009 the European Parliament and the Council of Europe adopted Brussels regulation (EC) No 661/2009 General Vehicle Safety. This made several active safety features mandatory in Europe, including ESC (in conformity with Annex 9 of regulation ECE R13H). ESC was made mandatory for vehicles in categories M1 (passenger car) and N1 (light commercial vehicle) from November 2011 for new models and from November 2014 for all new vehicles.

Application with the official technical service provider

Historically lateral vehicle dynamics have been more or less unregulated. Annex 9 of R13H therefore introduced by way of regulation a new technical area, though the terms of reference remained to be determined.

PSA and UTAC have been discussing this subject since early 2010 with a view to starting to define the conditions under which Annex 9 of R13H will apply to type approvals of the ESC system in vehicle projects. Among other matters, the text permits the use of simulation to obtain type approval (Appendix 1 of Annex 9 of R13H).

In its development process for the functions of ABS/ESP systems, PSA uses a numerical model correlated with each vehicle body applied. It was therefore natural to try to use this numerical tool to reduce the number of physical vehicle tests, since the text explicitly authorizes this.

The principle underlying HIL simulation involves integrating a 'real' ESC control unit into a real-time platform, simulating the whole of the vehicle, sensors and actuators used by the computer. This means that ESC is engaged as if it were in a real vehicle. This approach has the advantage of using a serial ESC control unit.

Validation of the PSA HIL simulation tool

In order to demonstrate the validity of this simulation approach and to define the ESC type approval process, a scheme to share the maturity level of this tool has been undertaken with UTAC.

The exchanges between PSA and UTAC have made it possible to define a working program in order to validate the simulation tool for type approval. Based on a project that is still at the development stage, a

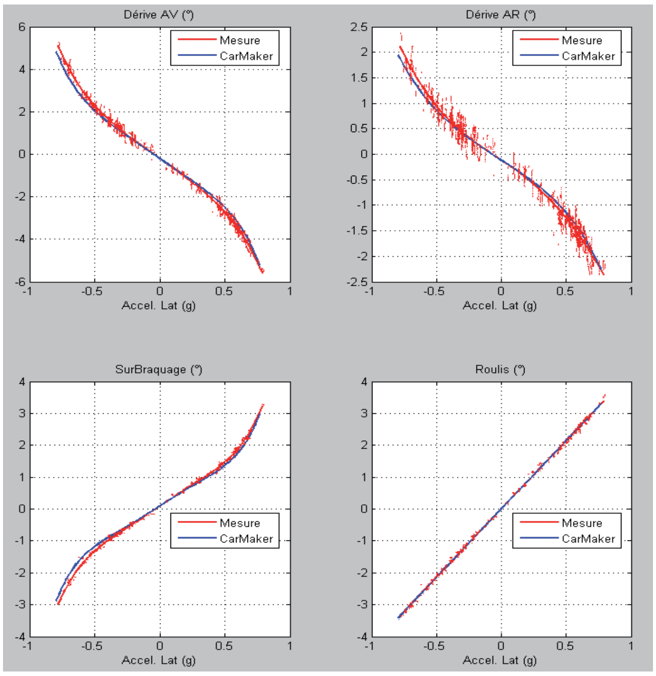


FIGURE 1: **EXAMPLE OF VALIDATION FOR STEADY STATE TEST WITHOUT ENGAGEMENT OF THE ESC SYSTEM**

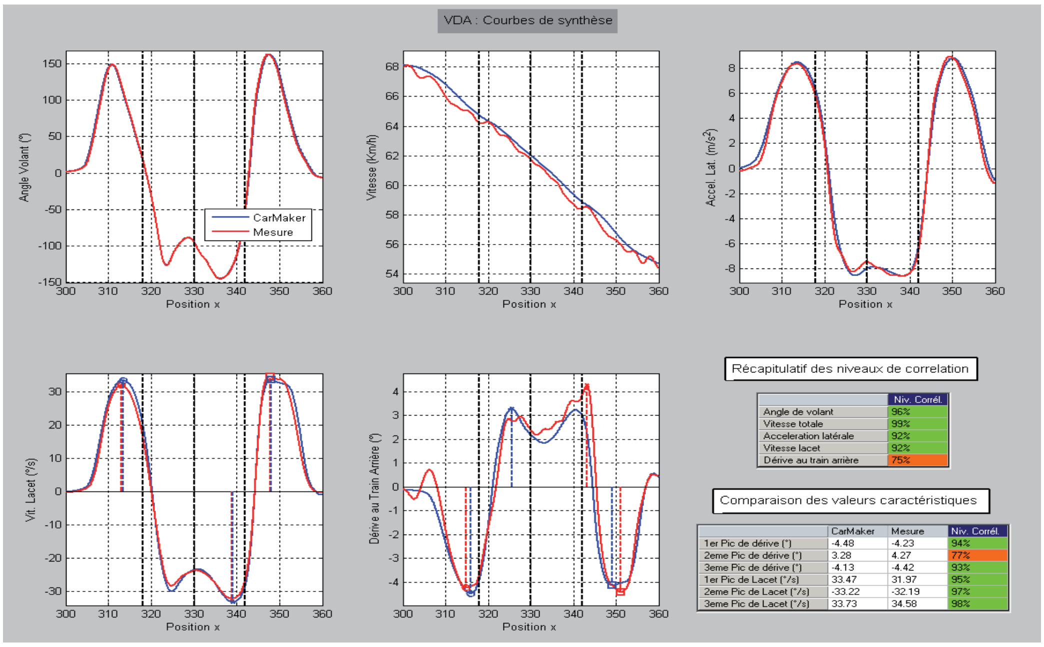


FIGURE 2 (ABOVE): **VALIDATION FOR A LANE CHANGE TEST WITH ENGAGEMENT OF THE ESC SYSTEM**

FIGURE 3 (BELOW): **SINE WITH DWELL MEASUREMENT FOR A STEERING WHEEL ANGLE OF 270°**

course of action has been implemented, (see sidebar p30, *Program for tool validation*).

PSA obtained representative models of real situations for vehicle dynamics applications integrating an electronic stability control (ESC) system. These models can therefore be subjected to extreme solicitations in situations close to the limits of controllability. The level of complexity involved in system modeling depends on how the model is to be used.

Given these objectives, the vehicle dynamics model selected by PSA for this type of simulation is CarMaker, from IPG Automotive. This model can be used in either a SIL (software in the loop) or HIL (hardware in the loop) environment.

PSA's choice was focused on HIL simulation, which produces realistic results and enables the user to test a standard ECU and its physical interfaces.

The first step in the model input data process concerns the recovery of functional characteristics necessary to define vehicle model parameters. This input data comes from different services within PSA or from external suppliers, and each item corresponds to a particular specification. This data includes architectural data (masses, inertias, center of gravity, wheelbase, etc.); chassis data validated against measurements on various characterization test bench (kinematics, compliance, flexibility spring, damping, anti-roll stiffness, tire characteristics, etc.); aerodynamic data derived from wind tunnel characterizations; parameterization of the braking circuit (absorptions, C*, diameter of master cylinder, etc.); and parameterization of the ESC hydraulic block (electrovalve cartographic maps, accumulators, attenuators, etc.).

For some projects, it is also necessary to develop and integrate specific subsystem models (e.g. hybrid powertrain, EPS, etc.) into the HIL test benches to be sure of having the required representativity.

Beginning with this initial step, it is possible to create simulations with or without an ESC system, however the representativity of the model at this stage is not guaranteed. To verify the representativity of the model, it is necessary to have access to dynamic measurements taken on vehicles fitted with instruments. There are five types of measurements.

The first is hydraulic measurements with passive pressure control. The upstream pressure is exerted by the driver. It is modulated in the receptors by the outlet and inlet valves (ABS, EBD functions).

For hydraulic measurements with active pressure control, the upstream pressure is exerted by the ESC pump. It can be modulated in the receptors by means of the outlet and inlet

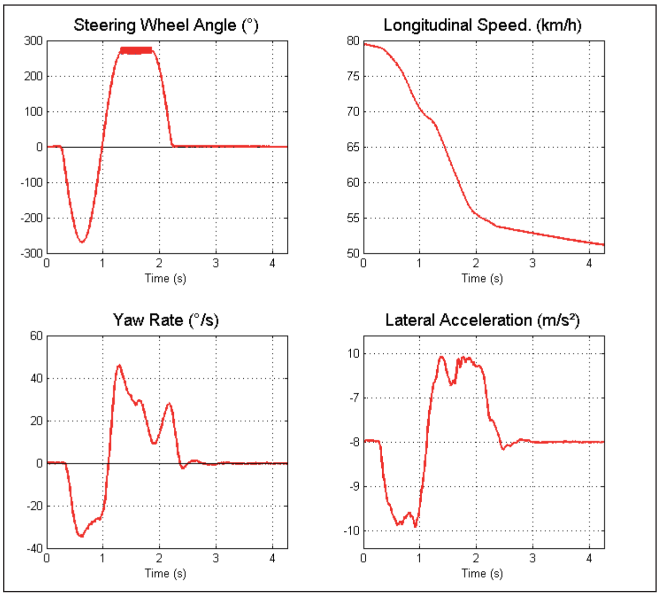


Table 1: ESC system disconnected		
Type of maneuver	**Validations**	**Acceptability criteria**
Driving with different steering wheel angles at constant speed	Steady state	Oversteering, roll, side slip angle of axle
Step steer for different levels of lateral acceleration	Steady state (after step steer) and transient state	Yaw rate, lateral acceleration, side slip angle
Steer with increasing frequency	Transient state	Frequency analysis of the transfer function between steering wheel angle and yaw rate
Slalom	Transient state	Phase shift in yaw rate and lateral acceleration
Power off in a straight line	Engine braking, rolling resistance	Longitudinal acceleration
Braking in a turn	Longitudinal/ lateral tire coupling	Yaw rate, lateral acceleration, side slip angle
Lane change	Longitudinal/ lateral tire coupling	Yaw rate, lateral acceleration, side slip angle

Table 2: ESC system connected		
Maneuvers	**Validations**	**Acceptability criteria**
Braking in a straight line with consistent grip	ESC longitudinal forces	Braking pressures, longitudinal acceleration, ESC regulation criteria
Braking in a straight line with asymmetric grip	Longitudinal/lateral tire coupling (ABS)	Braking pressures, longitudinal acceleration, ESC regulation criteria
Power off in a turn	Longitudinal/lateral tire coupling with engagement of ESC system	Yaw rate, transverse acceleration, drift, longitudinal acceleration
Braking in a turn	Longitudinal/lateral tire coupling with engagement of ESC system	Yaw rate, transverse acceleration, drift, longitudinal acceleration
Lane change	Longitudinal/lateral tire coupling with engagement of ESC system	Yaw rate, transverse acceleration, drift, longitudinal acceleration

valve, but also by the pilot valve (ESC, BASR functions).

Then there are vehicle dynamics measurements in steady-state behavior (ESC off); vehicle dynamics measurements in transient state (ESC off); and vehicle dynamics measurements in combined tire forces (lane change or braking in a turn).

From the model input data and measurement input data, the first level of correlation of the model can be established by comparing the curves produced by the simulation with those measured during the various maneuvers. When the results of the comparison are not satisfactory, a process to identify the model's parameters must be implemented to obtain the required degree of representativity.

This work is broken down into two distinct correlation activities – hydraulic model of the ESC system and the vehicle dynamics model.

Hydraulic model of the ESC system

After properly verifying the conformity of the parameters communicated by the PSA departments concerned, or by the suppliers, the action taken to identify parameters will concentrate on those that are not well known (e.g. pressure drop coefficients) in this model (made up of about 70 parameters).

A program of optimization based on experience gained in previous projects, coupled with a hydraulic model sensitivity study on each of the parameters, makes it possible to define a set of unique model parameters that accurately correlates all the measurements made on the vehicle (modulation of active and passive pressure, high and low pressure).

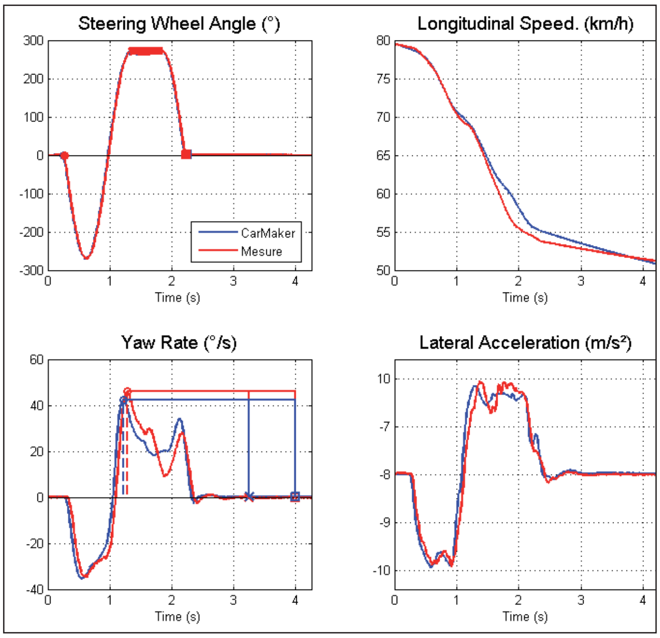
To carry out this process quickly, PSA has developed a tool that enables it to have representative simulation, undertake post-treatment and conditioning of the ESC signals recorded during vehicle maneuvers – necessary for reconstruction of electrovalve action – and activate the ESC pump, etc. The tool can also establish the initial level of correlation with the first parameters that are set, begin an identification procedure in order to improve the level of correlation, and validate the identified model.

Vehicle dynamics model

The objective of this phase is to obtain a vehicle dynamics model that corresponds to particular situations of life and where the validity conditions are known and limited (e.g. side slip angle of tire ≤18°, high level of grip, etc.). Sine with Dwell corresponds to these validity ranges.

To start, the performances of the model are evaluated with regard to the maneuvers measured on the vehicle. In the case of the HIL simulations, the maneuvers must cover a functionality range without and with engagement of the ESC system. This last case involves correlating the hydraulic model first. For each maneuver, specific technical criteria are used to determine the level of validity of the model (Tables 1 and 2).

If the model is not satisfactory, a process of identification to correlate the vehicle dynamics parameters is implemented in the same spirit as described for hydraulic model correlation, as explained above.



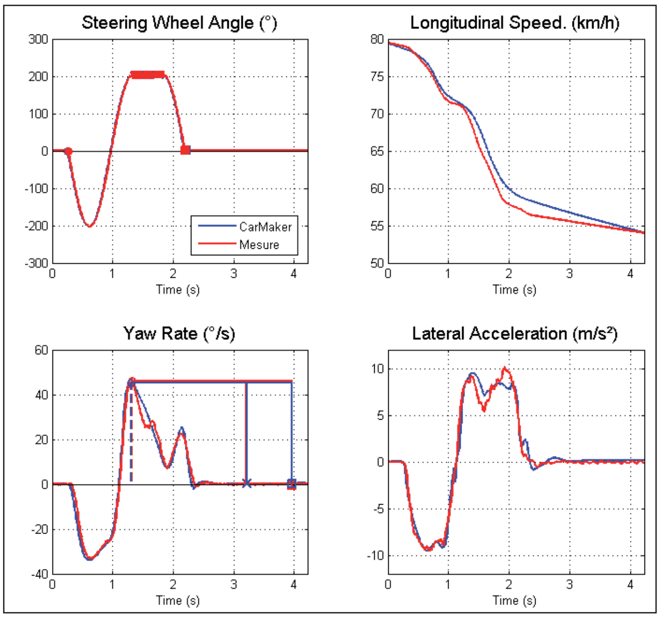


FIGURE 4A (TOP): **EXAMPLE OF TIME-BASED VALIDATION FOR STEERING WHEEL ANGLES OF 270°**

FIGURE 4B (ABOVE): **EXAMPLE OF TIME-BASED VALIDATION FOR STEERING WHEEL ANGLES OF 200°**

A compromise should be sought between the different operating points.

Figure 1 shows an example of the result of correlation for a steady state test. In the case of Sine with Dwell, it is important to have a good level of correlation for this type of maneuver because it is used directly to determine the initial angle A (SIS, Slowly Increasing Steer).

Figure 2 shows an example of the result of correlation for a transient state test (lane change). A good correlation on this type of test (ESC off and ESC on) is important, because it is representative of the demands of Sine with Dwell.

FIGURE 5 (RIGHT): **EXAMPLE OF A SIMULATION/MEASUREMENT COMPARISON FOR A SINE WITH DWELL TEST SERIES**

FIGURE 6 (BOTTOM): **COMPARISON OF THE EFFECTS OF CHASSIS/ MASSES (POWERTRAIN)/TIRES/ ADHESION**

SINE WITH DWELL MEASUREMENT

- **Data acquisition chain (minimum capture frequency: 200Hz)**
- **Motor-driven steering wheel measuring the angle, torque and speed of the steering wheel (precision: 0.25°, range ±300°)**
- **Vehicle speed sensor (precision: 1km/h, range 0 to 200km/h)**
- **Yaw rate sensor (precision: 0.15°/s, range ±50°/s)**
- **Lateral acceleration sensor (precision: 0.05m/s², range ±15m/s²)**
- **Body height lasers (precision: 0.6mm, range ±125 (front) or ±400mm (rear))**
- **'Brake pedal' position sensor**
- **Anemometer**

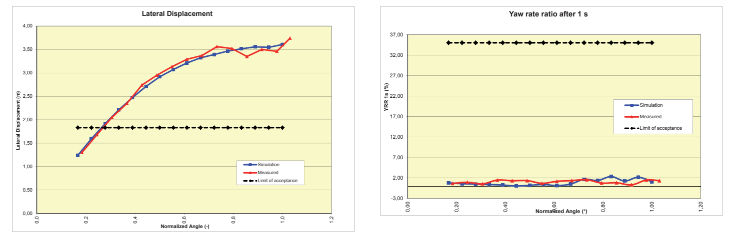


Figure 13: Example of a simulation/ measurement comparison for a Sine with Dwell test series

Table 3: Sine with Dwell results		
	Results	**Criteria**
Offset at 1.07s	~3 to 4m	>1.83m
% at T0+1s	~0%	<35%
% at T0+1.75s	~0%	<20%

When these two last steps are achieved the model should be ready for a Sine with Dwell simulation.

Sine with Dwell approval tests

A vehicle project undergoing development at PSA (C4 family) has been adopted to act as first line support for these Sine with Dwell correlation tests. To enable these tests to be carried out, the vehicle must conform to the specifications set out in paragraph 4.3 of Annex 9 of R13H. These specifications concern the conformity of the vehicle (defined in terms of chassis, architecture, aerodynamics, power steering and ESC software).

The measurement tools used by PSA to carry out the test are detailed in the sidebar Sine with Dwell measurement (above).

To perform the maneuvers described in section 5.9 of Annex 9 of R13H it is necessary to use a steering robot capable of producing the required steering wheel angles at a frequency of 0.7Hz.

After carrying out the processes of sensor calibration, brake and tires running-in and temperature setting, a series of Sine with Dwell measurements was started with UTAC in conformity with the specifications of section 5.9 of Annex 9 of R13H. Figure 3 illustrates an example of the result obtained during this test series (steering wheel angle = 270°).

The selected reference vehicle was the subject of the standard correlation process, as previously described. Based on this correlated model, the tests carried out in the presence of UTAC were reproduced on the HIL test bench. The exact test conditions (masses, grip, driver actions, speed, etc.) were incorporated into the model. In the first instance, a time-based comparison of the results was made.

Figures 12a and 12b show two of the Sine with Dwell maneuvers with their simulation results: steering wheel angle of 270°, the most unstable situation in the test series; and steering wheel angle of 200° with ESC intervention, giving a better level of stability.

Note that in the Sine with Dwell maneuver, after release of the accelerator, vehicle speed is the result of the model's functionality (engine brake, friction, braking action generated by the ESC system). Yaw rate and lateral acceleration are physical values useful for ESC type approval. A sufficient degree of precision must be achieved for all the simulated steering wheel angles.

Results of the Sine with Dwell test series on the right-turn direction are represented in Figure 5. These results – highlighted in Table 3 – show that the vehicle has no difficulty in fulfilling the type approval criteria.

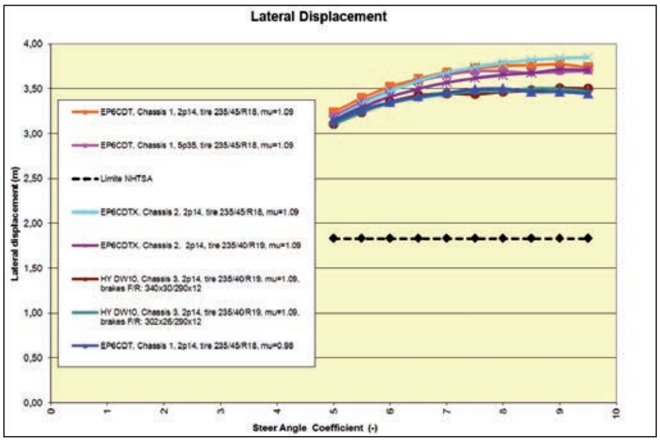
Study of the sensitivity of vehicle dynamics parameters with regard to type approval criteria

A sensitivity study on vehicle parameters affecting Sine with Dwell maneuvers was undertaken to illustrate the potential impact of different parameters on type approval criteria (mainly lateral displacement).

All the main parameters were modified. They mainly concern the vehicle architecture data, masses and their distribution, kinematics and compliance of the axle units, suspensions (flexibility springs and damping), anti-roll stiffness, braking characteristics, tire characteristics (tire slip stiffness, inflation pressure, etc.).

The result of this sensitivity study (see extract in Table 4) shows that in the vehicle studied, the position of the center of gravity in Y and X are the parameters with the greatest influence on lateral displacement at tBOS+1.07s. Modification of the front toe and the tire parameters (inflation pressure and tire slip stiffness) are also road-holding parameters that exert an influence. But overall, the impact of these modifications

Table 4: Sensitivity study results		
Road holding parameters	**Variation**	**Lateral offset**
Position of the C of G in X	±50mm	±3.50%(*)
Position of the C of G in Y	±10mm	±3.15%
Front toe	±0.25mm	±2.90%
Front tire pressure	±0.2 bar	±2.70%
Front tire slip stiffness	±5%	±2.10%
(*) An increase in the parameter entails a reduction of the criterion analyzed and vice versa		



remains weak. The discrepancies noted produce a delta for the maximum displacement of about 3%, which is very weak in view of the vehicle results and acceptability thresholds required by the regulations.

The coupling between the different parameters was not taken into consideration in this sensitivity study; an experimental plan will be required for this. In the meantime, this work is performed indirectly in each type approval when the variants of a project are qualified.

Figure 6 shows results regarding the displacement of distinctly different project vehicle variants (DS5 standard and hybrid). The details of the variants, to indicate their sensitivity to parameter changes, are shown in Table 5.

Figure 6 reveals that under the lateral displacement criteria of the vehicle, increasing the load (adding three passengers in the rear and 21kg in the trunk) for a given chassis definition, different chassis tuning (Chassis 1/Chassis 2) and a change of tires within a same range, each have a low influence with the same order of value. For the hybrid definition (in terms of axle definition and masses), its displacement curve is sensitive to parameter changes. It is also interesting to note that a slight reduction of grip produces an effect similar to moving from the standard to the hybrid definition.

Since the test needs to investigate high-level grip, it is important to control this parameter throughout the series of tests to obtain coherent results and good correlation. One of the advantages of simulation tools is to show this sort of tendency.

Results

The technical sharing implemented with UTAC has enabled the consolidation of a process to approve the entire range of a project. The process adopted and accepted by UTAC includes several stages. First is the definition of the reference vehicle within a project (which can define more than one), which will be physically tested and additional variants, which will be subject to type approval by HIL simulation.

The next stage is the creation of the HIL models corresponding to the whole group of these variants. Then come a series of physical Sine with Dwell tests on the reference vehicle, followed by validation of the HIL vehicle model by means of simulation/measurement correlation of the reference variant. HIL simulations for all the vehicle variants defined above are carried out before the results are formatted and the documentation PSA requires for ESC type approval is supplied.

A vehicle family (Peugeot 208, Citroën C4, Citroën DS5, etc.), can consist of a number of different body shapes (sedan, station wagon, coupe-cabriolet, etc.). For each of these body shapes there is a large number of vehicle configurations: axle units, suspension types, engine types, tires, brakes, etc.

Table 5: Vehicle variant details

	Engine types	Masses and distribution (front/rear)	Axle types	Tire sizes	Brake disc sizes	Tire / road surface adhesion
DS5 Standard	EP6C DT	2P14: 1690 (62% / 38%) 5P35: 1896 (56.4%/43.6%)	Front: PMP(*) Rear: Torsion beam	235/45 R18	Front: 302x26 Rear: 268x12	μ = 1.09 μ = 0.98
DS5 Standard	EP6C DTx	2P14: 1740 (60.8%/39.2%)	Front: PMP(*) Rear: Torsion beam	235/45 R18 235/40 R19	Front: 340x30 Rear: 290x12	μ=1.09
DS5 Hybrid	DW10	2P14: 1946 (57.4%/42.6%)	Front: PMP(*) Rear: Multilink	235/40 R19	Front: 340x30 302x26 Rear: 290x12	μ=1.09

(*) PMP: Pseudo MacPherson

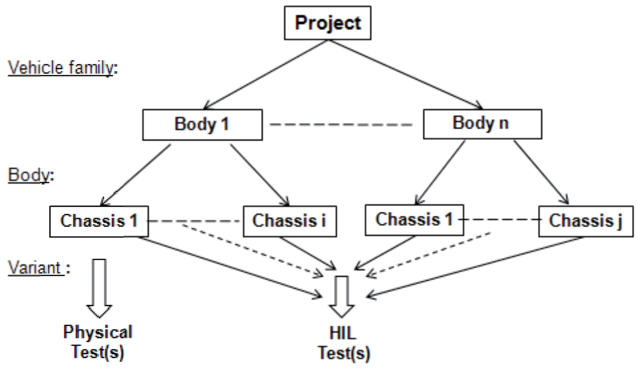


FIGURE 7: **VEHICLE VARIANTS DEFINITION**

The underlying principle is that a variant should correspond to a particular chassis tuning, and the engine chosen for each of these variants must cover all the brake system of the family. Figure 7 illustrates these views of variants.

To perform the ESC type approval process with 'industrial' HIL simulation, specific tools have been integrated into the PSA 'simulation tool box' so as to automate the launch of a series of Sine with Dwell simulations on the HIL test benches, post-process all the curves produced by the series of simulations, and generate the deliverables automatically in the format agreed with UTAC.

Conclusions and outlook

This article show that applying an internal simulation tool to the projects has enabled the fulfillment of the needs of R13H approval with limited effort. In addition to streamlining the number of tests, this approach also has the advantage of facilitating technical exchanges with the official laboratory, through visualization of the results and showing how they change according to the technical diversity of a vehicle range.

Since the formalization of this registration process by the UTAC, five projects have already been approved. The reliability of the results confirms that the degree of maturity reached by the simulation tools today is sufficient to meet this type of requirement.

Mixed with physical tests of correlation, this approach proves that simulation is an effective tool for this kind of field and that it could be extended to support other areas of approval or similar activities such as ISO 26262.

Right on cue

DRIVER INTERACTIONS ARE KEY TO HUMAN-AND-HARDWARE-IN-THE-LOOP (H2IL) SIMULATION. **PHIL MORSE** LOOKS AT ANSIBLE MOTION'S SIMULATOR SOLUTIONS

The human body is remarkable. Although we have only five limited-bandwidth channels with which to acquire data, and some clumsy appendages with which to enforce our commands, we have the ability to act as a sophisticated control system for complicated machines such as automobiles and airplanes (which are admittedly designed by us, to suit our own abilities. Good luck trying to pilot a vehicle designed by the inhabitants of, say, Planet X). We quickly get out of kilter, however, if we receive information that is not aligned with our expectations. Particularly sensitive is the human vestibular system, the canals and organs of the inner ear, which reacts to inertial stimulants and contributes to one's sense of balance and spatial orientation.

Although it would seem that an understanding of these details would be at the core of the human-machine interactions within driving simulator systems – or human-and-hardware-in-the-loop (H2IL) simulator systems, as they might more rightfully be badged – it is difficult to find anyone who can shine a light on this topic among the slew of hardware, graphics, and vehicle modeling experts who are actively engaged in the H2IL simulator world. Fortunately, Ansible Motion, based in Hethel, UK, has quite a bit to say on the subject.

Ansible Motion's driving simulators have a tendency to turn heads. These simulators simply look different from anything else around. The company has clearly ventured away from traditional simulator architectures by supplanting the 6DOF, variable-length strut, octahedral mechanism ('hexapod') with a completely new 6DOF motion solution of its own design, one that is as pleasing to the eye as it is remarkably compact. Ansible Motion's technical director, Kia Cammaerts, admits that this compactness is both a blessing and a curse for his company, which began designing H2IL simulator motion systems in 2009, and has provided solutions to a diverse field, from vehicle dynamics software companies to Formula 1 teams. Cammaerts reports, "We are happy when we receive compliments on the compact

"Particularly sensitive is the human vestibular system, the canals and organs of the inner ear, which reacts to inertial stimulants and contributes to one's sense of balance and spatial orientation"

MAIN: **AN H2IL SIMULATOR USING ANSIBLE MOTION'S 6DOF PLATFORM, HANDWHEEL LOADER, DYNAMIC VISION SYSTEM, AND CUEING STRATEGIES**

RIGHT: **H2IL ALLOWS ENGINEERS TO ACQUIRE DATA AS GENERATED BY A REAL DRIVER IN A CONTROLLED LAB ENVIRONMENT**

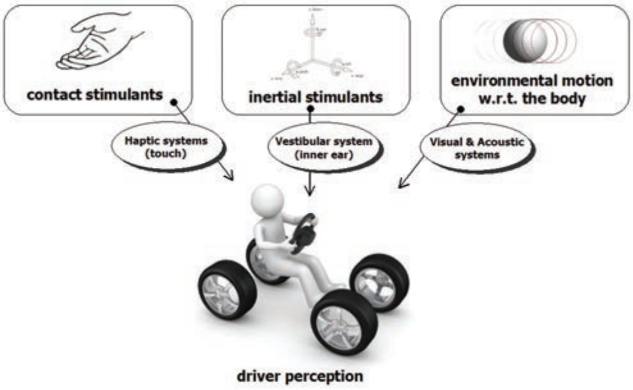
size of our motion platforms, but we are sometimes confronted with disbelief that they can replace hexapods – which are quite large and imposing, physically. As it turns out, our operating space is substantially larger than most hexapods'."

Cammaerts' assessment is true. Just as one cannot appreciate the capabilities of a bird by watching it perched on a fence post, appreciating Ansible Motion's 6DOF platform requires seeing it in motion. Applying power and commanding something as simple as a machine homing sequence provides an adequate view of this – the intricately layered linkage arrangement comes to life, breathes, expanding and contracting in and out of itself like an origami swan. Mesmerizing.

Pressing Cammaerts on how his motion platform came to look and function like it does quickly leads to a broader discussion, and circles back around to human interactions and perceptions. He explains the departure from the traditional hexapod by noting that, "Starting from first principles, it became evident to us that hexapods, like those seen in aircraft flight simulators, are actually poorly suited to automotive applications because they cannot capture vehicle transient dynamics. So we sat down and had a serious rethink. What you see – and do not see – is the manifestation of our thoughts on providing realistic motion cueing for the driver of an automobile."

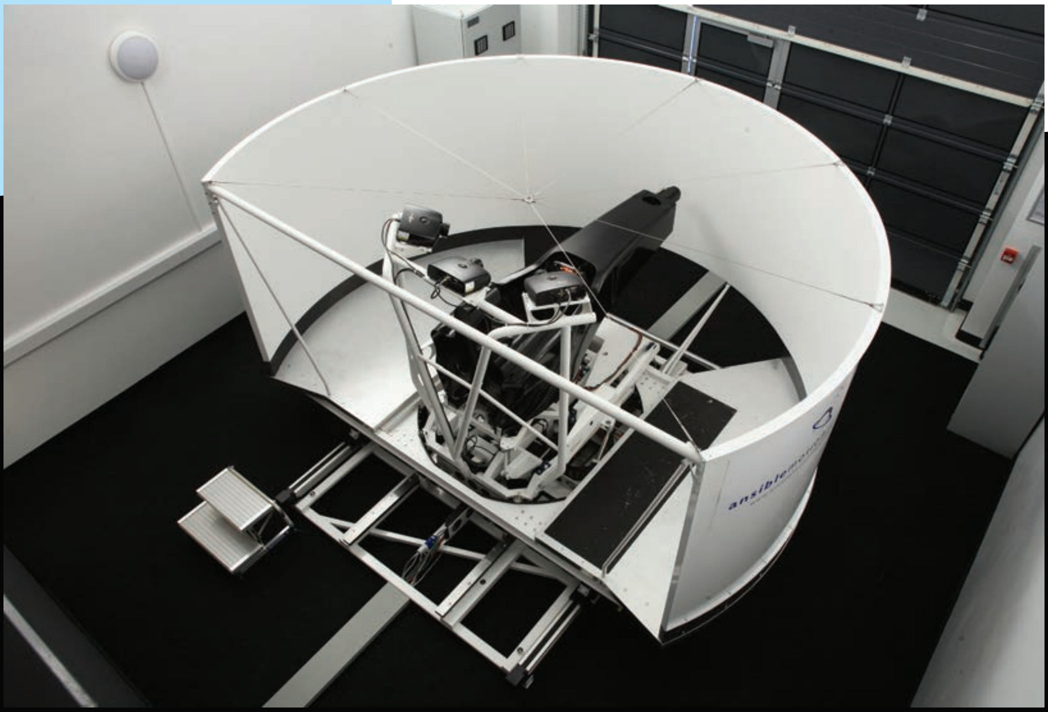
In the simplest terms, Ansible Motion's motion platform delivers independent, serial articulation authority for the three most important driver control cues. Yaw rotations, lateral movements, and longitudinal movements are each controlled by individual actuators and command logic. The implications of this for vehicle dynamics simulations are evident. Such a design avoids the two classic traps of the hexapod: the reduction of available motion bandwidth in other vehicle axes that occurs because all six actuators might use up a good part of their stroke to produce a single vehicle motion such as yaw, and the lack of independent authority over the interactions between motion axes that influence a driver's sense of directional stability. But the secret to Ansible Motion's approach is subtler than this, as alluded to by Cammaerts' oblique comment that there is something of importance that is "not seen".

In speaking of 'motion cueing', Cammaerts is referring to the algorithms that synchronize the movement of machine actuators with the vehicle physics and graphics. Cammaerts continues, "It is not as simple as extracting command signals from a vehicle dynamics model, and using them straightaway to drive our [motion platform and handwheel] motors and supplemental cueing devices. In fact, the goal is not to replicate actual vehicle motions at all, but to immerse drivers into compelling and consistent



"The goal is not to replicate actual vehicle motions at all, but to immerse drivers into compelling and consistent environments where they can interact with vehicles realistically from an engineering perspective"

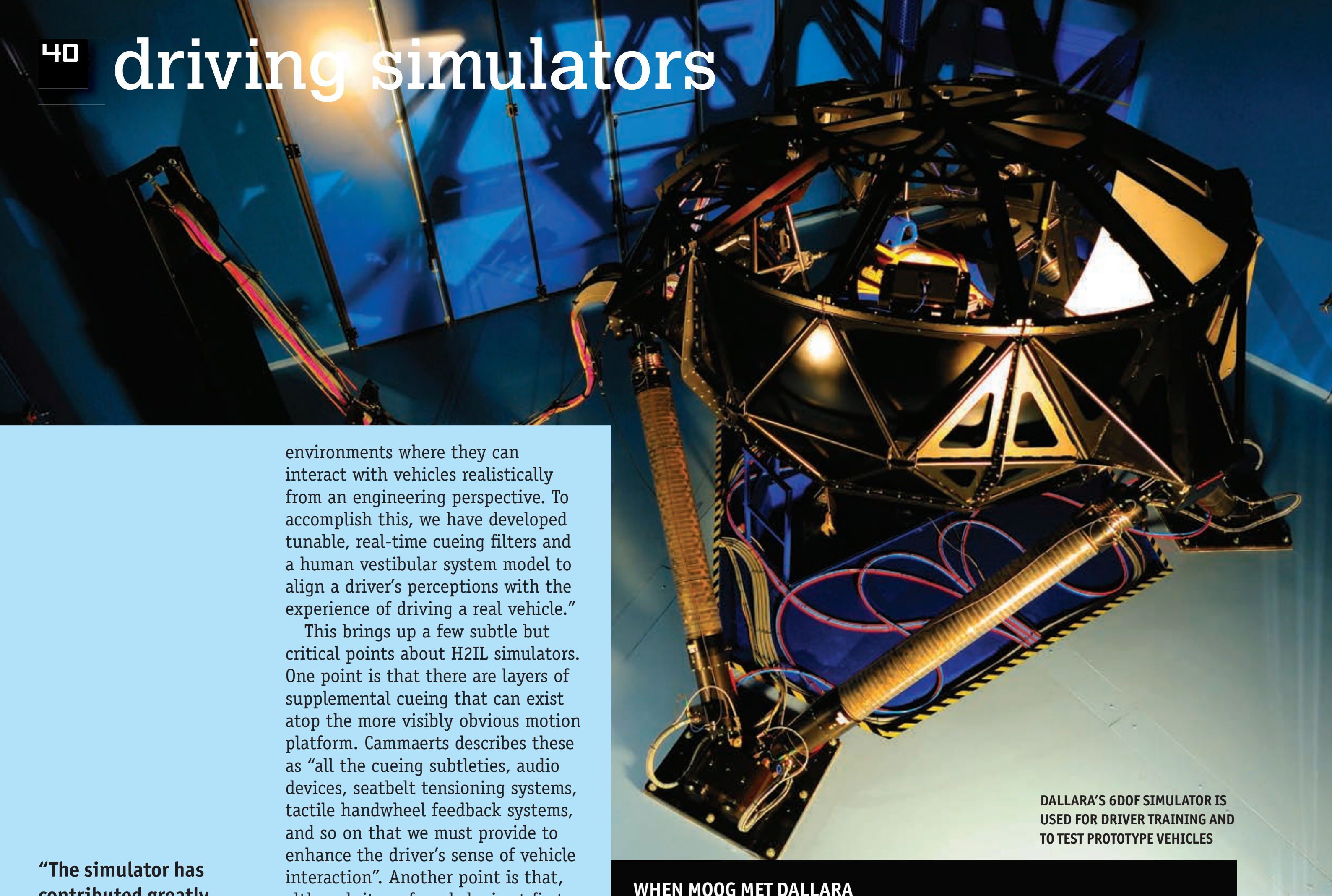
Kia Cammaerts, technical director, Ansible Motion



TOP RIGHT: **DRIVERS RECEIVE MULTIPLE INPUTS, FROM WHICH VEHICLE CONTROL DECISIONS ARE MADE**

CENTER: **A STANDALONE VARIANT OF ANSIBLE MOTION'S HANDWHEEL LOADER, AS MIGHT BE USED WITH A BENCHTOP H2IL SIMULATOR**

RIGHT: **ANSIBLE MOTION'S UNIQUE ARCHITECTURE FACILITATES THIS RATHER COMPACT H2IL INSTALLATION**



environments where they can interact with vehicles realistically from an engineering perspective. To accomplish this, we have developed tunable, real-time cueing filters and a human vestibular system model to align a driver's perceptions with the experience of driving a real vehicle."

This brings up a few subtle but critical points about H2IL simulators. One point is that there are layers of supplemental cueing that can exist atop the more visibly obvious motion platform. Cammaerts describes these as "all the cueing subtleties, audio devices, seatbelt tensioning systems, tactile handwheel feedback systems, and so on that we must provide to enhance the driver's sense of vehicle interaction". Another point is that, although it confounds logic at first glance, H2IL simulators are not replicating actual vehicle motions. They are, for lack of a better description, tricking a driver into interacting with a vehicle physics model in a realistic way. Engineers can extract useful data that assists with vehicle development and tuning, but a lateral acceleration trace (as a simple example) is extracted from the real-time vehicle physics model, not a physical sensor on the motion platform.

Further benefits, perhaps the primary ones, are derived from the ability to explore vehicle changes quickly, without the resource consumption (tires, fuel, travel, etc.), undesirable environmental variability, or changes in driver psychology that might accompany a physical part change or vehicle electronics system update on a proving ground or racetrack. Of course, drivers can also benefit tremendously from the 'training hours' that can rapidly accumulate in a properly engineered simulator. But all this is already evidenced by the proliferation of driving simulators within the automotive industry, where the case for H2IL simulators has, seemingly, already been made. The question now being tackled is how to do it right.

"The simulator has contributed greatly in the evaluation and refinement of car parts and bodies prior to the production phase. Most of the early prototyping can now be done by using sophisticated models and testing them in the simulator"

Andrea Pontremoli, CEO and general manager, Dallara

DALLARA'S 6DOF SIMULATOR IS USED FOR DRIVER TRAINING AND TO TEST PROTOTYPE VEHICLES

WHEN MOOG MET DALLARA

Dallara's high-performance driving simulator for race-car test and simulation became operational in 2010. Dallara had been looking for a way to effectively shorten product development time and reduce the cost of testing and driver training compared with track testing. Dallara asked Moog to provide the 6DOF motion system for the simulator and a solution was found in high-fidelity motion simulation, a technology that has been successfully used in systems for testing and training for the aerospace, defense, and automotive industries for decades.

The high-performance driving simulator was specially developed for test and simulation in motorsport. For this application, very low latency via higher acceleration and velocity are required in order for race car drivers to experience the most accurate feel of the car's behavior. For the 6DOF motion system, Moog designed new actuators that feature higher stiffness at lower weight in order to meet the ambitious frequency response specifications. The integration of a Moog control loading system to simulate the force feedback during steering, the special shape and construction of the dome, and the high-quality visual system, also helped improve the fidelity of the system.

"Over the two years that the simulator has been operational, it has proved to be an indispensable tool for us to reduce the development time and costs for new products, and the cost for car setup optimization and driver training as part of race preparations," says Andrea Pontremoli, CEO and general manager at Dallara. "It has contributed greatly in the evaluation and refinement of car parts and bodies prior to the production phase. For example, we have reduced the number of prototypes – most of the early prototyping can now be done by using sophisticated models and testing them in the simulator."

Including driver feedback early in the development process improves the vehicle model and reduces the need for design alternatives. The simulator allows feedback from less experienced drivers to be included, and it provides a safe environment for dangerous maneuvers.

For the Dallara simulator, Moog supplied a 6DOF motion system with high velocity and acceleration for optimal simulation of race cars; a motion cabinet (including real-time motion computer with motion cueing software); and a steering-wheel control loading solution for high-fidelity steering force feedback. The setup also comprises a very stiff dome with an optimized shape for the highest natural frequency response possible, to prevent false cues; a stiff visual screen integrated into the dome design, with a viewing angle of more than 180°; and an optimized projector mounting, resulting in what's reported to be excellent stable imaging during the roughest cues, to prevent blurry images and motion sickness.

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All-in-one

RACHEL EVANS LOOKS AT WHAT BHAI TECH'S IMPRESSIVE NEW DEVELOPMENT CENTER IN PADUA, ITALY, HAS TO OFFER VEHICLE DYNAMICS ENGINEERS

The Bhai Tech Advanced Vehicle Science Centre in Padua, Italy, which officially opened in June 2012, provides a variety of services, from vehicle and tire modeling to driver simulation and development.

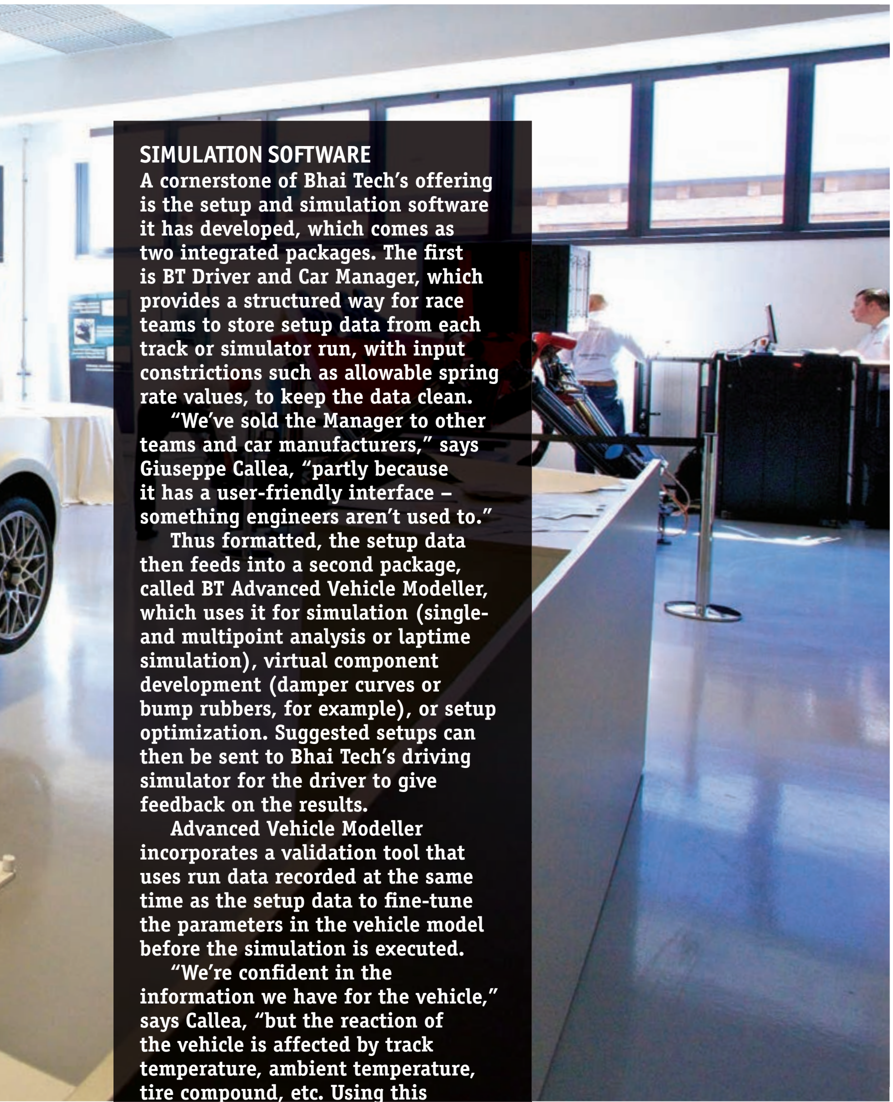
With two identical workshops, there is 800m² of space to work with, and each office space is around 500m². Each workshop provides the capacity to work on 10 cars at the same time. There are currently around 15 employees at the Bhai Tech facility, investment for which totalled the best part of US$6.3 million.

Roberto Costa, technical director at Bhai Tech, explains, "The facility is an all-encompassing tool. Under one roof we've housed a technical group that can do R&D, track work, future programs, code writing, simulation, modeling, and anything to do with vehicle dynamics in general. There is nothing else like this facility in Italy. Our intention is to work with all vehicles and to link this to the development of new race drivers and new components. We're aimed at motorsport teams, OEMs, drivers, engineers, and private engineers. We're accessible to all levels of motor racing. We're targeting teams more connected to GP2 toward LMP2 downward, rather than competing with an LMP1 or F1."

Bhai Tech also outsources when it makes sense to do so. "Tire modeling is one of the most sophisticated tasks in the development of vehicles," continues Costa, "and we do most of our tire work in the USA. We know a group of experts there who have developed all different strands of tire development modeling. We embraced this years ago and have continued to work with them on that."

Unique vehicle dynamics software has been developed exclusively for the Bhai Tech Advanced Vehicle Science Centre (see sidebar, right), as well as a unique driver simulator, designed and developed by Cruden in cooperation with Bhai Tech. GP2 and GT teams have been among the simulator's customers so far.

Also available is the Bhai Tech Racing aspect of the business, which is responsible for all on-track activities, offering manufacturers a platform for improving technically and logistically, as well as in their commercial activities. This includes testing, and support for vehicle- and driver development. For example, the firm has a wide-ranging contract with a GP2 team, which includes supplying it with a technical director and the use of the driving simulator in Padua. Bhai Tech has also raced its own McLaren MP4-12C GT3 as a



SIMULATION SOFTWARE

A cornerstone of Bhai Tech's offering is the setup and simulation software it has developed, which comes as two integrated packages. The first is BT Driver and Car Manager, which provides a structured way for race teams to store setup data from each track or simulator run, with input constrictions such as allowable spring rate values, to keep the data clean.

"We've sold the Manager to other teams and car manufacturers," says Giuseppe Callea, "partly because it has a user-friendly interface – something engineers aren't used to."

Thus formatted, the setup data then feeds into a second package, called BT Advanced Vehicle Modeller, which uses it for simulation (single- and multipoint analysis or laptime simulation), virtual component development (damper curves or bump rubbers, for example), or setup optimization. Suggested setups can then be sent to Bhai Tech's driving simulator for the driver to give feedback on the results.

Advanced Vehicle Modeller incorporates a validation tool that uses run data recorded at the same time as the setup data to fine-tune the parameters in the vehicle model before the simulation is executed.

"We're confident in the information we have for the vehicle," says Callea, "but the reaction of the vehicle is affected by track temperature, ambient temperature, tire compound, etc. Using this information to validate the simulation is the way to make the best use of it.

"Whether the car is running on the track or the simulator, the simulation and management tools are the same," he continues. "Simulator customers have started to log their data using the Manager software because they've found it easier to use than what they had before, providing a link between the simulation results and the changes to the car."

testbed for its ideas and software, not to mention a mobile billboard.

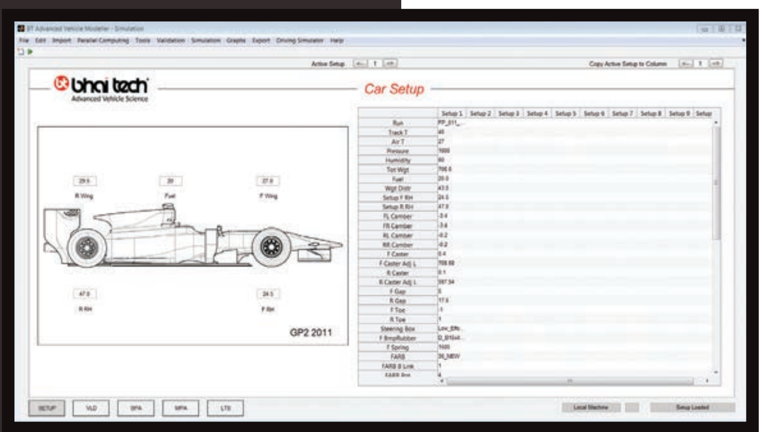
The accent isn't entirely on motorsport, however. Instead, the company has focused on the development of key aspects that it believes to be crucial to the research and development of any vehicle. Costa says, "We're not just focusing on motorsport, because manufacturers need to virtualize their cars, too. They can't just go from a CAD drawing; they have to be able to develop something at a much more sophisticated level than that – for safety, for economy, and for structural development."

As such, Bhai Tech offers services in research, analysis, and development in component design, and a computational grid for parallel computing.

With experienced engineers on board, such as head of vehicle dynamics Giuseppe Callea, it's all about who you know, according to the team at Bhai Tech. Costa emphasizes the importance of his contacts in running the business: "If a team feels that they have to do aerodynamic work, then there must be something they are not getting from the model they are using or the manufacturer of the car, or they don't have an engineering group big enough to carry out that work. Because we've done all this before and we have access to people in all those areas, we go directly to the people we've worked with in the past. We know what we're looking for and we know what they need."

"We're confident in the information we have for the vehicle, but the reaction of the vehicle is affected by track temperature, ambient temperature, tire compound, etc. Using this information to validate the simulation is the way to make the best use of it"

Giuseppe Callea, head of vehicle dynamics, Bhai Tech



BT ADVANCED VEHICLE MODELLER SOFTWARE USES SETUP DATA FROM THE MANAGER TOOL FOR SINGLE- OR MULTIPOINT ANALYSIS AND LAPTIME SIMULATION

Mass in transit

TOMASZ KRYSINSKI AND **JEAN-PAUL NAUZIN** (PSA PEUGEOT CITROËN) AND **FRANÇOIS MALBURET** (ARTS ET MÉTIERS PARISTECH) OFFER SOME FACTORS FOR CONSIDERATION WHEN DESIGNING A LIGHTWEIGHT VEHICLE

"Decreasing the vehicle's weight directly decreases the energy that has to be dissipated proportionately and therefore makes it possible to use smaller brakes"

The new worldwide accreditation cycle, the Worldwide Harmonized Light-duty Test Procedures (WLTP), will replace the current European cycle, the New European Driving Cycle (NEDC), as of 2020. It should be more representative of customers' actual use of vehicles. In particular, it will include more violent acceleration/deceleration phases and higher average speeds than the current cycle, with a view to bringing the measured consumption closer to that experienced by the customer.

The implementation of this new cycle will force automobile manufacturers to completely reassess the various impacts related to consumption. While the influence of some major factors in the NEDC cycle (downsizing or electrification of components, etc.) will be more limited in the WLTP cycle, the impact of factors such as weight will increase.

Figure 1 shows that the weight of vehicles has increased substantially over the past 15 years, a trend that is common to all manufacturers.

This change is due to a number of factors which, when listed in order of importance, are as follows: 40% due to passive safety; 15% due to the development of vehicle equipment; 15% due to improved soundproofing, as well as safety-related structural reinforcements; increased vehicle size; pollution reduction systems; and increased weight of other components such as suspension systems, brakes and gearbox systems, in order to offset the increased loads on vehicle axles.

The impact of weight on consumption has been shown in a statistical study carried out on 33 recent diesel vehicles. It roughly indicates that an additional consumption of approximately 0.5 l/100km can be expected per 100kg of extra weight. In practice, this is far more complex, as consumption depends on more than weight alone. Factors related to the vehicle's aerodynamic drag as well as its expected performance, which determines the size of the powertrain, must also be factored in.

Reducing consumption involves much more than engine development. All consumption sources – including tire rolling resistance and vehicle weight – as well as their reduction potentials, need to be assessed through a systematic, physical approach. It is also very important to account for the way in which the various actions are interconnected in order to achieve the best result.

Designing a lightweight vehicle

There are many ways to reduce vehicle CO_2 emissions by reducing weight: through direct effects (e.g. reducing the weight of components by using lighter materials) and indirect effects (virtuous cycle generated by initial weight savings).

In terms of direct effects, understanding the way in which weight and architecture are interconnected is key when it comes to lightweight vehicle design.

Vehicles are comprised of 'hard' items such as the engine and brakes, which constitute a mostly non-compressible weight when changing the vehicle's dimensions. Therefore, modifying the vehicle's dimensions will not have a significant impact on the weight of its components.

Put simply, on a constant equipment basis, the variation in weight from one vehicle to another is primarily related to the variation in the vehicle's dimensions. This is referred to as downsizing, or architecture fine-tuning, which ensures ample interior space by reducing the dimensions that are relevant for weight reduction purposes. To do this, the vehicle's weight is broken down into five parts, as shown in Figure 2. The statistical data on the same segment provides an average data for each segment, which varies according to the technologies used.

The value of these gradients is empirical and is used for pre-project calculations.

2

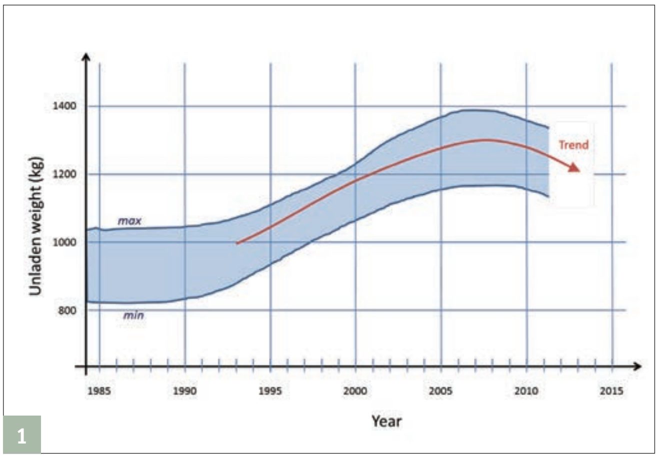


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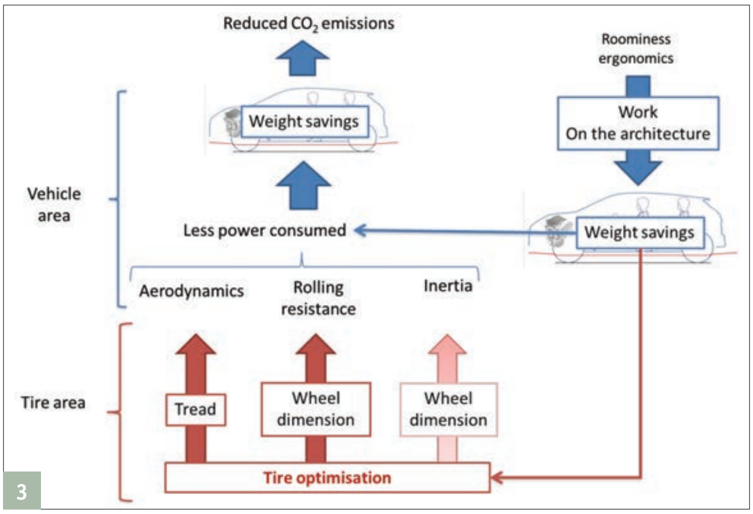
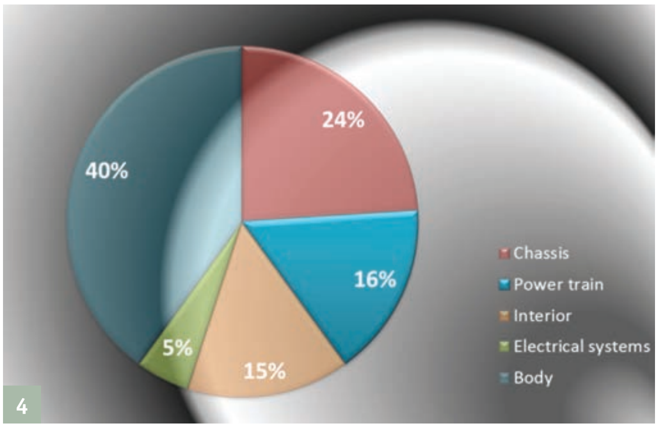
FIGURE 1: **CHANGE IN VEHICLE WEIGHT OVER THE PAST 30 YEARS**

FIGURE 2: **SIMPLIFIED MODELING OF THE VEHICLE'S WEIGHT BASED ON GEOMETRIC PARAMETERS**





Rolling resistance

Tires play a complex role in reducing fuel consumption, as they affect rolling resistance, aerodynamics, and the general architecture (Figure 3).

Rolling resistance is directly related to the wheels' tire geometrical data. The larger a vehicle's wheels are, the lower its friction coefficient is.

$$\mathbf{Crr} = \mathbf{Crr}_0 + \mathbf{f}\left(\begin{smallmatrix}\text{Tires}\\ \text{geometrical}\\ \text{data}\end{smallmatrix}\right)$$

With the f function dependent on the size of the tire and the technology used

The design of the tread layer can reduce the aerodynamic drag by a few percent. The design of the tread layer and the outer sidewall can reduce the air surface and optimize the airflow around the tire. When modeling in the preliminary vehicle design phase to define the impact of architecture on consumption, the effects of the aerodynamic pressure zones around the wheel area are neglected.

More generally, reducing consumption involves finding a compromise between the various factors related to aerodynamics, rolling resistance, and weight.

For example, increasing the wheels' diameter increases the vehicle's weight while maintaining the same interior space; reduces the rolling resistance coefficient; and increases the aerodynamic drag.

Overall, increasing the wheels' diameter increases the vehicle's consumption.

New materials and technologies

Once the vehicle's architecture has been determined, weight can also be reduced by introducing new materials and integrating components and functions in the same system. Figure 4 shows the weight proportions of the various subsystems of a vehicle.

A classical way to reduce vehicle weight involves the use of lighter materials such as aluminum, magnesium, plastic, or carbon fiber, as well as steel alloys. As mentioned above, reducing a vehicle's weight by 100kg results in savings of 0.5 l/100km. However, the large-scale distribution of these lightweight materials is not developed widely enough and is still too limited to high-end cars.

The use of new technologies can contribute to reducing vehicle weight directly. This can be the case by combining functions, optimizing components, developing intelligent solutions, or integrating functions such as third-generation bearings, one-piece parts, future mechatronics systems, or using nanotechnologies over the longer term. Given the large number of sensors and electronics in passenger vehicles, nanotechnologies have a strong potential to miniaturize and transfer data. A major hurdle is producing these components and integrating them into production lines, as well as recycling them.

Virtuous cycles

Following an initial weight-reduction phase by optimizing the architecture and choice of materials, a second weight-reduction phase can be obtained by assessing the effects of the first weight-reduction phase.

The first weight-reduction phase has effects on subsystems' loads. Each subsystem can then be optimized, thereby further reducing the weight of the overall structure.

Reducing a vehicle's weight and the inter-structure forces also has an effect on the vehicle's performance, which can be broken down into three main groups – impact, rigidity, and resistance. Each of these groups is affected by a set of performances that has a bearing on weight.

These performances need to be taken into account when optimizing each subsystem. This will often result in finding a compromise.

The downsizing of brake system components is one example of a virtuous effect. The required braking power at time t is expressed by:

$$P_{braking} = \underbrace{\frac{1}{2}\rho\, SCx\, V^3}_{\text{Aerodynamic}} + \underbrace{C_{rr} M\, g\, V}_{\text{Rolling}} + \underbrace{M\, \gamma\, V}_{\text{Inertia}} - P_{engine\ braking}$$

If we assume that the power losses related to rolling and aerodynamic forces are insignificant, as well as the engine braking action, then we can approximate the braking power output by:

$$P_{braking} \approx M\, V \frac{dV}{dt}$$

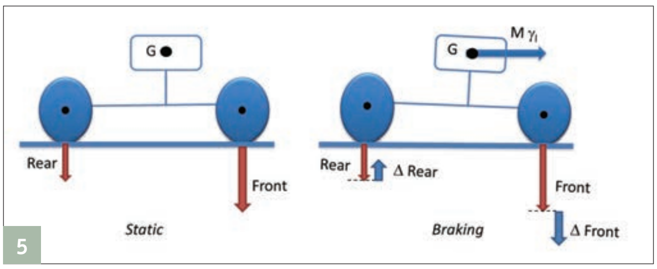


FIGURE 3: **THE TIRE'S INFLUENCE IN SAVING VEHICLE WEIGHT**

FIGURE 4: **VEHICLE WEIGHT SUBSYSTEM CONTRIBUTIONS**

FIGURE 5: **LOAD TRANSFER DURING BRAKING**

Material world

Magneti Marelli has been working on prototype composite suspension components for some years now, *writes Graham Heeps*. Having realized a mass-production-feasible, crash-safe, reinforced composite suspension arm that reduces component weight from around 1.5kg for an ultralight steel version to no more than 0.85kg, the firm's next target is to investigate the use of basalt- or glass-fiber to bring the cost down. The carbon-fiber version comes in at around five times the cost of the steel alternative.

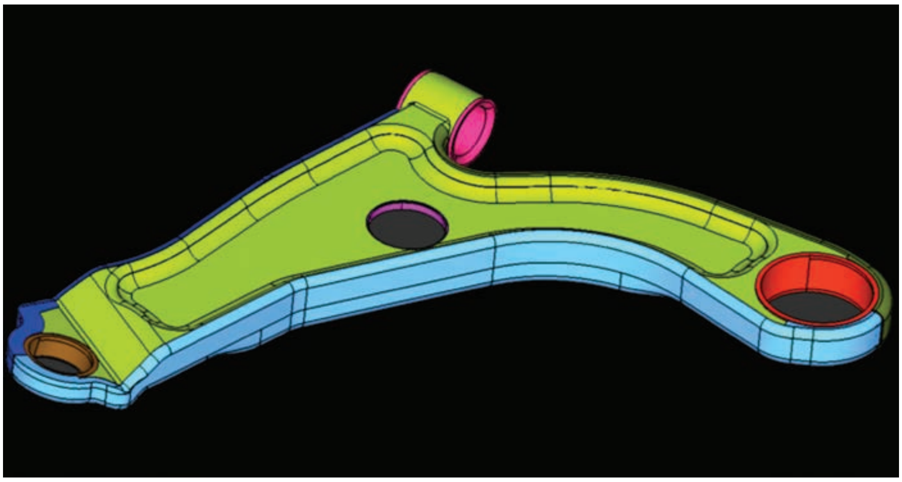
"In the next few years composite materials will play a very important role in automotive structural and safety components," says Piero Monchiero, suspension R&D manager at Magneti Marelli. "We are starting to understand which are the right technologies and which will be the right technical solutions to develop our parts and components with this kind of material.

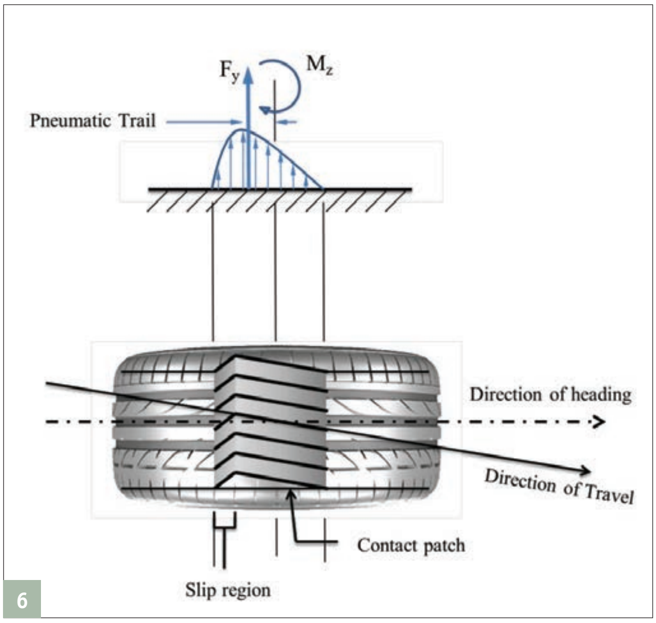
"We started with carbon fiber but along the way we've come to understand that it's probably not necessary. Basalt and glass-fiber will probably be good enough for conventional vehicles. The cost of glass and basalt is very acceptable but it is necessary to develop the technology a little bit to verify their reliability over the long term. Our parts have a long life, 15-20 years, so we have to verify their safety. But I think that we are near to having good, light materials [for suspension parts] using composites."

Magneti Marelli is studying the potential for composite knuckles and is also working on a front subframe made from a composite material. The latter probably won't need reinforcement from a ductile material, according to Monchiero.

"For components like the arms, which you have to sacrifice under crash conditions, yes it's necessary," he says. "But according to our first level of verification for subframes, it's probably not. These components have to be stiffer, more resistant, so it probably isn't necessary to use a hybrid [material] solution."

The rapid pace of development in this area provides a real opportunity for innovation, Monchiero believes. "The base plastics materials are being developed very quickly. I see that when I talk to suppliers of plastic materials over the course of a few months, they tell me that they now have new materials available. For example, when we started work on the composite arm, we used a thermoplastic material because at the time, this was the plastic material with the fastest curing time. But now thermoset materials are also achieving very short cure times, meaning that each operation needs less than one minute rather than the 10 or 15 minutes of before. We had to change to using thermoset material because the technology had developed so much."





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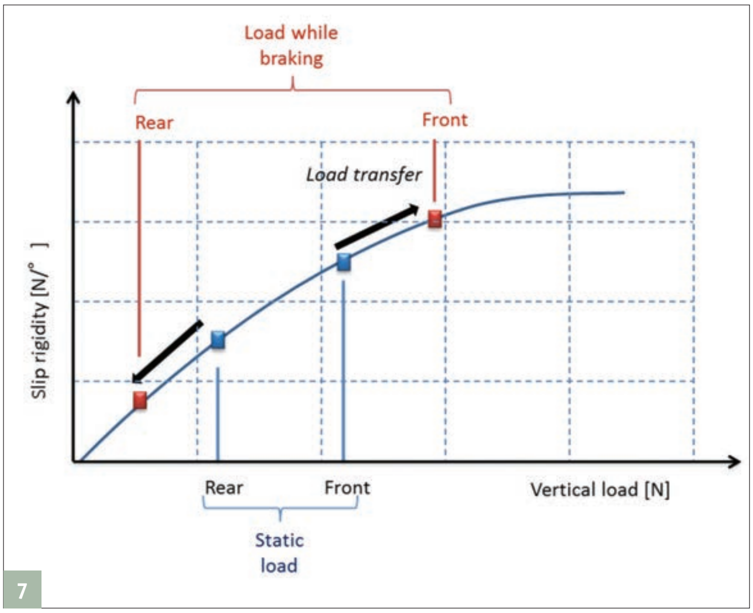
The energy dissipated over braking time T is given by the equation:

$$P_{braking}=\frac{1}{2}\rho\, SCx\, V^3+C_{rr} M\, g\, V+M\,\gamma\, V-P_{engine\ braking}$$

For example, a 1,400kg vehicle moving at 120km/h has to dissipate 776kJ of energy, which gives an average power of 141kW for a braking time of 5.5 seconds (deceleration of 0.6*g*). This energy that has to be dissipated has an impact on the force that has to be created inside the brakes. Decreasing the vehicle's weight directly decreases the energy that has to be dissipated proportionately and therefore makes it possible to use smaller brakes. The relationship between the energy dissipated and the size of brakes (and therefore their weight) is not linear or continuous. For example, changing from 283 x 26 front disc brakes with caliper 60 to 266 x 22 front discs with caliper 57, related to a 150kg reduction in gross vehicle weight, saves 5.7kg for vehicles in the C segment.

Weight reduction measures such as this can create problems, however, as the example of high-speed braking shows. A vehicle at a standstill has an initial load distribution on the various wheels. When rolling, and especially in braking situations, the load on each tire will vary. This results in an additive or subtractive force based on the stress placed on the vehicle. A number of parameters influence this weight transfer including the vehicle's weight, the position of the center of gravity, the wheelbase, the width of the road, the downforces, and the tires' performance.



The analysis of forces during braking shows that the rear end has a tendency to underload, and the front end on the contrary to overload. When we brake, we tend not to brake in a straight line, so a force is applied to the side of the tire. A lateral force develops in the contact area and the tire moves frontward at an angle a, called slip angle, with the direction of heading (Figure 6). This is how a tire's drift rigidity can be defined.

The approach is made more complex by the non-linear behavior of the suspension and tires (Figure 7).

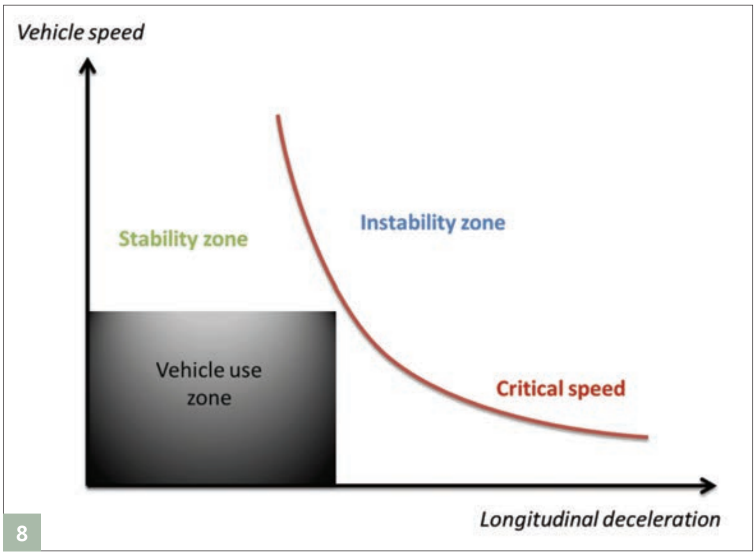
If the load discharged on the rear wheels becomes too little, the vehicle's performance will be affected. When braking, the load transfer is closely related to the longitudinal deceleration, the vehicle's speed, and the transverse acceleration (braking in a curve). Thus it is demonstrated, based on the distribution of these forces, that there can be potential risks of instability from certain vehicle speeds based on the deceleration (Figure 8).

This problem must be taken into account during the preliminary design phase for lightweight vehicles by modifying the position of the center of gravity, paying close attention to the front/rear weight distribution, or modifying the aerodynamic coefficient to hold the vehicle to the ground.

FIGURE 6: **SLIP ANGLE AND RESPONSE TO A LATERAL FORCE**

FIGURE 7: **TIRE'S DRIFT RIGIDITY BASED ON THE APPLIED LOAD**

FIGURE 8: **DIAGRAM OF STABILITY DURING THE BRAKING PHASE**



The outlaws

LOTUS F1'S REACTIVE RIDE-HEIGHT SYSTEM WAS THE LATEST IN A LONG LINE OF TRICK SUSPENSION TECHNOLOGIES TO BE BANNED FROM F1, AS **MATT YOUSON** EXPLAINS

As has become usual, Red Bull, Ferrari, and McLaren fought for the top positions in Formula 1 during 2012. But Lotus was right in among them, influencing the title battle, although a few tenths off the pace that would have made it a realistic contender for the title – its first since 2006.

It is therefore interesting to recall that the Lotus E20 was deprived of roughly that amount of lap time at the start of the year, when its reactive ride-height system was banned.

Lotus didn't utter any complaints about its innovative device being shuffled into obscurity by FIA decree – its installation was always likely to push the boundaries of legality – but it does demonstrate how the regulation of suspension occupies an ambiguous gray area where the spirit of the law and the written word are not always synonymous.

The system developed by Lotus for 2012 replaced the traditional rigid mounting of a brake caliper to an upright, with a pivoting attachment that, via hydraulic cylinders, would use brake torque to alter the ride-height of the car. The physics were straightforward, but the finesse required in application called for some very smart engineering.

"It was complicated," recalls Lotus technical director, James Allison. "Done wrong, you could get quite a nasty brake judder with this bouncing caliper. Engineering it so the car moves up exactly the amount that it would otherwise have dipped down by – that's all quite beautiful when you get it right. We were rather proud of having done it and were looking forward to using it."

Lotus ran the system at the Yas Marina Young Drivers' Test in November 2011, and again the following week in practice at Interlagos. Word spread, and over the close season – with little else to discuss – the media portrayed reactive ride-height control as the 'killer app' for 2012. It became a talking point, and FIA opinion, which had formerly been cautiously positive, was reversed and the technology was reluctantly put back in the box.

Muddy waters

It seems odd that, with a rulebook of technical regulations laid out in black and white, there's still sufficient ambiguity that F1 teams with increasingly limited resources are prepared to gamble on developing systems of dubious legality. In reality, F1's technical regulations have an organic quality, having evolved over many years to a point where original meanings are often confused, redundant, or contradictory. That's certainly the

LOTUS RAN ITS REACTIVE RIDE-HEIGHT SYSTEM AT THE 2011 YOUNG DRIVERS' TEST AT YAS MARINA

DAMON HILL DROVE THE ACTIVELY SPRUNG WILLIAMS FW15C TO VICTORY IN THE ITALIAN GRAND PRIX, MONZA, 1993

"Essentially, the entire suspension system is in breach of the overwhelmingly powerful Article 3.15"
JAMES ALLISON, TECHNICAL DIRECTOR, LOTUS F1

case with suspension, the rules for which laid out in Article 10 of the tech regs are inconsistent with a literal interpretation of Article 3.15, the catch-all aerodynamic reg, which states: "Any specific part of the car influencing its aerodynamic performance must be rigidly secured to the entirely sprung part of the car [and] must remain immobile in relation to the sprung part of the car."

Allison continues, "The arguments raised against our system were based around the overwhelmingly powerful Article 3.15. Essentially, the entire suspension system is in breach of this article; the default position is that all F1 cars are illegal… unless permitted by precedent and history, etc. The practical guideline adopted is usually that, if it complies with Article 10, then it's okay."

Lotus was prepared to defend its idea with a case that it didn't breach Article 10, making the argument that they didn't introduce any forces into the suspension system other than those generated in the normal course of braking.

"Our system didn't respond to brake pressure," explains Allison. "Sit in the pits and pump the brake, nothing happens; it responds only to the forces at the wheel. There has to be a wheel rotating on a track with a contact patch in order for these brake forces to do anything. And so the caliper reaction happens only as a response to the changing forces of the load applied to the wheels. It applies a load to the wheels – that's what the brake does – and then it responds as a result of that."

The counter-argument is made by F1 race director and head of the FIA's technical department, Charlie Whiting. "Normally these things are sorted out before a proper system appears on a car. With regard to the reactive ride system, I agreed in principle after James [Allison] showed me a highly schematic layout. Thinking about it some more, and seeing a much more detailed design, it became clear the primary purpose was undoubtedly aerodynamic – so my opinion on that was that it was contravening Article 3.15. I also felt that, because the position of the wheel was being affected by something other than vertical movement of its suspension travel, it could also be questioned under Article 10."

Whiting acknowledges that contradictions exist in the written word, but argues that practicality and the guidance of primary purpose can provide a consistency to decision-making.

"The fundamental relationship between Article 3.15 and suspension goes back to the World Motor Sport Council decision in the early 1990s that banned active suspension. The decision of the World Council said that, although it was acknowledged that suspension would affect the aerodynamics of a car, so long as that effect was incidental to the primary purpose of attaching the wheels and providing a proper suspension system, then it would be allowed.

"With that in mind, I think it's wrong to say that suspension is incompatible with Article 3.15. Clearly you need wheels, and clearly you need suspension – but where the questions arise are over matters such as reactive ride or, to take an example from a few years ago, the mass dampers. Initially they looked innocent – but it began to emerge that the primary purpose wasn't to even out bumps in the road, but to enable teams to run a totally different ride height regime. Hence we thought its primary purpose was more aerodynamic than ride control."

Worth a punt?

The F1 paddock is rarely shy in pointing to perceived illegality, but the extent to which it is pursued by rival teams, argues Allison, often involves a degree of calculation: "We felt reactive ride was worth a punt. We could quantify its advantages and it was going to give us a couple of tenths of a second; we were pretty sure we had the engineering right, so it was not going to give us any trouble; and we thought it could be copied but would be tricky to copy – and that's important.

"Invent something that can't be copied and you know you're always onto a loser," continues Allison. "No one is going to eat a whole season of not being able to build their own version, and so they're really going to attack it. On the other hand, with something that can be copied, they might eat a month or two of not having it and just get on with building their own – that influences decisions a bit.

"Mass damper was tricky for several reasons. I believe there was the rather awkward fact that some teams had asked about it previously and been given a negative answer. We [Renault] never asked specifically about it. We said what we were doing, but we'd never had the courtesy to go to him [Charlie Whiting] and ask if it was OK.

"But the system was cheap, it was simple, and it was very easy to copy… but there was also the fact that other teams – or at least McLaren – were already running a J-damper inerter-type technology. It was not as powerful, but was a useful technology that had a similar effect [as a mass damper] without the perceived illegality of the unconstrained mass. It meant the mass damper had much less allure for them and so it wasn't a train they wanted to jump on: they'd rather the train never left the station. It was wholly rational for them to attack it, leaving them with their inerter technology that we, and most of the grid, didn't have."

Both mass damper and reactive suspension were perhaps minor considerations given disproportionate attention. The same cannot be said for the decision to ban active suspension, which caused widespread consternation in a paddock that had adopted the concept wholesale (see *Active suspension*, p52).

WRC

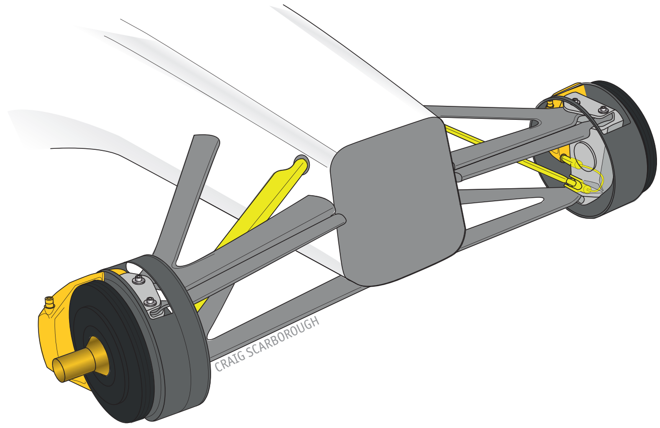
F1 wasn't the only sport to take a dim view of active suspension, though when WRC similarly banned it a decade later, it's reasoning was considerably more straightforward – albeit not to everyone's taste.

"It was wrapped up in the middle of one of our fairly regular – almost monotonously constant actually – streams of anti-cost, anti-technology hysteria," says Prodrive's David Lapworth, then technical director of the Prodrive-prepared Subaru World Rally Team. "From time to time cost control suddenly becomes the focus of WRC and the usual reaction to cost is to blame the engineers and so the sport responds by taking away their toys.

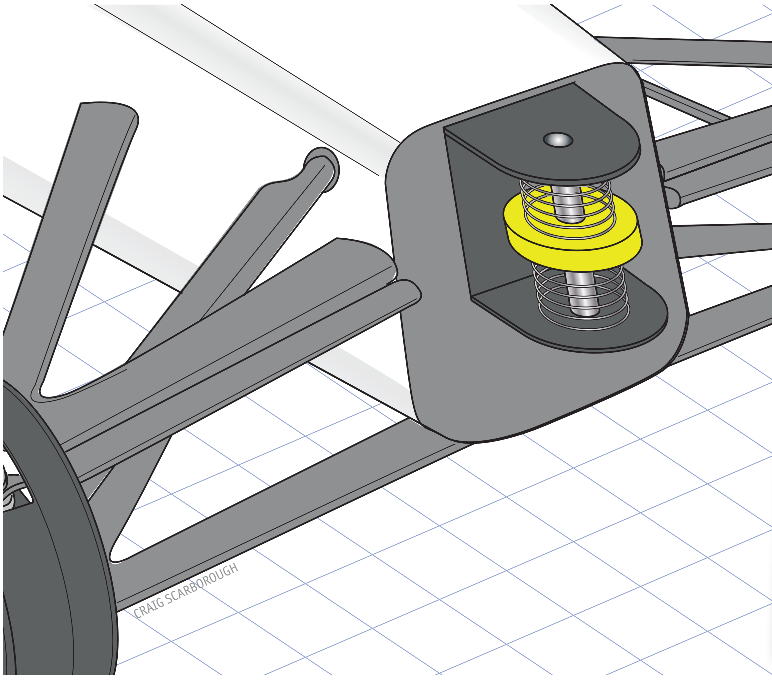
"I don't think that's a very sensible approach. I'd rather we looked at our branch of the sport, understood where the value is and then evaluated our technologies against that: are these relevant technologies? Are they going to enhance the sport and appeal to the fans and develop technologies that are relevant to road cars? If they are, they should be allowed. If they're completely specific to motorsport and don't add to the show, then there's a good case to ban them. Diving in and banning things on the basis that they look expensive isn't a very effective way of working."

Lapworth's ire is perhaps explained by the high hopes Prodrive had for the active system used by Petter Solberg on the 2003 San Remo Rally as a one-off test, shortly before the active suspension ban was announced. Unlike the F1 cars of a decade earlier, WRC was looking at the active suspension as an end unto itself, rather than as an aerodynamic aid, attempting to minimize body disturbance, keep tires presented to the road at the correct angle, reducing roll and pitch, and enable a car to run with soft suspension while retaining crisp body control.

Subaru's solution retained springs but ran a heavy hydraulic system at pressures approaching 200 bar. "We retained springs to support the



ABOVE: **BRAKE CALIPER MOUNTING AND HYDRAULIC CIRCUIT INSIDE THE UPRIGHT TO EXTEND THE PUSHRODS AS PART OF LOTUS'S OUTLAWED REACTIVE RIDE-HEIGHT SYSTEM**



LEFT AND BELOW: **THE MASS DAMPER HELPED FERNANDO ALONSO'S RENAULT R26 TO THE 2006 TITLE. THE DIAGRAM SHOWS THE MASS PLACED BETWEEN TWO SPRINGS, AS TAKEN FROM THE WORLD MOTOR SPORT COUNCIL EVIDENCE ON THE MASS DAMPER CASE**

RIGHT: PETTER SOLBERG'S ACTIVELY SUSPENDED SUBARU IMPREZA WRC ON THE 2003 SAN REMO RALLY

"Initially mass dampers looked innocent, but it emerged that the primary purpose wasn't to even out bumps in the road"
CHARLIE WHITING, HEAD OF THE FIA'S TECHNICAL DEPARTMENT

weight of the car, and a very soft, conventional damper arrangement to take control of the damping, mainly of the unsprung mass," explains Lapworth. "The control of the ride height, the attitude of the car, the roll of the car we did actively through the hydraulics. To all intents and purposes it was a full active car."

Legacy

McLaren technical director Paddy Lowe, who as joint head of electronics at Williams in the late 1980s and early 1990s was a leading light in the active suspension revolution, argues that the work done on active suspension in F1 was not a wasted effort, because the electronic systems and technological infrastructure that supported the active cars outlived the active era and formed a foundation for the technical advances that would follow.

Lapworth likewise says the research into active technology had benefits beyond its limited lifespan: "It wasn't completely wasted. Yes, with the benefit of hindsight we injected a lot of time and energy into something that was squandered, but the process that we went through specifying exactly how the system would work, understanding where the benefits would be, that probably helped us in our understanding of what we think rally car suspensions need to do. So that knowledge was in the bank and has helped us with what we've done since. But it would have been a lot cheaper to do that process as an R&D exercise rather than chasing it all the way through to a real car."

Motorsport technologies tend to be cyclical, but neither F1 nor WRC has shown the slightest interest in softening its respective stance on these banned suspension technologies (though F1 teams now use systems that are to all intents active suspension when straight-line testing, purely to save setup time). There is, however, a case to be made for raiding the ideas bank in the future. F1 seems to be committing to a road map of engine downsizing, with the aim of maintaining lap times from increasingly frugal powertrains. Energy recovery is in place, turbocharging is next on the menu, and the return of ground-effect cars has been mooted. But ground effect doesn't have the same technology transfer attraction as speed-sensitive ride height control.

In rallying, the appeal is perhaps more obvious. "The kind of active suspension we were playing around with is probably low on people's agendas – but more and more road cars have got electronic damping and variable suspension systems and torque modes," says Lapworth. "We'd be in favor of seeing more freedom in these areas because there is relevant technology that shouldn't be too expensive."

ACTIVE SUSPENSION

"It was the Canadian Grand Prix of 1993," recalls Charlie Whiting. "I was the technical delegate and my report basically declared them all to be illegal.

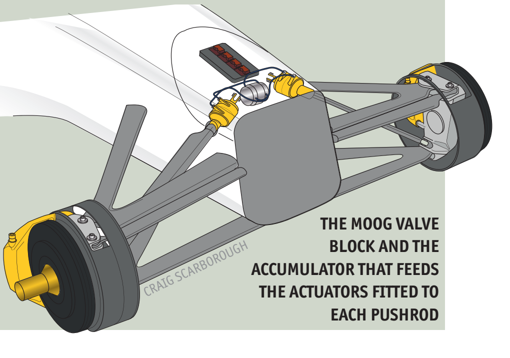
"The sense grew up over time, in a way very similar to the mass damper, where more and more facts emerged and you realized the primary purpose of the suspension system was to improve the aerodynamic performance of the car. Obviously the teams using it – Williams most notably – were not of that view, but the World Council concurred and the systems were banned."

At Williams in the late 1980s and early 1990s, Paddy Lowe had been instrumental in developing the FW14B and FW15C active cars that would dominate F1 in 1992 and 1993, with 30 pole positions and 20 victories. "It was a bit of a shock to everybody," he says. "It was all quite new, to have politics around the technical regulations. In the past the politics had been in the realms of teams cheating, or cheating and getting away with it. This was politics around developments that had been done in good faith and quite openly by teams that had made a big investment. The politics of having those banned overnight was a new experience for everybody. People hadn't gone into studying the rules in that level of legal depth. Afterward it became absolutely necessary."

The origins of the active car go back as far as the banning of ground effect in the early 1980s. After years of research, Lotus ran a fully active car for the whole of 1987. It added considerable weight and was responsible for a heavy parasitic power loss, but it did take victories on the bumpy street circuits of Monte Carlo and Detroit. Williams debuted a semi-active car (i.e. retaining passive compliance) later in the year and committed to it for 1988 – but neither team really made the technology work and it was ultimately abandoned. Lowe worked to refine the technology and eventually Williams brought it back for 1992.

"I don't think there was any set idea that it would take until 1992 to get back to racing it," recalls Lowe. "The reality was not so much needing to develop the active suspension itself; it was about needing to develop all the support infrastructure. Today that's taken for granted, but back then we needed to have a decent computer to put on the car that would be reliable for running the control. We needed the instrumentation, the harnessing, and the offboard software for analyzing data. Engine people were ahead of the game because they had been running electronic ECUs since the early 1980s, but we were still mechanical teams with people used to working on mechanical systems."

The task for Lowe was very difficult, but with a straightforward goal: build an active car that was quicker than the team's passive race car. In the winter of 1991/1992, Williams did back-to-back testing of an active and passive FW14. The active car was quicker and so the team committed to racing it in 1992 as the FW14B. Nigel Mansell won its first five races from pole position.



THE MOOG VALVE BLOCK AND THE ACCUMULATOR THAT FEEDS THE ACTUATORS FITTED TO EACH PUSHROD worldwide DEWETRON

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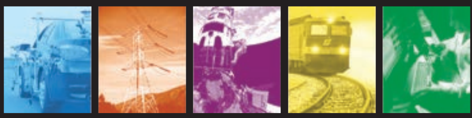

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Sprint endurance

GRAHAM HEEPS PROFILES ALFA ROMEO'S LONG-LIVED 'BERTONE COUPES', FROM THE MODEST SPRINT GT TO THE WILD GTAm CIRCUIT RACER

ALFA ROMEO AUTOMOBILISMO STORICO, CENTRO DOCUMENTAZIONE (ARESE, MILANO)

Like the Spider, with which it shares many of its underpinnings, the Alfa Romeo Giulia Coupe has passed into history as one of the Italian marque's most revered models.

Initially known as the Sprint GT, the first of these Bertone-styled coupes was launched at the Frankfurt Motor Show in 1963. The aluminum, twin-cam 1,570cc engine, with cast-iron cylinder liners, hemispherical combustion chambers, and sodium cooled exhaust valves, along with the five-speed gearbox, were proven items from the 105-series Giulia Ti and 101-series Giulia Sprint and Spider, which had been in production since 1962, but the Sprint GT had the additional benefit of a pair of Weber 40DCOE4 carburetors, improved camshafts, and a Bosch JF4 distributor. The combination not only contributed to additional power (106bhp in all), but made for a much smoother running unit than the single Solex predecessors.

The front suspension was independent with double wishbones, coil springs, and dampers with an anti-roll bar. The rear suspension consisted of a live rear axle well located by trailing arms and a reaction trunnion.

The car sat on 155 x 15 Pirelli Cinturato tires. There were servo-assisted Dunlop disc brakes all round. The body, penned by Bertone's young rising star, Giorgetto Giugiaro, was built at Alfa's factory in Arese, with the engines made at Portello.

The Giulia Sprint GT had not been on sale long before a serious fault appeared in the front suspension, causing seizure of the bottom wishbones, resulting in some cases in front suspension collapse on the road! The problem was caused by the

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MAIN IMAGE: **GTA RACE CAR AT SPA, AND ON THE GRID IN 1972** (BELOW). **LIKE OTHER GIULIA COUPE DERIVATIVES, IT SHARED MANY MECHANICAL PARTS WITH OTHER 105-SERIES ALFAS, INCLUDING THE SPIDER AND BERLINA**

ALFA ROMEO AUTOMOBILISMO STORICO, CENTRO DOCUMENTAZIONE (ARESE, MILANO)

bottom wishbone pivot pin bushes seizing and ripping the pin from the cross member. The bushes were of the 'sealed for life' variety with two spongy rings installed at the bushing end to prevent foreign matter from entering them. Not only did the rings dry out, but the bushes tended to seize very quickly. Cars that gave audible warning of seizure (squeaks and groans) were recalled to have their bushes lubricated with Shell Dentax initially, and later Calypsol AE grease inserted by a syringe.

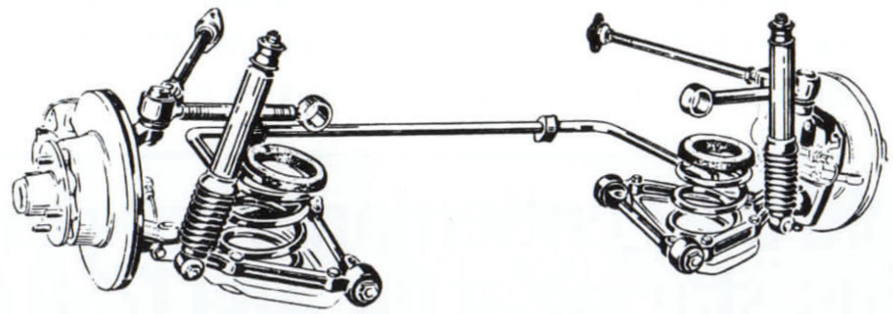
Alfa was quick to recognize this embarrassing fault by offering a repair kit, including a set of modified bushes, for use in instances where lubrication failed to unseize the bushes, or if the suspension broke completely. Fitting the repair kit was a major operation as it involved a brand new main cross-member complete with a pair of revised wishbone pivot pins. It was rather a hush-hush job for obvious reasons with clients being informed that their Sprint GT required a 'suspension operation', with Alfa bearing the brunt of the cost.

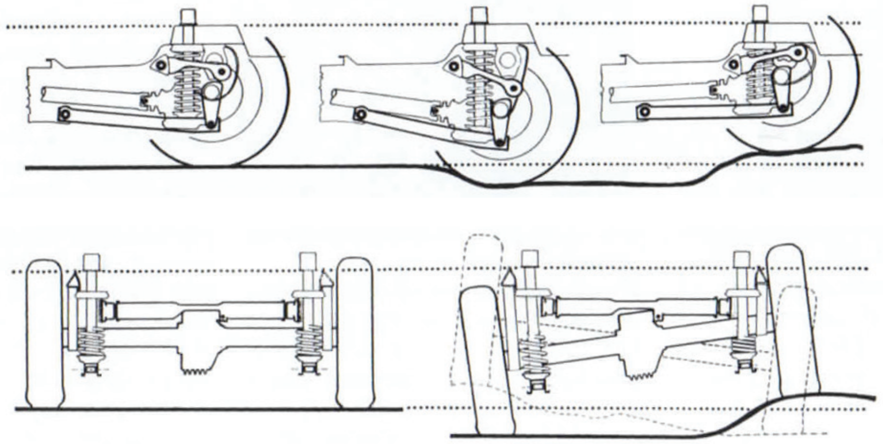
The all-round disc brakes were initially a problem, too. Very early cars had no servo at all, but even the single-circuit Dunlop setup with two-piston front calipers and a hydro-mechanically operated rear caliper was generally deemed inadequate for the job. It was phased out in 1967 in favor of a superior system from ATE.

During the next decade the car evolved into the Sprint GT Veloce (GTV) – initially with a 1.6-liter engine, later with 1750 and 2000

"The Sprint GT had not been on sale long before a serious fault appeared in the front suspension, causing seizure of the bottom wishbones"

SPECIFICATIONS
1963 Alfa Romeo Sprint GT
Engine: 1,570cc, twin-cam I4. 106bhp, 132Nm
Dimensions: 3,480mm (L) x 1,575mm (W)
Curb weight: 972kg
Suspension: Front double wishbones, coil springs and dampers with an ARB; live rear axle located by trailing arms. Girling or Allinquant dampers
Steering: Either cam-and-peg or recirculating-ball
Brakes: Dunlop. Front discs 286mm
Price new (1964): £1,849 (US$2,938)
Acceleration zero – 60mph: 11.2 secs
Standing ¼-mile: 18.5 secs
Maximum speed: 112mph





LEFT: **ALFA 105-SERIES FRONT SUSPENSION** (TOP) **AND REAR-SUSPENSION GEOMETRY** (BELOW)

OPPOSITE PAGE: **GTA 1300 JUNIOR. THE GTA HAD 266MM FRONT DISCS BECAUSE OF THE SMALLER WHEELS AND SLIGHTLY SOFTER FRONT SPRINGS BECAUSE THE CAR WAS LIGHTER THAN STANDARD GTs**

"Homologation for motorsport required a production run of 1,000 cars, but because Alfa had used up its run of aluminum-alloy bodies on the earlier, lightweight GTA variant, the GTAm was homologated around a steel-shelled, US-spec, fuel-injected 1750 GTV"

BELOW: **ONE OF THE VERY LAST GT 1300 JUNIORS, DATING FROM 1974. THIS MODEL WAS THE MOST NUMEROUS OF ALL GIULIA COUPE VARIANTS; SOME 91,964 OF THE 223,615 BERTONE COUPES MADE WERE GT 1300 JUNIORS**

ALFA ROMEO AUTOMOBILISMO STORICO, CENTRO DOCUMENTAZIONE (ARESE, MILANO)

motors – the GT Junior models were introduced, and highly successful racing derivatives were produced.

The ultimate Giulia Coupe race car was the GTAm (from GT America), built to compete in the European Touring Car Championship's Group Two from 1970. Homologation for motorsport required a production run of 1,000 cars, but because Alfa had used up its run of aluminum-alloy bodies on the earlier, lightweight GTA variant, the GTAm was homologated around a steel-shelled, US-spec, fuel-injected 1750 GTV (the engine was bored out to a 2-liter). A selection of lightweight body panels and windows ('optional' on the road car to get around homologation regulations) brought the weight down to target.

Chassis developments followed on from the GTA, with extenders to lift the top ball joint location on the front uprights, thus lowering the roll center, and shorter steering arms on the uprights to provide at the same time quicker steering response and greater clearance for wider (8in or 9in), 13in-diameter wheels.

At the rear, the roll center was also lowered, with lateral location of the axle courtesy of a low pivot fixed to the differential casing and running between two vertical guides attached and triangulated to the boot floor. The two standard trailing arms became fully rose jointed and the standard top T-arm was replaced by a cross bracket and a rose jointed link to the standard attachment on the top of the diff.

It was the combination of the low rear roll center, extremely stiff front springs, and the necessarily restricted droop movement on the front suspension, that produced the characteristic Alfa three wheeling stance in a corner. Ventilated front discs completed the package.

Back on the road, the final major evolution before the car was replaced by the Alfetta GT was the 2000 GTV from 1971, by which time the Coupe had twin-circuit brakes and smaller but wider wheels (5.5 x 14 wheels instead of 4.5 x 15s).

With thanks to Jon Dooley, David Edgington and Chris Savill



GROOVY, BABY

"There's this stretch of road that goes from your joint to someplace very groovy. It has to swing and it has to be at the end of this wild road that's all kinky and twisty and blacktop with white guard rails to protect you from the scenery. You'll need a girl. One with tawny Breck-Shampoo hair that shines like an illuminated waterfall and lavender eyelids and the right kind of pants, the kind that aren't, you know, lower-class tight, but sort of expensive-tight.

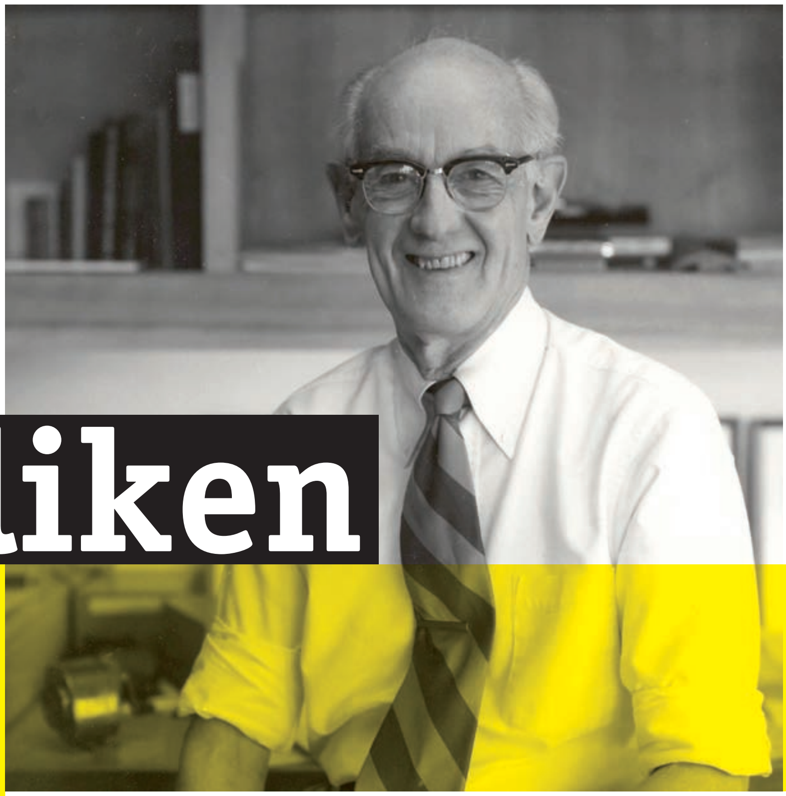
"Now you need a car, a very special teeny-weeny sort of grand touring car, Ferrari Lusso or Aston Martin or Mercedes 230 SL or Maserati 3500 GT? Come on, they're too big and they're only for wealthy illiterates and young greaseballs anyway, right? This has to be an exquisite little jewel box car that you can fling around corners at just about 1.3*g* with your arms out straight and your head cocked just like Innes Ireland's in all the pictures and an engine that goes eeeeee-yyyooowww when you accelerate and yyyooo www-eeeeee when you downshift with a nutty shift lever that's just like a long toggle switch.

"Make sure the engine has lots of aluminum pieces on it and, oh yes, double overhead camshafts. Double overhead camshafts are very big and they look so nice – I mean they just kill gas station attendants and guys who try to be friends because they owned an MGA once. And big brakes, don't forget big brakes, so that you can go yyyooowww-eeeeeeing into the corners and save the downshift until just before the tawny girl unbuckles her belt to bail out. Say hello to the Alfa Romeo Giulia Sprint GT."
(Car and Driver, April 1965)

ALFA ROMEO AUTOMOBILISMO STORICO, CENTRO DOCUMENTAZIONE (ARESE, MILANO)

"This has to be an exquisite little jewel box car that you can fling around corners at just about 1.3*g*"

"At the end of the war, engineers with expertise in aircraft stability began to apply mathematical modeling to similar problems in vehicle dynamics"



Bill Milliken

JOE WALTER OFFERS AN APPRECIATION OF WILLIAM F. 'BILL' MILLIKEN, THE PIONEERING VEHICLE DYNAMICIST WHO DIED IN JULY 2012 AT THE AGE OF 101

Adventure, innovation and risk were part of William F. 'Bill' Milliken's career in aircraft and vehicle dynamics for more than 75 years. He graduated from Massachusetts Institute of Technology (MIT) with a degree in mathematics, which was more attractive to him than the two years of German language study involved in an engineering degree, although he did devour elective courses in MIT's renowned aeronautics department.

Milliken spent most of World War II actively engaged in flight testing of prototype military aircraft such as Boeing's B-29 and Avion's flying wing (XB-79). At the end of the war, engineers with this expertise in aircraft stability and control began to apply mathematical modeling to similar problems in vehicle dynamics. Milliken joined the Cornell Aeronautical Laboratory (CAL) where he spearheaded the application of fundamental math and physics principles, first to the motion of aircraft, and then to automobiles. He was driven by necessity to tire testing because vehicle modeling required tire force and moment data.

This 'dynamic person' was well-known in racing circles for his basic research in automobile handling, but less well-known for his early creative work with tire testing and cornering behavior. For example, he demonstrated the validity of using quasi-static tire properties in modeling vehicles at highway speeds subject to non-steady-state steering commands. This fundamental understanding of tire-vehicle response underpinned subsequent research dealing with the Milliken Moment Method and the 'g-g' diagram.

His passion for race cars convinced him that you can't drive one successfully without becoming involved in the finer points of handling. Extant aircraft textbooks provided the necessary basic theory for motion in flight. Because the automobile had been around much longer than the airplane, he assumed that there must be equivalent texts for vehicles traversing roadways.

Other than a 1947 paper by Maurice Olley, Milliken found nothing that met his needs. He visited GM in 1952 in search of such information, but unexpectedly met Olley, and then received a sizeable grant from GM to undertake studies on transient behavior of tire-vehicle systems. This sponsorship continued until 1963 and was the beginning of the modern vehicle dynamics era in the USA. This built on Olley's seminal work of the early 1930s at GM dealing with steady-state behavior. Research was largely based on technology transfer from the aircraft field. Milliken laid great emphasis on physical understanding of equations because investigations of vehicle handling tended to be more art than science.

Milliken knew that the maneuvering forces for aircraft are aerodynamic in nature and could be measured in a wind tunnel; for automobiles, the forces principally arise from tire-road interactions and had to be obtained by other methods. Both aircraft and automobiles have operational limits based upon available external forces that, when exceeded, can result in dramatic changes in stability and control such as stalling and skidding. Thus, in 1952, Milliken secured a USAF contract to design and construct an on-road tire tester that enabled accurate measurements of the six tire-force and moment components over a large range of slip and camber angles under variable braking conditions. This machine was a major advance over previous tire testers and served as a basis for subsequent machines introduced elsewhere. One result obtained early on was that tire stiffnesses changed during service due to tread wear.

Toward the end of Milliken's career at CAL, the first commercially viable, flat-surface, high-speed lab tire force and moment machine – TIRF – was placed in operation, and is still running today.

Optimal vehicle control

The Integrated Vehicle Safety Department of TNO (Dutch Organization for Applied Scientific Research) investigates the application of modern control methods in the Integrated Vehicle Dynamics Control (IVDC) field, as a strategic research topic of the Beyond Safe framework. The aim of IVDC is to optimize the vehicle performance by maximizing the synergy between vehicle system like braking, suspension and steering systems and thereby enable improvements in the vehicle safety and comfort for all driving conditions.

This article presents the first steps undertaken by the authors to develop and experimentally evaluate an optimal IVDC strategy for stability control using only the braking system. More specifically it presents the structure of the controller, the vehicle and tire models used, the State Dependent Riccati Equation (SDRE) control technique and the experimental evaluation of the developed controller.

What is the optimal tire force distribution that controls the vehicle's yaw motion and has the least effect on its longitudinal dynamics? The answer to this question is challenging: considering that tire forces are highly nonlinear and the vehicle has more actuators (four braking forces) than control goals (yaw moment, deceleration), there may be more than one actuators setting for regulating the system. The State Dependent Riccati Equation (SDRE) is a suitable control technique to optimally regulate nonlinear over-actuated systems. Some first studies on SDRE control were done in the early 1990s by Krikelis et al (1992), while a more systematic development was carried out later by Cloutier et al (1996).

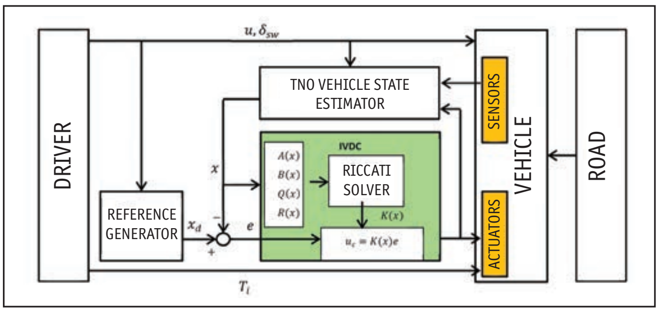
FIGURE 3: **THE CARLAB JAGUAR XF, WHICH IS EQUIPPED WITH A BRAKE BY WIRE SYSTEM AND THE STANDARD ESC SENSOR SET**

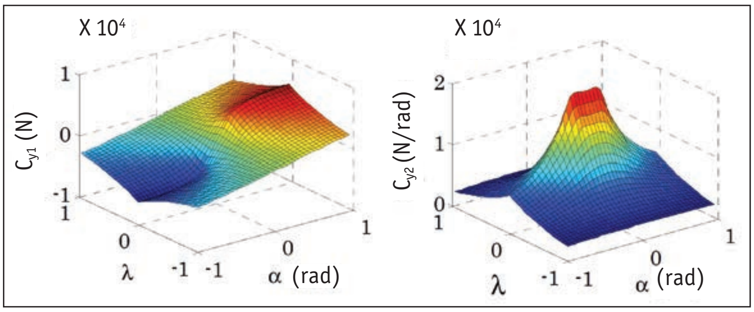
Figure 1 shows the structure of the developed control system based on the SDRE technique. A reference generator using a reference model and driver inputs (i.e. steering wheel angle δ_{sw} and longitudinal velocity u) provides the desired states x_d of the vehicle. TNO Vehicle State Estimator (VSE) estimates the states x of the vehicle at the current operating point. The error e between the desired and estimated states is the input to the IVDC controller. The IVDC control module includes: a nonlinear scheme of the system's state space equations on the basis of the combined slip TNO MF-Tyre model (as a product of TNO Delft-Tyre); the objective of the control, which is minimum tires' longitudinal slips and slip angles, in the form of mathematical cost function and a numerical solver for algebraic Riccati equation.

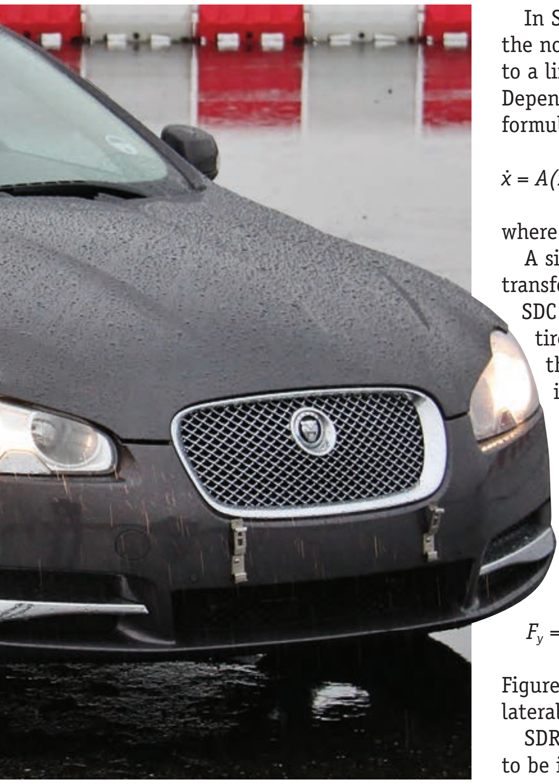
The nonlinear vehicle model is generated using a three-states model (i.e. longitudinal velocity u,

FIGURE 1 (BELOW): **STRUCTURE OF THE DEVELOPED CONTROL SYSTEM BASED ON THE SDRE TECHNIQUE**

FIGURE 2 (BELOW RIGHT): **REPRESENTS C_{Y1} AND C_{Y2} FOR LATERAL FORCES**







lateral velocity v and yaw rate r). In the model the acceleration in the longitudinal, lateral and yaw direction are nonlinear functions of the tire forces in the corresponding directions. The effect of roll dynamics on vehicle behavior is included by considering the load transfer in the calculation of tires' normal forces.

$$\dot{u} = vr + f(F_{xi}, F_{yi})$$

$$\dot{v} = -ur + g(F_{xi}, F_{yi})$$

$$\dot{r} = q(F_{xi}, F_{yi})$$

The MF-Tyre combined slip tire model is used for describing the tire forces' behavior (Pacejka 2002). There are two main reasons to use such a tire model. First it improves significantly the vehicle model accuracy at high g maneuvers. Second, to compute the optimal brake force distribution it is necessary to consider the interaction between the longitudinal and lateral forces (Van Zanten, 1998).

In SDRE it is needed to transform the nonlinear dynamic equations to a linear like form called State Dependent Coefficient (SDC) formulation (Equation 2).

$$\dot{x} = A(x)x + B(x)u_c$$

where x is the vehicle states $[u,v,r]^T$.

A significant difficulty in transforming Equation 1 into the SDC form is the combined slip tire behavior. To circumvent this, most of the researchers implement a decoupled tire model (Bonsen, 2010). The following representation is proposed to express the combined slip tire behavior using the MF-Tyre model,

$$F_x = C_{x1}(\lambda,\alpha)\lambda + C_{x2}(\lambda, \alpha)\alpha$$

$$F_y = C_{y1}(\lambda,\alpha)\lambda + C_{y2}(\lambda, \alpha)\alpha$$

Figure 2 represents C_{y1} and C_{y2} for lateral forces.

SDRE theory also allows constraints to be imposed on actuators (e.g. actuator saturation). The input of the control system u_c involves the wheels' longitudinal slip, which can be both positive and negative. A constraint was considered to prevent positive slip (acceleration).

The designed SDRE control minimizes a cost function,

$$J = \int (x^T Q(x)x + u_c^T R(x)u_c)dt$$

where $Q(x)$ and $R(x)$ are the state and input weighing matrices and in general are state dependent.

The state feedback law that optimizes the cost function (Equation 4) with respect to the constraints (Equation 2) is

$$u_c = -K(x)x$$

where $K(x)$ is the state feedback gain,

$$K(x) = R(x)^{-1}B(x)^T P(x)$$

and $P(x)$ is the solution of the Algebraic State Dependent Riccati Equation.

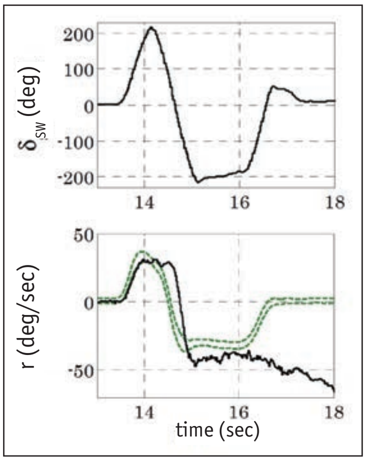


FIGURE 4: **THE YAW RESPONSE OF THE CARLAB WITHOUT STABILITY CONTROL AND THE DESIRED YAW RESPONSE**

$$A(x)^T P(x) + P(x)A(x) - P(x)B(x)R(x)^{-1}B(x)^T P(x) + Q(x) = 0$$

There are several numerical methods to solve the algebraic Riccati equation in real time. To prevent overloading of the real-time processor a modification of the Kleinman algorithm with an optimal number of iterations has been applied (Kleinman, 1968).

The proposed control strategy has been experimentally evaluated by implementing it on a real-time dSpace platform (dSpace Autobox 1005) in a driving car laboratory (CarLab) (Figure 3). The CarLab is a Jaguar XF that is equipped with a Brake by Wire system and the standard ESC sensor set. Also TNO Vehicle State Estimator (VSE) is implemented on the CarLab, which provides full state feedback for the SDRE controller. Furthermore, the effect of the nonlinear combined slip tire model on the optimal tire force distribution has been investigated by comparing the experimental results obtained using the proposed controller – denoted as TNO IVDC – with a SDRE controller that uses a decoupled slip tire model – denoted as Benchmark (Bonsen, 2010).

Several tests have been performed on a test track but for the sake of brevity only the Sine with Dwell maneuver will be shown. According to NHTSA, this performance test is the best able to excite an oversteer response of a wide range of vehicles (Forkenbrock, 2005). In Sine with

Dwell the vehicle is initially moving straight ahead with a velocity 75km/h. Then, at a certain time point, the steering wheel is started moving and δ_{sw} follows a 0.7Hz sine wave with a 500ms delay beginning at the second peak, as shown in the upper part of Figure 4. In the lower part, the yaw response of the CarLab without stability control and the desired yaw response are shown. As noticed the yaw rate starts deviating 1.2 seconds after the maneuver was initiated and does not return to zero even when the steering wheel is neutral, which means the vehicle has become unstable.

Figure 5 shows the steering wheel angle and vehicle states ($[u,v,r]^T$) for the proposed and benchmarked controller with respect to time. It is found that both controllers improve stability of the vehicle and thereby both satisfy the first goal of IVDC. However, the proposed controller demonstrates better performance in the yaw rate and lateral velocity response of the vehicle (2m/sec less lateral velocity). Moreover, it yields less effect on the longitudinal velocity compared with the benchmark and therefore shows a promising improvement in the second goal of the IVDC (i.e. less intrusive). It can be seen that CarLab with the proposed controller has a 15km/h less of a velocity reduction at the end of the maneuver. Figure 6 illustrates the brake torque for each wheel. It is well established that the braking force distribution is clearly different for the mentioned controllers. Considering the yaw rate response (Figure 5), from 13 to 15 seconds although the vehicle is oversteering the benchmark controller applies the same brake torque at front and rear wheels while the proposed one applies less at the rear. From 15 to 16.5 seconds the vehicle with proposed controller is understeered so the controller applies more brake torque in front wheels. It can be concluded that the proposed SDRE technique is more suitable for optimal brake force distribution (Van Zanten, 1998).

The experimental investigations have shown the ability of the proposed SDRE controller to stabilize the vehicle – under a variety of maneuvers – with an optimized braking force distribution. The controller, on one hand, improves the vehicle performance while keeping the lateral velocity less than 2m/sec and, on the other hand, has a minimum intervention – 15km/h less of a velocity reduction – compared with a benchmark controller.

The proposed SDC form for the vehicle with the combined slip tire model provides the appropriate framework to include more actuators. In the future, TNO's Integrated Vehicle Safety department will be looking at the active suspension system in the IVDC and also the ride comfort objective to the cost function.

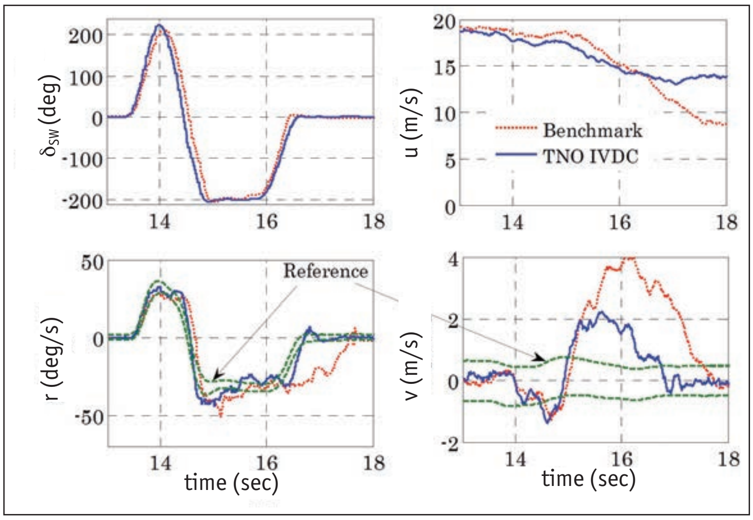


FIGURE 5: **STEERING WHEEL ANGLE AND VEHICLE STATES FOR THE PROPOSED AND BENCHMARKED CONTROLLER WITH RESPECT TO TIME**

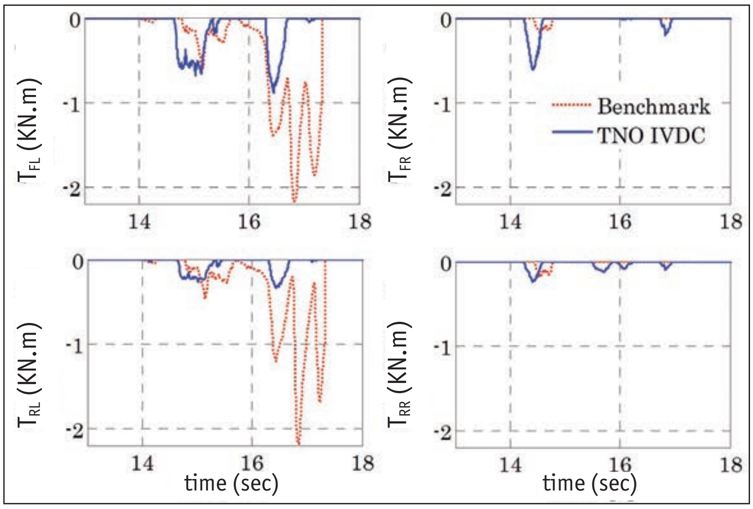


FIGURE 6: **BRAKE TORQUE FOR EACH WHEEL**

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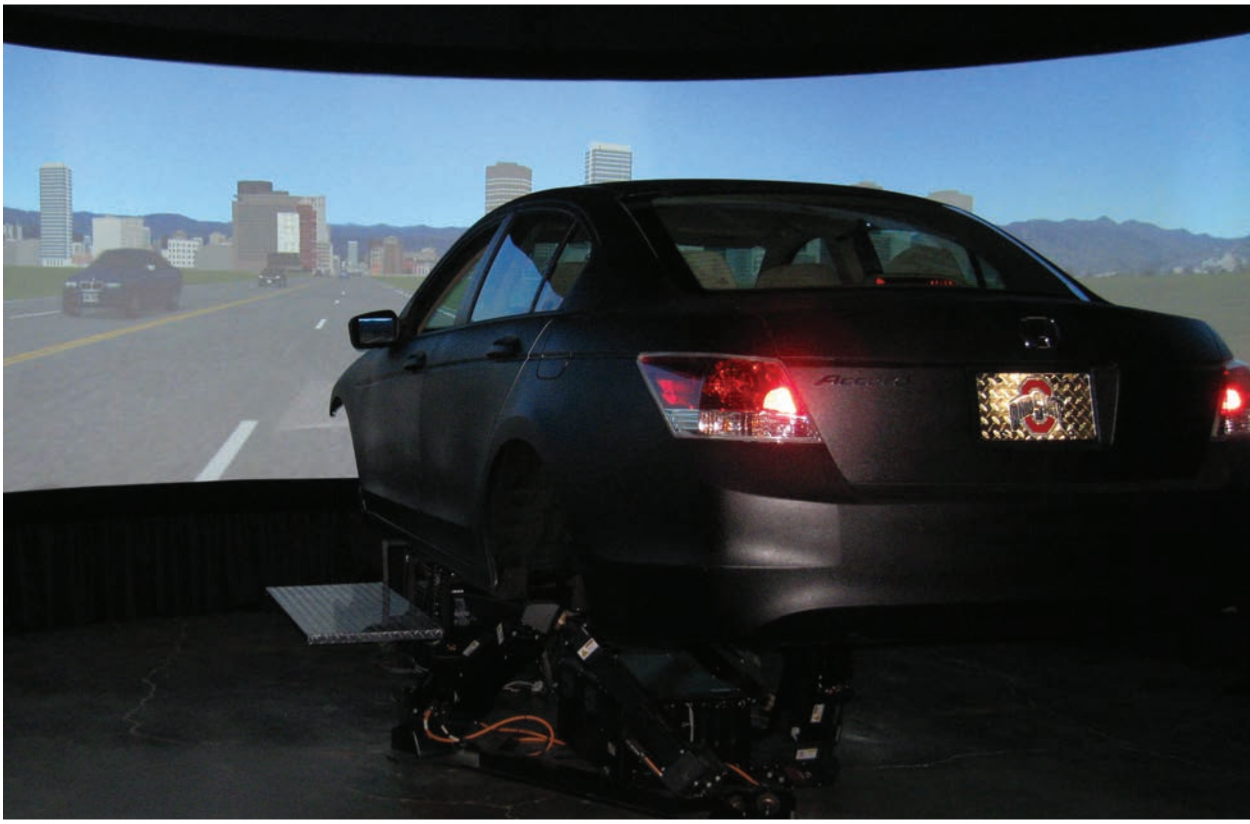
Advanced simulators

RIGHT: **USING THE DRIVING SIMULATOR, RESEARCHERS CAN STUDY VARIOUS DRIVER DISTRACTIONS**



This isn't your typical Honda Accord on the highway today, mainly because it's not on the highway. It's parked in a former warehouse on the campus of Ohio State University (OSU) in front of a 24ft-diameter seamless wrap-around projection screen. The vehicle is part of the university's new 5,800ft^2 driving simulator laboratory that enables researchers to study a variety of driver distractions such as talking on a cell phone, adjusting car controls, and sending text messages.

Jan Weisenberger, senior associate vice president for research at OSU, points out that the facility will provide an opportunity to investigate a variety of questions about drivers and vehicles. She says, "We plan to study driver behavior, including such measures as reaction times to unexpected events, cognitive workload when performing multiple tasks in the vehicle simultaneously, and decision making. In addition, driver preferences for different types of vehicle dynamic properties, such as steering feel, suspension, and drive train modifications, can be evaluated. Finding the best instrument designs for in-vehicle infotainment systems, ones that maximize ease of use while minimizing driver distraction, is another goal. Finally, researchers will look at vehicle warning systems, such as proximity sensors, to determine how best to alert drivers to impending hazards."

The new lab will feature two unique simulator environments. The larger environment features a full-size Honda Accord connected to Realtime Technologies' (RTI) SimCreator software, which allows for the creation of hundreds of different driving scenarios, from a rural road dotted with deer, to icy highways choked with traffic and emergency vehicles, to city streets with pedestrians crossing in front of cars. The vehicle is surrounded by a 260° panoramic screen that projects traffic scenes, while rear-view and wing mirrors on the cab display additional real-time images. Five projectors are used for the panoramic view out of the driver's windshield and side windows. The projected images overlap and are blended together to provide a seamless perspective. The vehicle's movement is controlled by the driver. The traffic around the vehicle is completely autonomous and surrounds the driver with the experience one would encounter in normal everyday driving. The vehicle is mounted on the latest Moog 6DOF

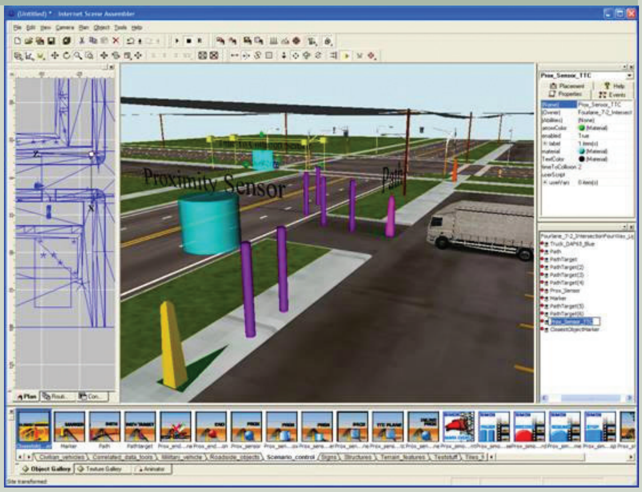
RIGHT: **THE DRIVER'S VIEW OUT IS HIGHLY REALISTIC, BOTH IN TERMS OF THE ANIMATED AND STATIC OBJECTS IN THE SCENE**

REAL-TIME SIMULATION AND MODELING SYSTEM

RTI has provided OSU with its unique and powerful SimVista scenario development software, along with SimCreator, used to develop the simulation software. The SimCreator package includes a complete visual simulation solution and SimVehicle, which is a high-fidelity vehicle model for varied engineering and human factors research needs.

The RTI-based visual simulation handles all real-time visual rendering processes, including animated and static objects in the scene. This makes the driver's view out of the window highly realistic. The audio software and hardware provide audio cues to match what the driver would expect under the conditions being simulated. Synthesized sounds include engine, wind, tire whine, and noise from other vehicles. The SimVista scenario system provides autonomous traffic simulation, pedestrian simulation, scripted events, and environmental controls. The SimObserver subsystem collects both video and data from the driving simulation that can be used for analyzing driving performance.

SimVehicle includes RTI's Vehicle Dynamics Editor, which allows input of powertrain, transmission, suspension, and tire parameters to quickly change the drivetrain and suspension parameters to enable users to experience driving different vehicles.



motion base that reproduces the actual car motions. The vehicle also has a high-fidelity steering motor that provides realistic steering feel based on vehicle speed and tire angle.

The second simulator enables testing of production vehicles using a 'drive on' turntable system under the front wheels and a single forward projected display. Vehicles can be driven onto turntables to allow the front wheels to move in response to steering input. The vehicle sits in front of a large flat projection screen. Sensors are attached to the brake and accelerator pedals. The vehicle can be driven in a simulated environment in a key-on engine-off condition to allow for testing of subjects using production vehicles and existing vehicle technologies. This data can be used to further understand differences between existing technologies in vehicles and new technologies proposed for tomorrow's vehicles.

Don Stredney, technical director of OSU's Driving Simulator Laboratory and director of the Ohio Supercomputer Center's Interface Lab, says, "We enthusiastically anticipate the use of the simulation environment to provide a high-fidelity, multisensory (visual, auditory, and motion) stimuli for our studies in vehicle-human communication. Integrated with quantitative eye-tracking and physiological monitoring, we look forward to working with a wide range of domains that research cognitive loads, including task, flow, and boredom, as well as additional existing and emerging distractions found within vehicles that could lead to human injury."

Upon its unveiling the lab was open to users from both universities and industries. Weisenberger notes that, "University researchers from Ohio State, OSU, and Wright State University are part of a consortium that will foster collaborative research projects in driving simulation. In addition, industry users, including auto makers and the companies that design and manufacture infotainment and warning systems, will be able to use the facility to evaluate new designs. Government labs can also use the facility to determine whether vehicles meet current safety guidelines, as well as to create new guidelines that enhance driver and passenger safety. We are hoping to create new partnerships among the university, industry, and government users to engage in exciting new research directions that will give us the road map for the automobile of the future."

RTI has deployed additional development and research simulators at the university and its collaborators to bring the total number of simulators to five. One of the simulators is being installed at the OSU Center for Automotive Research for use by the Control and Intelligent Transportation Research, which focuses on vehicle automation and vehicle safety systems, including sensing, control, situation awareness and understanding, vehicle-to-vehicle communication, and road and traffic infrastructure systems. OSU intends to interface the simulation with hardware-in-the-loop equipment and other experimental technologies and devices. This can also serve as a satellite facility for the large-scale simulator so that researchers can develop and test scenario design and preliminary experiments and analysis, and then move them to the motion simulator for full-scale activities. Keith A. Redmill, an assistant professor in the Department of Electrical & Computer Engineering at OSU, believes that, "This simulator will greatly increase our ability to prototype and test vehicle systems and evaluate driver behaviors before deploying systems for road tests on actual vehicles. And it will provide significant access to simulation technology for our students."

ABOVE: **RTI'S SIMCREATOR SOFTWARE ALLOWS FOR THE CREATION OF DIFFERENT DRIVING SCENARIOS**

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Effective ESC testing

MAIN: **THE ABD SR60 TORUS STEERING ROBOT WORKS IN COLLABORATION WITH VBOX DATA TO PRODUCE ACCURATE RESULTS**

With the recent introduction of mandatory fitting of electronic stability control (ESC) systems to passenger vehicles, manufacturers are faced with ever-increasing investment in advanced testing equipment.

The documentation that describes test standards FMVSS 126 (USA) and ECE R13H (EU) gives an indication of the type of equipment required to undertake the necessary procedures. This includes such items as three-axis accelerometer packs, laser or optical ride-height sensors, and radar speed measurement systems, for which the current market solution is expensive. The tests require that the steering response of the vehicle must be verified as falling within acceptable dynamic capabilities. This is done with a 'sine-with-dwell' maneuver whereby the vehicle, traveling at 80km/h, must deviate from its original centerline path by at least 1.83m (6ft) within 1.07 seconds of the initial steering input. The yaw rate ratios of the vehicle are also measured and compared at set time intervals, to prove that the ESC can control the heavy yaw movements without a spin being induced.

Measurement accuracy is critical, and accepted methods of capturing the data have employed not only the equipment already mentioned, but also calculation based on double integration of the accelerometer data.

The problem with double integration is that it is prone to error multiplication. If initial values aren't extremely accurate, the second order of calculations can render the data useless; consequently the accelerometer units themselves have to be of a very high specification (and therefore price) to ensure quality results.

To compensate for body-roll angle, it is suggested that laser ride-height sensors are used, which not only add to the overall cost, but also to setup time. Finally, speed is supposedly best measured using radar, which adds to the overall complexity of using several test components.

But there are alternatives. The VBOX GPS solution has major advantages before cost is even considered, simply due to the fact that less equipment is required and there is very much less to do in terms of installation. The 100Hz VBOX3i, coupled with an IMU, provides highly accurate speed, distance, heading, and yaw rate measurement at significantly lower cost than the combination of accelerometers, ride height sensors, and radar. Additionally, the margin for error is lower when using VBOX as only a single integration is required to accurately measure lateral displacement, thanks to the very accurate speed and heading measurements provided.

Taking readings from an antenna on the roof means that there is a need to compensate for vehicle roll – solved by mounting the VBOX inertial unit at the car's center of gravity and providing an automated calculation without the need for lots of post processing work with ride height data. This is where the Anthony Best Dynamics (ABD) steering robot comes into the equation.

Racelogic has teamed up with ABD to provide a comprehensive solution for engineers to carry out these tests, which require precise and repeatable control; the high torque and steering wheel speed required to perform the test means that a robot such as the ABD SR60 is essential. The ABD steering robot applies the desired steering wheel input throughout the test and captures the VBOX CAN output for speed, heading, yaw and roll rate at 100Hz. Firstly the car is 'characterized', by determining the necessary steering angle to achieve the required amount of lateral *g* (known as a 'slowly increasing steer' procedure). The actual sine-with-dwell test is then performed with increasing yaw and displacement, on both right and left turns.

The ABD software combines the VBOX data with that from the robot,

BELOW: **ESC TESTING BEING UNDERTAKEN VIA THE 'SINE WITH DWELL' TEST**

LEFT: **100HZ VBOX3I AND IMU PROVIDE A STABLE BASE FOR THE ABD SR60 STEERING ROBOT**

BELOW: **COMPLETE TECHNICAL SETUP, AS FITTED TO A HONDA CIVIC TYPE-R**

and produces an easily interpreted set of results that clearly show the level of displacement the car has achieved, at what steering angle, and over what period of time. This software also carries out calculations to compare yaw rate ratios at later phases of the sine-with-dwell test, which form the main body of evidence as to whether or not the ESC system passes or fails the regulation criteria. The combination of VBOX and ABD equipment ensures that overall test times, installation, and post processing is reduced when compared with suggested setups.

The ABD/VBOX setup has been benchmarked against two other systems: a high-expense accelerometer solution; and a Differential GPS Real Time Kinematic setup that gives 2cm positional accuracy.

The RTK DGPS solution is the ultimate in terms of measuring positional deviation, and under these particular test conditions where data is being used over very short time periods (1.07 seconds), the positional accuracy is a matter of millimeters. Consequentially the lateral displacement of the vehicle measured with the RTK DGPS data can be considered a reference, but a system such as this is also very expensive, so not a suggested solution for everyday ESC testing.

The benchmarking results prove that the VBOX GPS solution aligns with the reference measurements and also with that of the suggested equipment, and therefore meets the demands of the US and EU governing bodies. The difference is that with VBOX, the initial setup and outlay are much more palatable, no matter what size of organization is using it.

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Steering innovation

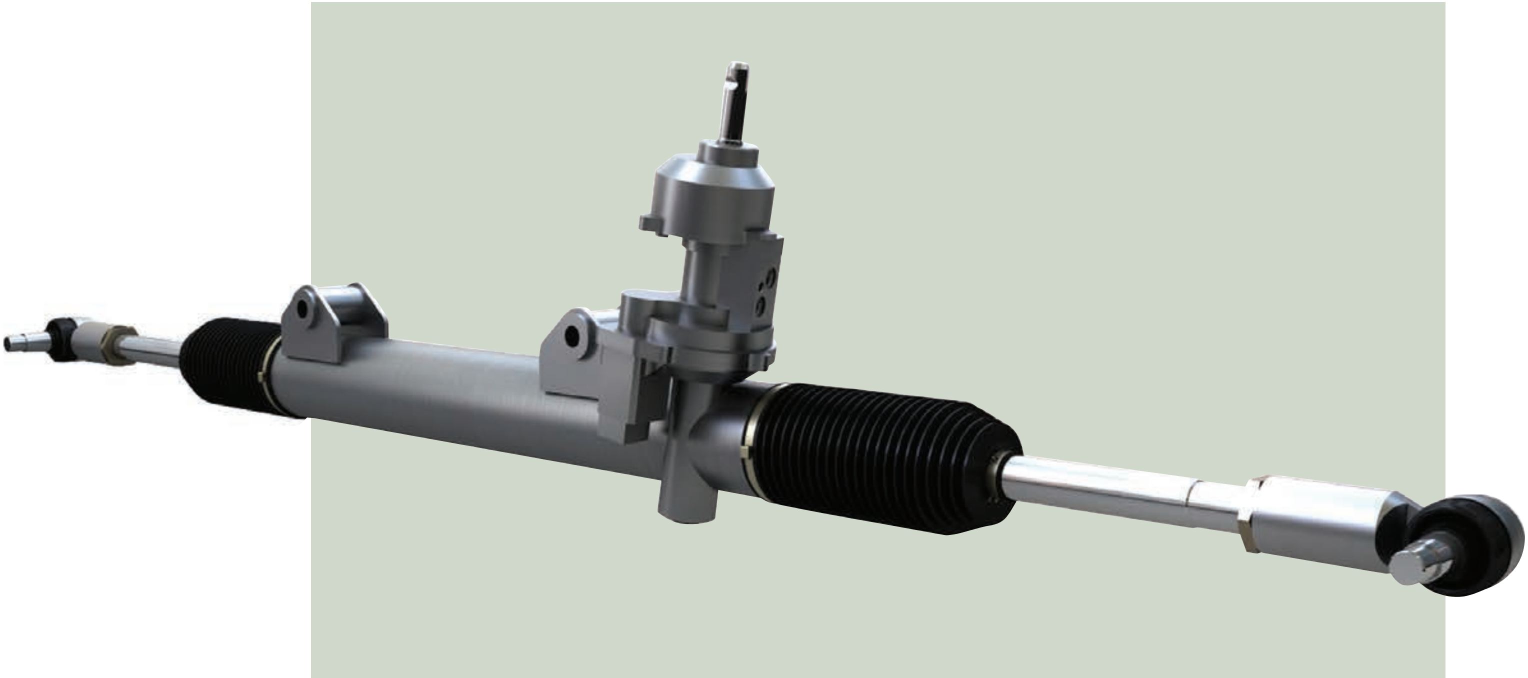

FIGURE 1: **TEDRIVE'S RACK-AND-PINION STEERING WITH ITS PATENTED IHSA MODULE**

At this year's IAA Commercial Vehicle Show, Tedrive Steering Systems GmbH launched two new and compelling technologies that hold a unique place in steering technology. One is the world's first truck rack-and-pinion steering gear with its patented iHSA module (see Figure 1). As a second innovation, Tedrive has implemented the iHSA module into recirculating ball steering systems for heavy-duty trucks and buses (see Figure 2).

With Tedrive's active recirculating ball steering systems, heavy trucks and buses can now be steered without driver input for active lane-keeping, and equipped with a range of additional functions such as commercial vehicle park assist, crosswind stabilization, as well as further safety and comfort features. With these new technological developments, the German specialist has established itself firmly as a comprehensive provider of automotive steering systems.

Active lane-keeping assistance for heavy commercial vehicles with intelligent hydraulic steering assist (iHSA) is a CO_2-optimized hydraulic steering system with all the safety and comfort functions of electromechanical power steering (EPS). It enables the integration of various active features – such as lane-keeping, crosswind compensation, trailer stabilization, and park assist – into hydraulic steering systems for heavy commercial vehicles and buses.

Tedrive is offering an iHSA application for both recirculating ball steering and rack-and-pinion systems (see Figures 3 and 4). Peter Heimbrock, head of development for Tedrive Steering Systems, explains: "In recirculating ball systems the steering input from the driver is transferred into a central block steering gear. This movement is then transmitted to the wheels via steering arm, push rod, and steering tie rod. While effective, it is also a complex system set-up with high part-count and of considerable weight. Rack-and-pinion systems can do the same job but with less part complexity, and are therefore also lower weight. Also the ride and handling performance will be improved significantly."

Taking a modular approach to its technologies across all vehicle classes, Tedrive has developed a number of different technologies to facilitate the use of rack-and-pinion steering with rigid axles, as well as with independent suspension. A new modular steel-housing rack-and-pinion design for HCV steering up to more than 7 metric tons front-axle load and the implementation of

FIGURE 2 (RIGHT): **TEDRIVE HAS IMPLEMENTED THE IHSA MODULE INTO RECIRCULATING BALL STEERING SYSTEMS FOR HEAVY-DUTY TRUCKS AND BUSES**

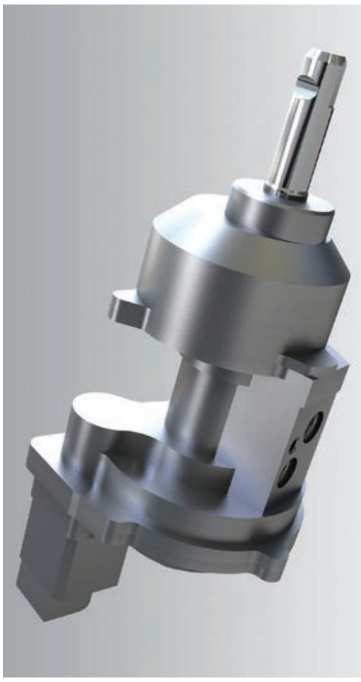

FIGURE 3 (FAR LEFT): **TEDRIVE'S IHSA MODULE FOR RECIRCULATING BALL STEERING SYSTEMS**

FIGURE 4 (LEFT): **TEDRIVE'S IHSA MODULE FOR RACK-AND-PINION STEERING**

further improvements in sealing and mounting also make it possible to meet extremely demanding durability requirements. In rack-and-pinion steering, the push rod and steering tie rod are directly replaced by the steering gear, thus significantly improving the steering feel. The number of components required is reduced, leading to a considerable saving in weight and cost, as well as less assembly complexity for Tedrive customers. Systems can now also be run at high pressures with a reduced minimum volumetric flow in the steering system, resulting in noticeable fuel savings and an associated reduction in CO_2 emissions.

It also makes sense to equip Tedrive rack-and-pinion steering with a Tedrive iHSA module. The use of this intelligent hydraulic steering assist enables the generation of steering input independently from the driver, making it possible to implement all the safety and comfort features familiar from the passenger car sector in the heavy vehicle classes for the first time using a 'plug-and-play' approach. This means, for example, that HCVs can also be equipped with an active lane-keeping assistant for the avoidance of serious accidents.

The hydraulic technology is variable, independent of front axle load, and environmentally friendly. Alongside the improved steering functionality, plus-points include optimized installation, cost and design benefits for platform strategies, and the CO_2 savings potential from the pump and steering gear. If the steering system is connected to the associated driver assistance systems via an interface, iHSA is able to perform the kind of comfort and safety functions that were previously the preserve of EPS systems from the passenger vehicle sector.

Whether in connection with rack-and-pinion, or with the conventional recirculating ball steering gears using plug-and-play, the use of iHSA technology represents a new approach to realizing active lane-keeping assistance beyond the familiar lane-departure warning systems. It is now possible to initiate steering without driver input and to implement all the safety functions familiar from the passenger car sector. With iHSA technology, Tedrive is helping commercial vehicle/bus manufacturers and fleet managers to comply with EC regulation 661/2009, which makes lane-keeping assistance mandatory for vehicle classes M2/M3/N2/N3 as of November 2013. This requirement is currently being met by an acoustic, visual, or haptic warning signal to the driver. However, the implementation of iHSA also facilitates active lane-keeping. This means that HCVs and buses can now be kept in lane without input from the driver, i.e. actively, thus significantly reducing the risk of an accident.

The Tedrive innovation also compensates for surface ruts and provides crosswind stabilization. Steering assistance adapts to the prevailing conditions, thus delivering considerably lighter and more precise steering characteristics. Alongside these functions, however, it is now also possible to implement new comfort and convenience features such as assistance with parking and maneuvering.

The packaging space required in either set-up is virtually the same as that of the conventional system. The scalable modular solution of Tedrive's recirculating ball steering system offers commercial vehicle and bus manufacturers great flexibility, as well as increased comfort and safety, paired with a high degree of performance density and environmental compatibility.

In order to expand its product portfolio to include recirculating ball steering systems, Tedrive has also acquired Chemnitz-based steering specialist RBL Brems- und Lenksysteme GmbH, making Tedrive a fully comprehensive provider of steering technologies. Tedrive steering systems now span all vehicle segments, from lightweight design for e-mobility, through to recirculating ball steering for heavy commercial vehicles and buses, making Tedrive a one-stop-shop for every OEM's steering needs.

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Making useful models

RIGHT: **VEHICLE SUBJECTIVE EVALUATION – THE VEHICLE IS EVALUATED THROUGH DIFFERENT EVENTS AND CONDITIONS TO BETTER UNDERSTAND THE PERFORMANCE CHARACTERISTICS OF THE VEHICLE**

George Box famously said, "All models are wrong, but some are useful," so how do you ensure that your models are useful? For a recent project, Altair ProductDesign (a division of Altair that provides engineering solutions, including physical and virtual vehicle development) worked with a customer to create a baseline design for a new entry into the mid-size sedan market that leveraged a competitive vehicle model heavily. A strong process for building high confidence models was essential to the success of this project and it required chassis design and development experience combined with both testing and CAE to be successful.

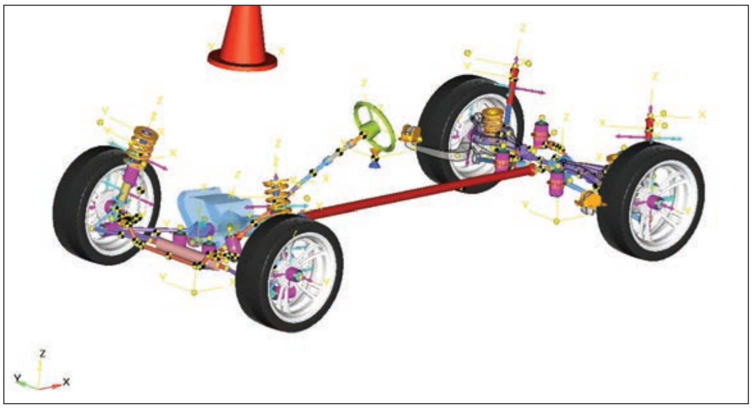
The customer had identified a market-leading target vehicle and the project started like a traditional design project, by determining targets and specifications. Altair was able to accomplish this through the process of subjective evaluation on public roads and at a proving ground, objective measurement in the lab and at proving grounds, kinematics and compliance (K&C) measurement, inertial measurement, hard-point measurement, and vehicle dynamics experience.

The design process started with the creation of suspension models using a multibody dynamics (MBD) analysis package – in this case, Altair's MotionView and MotionSolve, which enable engineers to virtually build the chassis configuration. Although MotionView and MotionSolve include numerous suspension build templates and standard analyses, the model must be further developed with experimental rigor and chassis experience to ensure its validity.

In addition to the use of a coordinate measurement machine to record the baseline vehicle suspension hardpoints, significantly more data needs to be acquired to create a useful model. This includes the center of gravity and inertial property measurement of key suspension components. A trifilar torsional pendulum is not an expensive or complex piece of equipment, but the examples in the Altair development laboratory are well used and essential for model population. Accurate vehicle bushing rates and parameters are critical to MBD models. Experimental values are required for all suspension, steering, and isolated subframe bushings.

There are some rules that help Altair ProductDesign achieve good correlation of its models to test results. These include always measuring hardpoints at the same loading condition and ride height as the K&C test. This is just good experimental practice, but can often be overlooked. Also, build a preliminary kinematics model to obtain preloads and deflections to define your bushing test specifications. This ensures you get a good model and good value from your testing expenses. In addition, be sure to include non-bushing compliances (such as those in the steering system) and MacPherson struts and hubs in your model.

The first level of model correlation needs to ensure line-on-line half-car MBD model agreement with experimental K&C results by including the correct amount of model fidelity and making decisions based on vehicle dynamics experience. Once this is achieved, it creates the primary baseline for iteration and design sensitive studies. It is also the gateway to assembling the initial full vehicle model, and leads to the second level of model correlation – comparison of simulation results with data acquired during vehicle dynamics events. Once you can accomplish that, you'll know your models are truly useful.



RIGHT: **ALTAIR MOTIONVIEW MODEL – TO BE USEFUL THE MODEL'S COMPLEXITY MUST BE MATCHED TO THE LEVEL OF OUTPUT DATA REQUIRED**

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Beyond the K&C basics

Vehicle developers worldwide understand the critical role MTS kinematic and compliance deflection measurement (K&C) systems can play in achieving desired handling and stability characteristics. What they may not appreciate, however, is that the current design of this platform also delivers a range of dynamic testing capabilities to help address a far broader spectrum of vehicle development needs.

The result of a recent full-system design enhancement, the current MTS K&C system features a bundle of optional dynamic testing capabilities not available together on any other K&C system. These capabilities are included in *Optional K&C Testing Capabilities* (see below).

With the exception of dynamic simulation, most later model MTS K&C systems can be upgraded in the field to incorporate all of these capabilities.

Vehicle developers gain three important advantages by utilizing an enhanced MTS K&C platform. First, the ability to perform a wider variety of tests with a single multipurpose platform creates economic advantages, eliminating the need to purchase, operate, and maintain discrete systems for each test.

Second, these capabilities enable the generation of accurate, meaningful data earlier in the development process. This is particularly important for fine-tuning virtual models, which often struggle to predict dynamic behavior. Generating accurate, repeatable dynamic data in the lab not only helps validate models – it also gives suspension engineers valuable insight into subsystem performance that can be acted upon prior to the proving ground.

Finally, by extending the functionality of its standard K&C platform, MTS gives vehicle developers more choices for equipping a given lab: use a traditional K&C system for pure characterization; add select dynamic capabilities to extend its utility; or choose the MTS Dynamic K&C system for comprehensive dynamic simulation.

Speed to market is essential at every stage of vehicle development: virtual developers want to validate models before components are designed; subsystem engineers want insight into new designs before vehicles reach the proving ground. No matter what your role on the vehicle development team, extending quasi-static K&C testing into the dynamic realm can augment your ability to bring prototypes to the test track faster, more efficiently, and with less rework.

THE LATEST DESIGN OF THE MTS K&C PLATFORM DELIVERS A RANGE OF DYNAMIC TESTING CAPABILITIES

OPTIONAL K&C TESTING CAPABILITIES

- **Time history playout: this enables vehicle developers to replicate road data or program and drive synthetic inputs while recording the suspension response on the MTS K&C.**
- **Chassis torsional rigidity testing: this enables vehicle developers to measure the torsional stiffness of a vehicle on the MTS K&C.**
- **Cornering simulation: designed for motorsports and performance vehicles, this feature applies roll, braking, and downforce movement independently at each wheel on the MTS K&C to study full-vehicle effects during cornering maneuvers.**
- **Center of gravity and vehicle inertia measurement: this full-vehicle test accurately measures center of gravity and vehicle inertia properties.**
- **Dynamic simulation: this feature enables engineers to explore suspension response behavior during transient handling maneuvers or synthetic dynamic inputs to each corner of a vehicle independently at up to 20Hz. It is ideal for mechanical hardware-in-loop testing, which integrates the MTS K&C and a virtual vehicle model with the objective to drive the vehicle suspension with a modeled vehicle on a modeled track as if it were on a real vehicle and track, measuring the performance of the modeled vehicle with a real suspension.**

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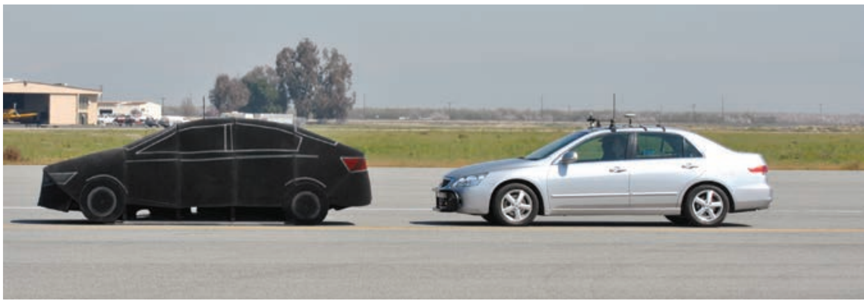
Active safety testing

TOP RIGHT: **GSTV WITH BLACK FOAM BODY PRIOR TO IMPACT**

RIGHT: **GSTV FOAM BODY PANELS SEPARATE DURING IMPACT. THE TEST VEHICLE DRIVES OVER LPRV**

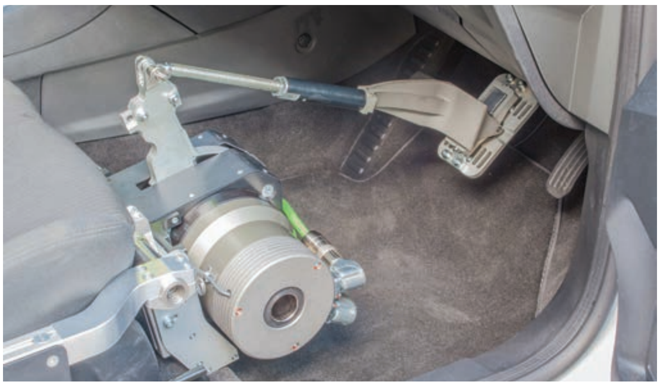


Anthony Best Dynamics (ABD) is a leading supplier of driving robot systems used for vehicle development. ABD driving robots are used by 17 of the top 20 most successful manufacturers in the world to develop their vehicles.

The testing of active safety system technology is becoming increasingly important for automotive manufacturers, and ABD has been working on new products to help satisfy customer requirements. ABD has recently signed a collaborative agreement with California-based Dynamic Research Inc (DRI) to jointly supply guided soft target vehicles (GSTVs) for active safety development.

The GSTV enables high-speed impacts to be performed on a test track without damage to the test vehicle. DRI specializes in applied research, development, and consulting in various fields including vehicle dynamics and control, biomechanics, and accidentology. The GSTV system developed by DRI comprises a hardened low-profile robotic vehicle (LPRV), which serves as a means of conveyance for a soft foam car body that acts as a realistic moving collision partner. In the event of a collision with the GSTV, the lightweight body panels of the soft foam car separate from each other and from the LPRV. The subject vehicle then drives over the LPRV, minimizing risk to test personnel and damage to expensive test vehicles. The LPRV fitted with a foam body is capable of speeds in excess of 70km/h (43mph). Braking actuators fitted to all wheels enable rapid deceleration of the LPRV.

BELOW: **RBR1500 FITTED TO A FORD MONDEO**



A jointly developed version of the GSTV fitted with ABD driving robot controller hardware and software enables the system to be used in conjunction with other vehicles fitted with ABD driving robots. ABD robot controllers enable multiple vehicles to be coordinated in position and time with centimeter-level accuracy. Other modes of operation enable the GSTV to be synchronized with a human-driven vehicle.

Joe Kelly, senior engineer at DRI, says, "The collaboration between DRI and ABD uses our innovative, patent-pending, low-profile vehicle design, and combines it with ABD's proven robot driver technology to produce a technically unsurpassed product that enables customers to precisely conduct vehicle-to-vehicle collisions."

Safety is of the utmost importance when using remotely controlled vehicles such as the GSTV. Multiple electronic controllers check that each other are functioning correctly. These are coupled with an innovative redundant braking system that ensures the GSTV can always be stopped quickly in an emergency.

In other driving robot news, ABD has developed a new brake robot actuator – its most powerful brake robot ever. The RBR1500 is a compact high-performance unit that can apply more than 1,200N to the brake pedal at speeds in excess of 1m/sec and has a peak force in excess of 1,500N. In common with all ABD brake robots, the RBR1500 can be used for precision control of pedal position and force, as well as vehicle deceleration using external feedback. The RBR1500 can also be used in conjunction with ABD steering robots.

ABD driving robots can be used with GPS-corrected inertial motion packs from OxTS, GeneSys, and iMAR, for feedback on path following and general robot test triggering. A recently developed software module now enables vehicles fitted with motion packs from any of these suppliers to be combined for vehicle-to-vehicle range control.

ABD has also recently completed work with Racelogic to enable its VBOX products to be used as feedback for forward-collision warning testing in conjunction with ABD's compact pedal actuators.

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Dynamics testing tools

Accurate determination of all movements is an important requisite during vehicle dynamics testing. The ADMA (Automotive Dynamic Motion Analyzer) Gyro system has been specially developed for such applications. The ADMA-G system consists of three optical gyroscopes, three servo accelerometers and an internal GPS receiver for absolute positioning with WAAS or RTK-DGPS correction. A 32-bit DSP processor unit continuously calculates the speeds and position in all three spatial axes, as well as pitch, roll, heading and side slip angles, from the sensor signals and external information.

Advanced driver assistance systems (ADAS), such as where a cruise control system also measures the distance to a vehicle in front and reduces speed as necessary, must be evaluated during the development phase. For the validation process in road trials, a test system is used that was developed by GeneSys Elektronik in cooperation with Dewetron and TÜV Süd Automotive.

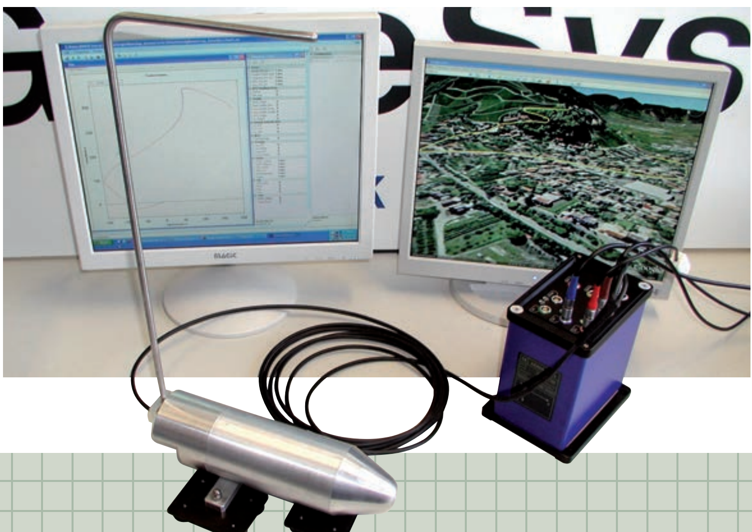
It is notable that this system, consisting of a combination of a GPS system and the ADMA inertial system, registers the positions and movement of multiple vehicles synchronously. In addition, further data (such as video data, data from the CANbus of the vehicle or other analog or digital data) can be synchronously registered and displayed. At the same time, the data of all the vehicles is synchronized. Other applications such as lane departure warning (LDW) and forward collision avoidance (FCA) are also addressed.

Electrical/electronic systems in automobiles must operate safely. In order to prove the functional safety of ABS/ESP systems during road trials according to ISO 26262, a number of situations are simulated; for example, a sensor failure. This is a measure to examine the extent to which the vehicle can still be controlled in such a case. Dewetron is currently introducing a new test system for such trial runs, which displays all vehicle values, exact position data and the internal parameters of the ABS/ESC system.

The test system consists of a DEWE-511 data recorder and a special plug-in for the DEWESoft 7 data acquisition software. GeneSys Elektronik's ADMA system is also used to obtain exact vehicle position and movement measurements. With a combination of GPS and inertial sensors, the system delivers very precise position data with an accuracy of a few centimeters.

One prerequisite of driving tests as part of vehicle development is to precisely determine the vehicle's position. In such applications, the ADMA delivers optimized and highly precise data. To ensure precise positioning even under difficult GPS reception conditions, GeneSys now presents the new ADMA-PP post-processing software, which allows optimization of ADMA data recordings and inclusion of GPS correction data after the test drive.

The software's core is a Kalman filter that perfectly combines GPS and inertial data. While the real-time option continues to be provided by the ADMA system, off-line calculation has decisive advantages. The easy-to-use package is completed by an auxiliary module with a barometric altitude sensor allowing accurate measurements of critical height-related data.



LEFT: **ADMA-G: GYRO SYSTEM FOR DYNAMIC TESTING WITH GPS SUPPORT**

BELOW LEFT: **ADMA-PP: POST-PROCESSING SOFTWARE FOR ROAD TESTS**

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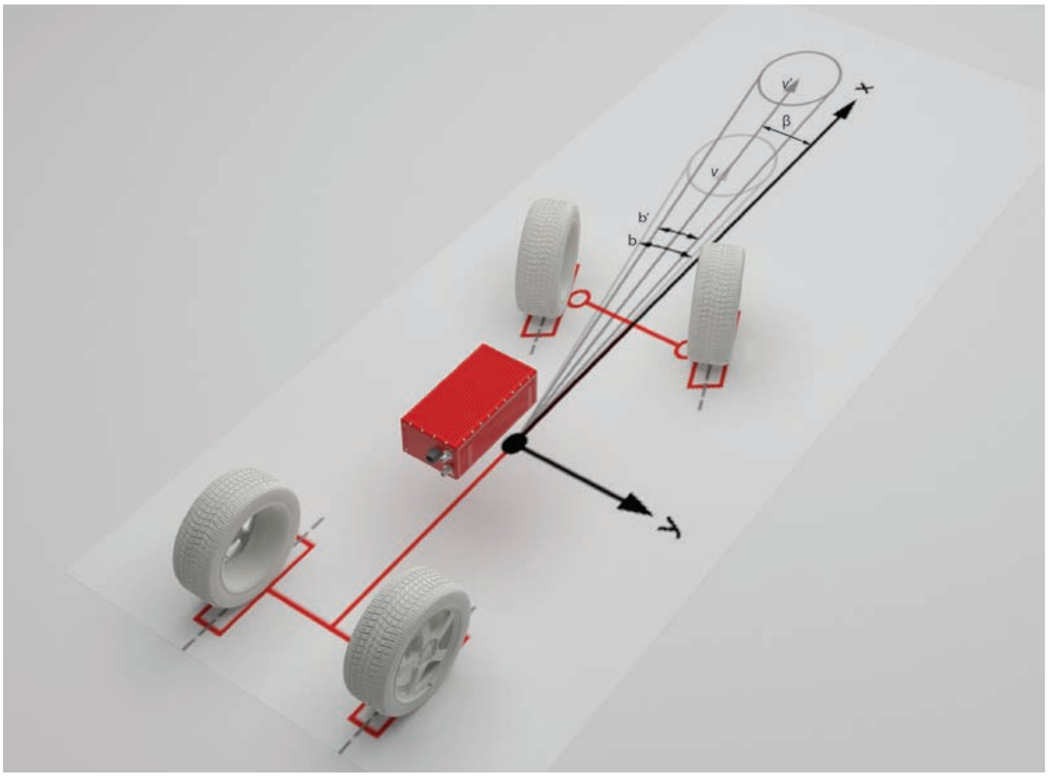
Slip angle accuracy



LEFT: **FOR A GIVEN SLIP ANGLE β, IT IS CLEAR TO SEE HOW THE RELATIVELY FIXED VELOCITY ERROR OF GPS AND INERTIAL NAVIGATION SYSTEMS (REPRESENTED BY THE CIRCLES) AFFECTS SLIP ANGLE ACCURACY (*b* AND *b'*) AS THE VELOCITY VECTOR (*v* AND *v'*) INCREASES. THIS IS WHY OxTS STATES SPEED WITH SLIP ANGLE ACCURACY CLAIMS**

SPEED (KM/H)	SLIP ANGLE
10	0.59°
20	0.32°
50	0.19°
100	0.16°
200	0.15°

Oxford Technical Solutions (OxTS), a leading manufacturer of GPS-aided inertial navigation systems for vehicle dynamics testing, has racked up another industry first by becoming the only manufacturer to routinely state speed alongside slip angle accuracy. "It's not immediately obvious to new customers why slip angle accuracy changes in relation to speed," says Keith Dear, sales manager at OxTS. "But slip angle is a function of a vehicle's forward and lateral velocities, and is therefore limited by the accuracy of those measurements, regardless of what type of system is used to capture it.

"As the need to accurately and confidently measure slip angle increases, we constantly strive to help customers understand what level of accuracy to expect and how best to achieve it," he continues. "By clearly stating the speed at which our slip angle accuracy is achieved, OxTS hopes to make it easier for customers to understand whether a system suits their requirements or not."

For years, the rule of thumb has been that slip angle accuracy can never be greater than heading accuracy. While this fact remains true, it fails to address why accuracy is affected by speed. This is explained when you consider that slip angle is actually the vehicle's velocity vector, and the error on GPS-based velocity systems – including GPS-aided inertial navigation systems – is relatively constant regardless of how fast the vehicle is traveling. At lower speeds, therefore, the significance of that error is proportionally much greater than at high speed.

The expected slip angle at any speed can easily be calculated as follows:

slip angle accuracy
$= \text{heading accuracy}^2 +$
$\tan^{-1}\text{velocityaccuracyspeed}^2$

Assuming a heading accuracy of 0.15° and a velocity accuracy of 0.1km/h, the table (above right) shows how slip angle accuracy improves with speed, and why it is important to know what speed a given slip angle accuracy was achieved at.

"There are many potential sources of error to control when attempting to measure slip angle accurately, no matter what solution you use," explains Dear. "That's one of the reasons OxTS spends so much time helping customers, because it's not just about buying the right equipment. By spending the time to accurately mount the measuring device close to the point of interest, and understanding the importance of how the measurement is made, customers can be confident in the data they collect. Clearly stating the speed at which our slip angle accuracy is achieved is just one way in which we hope to make life a little less complicated for all customers."

OxTS's RT Inertial and GPS navigation systems can measure slip angle to an accuracy of 0.15° RMS at 50km/h; have an update rate up to 250Hz; and have wide bandwidth. All outputs are computed in real-time with very low latency. A more detailed explanation on achieving slip angle accuracy can be found on the OxTS website.

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Chassis systems expertise

In the two years since BWI Group became an independent company, it has acquired a global reputation as a leading supplier of premium chassis systems. The product portfolio is split into two complementary groups: ride and handling technologies, and braking technologies. In both areas, the offering ranges from high-quality volume items such as twin-tube dampers and brake system components, through to highly sophisticated active systems and the expertise required to integrate them with high-end vehicle programs.

The company believes that it is the only major supplier to offer a range of technologies in both damping and braking, giving it a unique capability as increasingly sophisticated electronic controls allow tighter integration of vehicle systems. With an extensive in-house electronics capability and substantial vehicle-level experience, BWI can deliver complex, integrated solutions reliably and quickly in any region.

It's a capability that is in increasing demand. As vehicles acquire a growing range of sensors and an increasing ability to share data between systems, the benefits of tighter integration are increasing. BWI products are building on these opportunities by offering compatibility with open architectures and a modular structure that takes time and cost out of development programs.

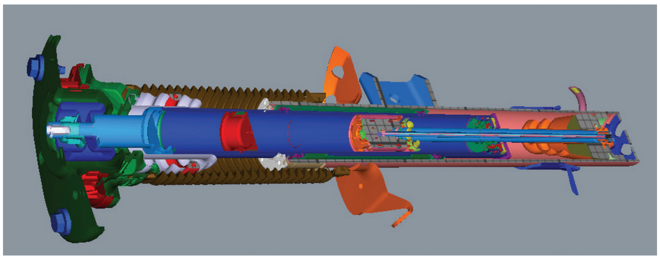
At a component level, BWI's strategy of delivering a comprehensive, well-supported portfolio in each of these sectors allows simplification of a vehicle manufacturer's engineering, purchasing, manufacturing, and logistics. Simplification is increased by the high level of electronic configurability, allowing broad model ranges and global platform strategies with minimum impact on complexity.

BWI'S PRODUCT PORTFOLIO

- Adaptive powertrain mounts
- Monotube dampers/struts
- Twin-tube dampers/struts
- MagneRide ride control system
- Coil springs
- Air suspension
- Corner and axle modules
- Brake components and assemblies
- Friction components
- Brake control systems
- Brake apply systems
- Electronic stability systems
- Roll-control systems
- Systems integration

The belief that complication must be minimized is reflected in the design of all BWI systems, which demonstrate the ability of successful innovation to deliver the best-possible performance using the simplest, most elegant technologies. An excellent example of this is MagneRide, which is widely recognized as one of the world's most advanced production ride control system. Using fixed-orifice dampers whose response can be changed by electromagnetically controlling the rheological properties of the damper fluid, MagneRide enables vehicle engineers to achieve an exceptional combination of ride and handling performance. Unlike conventional, valve-based, semi-active suspension systems, MagneRide is mechanically simple, with no valves or other small moving parts.

MagneRide illustrates BWI Group's strategy of applying robust innovation to solve each customer's unique challenges at an affordable price, whether that requirement is to deliver groundbreaking dynamics, or to simplify vehicle assembly. Combined with the company's in-depth electronics capability and vehicle-level integration expertise, it's also the strategy that will help vehicle manufacturers take the next step, further improving dynamics, refinement, and safety by building on the growing synergies between vehicle systems.



BWI CHASSIS PRODUCTS: ADAPTIVE POWERTRAIN MOUNT (BELOW LEFT)**; ACTIVE STABILIZER BAR SYSTEM** (BELOW CENTER)**; RANGE ROVER EVOQUE'S MAGNERIDE DAMPER** (ABOVE)

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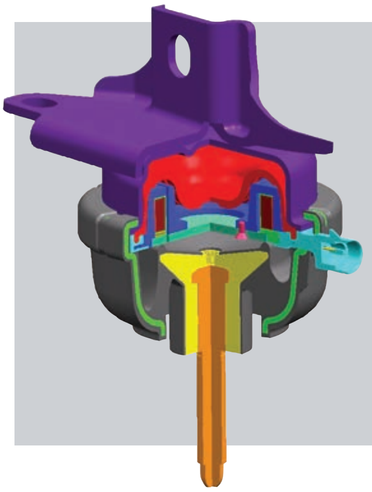
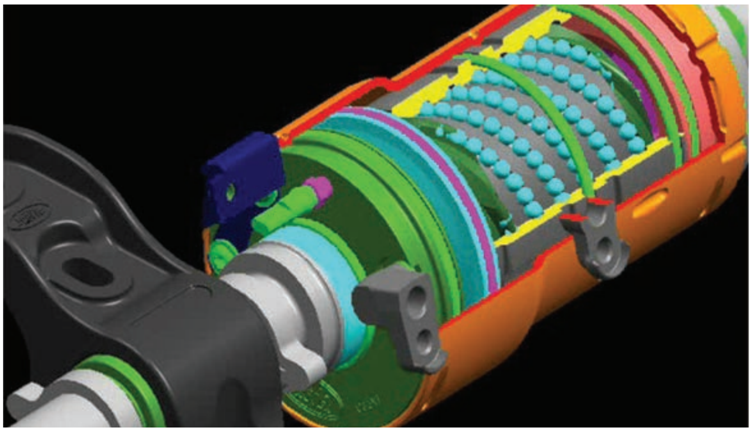




Collision warning testing

ABOVE: **DEWE2-M4 WITH ADMA AND MOB DISPLAY IS A COMPLETE TURNKEY EXPERIENCE**
BELOW: **DEWE2-A4 MODULE**
BELOW RIGHT: **DEWESOFT SOFTWARE WITH POLYGON PLUGIN**

The growing number of sensor and actuator clusters in modern vehicles allows for complex driver assistance systems. One of these is forward collision warning (FCW), which warns the driver of an imminent frontal collision using acoustic, haptic, or optical warnings.

NHTSA defines three driving scenarios for evaluating FCW systems. In the first test, a subject vehicle (SV) approaches a stopped principle other vehicle (POV) in the same lane of travel. The second test begins with the SV initially following the POV at the same constant speed; after a short while, the POV stops suddenly. The third test consists of the SV, traveling at a constant speed, approaching a slower moving POV, which is also being driven at constant speed.

The goal of these tests is to verify that the FCW system warns the driver early enough to avoid a potential crash. Time to collision is the key factor to be determined.

While this sounds easy to evaluate, the tests are in fact challenging. For each vehicle, data for multiple sensors and sources must be acquired synchronously: GPS/INS data for the vehicle positions and trajectories; CANbus data for vehicle speed and yaw rate coming from the vehicle; analog data for accelerations and acoustic FCW warnings; and video data for optical FCW warnings.

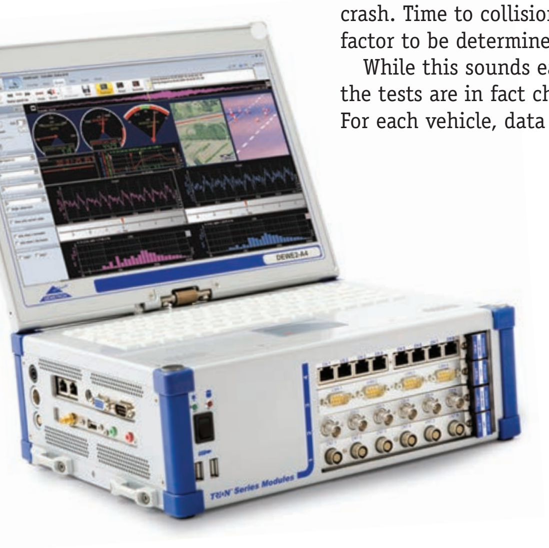
But what is even more challenging is to have full synchronization between the two vehicles during the measurement, in order to accurately calculate the relative distance between the vehicles, their trajectory, and their speed. With its Sync-Clock technology, Dewetron provides a turnkey solution for this challenging task.

The powerful recording software shows live data online during the measurement in freely customizable data visualization instruments (meters, bar-graphs, recorders, GPS map, etc.). The 3D display is capable of showing the relative position between the two vehicles, online.

The acoustic warning of the FCW system is measured using an analog ICP microphone. While the microphone acquires not only the sound of the warning, but also environmental noise (motor noise, talking, etc.), an online FFT calculation filters and isolates the input signal so only the acoustic warning is detected. The optical FCW warning is acquired with a synchronized camera.

The synchronization of the measurement systems between the two vehicles is done by GPS-Sync. This technology uses the pulse-per-second from the GPS signal and a highly precise clock in each data acquisition system. Since it is necessary to see important values, such as the relative distance between the two vehicles or vehicle speed during the tests, live measurement data is transferred between the two vehicles using a wireless network. The calculations and combinations of the measurement data of the two vehicles are executed directly in the subject vehicle and can be visualized so that the driver or measurement technician can immediately see all important values during the measurement.

The sequencer is used to automate the tests and to monitor characteristic values to immediately indicate whether the test was successful or if it needs to be repeated, or to generate final reports after the test cycle has been completed.



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Rear view

JOHN HEIDER OFFERS SOME THOUGHTS ON WORKING WITH REAR-WHEEL DRIVE

"One of the not-so-well-kept secrets of the vehicle dynamics engineer is that tuning and developing RWD vehicles is more fun than FWD"

"Just once in my career I'd really like to work on a rear-wheel-drive vehicle!" was the proclamation from an exasperated development engineer who'd spent the first 10 years of his career tuning a steady stream of very good front-wheel-drive vehicles. Unfortunately for him, his employer stopped making RWD passenger vehicles in the mid-1990s shortly before his career began. The occasion of his remark was upon exiting a 400bhp+ RWD American-made V8 performance car, which had found its way to a proving ground in Europe. Not an unusual reaction given the circumstances.

Vehicle dynamics development for OEMs is an extremely challenging and complex process. Using CAE tools to their fullest extent; balancing subjective evaluation results with objective testing; meeting functional, cost, quality and weight targets; and fulfilling sometimes fleeting management expectations is a daunting task. This complex process, however, is basically the same whether you are developing a small FWD economy car, a large SUV people hauler, or any vehicle in-between. Having said that, using this logic on numerous young engineering graduates in the department who invariably asked to work on Mustang as opposed to minivan programs was usually met with quizzical stares. Whether they knew this at the time or not, they were about to learn one of the not-so-well-kept secrets of the vehicle dynamics engineer: tuning and developing RWD vehicles is just more fun than FWD vehicles.

Those of us somewhere between those engineering graduates and an 'old guy' remember when popular thinking was that RWD platforms were dead and the future consisted of nothing but fuel-efficient, lightweight, FWD platforms in all passenger car categories. Many development engineers similar to the one quoted above have spent entire careers mastering the development of MacPherson strut front suspensions, twistbeam rear suspensions, and – if they're lucky – non-driven, multilink rear suspensions. These engineers have developed some good, some very good, and some outstanding vehicles. None of them have ever had the pleasure of powering one of their prototypes out of a corner, adding counter-steer to control the rear slip angle, and using the throttle and steering wheel to maintain a smooth, controlled oversteer attitude onto the ensuing straight. Pity them... it may not be the fastest way through a corner, but it certainly is the most fun.

To their credit, BMW, Mercedes, and other smaller OEMs steadfastly continued to develop and refine their RWD platforms when others believed the popular FWD thinking. Looking at the mid- to high-end luxury segments today, RWD platforms are the norm, with ever-increasing sales of AWD variants of these RWD vehicles (perhaps a nod to our Audi friends and their product planners from the 1980s is in order as well). The current RWD/AWD platforms feature the latest technology that vehicle dynamics engineers know and love: good weight distribution; integral link rear suspensions to better balance castor and recessional compliances; continuously variable dampers; active differentials; and stability control systems for those non-dynamically inclined. Coupled with the latest generation of high-horsepower, high-torque powertrains, the number of vehicles currently for sale which fall into the nebulous 'fun-to-drive' category has never been greater.

Will the new fuel economy standards see a whole new generation of development engineers destined to a career worrying about torque steer acceptability instead of power-on oversteer acceptability? Time will tell.

John Heider is from Cayman Dynamics LLC, providing vehicle dynamics expertise to the transportation industry: www.caymandynamics.com

STEERING THE FRONT WHEELS AND DRIVING THE REAR WHEELS WILL ALWAYS HAVE A PLACE IN THE HEARTS OF BOTH THE PUBLIC AND VEHICLE DYNAMICS ENGINEERS

ON THE WEB
Further comment from Heider at vehicledynamicsinternational.com

Is this a setup?

CARS WE DROVE RECENTLY THAT DIDN'T BEHAVE AS THEY SHOULD

CASE 25: HONDA CIVIC, BY GRAHAM HEEPS

SPECIFICATIONS

Honda Civic 1.8 i-VTEC ES
Dimensions: 4,300mm (L) x 1,770mm (W) x 1,470mm (H). Wheelbase 2,595mm, track 1,540mm
Curb weight: 1,270kg
Suspension: MacPherson strut front with ARB; torsion beam rear with ARB
Brakes: Front 282mm ventilated discs, rear 260mm solid discs
Steering: EPS. Turning circle 5.63m (at body); 2.62 turns lock-to-lock
Tires: Continental Premium Contact 2, 205/55 R16

It's news to no one that the small family car, or C-segment, is perhaps Europe's hardest fought. Our belief that the Ford Focus and Volkswagen Golf (even on the eve of the Mk6's replacement!) remain the benchmark here was recently confirmed by a Toyota engineer charged with bringing the hitherto uninspiring Auris up to the class benchmarks.

Meanwhile, Honda has introduced its latest pretender to the throne, a thoroughly refreshed Civic. The fallout from the Japanese earthquake and tsunami, on top of the global recession, has meant a tough couple of years for Honda, which has one of the best engineering reputations in the business. The Civic is the car to start the fightback in Europe, followed by the new CR-V (see page 8).

The first signs are good. The exterior styling is sharp and the interior is comfortable and stylish – we particularly like the sweeping dashboard with its digital speedo. The build quality feels right and there's plenty of space, including in the trunk, though the rear window remains a visibility black spot.

Honda's high standard in powertrain development is maintained by the 1.8-liter petrol motor and sweet-shifting gearbox, although the current engine line-up is firmly out of step with the class norm. A 1.6-liter diesel being introduced as these words were written will help redress the balance, but we also think it's time to embrace the downsized petrol-turbo trend.

The real problem is the ride, however. Frankly, we're a little tired of poor-riding cars that really have no excuse to be so. Honda claims that the new, stiffer rear torsion beam and fluid-filled compliance bushing should improve this Civic's ride considerably, and the 205/55 R16 tires on our mid-range ES-grade test car were plenty generous. So why, then, must buyers tolerate a lack of suppleness and a refusal to settle on bobbly UK surfaces, yet still have to put up with body roll?

The class best manage to soak up the patchy surface of a typical British road without complaint. Granted, they also have multilink rear suspensions, but it must be possible to do better than this with the Civic's torsion beam – a feature that's particularly noticeable in the disparity between front and rear behavior over ruts and ridges. At least the noise suppression is better than before.

Given that the Civic is built in the UK, the ride and the disappointingly vague steering are a let-down. Honda's European R&D is led from Germany, a land of smoother roads than ours. But if other OEMs can do better, surely Honda should be able to, too?

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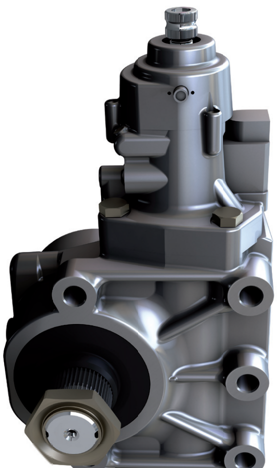

In the commercial vehicle sector, tedrive premieres its patented iHSA® module, the intelligent Hydraulic Steering Assist. The iHSA® module can be integrated into tedrive rack & pinion and tedrive recirculating ball steering systems, offering commercial vehicle manufacturers a high degree of design and functional flexibility for all hydraulic steering systems. The result is a compelling combination of optimized ride and handling performance, comfort and safety, with the optional iHSA® module facilitating functions such as active lane-keeping aimed at reducing serious accidents. With the iHSA® torque overlay, heavy trucks and buses can now be steered without driver input for active lane-keeping, and equipped with a range of additional functions such as CV park assist and crosswind stabilisation.

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