Mobility for tomorrow

Schaeffler Symposium 2018

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On the Path towards Mobility for Tomorrow

Foreword

There is a solution to almost every problem. If there isn't, we find or invent one. This is how we define being "innovative" at Schaeffler, which is one of our four core values. This definition also demonstrates our understanding of innovation: It is not an end in itself but serves our customers and society. With our strategy "Mobility for tomorrow" that we put in place two years ago, we give direction to the inventive spirit of our engineers. As a systems partner, Schaeffler contributes to shaping the growing demand for mobility in a sustainable way, whether in urban or interurban transport of passengers and goods.

In this regard, we are building on the foundation of a long tradition of innovation that has been documented in the conference proceedings of the Schaeffler Symposium which has been taking place every four years since 1978. Environmentally-friendly drives have been the focus from the very beginning. Whereas optimizing conventional powertrains in terms of minimum fuel consumption and high comfort levels were the main issues in the beginning, hybridizing the powertrain was already the topic of discussions at the 2002 Symposium. Since then, electrification has become increasingly important, which is not only reflected in today's volume-production projects for hybrid modules and electric axle drives, but also in the conference program of this year's Symposium: For the first time, we will be presenting "electric and hybrid drives" in a separate section. At the same time, we keep in mind that internal combustion engines and transmissions will have a place in the drive-system mix of the future, which is why we will also be presenting innovations on these topics.

We also want to look beyond what we can do today, of course, at this year's Symposium. I invite you to read the chapter on "people movers", in which we merge the drive and the chassis into one unit.

On behalf of my fellow Executive Board members, I would like to wish you an interesting Schaeffler Symposium 2018 and hope you will find it inspirational at the same time.

Klaus Rosenfeld
Chief Executive Officer

Schaeffler
Symposium
2018

Mobility for Tomorrow

Prof. Dr.-Ing. Peter Gutzmer
Uwe Wagner
Matthias Zink

What moves us: Future mobility concepts

There are many drivers for tomorrow's mobility: Climate change is just as important as urbanization, digitalization and automation. All of these trends are concentrated in innovative, visionary development projects such as Neom in Saudi Arabia [1] and Chengdu Tianfu District Great City in China [2]. When planning such cities, the same points are always in focus:

1. New, connected mobility solutions allow fast and highly automated transport within an urban area.
2. Almost zero emissions result in a high quality of life.
3. Only renewable energy is used.

In today's major cities, congestion is a part of everyday life. According to data by GPS provider Tom Tom, every car driver in "Mexico City" loses 59 minutes of their lifetime in traffic jams – calculating only the required additional driving time caused by heavy traffic. The expansion of local public transport is therefore being promoted worldwide in all major cities. However, this should not only prevent traffic collapse, but also reduce the dominance of car traffic in major European cities. Easing the traffic situation by means of individualized, pedestrian-oriented mobility leads to more livable downtown areas with less traffic. Added to this is the fact that intra-urban good transportation has been increasing significantly due to online shopping. For these types of traffic new vehicle concepts are needed, which achieve high flexibility with low space requirements.

The electrified bicycle (pedelec) is an ideal intra-urban mode of transport. Globally rising sales increasing the share of bicycles in traffic. However, even a high-tech pedelec has the same classic disadvantages of regular bikes. It lacks a weather protection for the user. The load transportation is severely limited in a single-track vehicle. Against this backdrop, Schaeffler developed the concept study "Bio Hybrid" and presented it for the first time in 2016. It transmits the pedelec idea to a two-track vehicle that supports the rider by a low-voltage drive of 250 to 750 watts. The braking energy is recuperated as much as possible, as is usual with a hybrid drive. The electric drive has a reverse gear to ensure a good maneuverability of the lightweight vehicle. The track of the Bio Hybrid is designed with 800 mm so that public bicycle paths can be used. As a result, not only passenger transport, but also transport of goods is environmentally friendly and can be implemented independently of car traffic. A separate limited liability company (GmbH) was founded in 2018 to develop the vehicle to volume production readiness in a start-up atmosphere and to market it quickly.

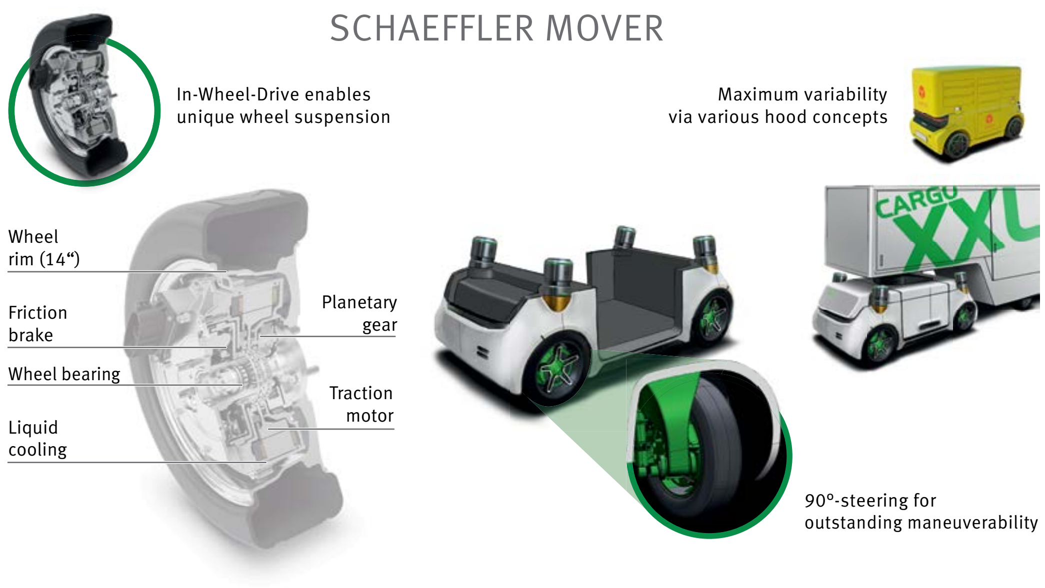
Another potential solution to ease the traffic situation is a people mover, also called a robo cab or robo taxi. This generally refers to an autonomous and electrified vehicle that replaces today's car sharing concepts, primarily in an urban environment. These mobility solutions will initially be used in separate areas and later extended to public traffic. According to several studies, the rapid progress in autonomous driving enables these mobility services to be launched rapidly and in large volumes [4; 5]. Southeast Asia's major cities (such as Singapore) offer ideal conditions for this. An infrastructure is currently being developed there, and society is open to the idea of a sharing economy. In all likelihood, these concepts will complement or even entirely supersede local public transport.

At the colloquium 2018, the study "Schaeffler Mover" will make its debut as the technical basis for such a mobility concept. This vehicle, which is consistently designed for connected operation, can transport up to four people Thus, it fills a ma-

jor gap in the automobile industry's portfolio. The Schaeffler Mover enables separation of the body from the vehicle platform. The body, which can be converted for the desired application, can be quickly separated from the platform that contains the entire technology required for driving. Only the sensor system needed for autonomous driving is partially integrated into the body. The core of the platform consists of four "Schaeffler Intelligent Corner Modules" that combine all drive and chassis components in one unit: in-wheel drive, wheel suspension including suspension and the actuator for electromechanical steering [6].

Reliable, flexible and comfortable service with a high level of availability are the essential customer requirements for a highly automated vehicle for urban use. The Schaeffler Mover meets these requirements. The vehicle dynamics control, for instance, allows each Schaeffler Intelligent Corner Module to be controlled individually. This makes it possible to change lanes with optimized lateral forces, which is very pleasant for, a reading passenger, for example. At the same time, this drive concept guarantees very high level of availability and reliability. The chosen wheel suspension allows a steering angle of up to 90°, providing excellent maneuverability in narrow streets as well as fast and targeted stopping for passengers getting in and out.

The high degree of integration featured in the Schaeffler Intelligent Corner Module has additional benefits. The drive and chassis require less installation space overall – passengers and components (battery and accessories) use the space they release. In addition, the module



1 **The Schaeffler Mover with in-wheel drive is a vehicle platform for a wide variety of vehicle concepts and applications, such as robo taxis. The Schaeffler Mover is designed for a wide variety of hood concepts, thus enabling maximum flexibility. Maximum vehicle maneuverability is ensured by the combination of 90 ° steering system and in-wheel drive.**

2 **Primed for change: Clean powertrain solutions and innovative transportation concepts are decisive for the mobility world of tomorrow. Schaeffler's activities range from energy-efficient technologies for low-emission and zero-emission driving to new forms of mobility for the city of the future.**

1 **Efficiently on the move**
With production solutions for modern vehicle architectures Schaeffler helps to reduce air pollution in urban areas. The product range extends from thermal management modules to engine valve control to 48-volt hybrid technologies and electric clutch systems.

2 **Headed for an electrified future**
With all-encompassing solutions Schaeffler makes fully electric powertrain technologies possible, such as electric axles and high-voltage hybrid modules.

3 **Schaeffler Mover**
The Schaeffler Mover with wheel hub drive offers a platform for diverse vehicle concepts. The powertrain and chassis components are combined in a single, space-saving unit. This enables 90° steering and maximizes cabin space: optimally suited for autonomous and electrified mobility solutions such as robot taxis or transportation vehicles.

4 **E-Board**
This ideal means of transportation from the parking lot to the office boasts handy dimensions and a range of 25 km.

5 **Bio-Hybrid**
This compact vehicle with four wheels and an electric traction motor like that of a pedelec offers high levels of vehicle dynamics, tracking stability (roadholding) and weather protection. The concept is designed as a platform so that, in addition to a passenger vehicle, body styles such as a cargo version are possible as well.

allows a simple scaling of the people mover since wider and longer variants can be produced without changes to the drive and chassis.

Schaeffler showed the vision of the in-wheel drive already in 2013. Its development was part of the MEHREN research project. At that time, a Ford Fiesta was used as a demonstration vehicle to show the space-saving and functional advantages of an in-wheel drive [7]. A variant of the drive developed then is now used in the Schaeffler Mover.

In a people mover, connectivity is the essential prerequisite for smooth operation. This is achieved by the vehicle's digital twin which represents a copy of the real vehicle in the cloud. The continuous analysis of operating and condition data is used to identify future maintenance requirements at an early stage. It is also helpful for analyzing usage profiles and using them as a basis for further optimization of the vehicle. Connectivity also provides additional customer value through personalized, digital services.

What guides us: The mobility energy chain

The goal has been defined. To limit global warming to 1.5 to 2 °C, the international community has committed itself to drastically reduce man-made CO_2 emissions of which around 70 percent are caused by the burning of fossil fuels [8]. That is why the two-degree goal can only be achieved at acceptable cost if the energy transition is initiated in all sectors by 2030 [9].

Globally, the traffic sector is an important source of anthropogenic CO_2 emissions.

The share of road transport in the total emissions of developed economies stood at 26.6 % in the OECD average in 2016 [10]. There are considerable variations here, depending, among other things, on political provisions, economic structures and, last but not least, geographical conditions. But even in countries in which initiatives to increase energy efficiency and to change to renewable energy sources are well advanced, the traffic sector has so far not contributed significantly to reducing CO_2 emissions. In Germany, for instance, the emissions in all sectors, converted to CO_2 equivalents, were reduced by 27 % between 1990 and 2015. Savings were particularly high in the buildings sector (-34 %) and in manufacturing (-32 %). The traffic sector, by contrast, only achieved a reduction of 2 % that even decreased to 0 % by the end of 2016. Since the distance-related fuel consumption of individual vehicles has verifiably decreased during the same period, the undiminished amount of emissions must be attributed primarily to the growing demand for passenger and freight transport. Traffic will continue to increase: Relative to the baseline year of 2010, the number of passenger kilometers in motorized individual traffic is expected to increase by 12.9 % by 2030, while tonnage kilometers in freight traffic are expected to increase by 16.8 % [11]. Compared to this rather moderate growth in Germany, road traffic is likely to increase at a downright explosive rate in regions of the world with dynamic economic growth. According to a study by Shell, the number of cars worldwide will increase to around 2 billion vehicles by 2050 [10].

Achieving Paris' ambitious climate goals requires a major shift in future powertrain technologies. Based on market analyzes and its own calculations, Schaeffler has developed a scenario in which the following market shares will be attributable to the different powertrains in the year 2030:

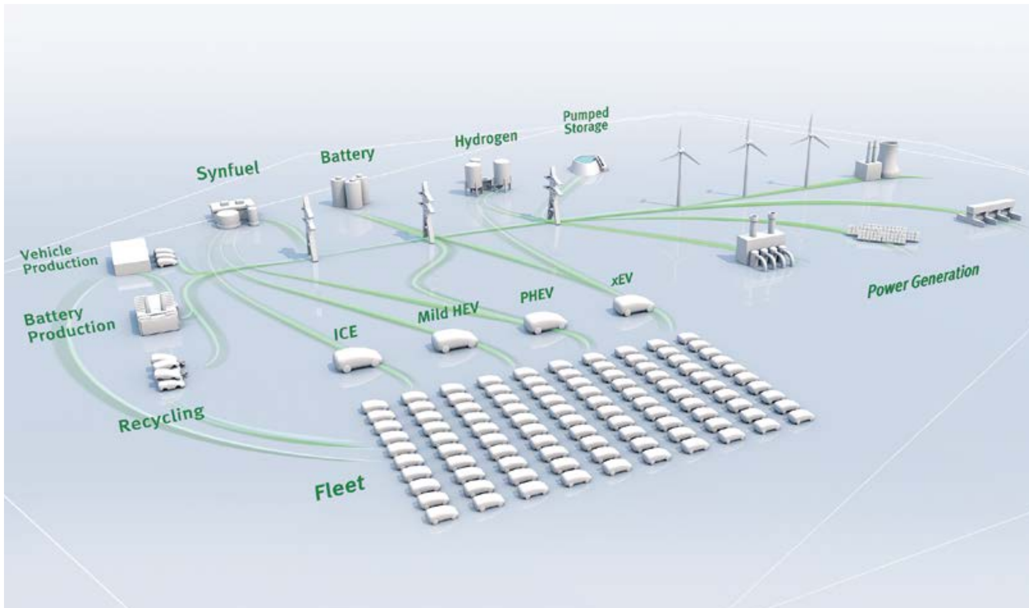
- 30 % of passenger cars produced will be equipped exclusively with an electric drive. Given an annual global production volume of 120 million units, this is equivalent to 36 million electric vehicles. In comparison with the 2030 predictions: According to the International Energy Agency, around 750,000 new battery electric vehicles were registered worldwide in 2016 [12].
- 40 % of the production volume is made up by hybrid vehicles.
- In the remaining 30 % of all vehicles, an internal combustion engine is the only power source.

Of course, this is only one scenario of many that is not likely to happen exactly in the way described. And yet it shows in which direction the future powertrains will develop.

When looking at CO_2, consideration should not only be given to emissions during the utilization phase but also to those portions that are generated by energy supply and the vehicle's manufacture, including the drive and electric energy accumulator. The energy chain can thus be divided into the following three sequences, whereby long-term solutions must be optimized with regard to all parts, including their interaction.

Tank to Wheel

Only the internal combustion engine is responsible for this portion of the CO_2 emissions. However, the worldwide legally required target values for 2030 cannot be achieved by the internal combustion engine alone. In addition, it requires electrical driving for certain periods and the use of recuperation through hybridization. Nevertheless, development work on the internal combustion engine powertrain also continues in order to reduce CO_2 emissions and pollutants in the operating maps expanded by WLTC



3 **The energy chain of mobility for tomorrow**

and RDE. It seems possible to achieve a specific fuel consumption of 200 g/kWh for gasoline engines and thus an efficiency of approx. 45 %. The diesel engine has already reached this value; here the task will be to lower pollutant emissions below specified limits without massively impacting efficiency. And finally, natural gas offers the potential to reduce CO_2 emissions by approx. 25 % compared to gasoline operation. Here the challenge lies not so much in the vehicle technology as more in the construction of an appropriate infrastructure in order to achieve a buyer acceptance that is appropriate to its potential.

Well to Tank

When the greenhouse gas emissions caused by the use of automobiles are viewed holistically, all emissions caused directly or indirectly by energy supply must also be taken into account and converted to the number of kilometers driven. Fossil fuels usually have a very favorable footprint, which must be primarily attributed to the fact that chemical refinery processes but no conversion from one type of energy to another are required. In order to use electricity from renewable sources in the traffic sector, it is a good idea – purely from an efficiency perspective – to use it as directly as possible, that is, in battery electric vehicles. When converting to chemically bound energy, the following sequence should theoretically be chosen in accordance with the number of steps and the multiplying efficiencies:

1. Hydrogen generated via electrolysis and its use in the fuel cell (an option pursued primarily in Japan with the first fuel cell production vehicles)
2. Synthetically generated gaseous or liquid fuels of which e-methane as a substitute for natural gas – that can be added in any amount – represents the energy source with the smallest efficiency losses in production.

In reality though it is not only the best efficiency level that is relevant but also the opportunities for using a specific combination of energy generators, energy accumulators and energy sources. If, for instance, the operating profile of a vehicle does not make it suitable for electrification due to very long distances driven and very brief refueling stops, it makes sense to think about alternative ways to achieve the zero emissions goal. Generally, energy produced from renewable sources is not always available in the right place at the right time, so that the conversion to alternative energy sources such as hydrogen or synthetic fuels could be a reasonable solution. The extent to which second-generation biofuels – that is, without competition with food production – play a role is part of an ongoing public debate.

Production and disposal

Unlike vehicles with an internal combustion engine, the production of the energy accumulator has a significant impact on overall emissions of battery electric vehicles. These are determined with the help of life cycle analyses that take into account the material and energy flows during the production and disposal of all drive components. If this type of analysis is used for an electric vehicle with the electricity mix currently used in Germany, a cradle-to-grave evaluation would reveal that the additional energy required for production is not compensated until more than 150,000 km have been driven when compared to a diesel vehicle [13] (Diesel ICEV vs. BEV100). Battery size is essential in determining when exactly this intersection is crossed. If the share of renewable energy in electricity production increases dramatically, however, electric vehicles will show better results much earlier in the overall climate footprint.

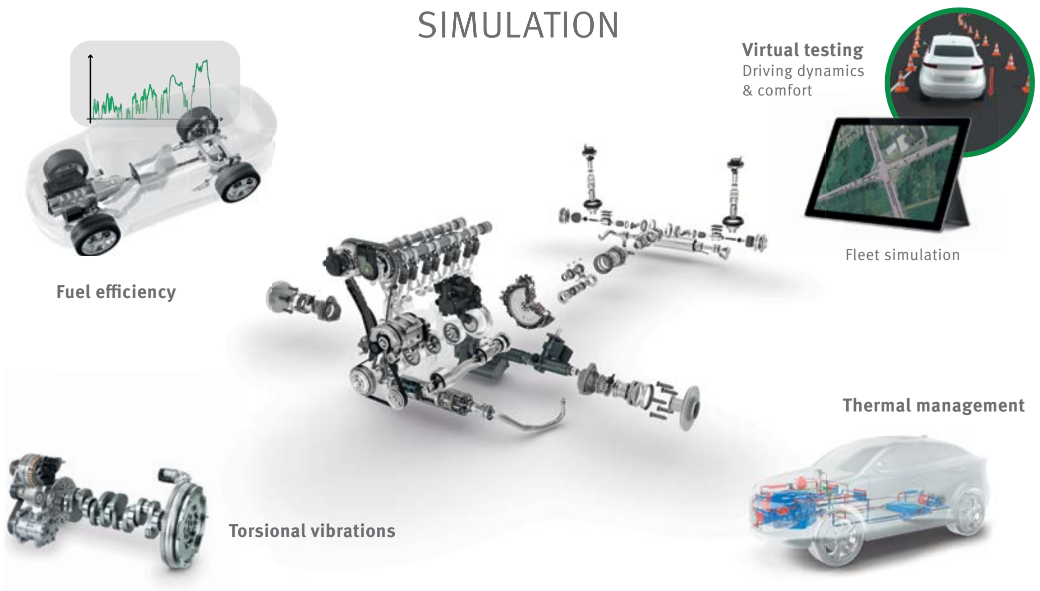
In this environment, Schaeffler is using the scenario described above, according to which around 30 % of all vehicles will be fully electrified by 2030 and 30 % will run with only an internal combustion engine. The remaining 40 % consist of various hybrid powertrains. In addition to developing the right technical solutions for this scenario, the focus will continue to be on observing the following trends:

- Continuing development of accumulator technologies
- Availability of required resources
- Development of production and distribution infrastructure
- Political support
- and, last but not least, the development of traffic demand (quantitative and qualitative).

It is only by combining innovative product developments with realistic scenarios that the traffic turnaround can be successfully managed.

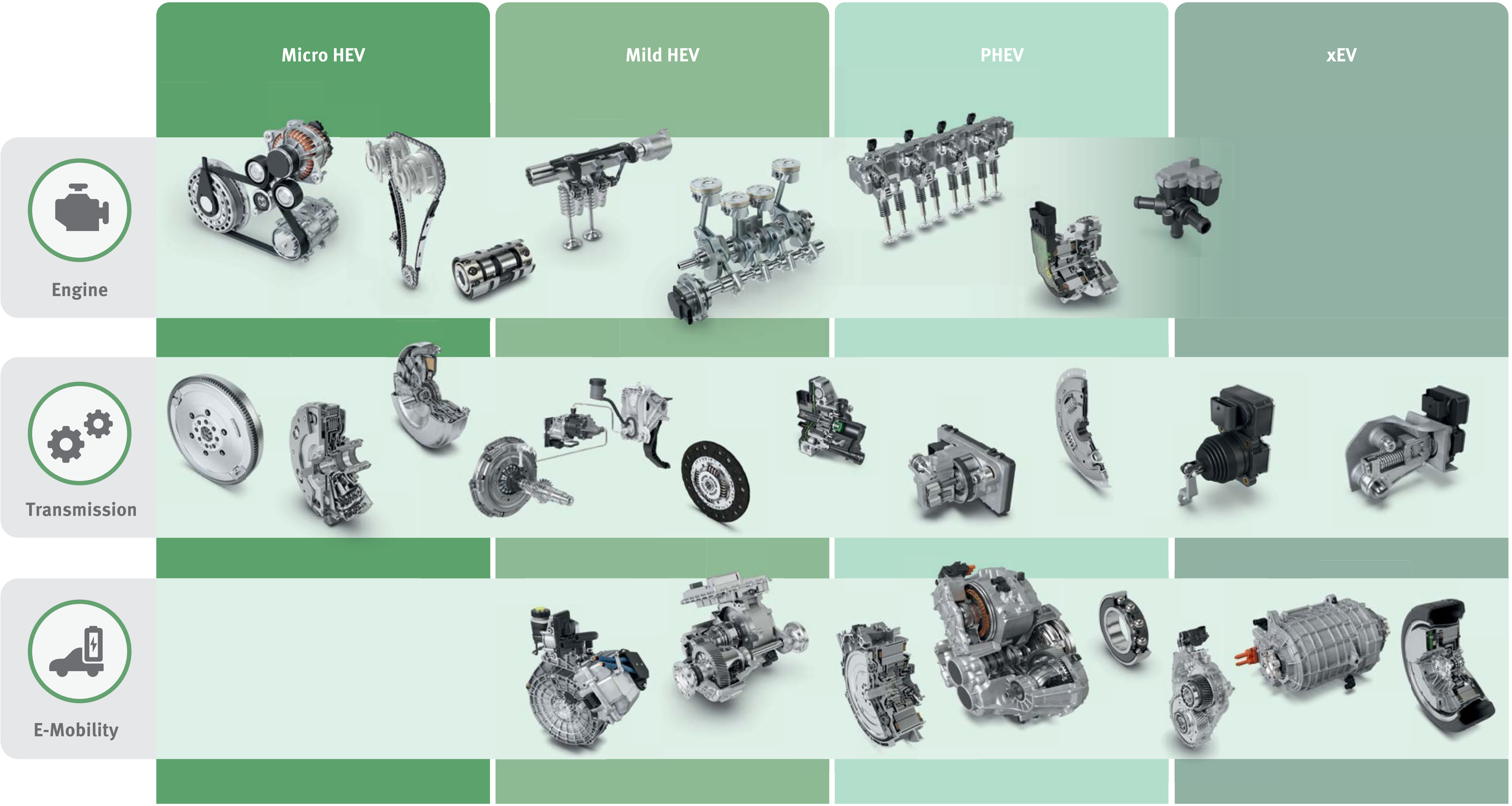
What drives us: The Schaeffler Powertrain Matrix

The key challenge for all conceivable future forms of mobility will be the efficient conversion of stored energy into kinetic energy. A sustainable evaluation of the “right” method of propulsion must include the real use of the vehicle under various driving conditions, in various markets and regions. This also requires the use of a homologation cycle as a foundation to represent this reality in the best way possible. Initial steps have already been taken through the use of the WLTC instead of the NEDC, as well as through the definition of RDE operating points. These new more realistic homologation conditions in combination with ambitions to reduce both CO_2 and pollutants pose considerable new challenges for the automotive industry.



4 **Managing complexity through simulation and development competence**

POWERTRAIN MATRIX



5 The Schaeffler powertrain matrix – solutions for all powertrain concepts

In view of today's primary energy mix along with infrastructure and storage limitations, it is important to develop technically and economically feasible short-term solutions based on real driving conditions and current powertrains, in addition to electrified solutions. A high level of vehicle and powertrain expertise is now required to develop the technically and economically best solutions for incorporating the variety of energy storage solutions mentioned (fossil fuels, e-fuels, batteries and fuel cell); the current diversity of drive units in the form of three engine types (Otto, diesel, electric motor); five different types of transmission; and at least six different hybrid types and positions. In order to play an active role in this transformation of the automotive industry, Schaeffler has moved away from the classic approach separating the "engine" and "transmission" units and is now focusing on the development of the overall system. This supports innovation at the level of the powertrain as well as the entire vehicle while also enabling more sustainable analysis.

Based on the degree of electrification – "micro-hybrid", "mild/full hybrid", "plug-in hybrid" or "xEV" – a powertrain matrix will be used to continue to develop tomorrow's engine, transmission and electric drive subsystems.

What all approaches have in common is that the optimum solution can only be achieved if the entire powertrain is analyzed, including all physical interactions between the internal combustion engine, the transmission and the electric motor; thus, not only the flow of forces but also acoustic and thermal phenomena are taken into account. This analysis process is built on Schaeffler's simulation and development competencies across various product areas, in addition to expertise in the implementation of high-value add product ideas.

New concepts, such as the Schaeffler Mover introduced at the Symposium, offer the basis for completely new opportunities within the mobility of tomorrow by merging the powertrain with the chassis to form the "Rolling Chassis."

The new diversity

There is no uniform technical approach to the mobility of tomorrow; neither for the development of energy sources used in traffic nor for powertrains or automotive components. The ambitious goal of pursuing continued climate protection while simultaneously maintaining mobility as the basis of social and economic development can only be achieved if several paths are pursued in parallel. As a technically innovative partner to the entire automotive and mobility industry, Schaeffler enables this new diversity.

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Engine

1

The Combustion Engine

A Drive with a Future

Dr. Martin Scheidt

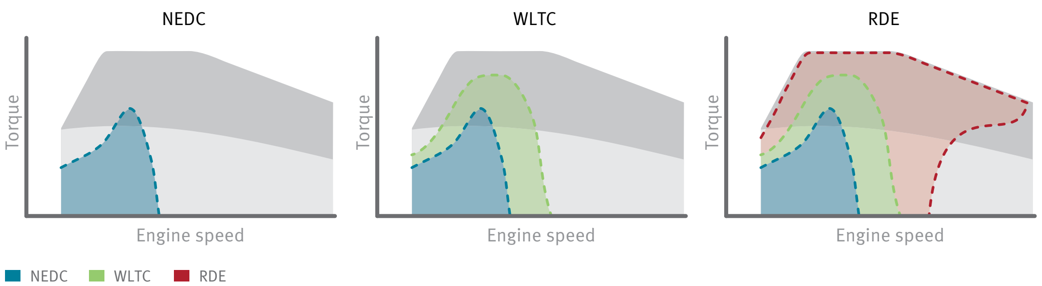
Introduction

For more than seven decades, Schaeffler has been collaborating with automotive manufacturers on components and systems that reduce consumption and emissions while improving performance and driving pleasure. One of the main drivers behind further development are the tightening global emissions requirements. In parallel the reduction of fuel consumption and, thus, CO_2 emissions has also seen increasing focus in the last few years. Meeting both these requirements has been a challenge for the engineers as minimizing one exhaust component often leads to an increase in the other. Future developments in the powertrain will therefore have to address both targets – i.e. reduce harmful emissions as well as CO_2. As this article shows, the combustion engine offers considerable improvement potential both as a singular power source and embedded in a hybrid system.

Another possibility for sustainably reducing the CO_2 emissions of the combustion engine in the short and mid-term are alternative fuels such as compressed natural gas (CNG) and so-called e-fuels, which are manufactured from renewable energy sources by way of a hydrogen chain using CO_2. Fossil CNG is available today and produces approx. 25% less CO_2 compared to gasoline; in future CNG can be produced using biomass or electrolysis/synthesis. The latter two options truly make CNG CO_2 neutral. E-fuels can also act as a suitable replacement for current gasoline and diesel fuels and thus be used in entire vehicle fleets, or be formulated with a blend of oxygen molecules (so-called oxygenates) that could facilitate nearly 100% soot-free and low-pollutant combustion. These fuels do, however, require adapted engine technology to work.

Legislation and technical development

In the last 15 years, combustion engines – in particular gasoline engines – have made enormous progress. Driven by tightening CO_2 fleet consumption limits, engineers followed the trend set by diesel engines and upgraded gasoline engines with direct injection and turbocharging. Using this technology enabled downsizing the engines – with lower displacement and fewer cylinders – offering the same output as their larger predecessors. Additionally, the higher specific engine torques made it possible to mate the engine to a transmission with longer gear ratios as well as more gears allowing a downspeeding concept, i.e. engine operation at lower speeds.



1 **Schematic depiction of the NEDC, WLTC and RDE-relevant performance map ranges**

Promoted by the New European Driving Cycle (NEDC) as the basis for emissions and fuel consumption ratings, the engines were trimmed for optimal CO_2 emissions at low speeds and loads (Figure 1, left). This, in turn, yielded exceptionally low fuel consumption when certifying the vehicle. In reality, however, these vehicles consumed far more fuel in daily use.

Since September 1, 2017 the WLTC (Worldwide Harmonized Light-Duty Vehicles Test Cycle) went into effect for new passenger car models in Europe, and starting on September 1, 2018, WLTC measurements must have been taken for all newly registered cars. The WLTC entails higher average vehicle speeds, more aggressive acceleration and deceleration, stricter testing guidelines as well as consideration given to optional equipment to more accurately depict fuel consumption for consumers (Figure 1, center).

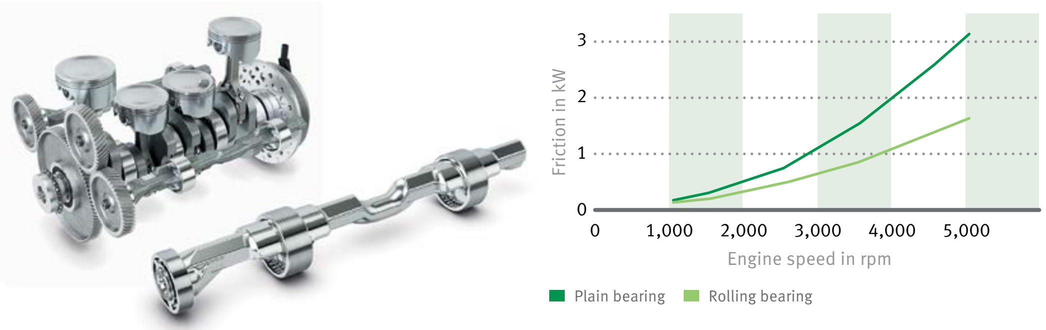
The biggest change for Europe, however, is the new, mandated measurement of pollutant emissions in real-world operation, commonly known as the Real Driving Emissions Test, or RDE (Figure 1, right). This test requires pollutant emissions (NO_x, PM, CO and HC) from a vehicle to be measured and quantified not in the laboratory, but on the road. Since this test does not follow a fixed procedure, but is driven in a stochastic, or random manner, the emissions of the combustion engine must correspond with the legal limit values across the entire engine operating range. For gasoline engines fitted with a three-way catalytic converter, this new regulation requires that the engine is operated with stoichiometric air/fuel ratio at all points in the engine map. Scavenging, i.e. flushing the combustion chambers with fresh air to boost torque, and running a rich fuel-air mixture under full-load conditions to protect components, can therefore no longer be implemented. For constant engine displacement, this means that the rated torque and power output are reduced if no countermeasures are put in place.

One measure that can help is “rightsizing”, or increasing displacement to maintain the same level of engine performance. This is limited by engine architecture and available installation space in the vehicle. Other options for improvement under full-load conditions are cooled exhaust gas recirculation and water injection. An even better idea is to leverage the options afforded by variable valve timing. Implemented correctly, engineers cannot only optimize full-load performance, but also reduce pumping losses and maximize combustion efficiency under part-load conditions. Combined with an adapted turbocharging setup, this results in clean combustion and added potential for reducing fuel consumption without losing engine power.

Another practical measure for reducing fuel consumption that can be used on any engine is to further reduce mechanical friction in the drivetrain. In light of the potential for improvement afforded by avoiding friction losses, this area will continue to be important for Schaeffler when it comes to further developing drivetrain solutions.

Friction reduction

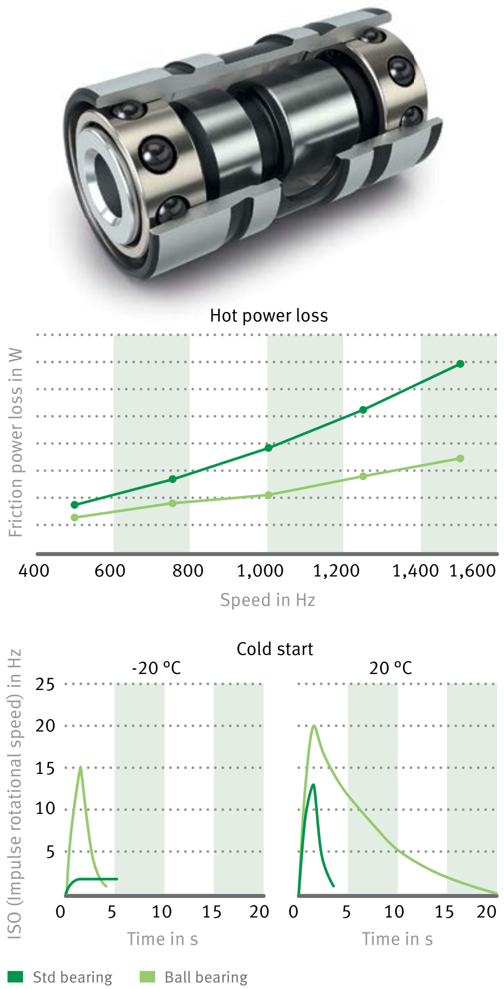
Rolling bearings have already considerably lowered friction levels in accessory drives and successfully replaced plain bearings in camshafts and balancer shafts, turbochargers and finger followers. In balancer shaft systems for four-cylinder engines comprising two shafts rotating at twice the engine speed, the frictional loss associated with plain bearings can be roughly halved by using rolling bearings instead (Figure 2).



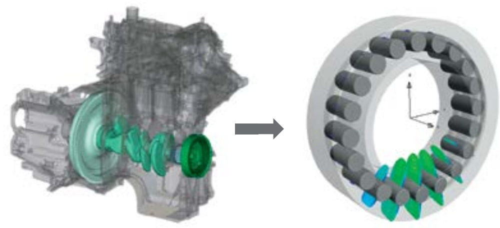
2 **Savings with balancer shaft systems with rolling bearings**

An important factor for the dynamic response of the engine are the potential savings associated with the turbocharger (Figure 3). During a cold start, rolling bearings in the turbocharger can reduce frictional loss by up to 80 %, thereby ensuring good engine response. The turbine in the turbocharger also spins up faster under acceleration so that more boost pressure is realized in a shorter period. Testing has shown that this setup not only improves efficiency by around 2.5 %, but also allows the engine to build up torque and produce its power output much faster. As an alternative to boosting power and torque, the improved dynamic response of the turbocharger can be used to reduce fuel-rich mixtures to minimize NO_x and soot emissions during dynamic engine load changes.

The next level when it comes to replacing plain bearings with rolling bearings in the engine could be the crankshaft main bearing [1]. Schaeffler investigated the use of roller bearings together with Ford on a production engine. A fuel economy improvement of 1 % was confirmed simply by using a rolling bearing at the first main bearing position located the furthest away from the flywheel. The benefit of this approach is that it is easy to fit a rolling bearing right here. Figure 4. The approach is particularly well suited to combustion engines with a belt-driven starter-alternator as this is



3 **Benefits of a turbocharger with rolling bearings**



4 **Three-cylinder engine with a rolling bearing at the first main bearing position**

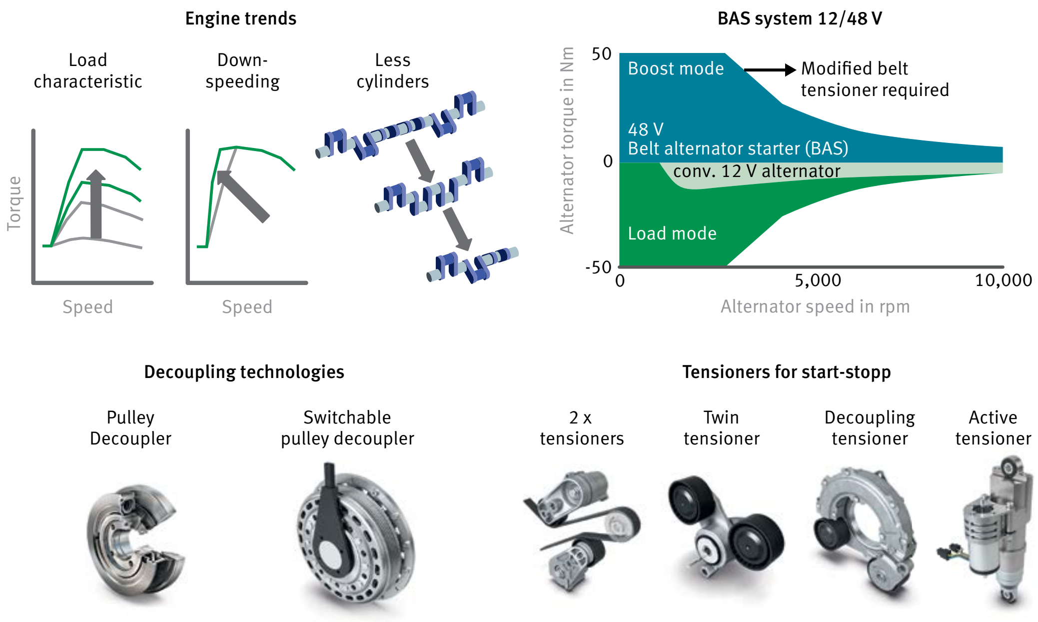
where the first crankshaft main bearing is subjected to very high loads, some of which are caused by the belt-driven restarting sequences of the combustion engine.

Regenerative braking via the belt-driven starter-alternator

Using a belt-driven starter-alternator is an appealing additional step to minimize energy losses in the overall system and in so doing reduce consumption and CO_2 emissions. A P0 hybrid configuration can mostly be conveniently implemented in the belt drive of new and existing vehicle and engine architectures (Figure 5).

In overrun mode, the belt-driven starter-alternator can effectively harness the kinetic energy of the vehicle and channel it into the battery via the on-board electrical system. This energy can then be used to restart the engine after a start/stop sequence or when “sailing” (engine is mechanically decoupled from drivetrain). The energy can also be used to provide extra boost during acceleration – either directly or by way of electrical charging systems. Start/stop and sailing concepts necessitate frequent engine starts and stops. Experience has shown that consumers are more likely to accept such operating strategies when they go largely unnoticed by the driver. Compared to conventional pinion starters,

5 **Accessory drive with P0 hybridization**



6 **Tensioning systems and decouplers for P0 systems**

belt-driven starter-alternators allow the combustion engine to be started faster and more smoothly. This, in turn, enables high availability of this function such that the combustion engine equipped with a belt-driven starter-alternator can generally be stopped and started more frequently. The P0 hybrid with belt-driven starter-alternator requires a bi-directional tensioning system that enables the transition between the tight and slack states during periods of generation/recuperation and starting/boosting. For this purpose, Schaeffler has developed different solutions that range from systems with two single tensioners to electrically actuated concepts (Figure 6).

Isolating the torsional vibration transmitted by the crankshaft is also required to reliably transmit the belt forces with moderate pretension loads. Here, too, several solutions exist, whereby the crankshaft decoupler provides the best isolation for the P0 system. To further improve NVH characteristics and, thus, comfort levels for vehicle passengers, additional engine-specific measures should be implemented that help the combustion engine start and stop smoothly by actively reducing compression. With the electrically actuated camshaft phaser and the eRocker – an electromechanically actuated roller finger follower – Schaeffler offers systems that are specifically designed to meet these requirements.

Variable valvetrain

Schaeffler has for years been supplying automotive manufacturers with a wide variety of variable valvetrain components ranging from camshaft phasers and two-stage valvetrains in various different configurations, to the UniAir continuously

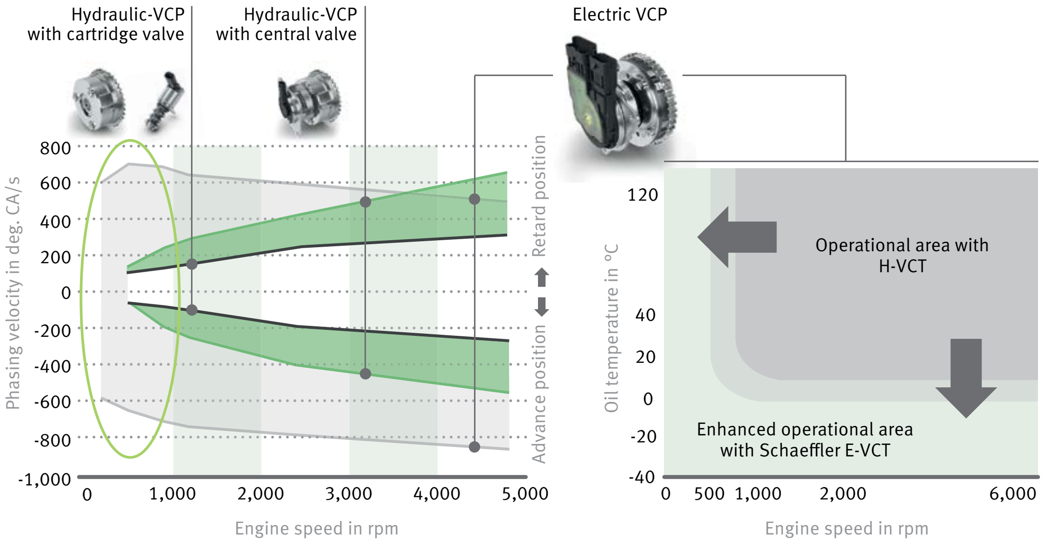
variable valvetrain, which offers the highest level of engineering freedom when it comes to actively controlling the air path (Figure 7).

Camshaft phasing systems allow the valve timing to be adapted to the respective operating conditions of the engine. They are typically used on the intake and exhaust side. Schaeffler develops and supplies both hydraulic and electrically actuated systems. The benefit of the hydraulic concept is its robust design and low cost. The system does, however, require a minimum oil pressure level in the engine. By contrast, the electric camshaft phaser can also adjust the valve timing when the engine is stopped as it does not rely on oil pressure (Figure 8) (left). During a start/stop sequence or when the vehicle is operated in sailing mode, the electric camshaft phaser can prepare the combustion engine for a restart in advance by adjusting the timing such that the engine is decompressed when restarting. The starting system therefore does not need to produce as much torque and can engage more smoothly. This also reduces the loads on the main bearings, thereby reducing friction and wear.

Compared to the hydraulic camshaft phaser, its electric equivalent also expands the temperature range that can be used for phasing. By realizing very short phasing times (Figure 8, right) and precise angular adjustment, the electric camshaft phaser also makes it possible to save fuel during dynamic engine operation. Replacing a hydraulic camshaft phaser with an electric one typically allows to reduce the output of the oil pump. This, in turn, further reduces fuel consumption. Simulations on a 1.0-liter gasoline engine with direct injection and turbocharging confirm a reduction in fuel consumption of up to 2 % as a result of these effects.

We have recently seen a resurgence of engine concepts in development whose compression ratio can be variably adjusted. In this field, Schaeffler

7 **Variable valvetrain – Schaeffler's portfolio**



8 Electric camshaft phaser: Expansion of operating ranges (left) and higher phasing dynamics (right)

is working on electromechanical actuators for variable compression ratio (VCR) systems. As varying the compression ratio has a direct impact on the combustion in the cylinders of the engine and therefore also affects emissions and fuel consumption, the actuator system must meet highest requirements when it comes to adjustment speed, accuracy and, above all, reliability. In developing the VCR system into a mature, production-ready solution, Schaeffler is able to draw on the extensive experience it has gained with electromechanical camshaft phasers as the technology incorporated in both actuator concepts is very similar.

Cylinder-individual valve control with switchable roller type finger followers and the UniAir system

While camshaft phasers adjust the valve lift curve on the whole, switchable finger followers enable toggling between two valve lift curves – e.g. for combustion concepts with early intake valve closing (EIVC) or late intake valve closing (LIVC) – that de-throttle the engine and thereby reduce fuel consumption. Valve or cylinder deactivation can also be realized by switching between full lift and zero lift. The actuation of the switchable finger followers on current systems is effected by using the engine oil pressure and a shift valve in the adapted oil circuit of the cylinder head. Easier integration is made possible by the new electromechanically switched eRocker roller finger follower-system. It is actuated mechanically via a centrally located electric actuator.

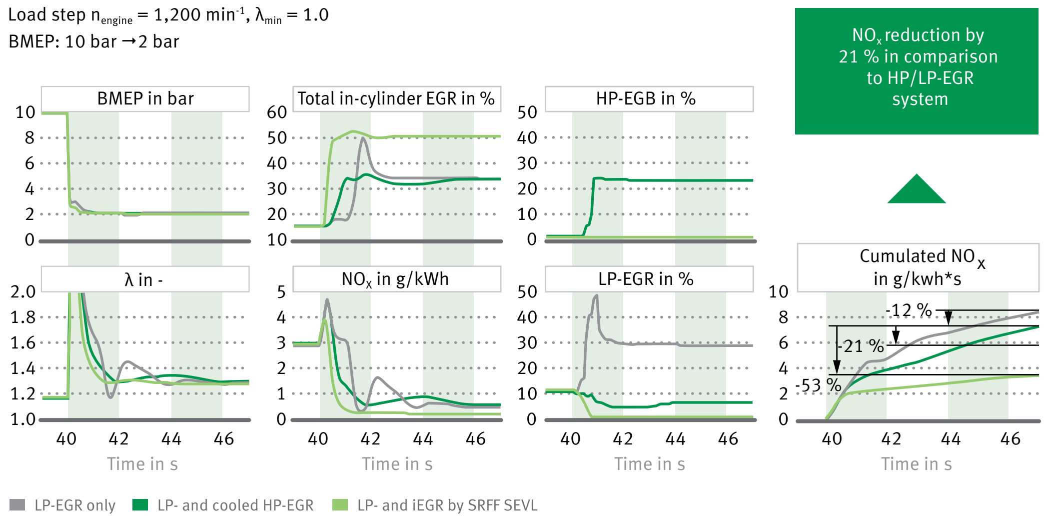
While variable valvetrain concepts in the form of switchable roller finger followers used to be the domain of the gasoline engine, recent diesel engine applications are also making increased use of this setup. Switchable roller finger followers facilitate valve lift strategies with a second, auxiliary opening lift of the exhaust valves. They positively impact the diesel engine in several ways: First, they reduce the warm-up times of the exhaust gas aftertreatment components as hot exhaust gas can be drawn back into the cylinder during a cold start (internal EGR). Second, unlike an external configuration, the faster internal EGR offers faster control during transient operation. When the engine load changes instantaneously, for example, the internal EGR rate ramps up more quickly, which reduces peak NO_x emissions by 21 % during simulation exercises (Figure 9).

Additionally, switchable finger followers on the intake side also make it possible to further reduce CO_2 and exhaust emissions by e.g. minimizing the effective compression ratio via EIVC or LIVC, thereby lowering peak pressure levels and combustion temperatures under high-load conditions with a corresponding reduction in engine-out NO_x emissions.

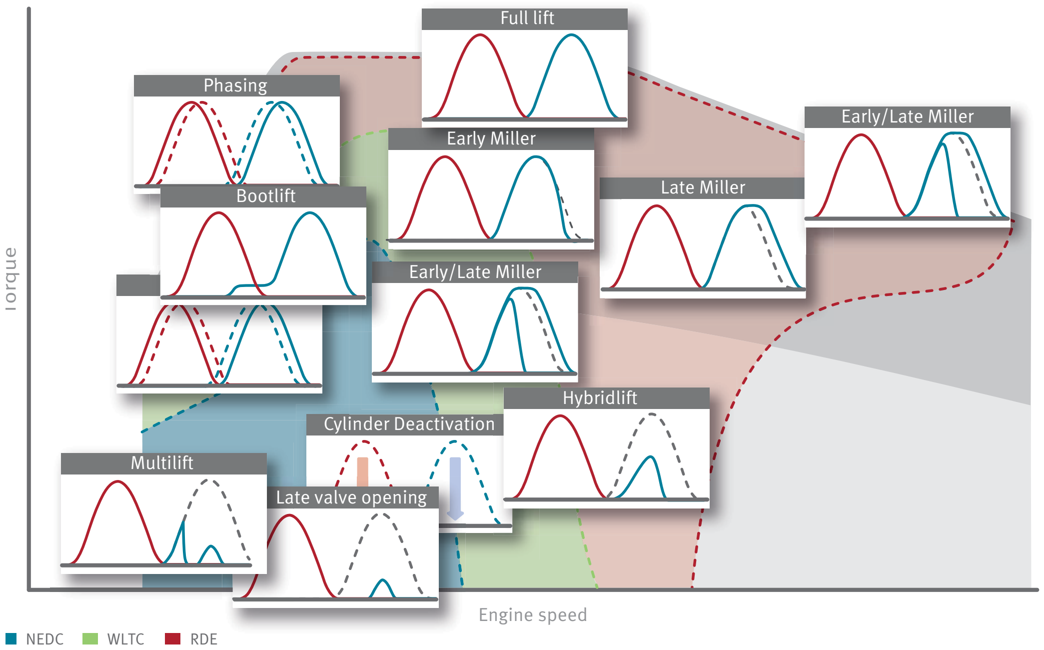
Out of all the current variable valve trains available for series production gasoline engines, the electrohydraulic UniAir system is the most versatile. The fully variable system allows the timing and valve lift to be individually adapted to the respective engine operating conditions so that EIVC or LIVC concepts can be easily implemented (Figure 10). This, in turn, gives engineers more freedom to optimize the combustion process, which can be leveraged to push back the knock limit and avoid enrichment at high loads. Both factors allow to extend the Lambda-1 operation across the entire engine map and positively contribute to meeting the requirements of the new RDE legislation.

Rolling cylinder deactivation improves efficiency of small engines

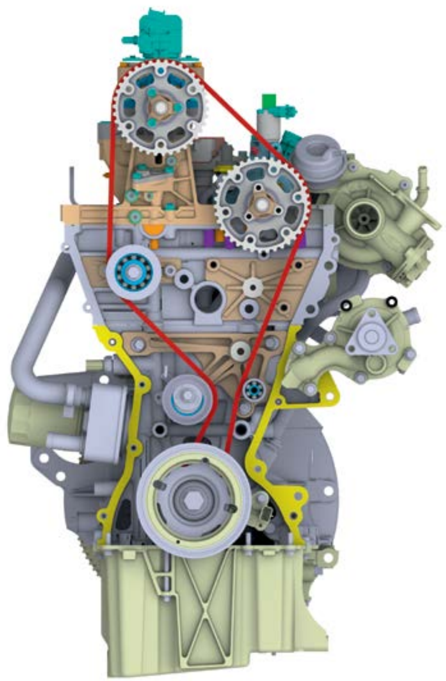
An example of efficiency-improvement with variable valvetrain systems is cylinder deactivation. In the past, this concept was largely reserved for engines with four, six or eight cylinders. In the future, three-cylinder engines will also profit from the CO_2 benefits enabled by this function. Ford, for example, recently launched a production solution that integrates switchable roller finger followers from Schaeffler [2, 3].



9 Switchable roller finger followers considerably reduce engine-out NO_x emissions on a diesel engine



10 UniAir variable valvetrain for all requirements



11 Prototype engine used to investigate the rolling cylinder deactivation system

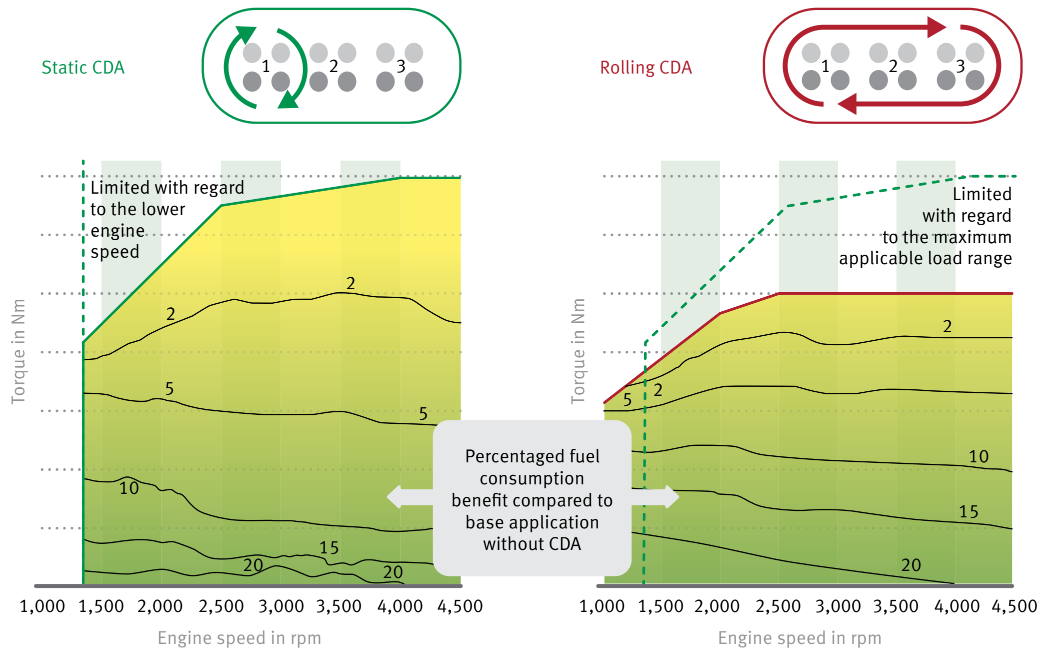
In general, the potential for reducing CO_2 emissions by cylinder deactivation largely depends on the engine load and the usable engine speed range for the deactivation. At low speeds, the possibility of deactivating a cylinder is typically limited by the NVH characteristics of the drivetrain. This is where an optimized dual-mass flywheel (DMF) can be used to isolate torsional vibrations in conjunction with a matching centrifugal pendulum (CP) or, on engines with lower torque outputs, a clutch disc with CP. As a cost-effective alternative to a DMF, in certain applications the clutch disc with CP also effectively eliminates the unavoidable torsional vibrations in the drivetrain that are negatively perceived by vehicle occupants.

At the 2014 Schaeffler Symposium, the concept of rolling cylinder deactivation, or RCD, was introduced on a three-cylinder engine. As the name implies, this concept involves deactivating one cylinder after another, in succession, from one engine cycle to the next, whereby each cylinder fires at every fourth rotation of the crankshaft. Schaeffler has since verified the concept during practical testing. In the process, static cylinder deactivation, which involves deactivating the same cylinder every time, was compared with rolling cylinder deactivation. The prototype engine used (Figure 11) was fitted with the UniAir fully variable valvetrain on the intake side and switchable valvetrain elements on the exhaust side. Integrating the UniAir system made it possible to further optimize the thermodynamics of the engine, e.g. by adjusting the residual fuel-air mixtures in RCD mode and reducing charge cycle losses created by EIVC and LIVC valve timing.

The two types of cylinder deactivation reveal different functional limitations: While the limiting factor for the static cylinder deactivation system is low engine speed operation and the associated NVH excitation of the drivetrain, rolling cylinder deactivation is restricted by the maximum attainable load level as in this engine application, only 1.5 cylinders are being used, whereas two cylinders are used with the static setup (Figure 12). In practice, the optimal concept is chosen considering the weight of the vehicle and the size of its engine as this determines the most frequently used load points on the engine map and, ultimately, the potential for saving fuel by deactivating cylinders.

The ideal combustion engine in a full hybrid system

The combustion engine will continue to be the dominating power source in vehicles for many

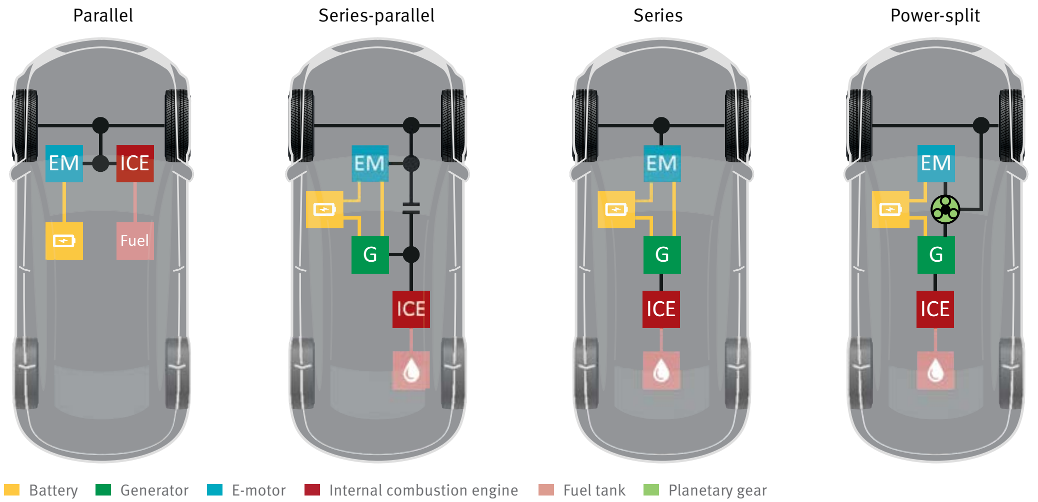


12 Comparison of operating ranges for static and rolling cylinder deactivation

years to come. This does not conflict with the ongoing trend toward electrification as hybrid systems also integrate a combustion engine in their powertrain in addition to the electric motor. When it comes to full hybrid drive systems in particular, however, the question is raised as to whether it would be possible to simplify the combustion engine as high dynamic driving requirements are largely fulfilled by the electric portion of the drivetrain. To analyze this, two current hybrid concepts and the resulting technical requirements for the best suited combustion engine will be examined: the P2 parallel hybrid and the power split – a continuous power splitting serial parallel hybrid (Figure 13).

The combustion engines used in a power split serial parallel hybrid are typically naturally-aspirated engines with relatively large displacement and a low specific output. As they are only operated at the lower and medium speed ranges on the engine map, they have a higher compression ratio and are specifically optimized for low friction. They typically are designed as a long-stroke engine and may in future possibly integrate rolling bearings on the crankshaft. The engines are operated in the EIVC or LIVC mode, i.e. their high geometric compression is effectively adapted via variable valve timing depending on operating conditions. This active adjustment can be carried out using an intake camshaft phaser. A cylinder head with separate camshafts for the intake and exhaust valves is required for this configuration, in order to use phasing on the intake side independently of the exhaust side. In addition to the increased design effort, the friction level in the system also rises. An alternative way to achieving a valve timing variability that is also less design effort and more efficient is that of cylinder-selective variability such as the UniAir system, for which a single camshaft suffices for the intake and exhaust timing sequences.

In pursuit of a cost-saving modular strategy, these engines are also frequently offered in certain markets in parallel as conventional, non-hybridized versions that then feature a lower compression ratio and a higher specific output. Due to the compact overall package of engine and planetary transmission these drivetrains are well suited for transversal installation in the vehicle.



13 **Comparison of different hybrid architectures**

The rated outputs of the combustion engine and the electric motor as found in a power split hybrid must be very well coordinated due to their mechanical connection via a planetary gearset. In this context, the electric motor fitted must have its output power well matched to the engine to ensure that it can balance or “hold” the torque of the combustion engine. This approach practically limits the total system output of the drivetrain, which is why power-split hybrid drives are typically found in compact and mid-sized passenger cars as well as entry-level SUVs.

Unlike the serial parallel hybrid, the combustion engine and electric motor in a parallel hybrid can be scaled in size independently of each other. This platform also makes the system suitable for use in heavier passenger cars and full-sized SUVs. One benefit of the parallel hybrid concept is the high degree of flexibility as engine families can be used in conventional and hybridized drivetrains with just a few minor modifications. This allows to optimize production capacities, particularly when introducing electrified powertrains in select markets. In order to maximize overall system efficiency, it is advisable to implement all combustion engine-specific measures that realize clean combustion and highest efficiency such as variable valve trains, turbocharging, direct injection and, in the future, even variable compression. Due to the design proximity to the engines for conventional drivetrains, it makes sense to utilize modular solutions and platform concepts.
The outcome of the comparison shows that there cannot be a single combustion engine concept that is particularly well suited to hybrid applications, since the requirements the engine must meet in the different hybrid configurations are too varied. The combustion engine used in a specific application must be designed for or adapted to the target hybrid concept to ensure that it can provide the best possible results in the overall drivetrain system.

Summary and outlook

Ever stricter emission limits and increased CO_2 requirements can only be met by further developing and advancing the drivetrain. By integrating low-friction components and systems such as belt-driven starter-alternators for recuperation purposes, Schaeffler makes it possible to minimize energy loss and improve the efficiency of the drivetrain. Variable valve train systems ranging from a basic camshaft phaser all the way to the fully variable UniAir system are tools for optimizing the combustion process and reducing fuel consumption and engine-out harmful emissions. This applies not only to conventional combustion engine drivetrains, but also to hybrid systems. To ensure that the latter can achieve maximum efficiency, the combustion engine must be designed and tuned to harmonize with the overall drivetrain system. Thus, the optimization of the combustion engine and its subsystems will continue to be at the heart of development strategies that pursue high efficiency and low emissions.

Alternative fuels offer an additional approach that goes beyond engine design for reducing CO_2 emissions in the short and mid-term. Today, CNG is already available as a gaseous fossil fuel that produces around 25 % less CO_2 than conventional gasoline. In the mid and long-term, it will be possible to synthesize methane gas in a PtG (power to gas) process. Diesel engines will not be left behind either as there is currently discussion on offering synthetic fuels based on a PtL (power to liquid) process; the first pilot plants are already starting to produce such fuel. If the primary energy required during this process also comes from renewable sources such as wind power or photovoltaic systems, these fuels can be regarded as CO_2 neutral, since the amount of CO_2 emitted when they are combusted is roughly equivalent to that which is chemically bound during production.

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2

Airpath Flexibility

Unlocking the Full Potential of the UniAir System

David Kehr
Daniel Wolf

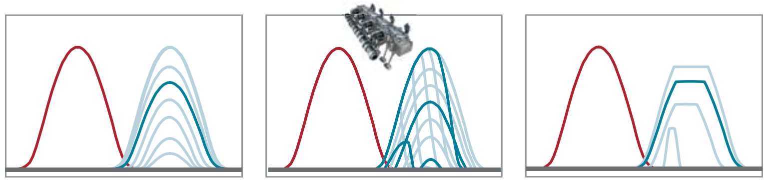
Introduction

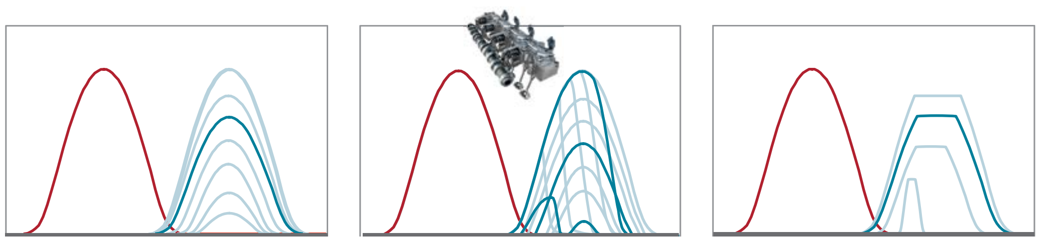
Ever stricter global emissions control legislation and lower target CO_2 values not only characterize the development of drivetrains offered by automotive manufacturers currently, but will also be a dominant factor in the future. To ensure that all mobility needs are catered to, the combustion engine will continue to play a major role in the drivetrain mix in both the short and medium-term and in so doing make an important contribution toward achieving CO_2 emissions targets.

To this end, legislation in Europe underwent a fundamental paradigm shift in the last few years. Previously more stringent emissions laws were implemented by mandating lower limit values that were cross-checked in the laboratory by running a test cycle that had not been changed years. This so-called NEDC (New European Driving Cycle) only covered a relatively narrow spectrum of engine operation – a comprehensive set of engine speeds and engine loads was never measured. This has changed with the advent of the new Worldwide Harmonized Light Vehicles Test Cycle (WLTC), which significantly broadens the measurement testing range. Adding to this is the even more intriguing RDE (Real Driving Emissions) test, which involves measuring pollutant emissions in real, actual driving conditions on public roads.

As such, emissions certification covers a much broader range of engine speeds and loads. In addition, average fleet emissions of CO_2 must be reduced to 95 g/km by 2021 and by a further 30 percent before 2030. The changed constraints have led to new requirements for future combustion engine concepts.

- To minimize charge cycle losses, de-throttling continues to gain in importance.
- In this context, active emissions regulation controls must be able to quickly respond to changes in engine speed and load during transient operation.
- In light of RDE requirements, gasoline engines must be operated stoichiometrically across the entire engine performance map (λ=1); running a rich mixture under full-load conditions in order to protect componentry will no longer be possible.





1 **Current fully variable valve train systems**

Based on this requirements profile, fully variable valve train systems quickly prove to be the ideal technology for the job. As at each operating point, the correct air mass in the combustion chamber is actively regulated in the intake ports of the cylinders by way of valve port management, the throttle valves typically found in the induction tract of conventional gasoline engines can be fully opened or even be omitted altogether.

Fully variable valve trains can generally be subdivided into mechanical, electrohydraulic and electromagnetic systems based on their actuator control (see Figure 1).

Schaeffler introduced the UniAir technology as a production-ready solution back in 2009. The same year saw its first application in the Fire MultiAir engine from Fiat as used in the Alfa Romeo Mito. Schaeffler supplies the UniAir as a modular system comprising:

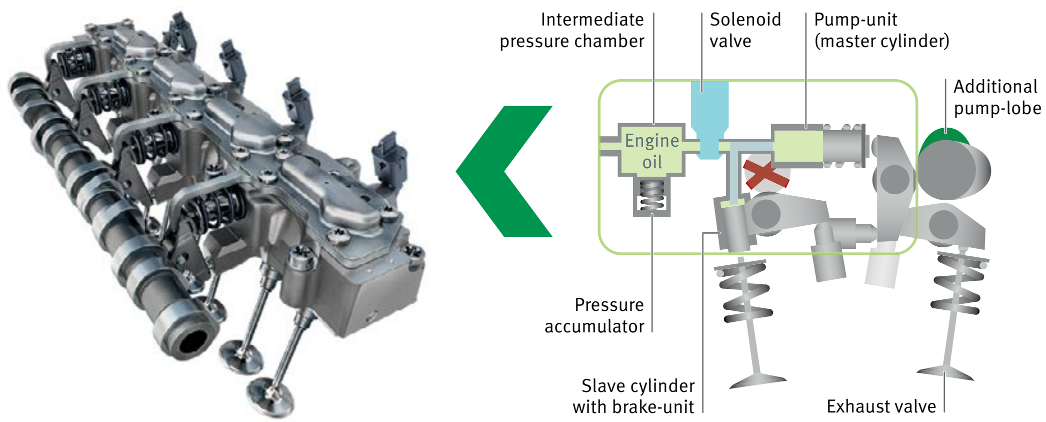
- Electrohydraulic actuator elements
- Control unit software module for engine management
- Application-specific calibration data set

Since the start of production, Schaeffler has manufactured more than 3 million units of the UniAir system as part of a global production network. Used in different engines, the system has continually been upgraded and improved with new functions. By ensuring compatibility of the UniAir software module with the engine management systems of established manufacturers such as Continental, Magneti Marelli and Bosch, Schaeffler realizes a high degree of flexibility when it comes to selecting the components of an engine.

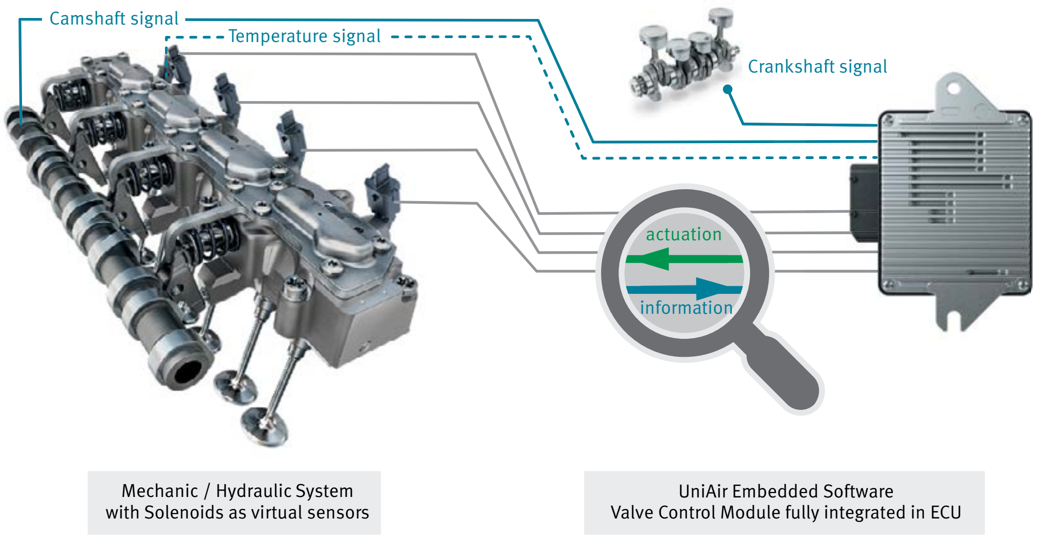
The latest generation of the UniAir system not only makes it possible to reduce charge cycle losses, but also optimize combustion and achieve transient torque control via the air path.

System design

Figure 2 illustrates the design of the electrohydraulic UniAir system. Instead of using rigid connecting elements between the camshaft and en-



2 **Design of the electrohydraulic UniAir system**



3 **Interlinking of actuator technology and engine control unit**

gine valve, a defined oil volume encapsulated in the high-pressure chamber transfers the cam (lobe) contour to the engine valve. A pump driven by the camshaft via a finger follower builds pressure in the high-pressure chamber.

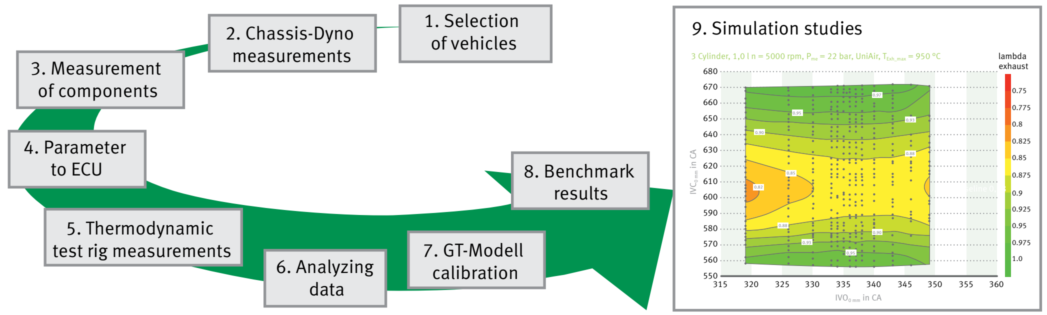
When the solenoid valve is closed, the oil pressure acts on the engine valve by way of a piston, and the valve opens. As soon as oil flows through the open hydraulic valve out of the high-pressure chamber, the force level acting on the engine valve against the valve spring assembly drops, and the engine valve closes. When the solenoid valve opens at cam lift, the oil flows out of the high-pressure chamber into a buffer known as the intermediate pressure chamber. When the high-pressure chamber is subsequently filled in the base circle phase, the oil returns. This minimizes the amount of energy required to establish oil pressure in the system.

The control system, in addition to the electro-hydraulic actuator technology, represents a key module of the UniAir system from Schaeffler. The control algorithms developed by Schaeffler are provided to automotive manufacturers as a software module, which is then integrated as a control module in the engine control unit (see Figure 3).

The electronics utilize sensor signals quantifying the position of the camshaft and crankshaft as input factors. This information is augmented by data on the oil temperature in the UniAir module. The requirements defined by the engine management system are then implemented, and the solenoid valves of the UniAir system are energized as needed. The on and off times of the solenoid valves are used by the control system to monitor the cylinder activation of the engine valves, and the system compensates for tolerance and age-specific changes in system response as well as balances out the effects of fluctuating oil quality.

Development and application process

A few years ago, Schaeffler embarked on a collaborative project with a development service provid-

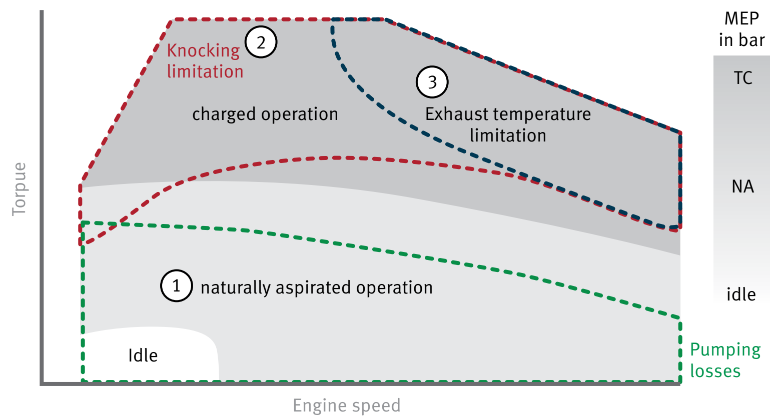


4 **Application process**

er to leverage the benefits of the UniAir system in practical operation. The objectives of the venture were to define and establish a solid standard development method for integrating UniAir in existing engine-vehicle combinations as well as conduct follow-up investigations on vehicles that have been modified in-house by Schaeffler. Figure 4 provides an overview of the current UniAir application process. After a suitable test vehicle is selected, all vehicle parameters that are critical to the investigative work are quantified on the roller dynamometer. The engine is then analyzed in detail on the thermodynamic engine dynamometer. The result serves as the benchmark for the UniAir system application. To this end, actual engine data is integrated in a GT-Power model and UniAir-specific adaptations are made. It is on this basis that the reduced fuel consumption is plotted across the engine map during simulations and in some cases verified in dynamometer test runs. This proven application process also allows Schaeffler to help customers looking to integrate UniAir into their engine system by offering additional development services for adapting the engine components and software parameters to the fully variable valve train system.

Benefits of UniAir at steady-state operating points

Figure 5 plots a load curve on the performance map of a gasoline engine with forced induction. The areas, or ranges in which UniAir can bring about considerable benefits under different loads in steady-state engine operation appear as dotted lines:



5 **Load curve on the performance map of a gasoline engine with forced induction**

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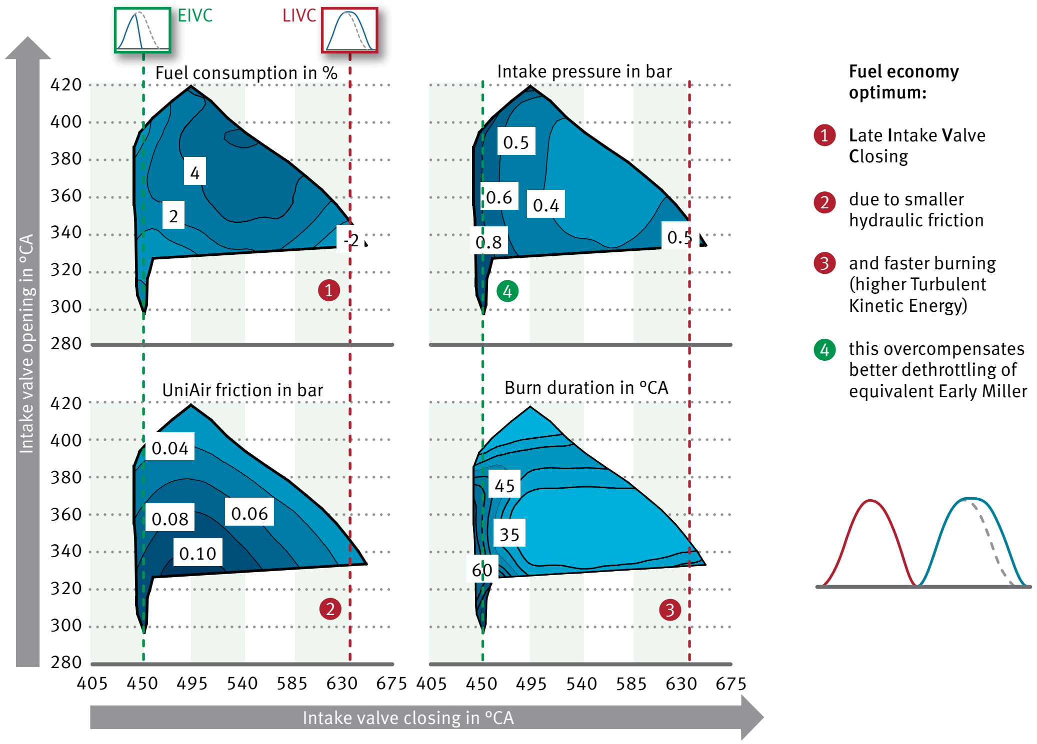
- Under low-load conditions, throttle losses can be reduced (point 1 in Figure 5).
- By leveraging strategies with early intake valve closing (EIVC) or late intake valve closing (LIVC), the knock limit can be shifted higher in the load range without having to take worsening combustion into account (point 2 in Figure 5).
- When operated under load conditions, EIVC or LIVC, together with adapted forced induction, elimination of wide-open throttle enrichment is possible. Subsequently, the engine can be operated throughout the entire performance map with λ= 1, which coincides with the optimal range for the three-way catalytic converter (point 3 in Figure 5).

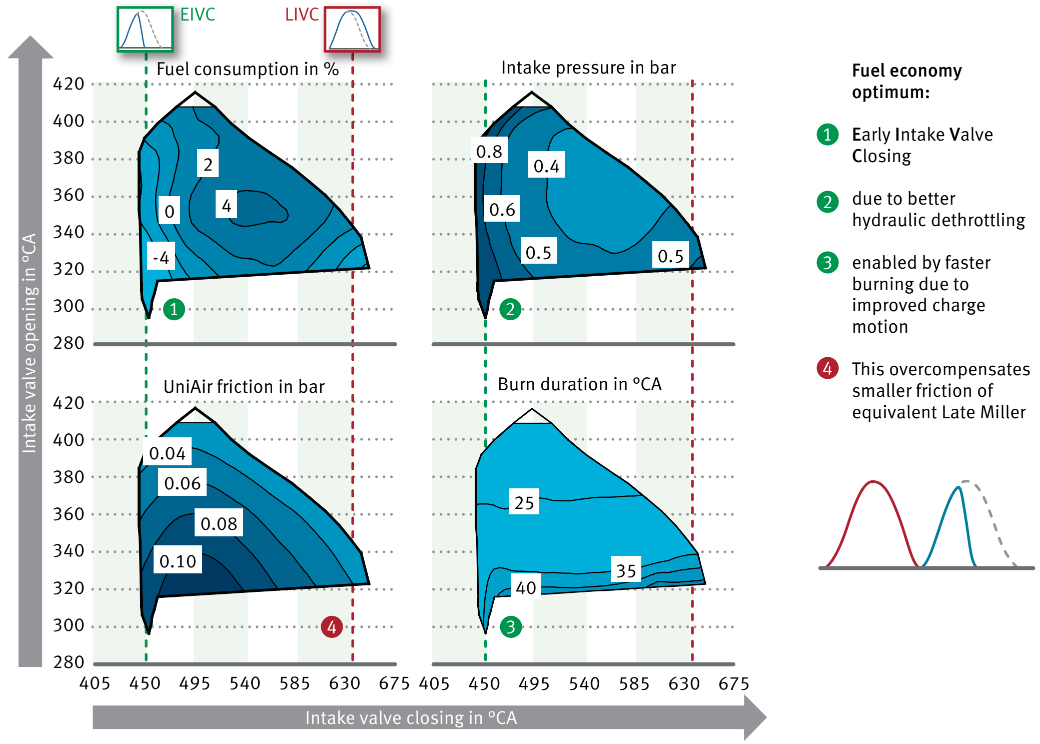
Practicing the development method outlined, Schaeffler carried out investigative exercises in steady-state engine operation for the above three load scenarios utilizing EIVC and LIVC technology. For the performance map range involving low engine loads, the characteristic operating point of 2,000 rpm and 2 bar of mean boost pressure was examined. Figure 6 reveals the results of the simulation for specific fuel consumption, friction intake manifold pressure and combustion duration.

The green line represents the EIVC and the orange line the LIVC strategy. As is indicated by the plotted consumption curve in the upper left corner of Figure 6, LIVC reduces fuel consumption by around 2 % compared with the production standard, while EIVC generally increases fuel consumption. This is particularly remarkable given the fact that engine de-throttling improves with early intake valve closing as opposed to late closing, as the intake manifold pressure in the upper right corner of Figure 6 indicates. The lower right quadrant of Figure 6 provides an explanation: The combustion duration with EIVC is much longer and therefore less fuel efficient than with LIVC. Early intake valve closing also leads to higher friction levels (lower left quadrant of Figure 6). These two factors overcompensate the benefits of reduced throttle losses with EIVC and are responsible for allowing the LIVC strategy to reduce fuel consumption. The situation changes if the intake valves have masking, or additional covers at the valve seat. Figure 7 shows the results under otherwise identical constraints as for the previous simulation.

Now the EIVC strategy considerably reduces fuel consumption as the masking leads to a much shorter combustion duration and the improved de-throttling response really comes into its own thanks to the early intake valve closing. Despite the still higher friction levels of early as opposed to late intake valve closing, EIVC is the better choice. Generally speaking, the effect of the charge motion is key to combustion efficiency and cannot be overlooked during the operative application. Optimal results can be achieved if the overall engine is tuned to the new operating conditions when the UniAir system is integrated. This includes the engine valve springs, which can generally be lowered with the UniAir system and lead to a further reduction in fuel consumption, particularly in the low load range. Schaeffler currently is investigating the potential of this design measure in simulation exercises as well as on the test stand.



6 **Results of engine tests at an operating point of 2,000 rpm with 2 bar of mean boost pressure, without masking**



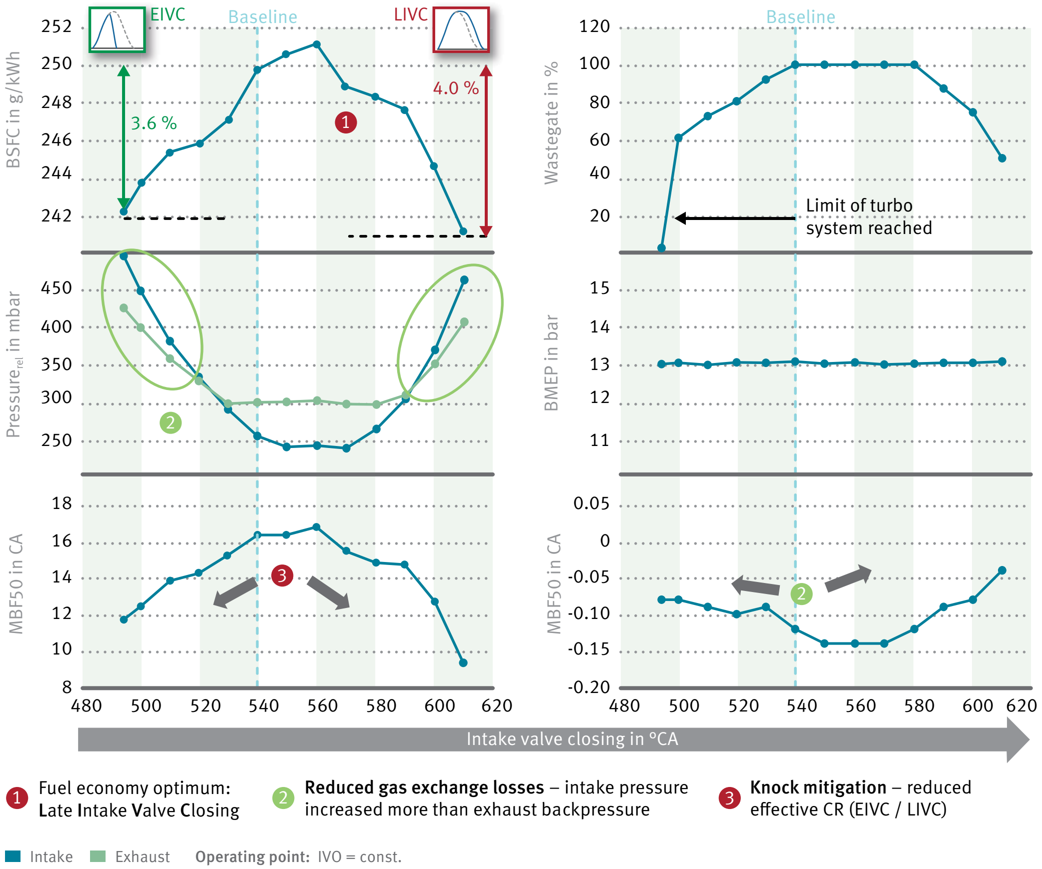
7 **Results of engine tests at an operating point of 2,000 rpm with 2 bar of mean boost pressure, with masking**

For tests conducted at the knock limit of the engine, a low-end torque point of 2,000 rpm and 13 bar under high-load and low operating speed conditions was chosen (point 2 in Figure 5). In Figure 8, the blue line represents the intake valve closing of the standard engine, while the diagram values to the left and right result from early and late intake valve closing, respectively.

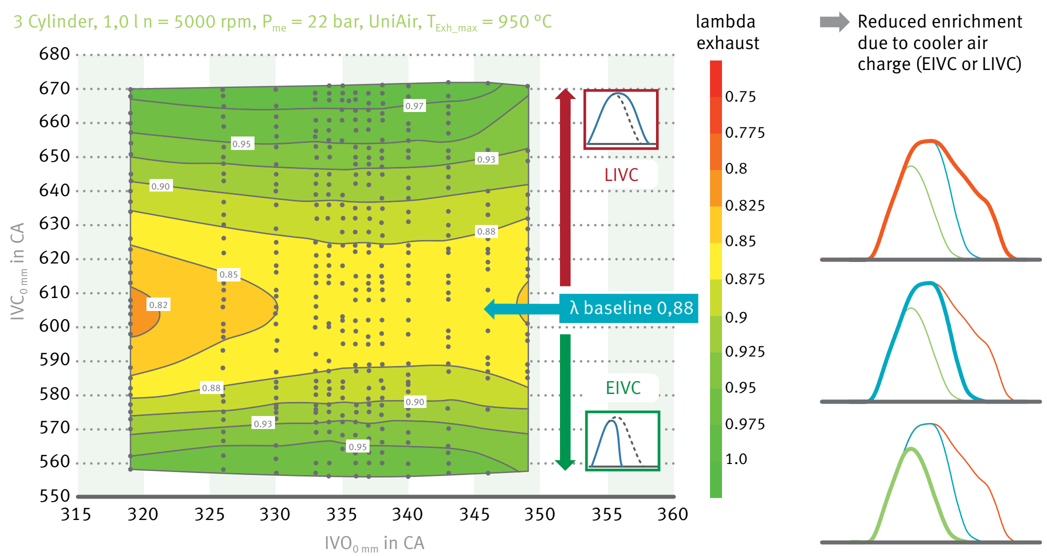
Both approaches reduce the effective compression, thereby shifting the knock limit of the engine further upwards. Fuel consumption at the operating point in question can be considerably reduced with early as well as late intake valve closing. In the process, late closing offers slightly better performance (with up to 4 % less fuel consumed) than early closing (3.6 % at the optimum point) (top left diagram in Figure 8). One reason for the fuel savings is the better center of combustion as shown in the lower left quadrant of Figure 8: For the standard engine, the 50 % conversion point is at an unfavorable 16° Crank Angle (CA) after TDC. Utilizing EIVC and LIVC, this point approaches the theoretical optimum of 8° CA at approximately 12° or 9.5° CA, respectively. The center left diagram in Figure 8 plots the differential pressure in the intake and exhaust tract. A positive scavenging gradient can be observed in the gas-exchange loop during early and late intake valve closing. Even when pumping losses are factored in as shown in the lower right quadrant of Figure 8, EIVC and LIVC yield better performance over the standard engine. The position of the wastegate flap as depicted in the upper right quadrant of Figure 8 plays a key role at different engine operating points. While this flap is fully open on the standard engine, with the EIVC setup, the incoming air mass is cut off by the intake valve control system, whereas with the LIVC configuration, this mass is partially displaced so that the wastegate flap does not need to be opened all the way. This also means that the turbocharger in the overall engine system must be tuned to the specific operating requirements of the EIVC or LIVC strategy, whereby the engine must be run at higher boost pressures to ensure that the necessary air mass enters into the cylinder.

An operating point of 5,000 rpm and 22 bar of mean boost pressure were chosen for testing under full-load conditions (point 3 in Figure 5). In many cases, the maximum exhaust temperature under high-load conditions represents a limiting factor because engine components must be protected when exposed to extreme thermal loads. For the standard engine analyzed, an exhaust temperature limit of 950 °C to protect components was considered. Under a full engine load, the air-fuel blend must be considerably enriched to cool the combustion chamber and meet the 950 °C exhaust temperature limit. As Figure 9 shows, the standard engine is operated with λ= 0.88 at the operating point in question.

In Figure 9, the intake valve closing is superimposed onto the ordinate; LIVC displaces the value upward, while EIVC displaces it downward. Both approaches can be used to configure a quasi-stoichiometric combustion recipe while taking the maximum temperature into account. Figure 10 shows the resulting effect on specific fuel consumption.

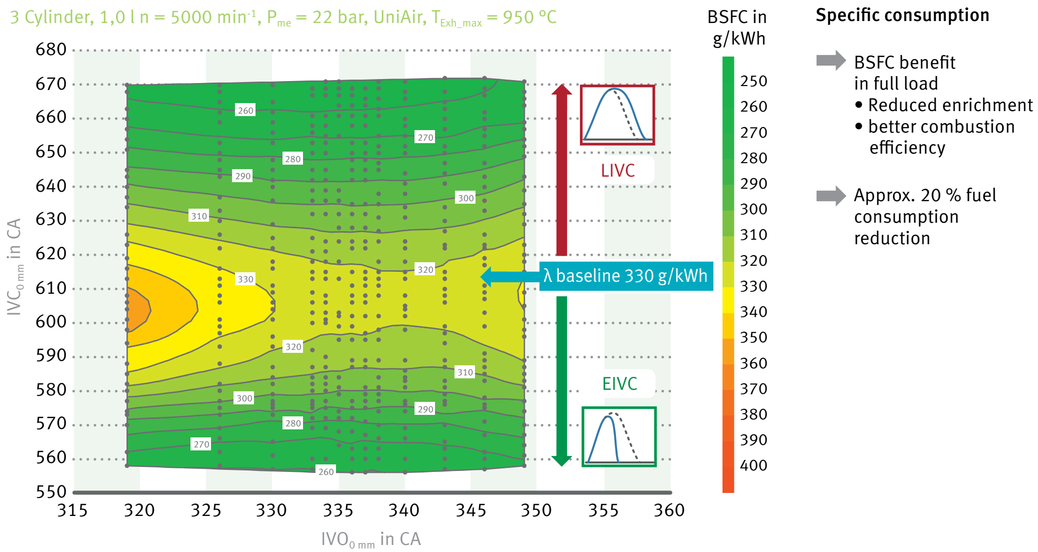


8 **Results of engine tests at an operating point of 2,000 rpm with 13 bar of mean boost pressure**

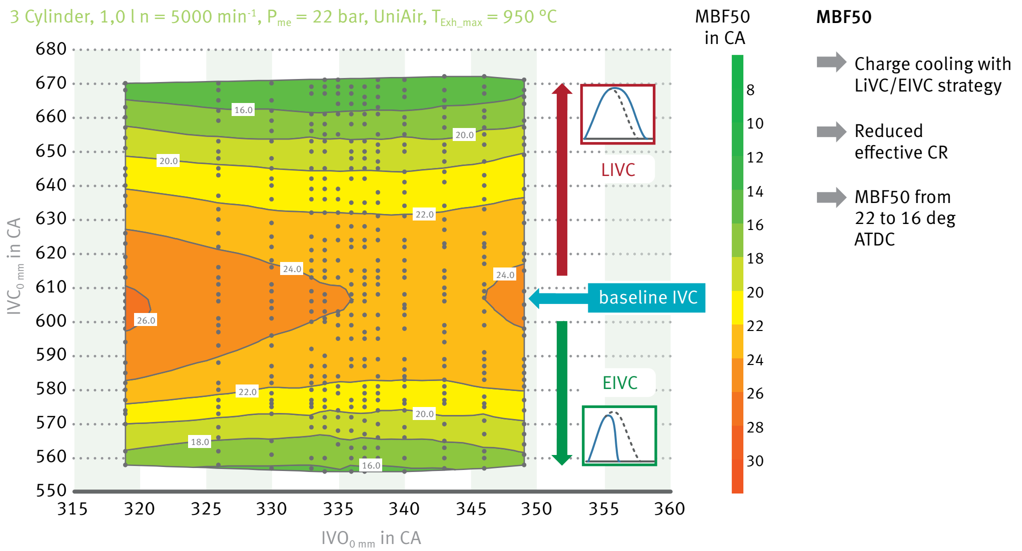


9 **Air-fuel blend at an operating point of 5,000 rpm with 22 bar of mean boost pressure**

With a starting point of 330 g/kWh, fuel consumption can be reduced by 20 % to 260 g/kWh with EIVC or LIVC. In addition to the quasi-stoichiometric combustion, the better center of combustion during early and late intake valve closing also contributes to the fuel savings achieved. As a result of the fuel enrichment-induced cooling required, the 50 % conversion point of the standard



10 **Specific fuel consumption at an operating point of 5,000 rpm with 22 bar of mean boost pressure**



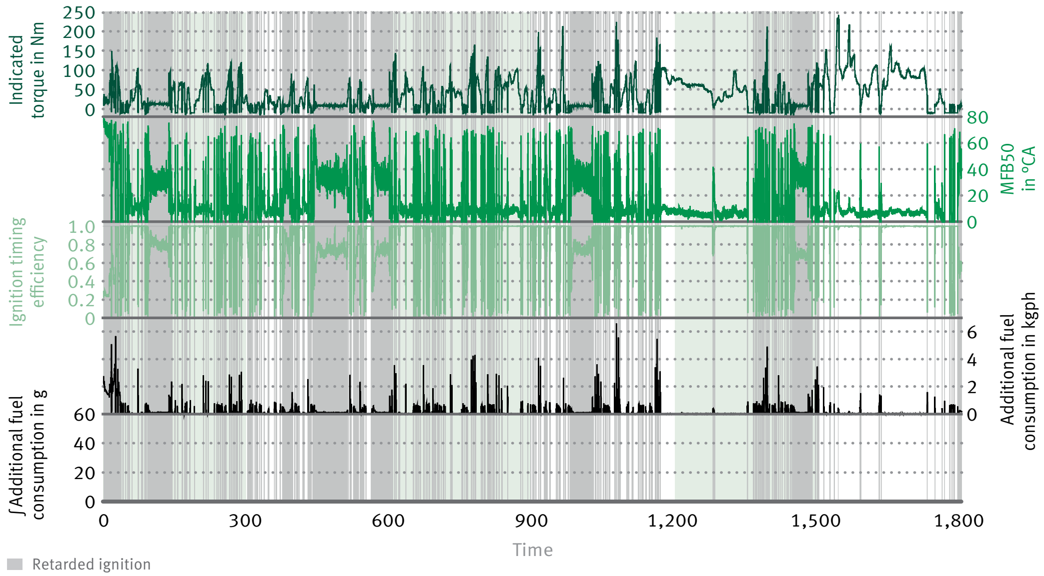
11 **50 % conversion point at an operating point of 5,000 rpm with 22 bar of mean boost pressure**

engine shifts to an unfavorable 22° CA after TDC. Active control of the effective compression enabled by the UniAir system ensures lower compression temperatures such that the engine remains further from the knock limit. As such, the ignition point and center of combustion can be advanced. Instead of 22° CA after TDC, the 50 % conversion point moves to 16° CA after TDC (Figure 11). As is the case with the low-end torque point observed, here, too, the turbocharging system must be adapted for the system application so that the increased air mass required by the EIVC or LIVC setup can be accommodated.

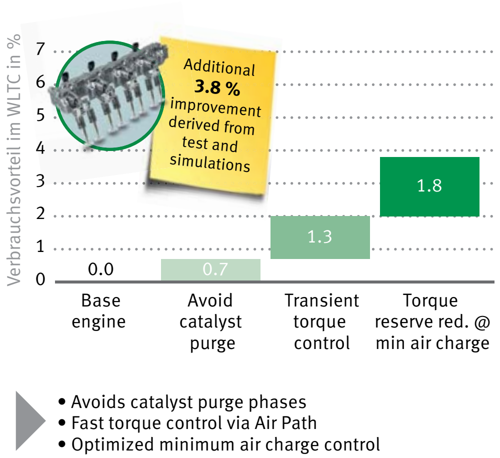
Benefits of UniAir in transient operation

In light of the WLTC and RDE legislation, the transient response of a combustion engine will play an ever-increasing role when it comes to evaluating fuel consumption and emissions. In conventional gasoline engines, highly dynamic transient load shifts are typically controlled by momentarily altering the ignition sequence, such as is the case when shifting gears in a dual-clutch transmission. With the cycle-specific control logic, the UniAir system opens up new possibilities for altering torque output via the air path as the air mass can be adapted almost as quickly as the ignition point. Examples of using the air path for fast torque control include not only rapid load shift and gear changes, but also engine idle speed, inertia fuel shutoff and cylinder deactivation.

Figure 12 plots the progression of a WLTC test for a standard engine. The phases during which the ignition sequence is retarded to realize dynamic load shifts are highlighted in gray. You can easily see that these phases occur very frequently during WLTC testing. By retarding the ignition, the 50 % conversion point (shown in blue in Figure 12) is also shifted from typically 10° CA after TDC back to up to 70° CA after TDC. At this time, combustion is almost 100 percent inefficient for a brief period (bright green progression plotted in Figure 12).



12 **Impact of altered ignition sequence on fuel consumption during WLTC testing**

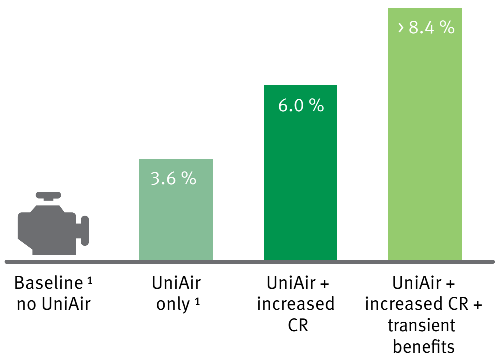


13 Fuel-saving potentials afforded by transient air path regulation in the WLTC test cycle

This has a measurable negative impact on overall fuel consumption. The black progression at the top in Figure 12 indicates the momentarily increased fuel consumption caused by the inefficient combustion sequence, while the black progression below cumulates the values. In the WLTC test cycle, this increased fuel consumption adds up to a noteworthy 50 g.

By regulating the torque output via the air path, the ignition point and center of combustion can be maintained in the optimal window of operation when in transient operating mode. The air mass is regulated by the EIVC control similarly to the aforementioned processes in steady-state operation. This gives rise to three fuel-saving potentials in the WLTC test cycle (Figure 13).

At idle speed and under low-load conditions, the throttle valve linked to conventional engine control systems cannot throttle back the air as much as the engine would actually require, and the ignition is delayed accordingly. Due to its fine-tuned air regulation logic that lifts the valve by only 0.5 mm, the UniAir system also does not need to alter the ignition sequence under low-load conditions, such as at idle speed. This, in turn, reduces fuel consumption by up to 1.8 % in the the WLTC cycle. Adding to this are the benefits of the aforementioned torque reserve, which can be accessed during rapid transient load requests or gearshift changes without moving the center of combustion to a negative operating point. This reduces fuel consumption by 1.0 % in the WLTC test cycle. The third effect is the purging of the catalytic converter following periods of deceleration. On conventional engines, fresh air is pumped through the cylinders into the catalytic converter when the vehicle is coasting. When combustion is restarted, the engine must first be run on an extremely rich mixture with λ = 0.8 to 0.9 for 5 to 10 seconds in order to remove the excess oxygen from the catalytic converter; otherwise, it will not be able to chemically neutralize the pollutant emissions. As UniAir is able to close the intake valves during periods of deceleration, no pump effect is encountered nor does oxygen enrichment take place in the catalytic converter that then needs to be removed during subsequent firing events by running a rich combustion mixture. In the WLTC test cycle, this single effect reduces consumption by approximately 1.0 %. Combining all of the potential savings with aforementioned measures in transient operation, a total of 3.8 % reduction in fuel consumption can be achieved.



1 WLTC simulated with measured 3 cyl. DOHC, VVT engine maps (with & without UniAir) and based on C-segment passenger car

14 Fuel-saving potentials afforded by the UniAir system in the WLTC test cycle

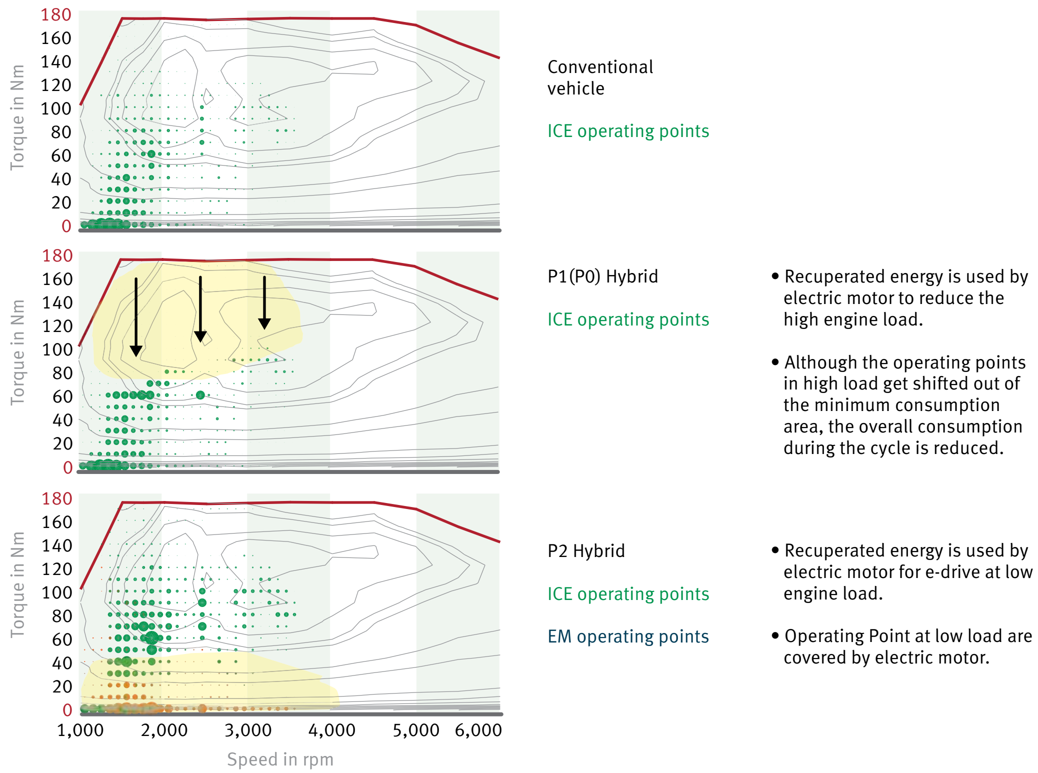
Schaeffler has carried out a series of simulation exercises to illustrate the overall benefits of UniAir for a three-cylinder turbocharged engine subjected to the WLTC test cycle (Figure 14). Due to the reduced throttle losses, the system minimizes fuel consumption by 3.6 % as compared with conventional throttle valve regulation. If one utilizes the higher knock limit afforded by the UniAir system to increase the compression ratio, the fuel burn is improved by 6 %. Together with the improvements in transient operation, the total reduction in fuel consumption of 8.4 % in the WLTC test cycle can be achieved, depending on the respective application scenario and application-specific constraints.

UniAir in a hybrid system

The potential of the UniAir system for improving a combustion engine in an electrified drivetrain depends on the hybrid concept utilized. Figure 15 shows the performance map results in the WLTC test for the three different configurations

- 100 % combustion-engine drivetrain
- P0/P1 hybrid (starter-generator)
- P2 hybrid (hybrid module on the crankshaft with mechanical disconnection from the combustion engine)

The green dots in Figure 15 indicate the dwell time at a given operating point. Without hybridization, medium and high loads are run in particular (Figure 15, top). P0/P1 hybridization shifts



15 Performance map results for a 100 % combustion engine drivetrain and hybrid systems in the WLTC test

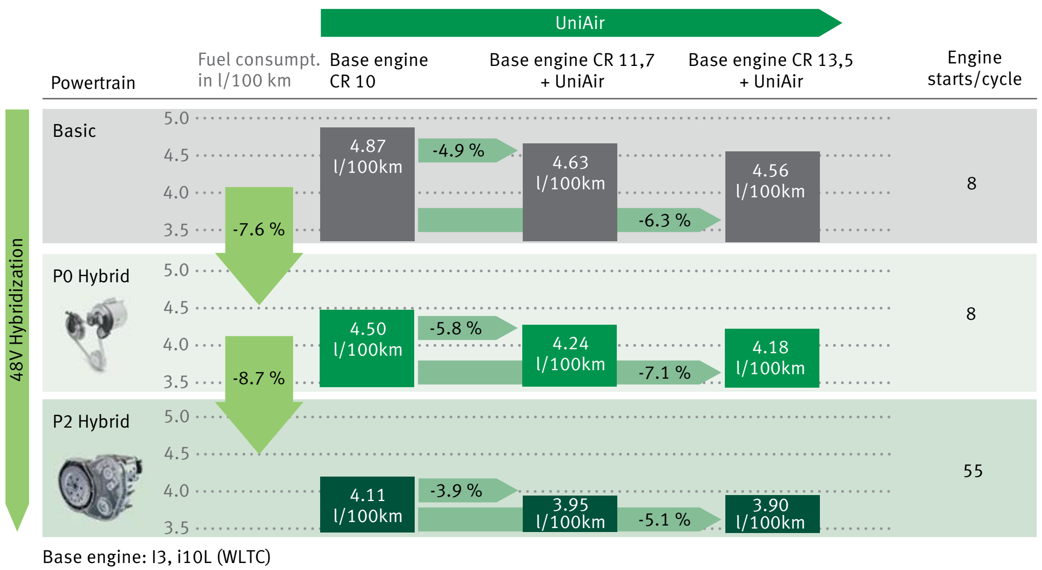
operation to lower load levels for the combustion engine as the recuperated energy is routed through the electric motor into the drivetrain to “soften” peak engine loads. Even if the combustion engine is run in a less favorable range in the WLTC test, the fuel consumption of the overall system is reduced thanks to recuperation, or regenerative braking. The benefits of de-throttling under low and medium-load conditions as offered by the UniAir system are particularly effective in this hybrid setup as the combustion engine frequently enters this operating range.

Compared with a P0/P1 hybrid, the electric motor in a P2 hybrid typically has a higher rated output. This means that a large portion of the low-load operating range can be covered using electrical energy only, whereby the kinetic energy harnessed during regenerative braking is channeled back into the drivetrain so that the combustion engine can be switched off. The combustion engine is largely responsible for propelling the vehicle in the higher load range (Figure 15, bottom).

Here, the UniAir system can also be used to optimize the engine under these conditions. Not only can the system de-throttle the engine in the medium to high-load range; it also reduces consumption under full-load conditions by shifting the knock limit and center of combustion as previously described.

In-house analyses at Schaeffler point to the practical potentials of the UniAir system in a WLTC simulation for a 48 V hybrid drivetrain as used in a P0/P1 and P2 configuration (Figure 16).

The starting point for the tests was a conventional 1.0-liter three-cylinder gasoline engine without hybridization or a fully variable valve train. By applying UniAir system and tweaking the compression from 10 to 11.7, a fuel savings of 4.9 % could be achieved (Figure 16, top). If the compression is increased still further to 13.5 %, fuel consumption can be reduced by an entire 6.3 %. The graph at the center of Figure 16 shows the potential of the UniAir system for a P0/P1 hybrid powertrain.



16 **Fuel-saving potentials afforded by UniAir in a hybrid system**

Here, the starting point is the engine without the UniAir system but which has been fitted with a hybrid module. By integrating the UniAir system, together with the compression ratio of 11.7 %, 5.8 % less fuel was burned. A high compression ratio of 13.5 % reduces fuel consumption by 7.1 % as compared with the base version. The graph at the bottom of Figure 16 plots the results for the P2 hybrid. Here, too, the technological leap afforded by the UniAir system, together with a higher compression ratio, yields a fuel consumption savings of between 3.9 % (11.7 % compression ratio) and 5.1 % (13.5 % compression ratio).

Summary and outlook

Schaeffler has been mass producing the fully variable UniAir valve train system since 2009 and has since delivered over three million units to customers around the world. The system has also been continually improved and optimized with more functions. Schaeffler has devised a development method that also makes it possible to quickly and easily integrate the UniAir system in existing engine designs. In this context, in order to fully exploit the performance capabilities of the UniAir system, it is important that the overall engine with turbocharging system be adapted to the requirements of the fully variable valve train. With this setup, Schaeffler was able to reduce fuel consumption by a minimum of 8.4 % during WLTC testing. Particularly in transient engine operation, the cyclically variable air path regulation logic of the UniAir system considerably lowers fuel consumption not only during cycle-based measurements, but also in operation on public roads.

When it comes to future drivetrain concepts involving P0/P1 hybridization, UniAir represents a key module for coordinating combustion to the specific requirements of the various concepts. Additional engine tests at Schaeffler also point to the fuel-reducing capabilities of the UniAir system when the combustion engine is restarted, which are realized by not having to recondition the three-way catalytic converter. The pronounced flexibility of the UniAir system likewise facilitates cost-effective modular concepts as the engines can be easily adapted to different applications by fine-tuning the engine software.

Preliminary tests are currently being run on a UniAir system fitted with an additional pressure accumulator. The goal with this configuration is to reduce the hydraulic losses associated with EIVC concepts. Instead of simply dissipating the oil pressure in the UniAir actuator via the return line, the pressure accumulator utilizes the excess energy to tension a spring, which then releases the pressure back to the hydraulic system during the next cycle. Initial results indicate that this improvement can reduce hydraulic friction levels by upwards of 30 %.

Literature

[1] Haas, M.: UniAir – The First Fully-Variable, Electrohydraulic Valve Control System. 9. Schaeffler Kolloquium, Baden-Baden, 2010

[2] Haas, M.; Piecyk, T: Valve Trains for Implementing Innovative Combustion Strategies. 10. Schaeffler Kolloquium, Baden-Baden, 2014

3

Fast and Precise

Electrical Cam Shifting

Andreas Thölke

Introduction

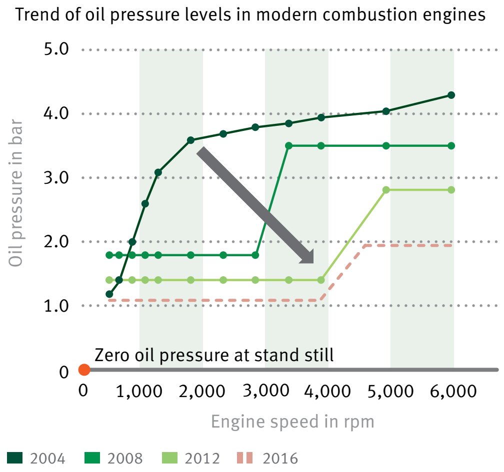
Due to the ever stricter emissions legislation, reducing pollutant emissions fuel consumption will play a key role in developing internal combustion engines in the future. In this context, not only does treating exhaust emissions downstream of the engine form a critical component of the emissions concepts for internal combustion engines; it is equally important to minimize emissions inside the engine. Optimizing the charge cycle is a promising approach as it impacts the combustion process and affects fuel consumption and emissions. Since combustion engines in vehicles are not operated under steady-state conditions, but rather at different speeds and loads, the charge cycle must be adapted to the operation situations. Camshaft phasing systems have for years proven to be a sufficient means of achieving this objective as they allow the valve timing and charge cycle to be aligned with the given operating point. During transient operation, when the engine transitions to a different operating point in the performance map, the phasing system must switch to the new camshaft timing as quickly as required. If the phasing speed is insufficient, the ECU must actively intervene in the ignition and fuel injection sequences. Though this works, but typically the engine efficiency is decreased during the transient. Phasing speeds of up to 500° CA/s are currently required. In case the exhaust side is additionally equipped with a cam phaser, it is possible to internally recirculate the exhaust gas to reduce untreated emissions and minimize fuel consumption. For this purpose, the variable cam timing must control both camshafts as precisely as possible. The target is to achieve a maximum deviation of less than 1° CA from the set point defined in the mapping [1]. With the advent of hybrid and start/stop systems, engine start-ups are coming more and more in focus due to the increased number of engine starts. Engine start-ups are critical for emissions, ensuring flexible and precise adjustment of valve timing represents an efficient measure for reducing emissions.

Variable camshaft phasing systems were previously the domain of the gasoline engine. This has lately changed, as now first diesel engines are equipped with camshaft phasers. The main goal of such systems is to reduce emissions by delaying the closing time of the intake valves, which decreases the effective compression ratio.

Hydraulic camshaft phasing systems

Current camshaft phasing systems typically employ hydraulically actuated swing motors (Figure 1). The inner part of the camshaft phasing unit contains a vane-type rotor that is connected to the camshaft. The outer part (stator) is driven by the crankshaft a chain, belt or gears. The individual segments in the stator and the vanes on the rotor form oil chamber pairs. When oil flows into one of these chambers, the camshaft timing is changed [2].

1 **Design of a hydraulic camshaft phasing unit**



2 **Oil pressure of different generations of an engine family from 2004 (dark green) to 2016 (light red)**

The oil flow is controlled by a proportional valve. Together with position sensors at the crankshaft and camshaft, the system forms a closed-loop control circuit. This allows to adjust all demanded angular positions continuously.

Although the hydraulic concept is robust and has proven itself in numerous applications, it is quickly reaching its technical limitations. Active adjustment is linked to the engine oil pressure. To reduce fuel consumption, modern engines are being designed to be highly efficient. This also affects the lubrication circuit. The oil pressure is reduced to minimize the input energy required by the oil pump. Figure 2 shows the reduction of oil pressure of different engine generations of the same engine family from 2004 to 2016.

In the current engine variant from 2016, the oil pressure is sometimes just 1 bar, which makes it even more difficult to achieve the required phasing speeds in transient operation. This constellation will become more critical as the Worldwide Harmonized Light Vehicles Test Cycle (WLTC), which went into effect in the EU in 2017 to more accurately conduct emissions tests, covers a much broader speed and load range of the engine and includes more dynamic portions with hard acceleration than the previous New European Driving Cycle (NEDC). This elevated dynamic level necessitates considerably more actuating routines. Every deviation from the target angle can lead to an increase in raw emissions.

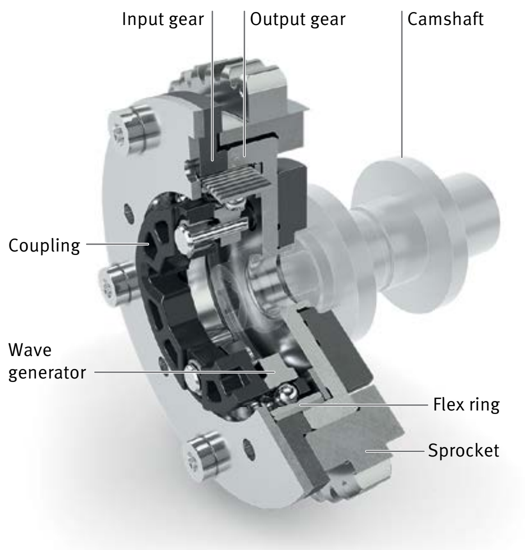
When switched off, the combustion engine has no longer any oil pressure for camshaft phasing. As immediately after the engine is re-started, there is only little or no oil pressure available for actuating, the cam phase is locked in a park position. This is why hydraulic cam phasers only adjust one position during engine start. Start/stop functions, which switch off the combustion engine when the vehicle is at a standstill (e.g. at a stoplight) and automatically switch it back on when the driver wants to drive, increase the number of engine restarts during operation. To optimize fuel consumption and emissions in these situations, a phasing system will be required that can realize different timing sequences for individual start-up conditions.

Electric Cam Phaser

By integrating an electromechanical camshaft phasing unit (ECP) (Figure 3), which Schaeffler has been manufacturing since 2015, it is possible to fully decouple phase adjustment from the engine.

In an electric camshaft phasing system, a brushless DC motor (BLDC) and a reduction gearset are used in place of the hydraulic actuator to adjust the phase angle between the crankshaft and camshaft. Compared to conventional brush motors, BLDC motors are more efficient and maintenance-free. In combination with a high-ratio, three-shaft gearset, the electric motor forms the phase system. The gearing comprises two hollow gearwheels and an oval rolling bearing that forms

3 **The electromechanical camshaft phaser**



4 **Design of the gear-set for the electromechanical cam phaser**

the wave generator in conjunction with the flex ring (Figure 4).

The flex ring of the wave generator connects the sprocket via the input gearwheel and the output gearwheel with the camshaft. The output gearwheel has two more teeth than the input gearwheel. The wave generator presses the toothed flex ring into both gearwheels. As the wave generator rotates, the different number of gear teeth on the gearwheels produce the reduction ratio of the gear set [2].

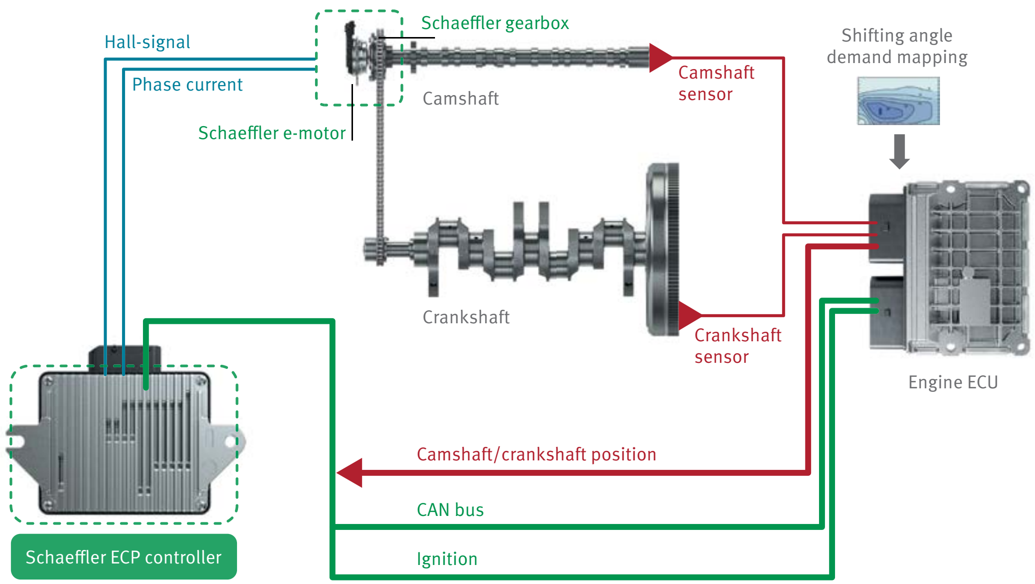
The electric motor is connected to an ECP control unit, which regulates the motor's operating speed and processes the Hall sensor signals from the motor (Figure 5).

The sensors integrated in the electric motor detect the position of the rotor and monitor temperatures. The ECP control unit communicates with the engine control unit. The target angle values of the camshaft are sent by CAN bus to the ECP control unit, which compares it against the current position. The electromechanical cam phaser switches between the three operating modes of advance timing, constant phase angle and retard timing. To realize an advance timing adjustment, the e-motor rotates more quickly than the camshaft, and to retard timing it rotates at a slower speed. The constant angle is maintained by rotating the output shaft of the electric motor at camshaft speed.

As the level of hybridization increases, the available space in the engine compartment of modern vehicles decreases. Therefore one of the main development targets was to engineer an actuator system that is very compact.

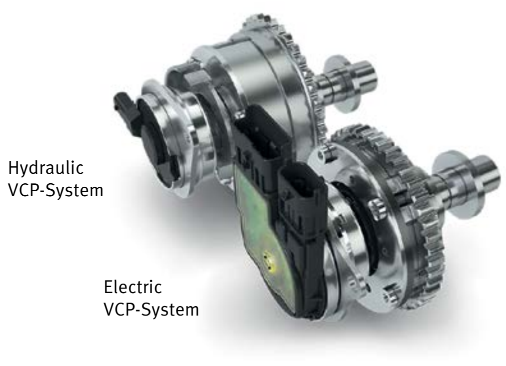
The electric cam phaser fits within the space of a conventional hydraulic actuator – without the need for modifications (Figure 6).

Modular interchangeability makes it easy to switch between hydraulic and electric camshaft



5 **Topology of the electromechanical cam phaser**

phasing units and facilitates platform concepts for equipping the individual variants of an engine family with different actuators.



6 **Hydraulic and electromechanical camshaft phasing units require the same amount of installation space**

Properties of the electromechanical cam phaser

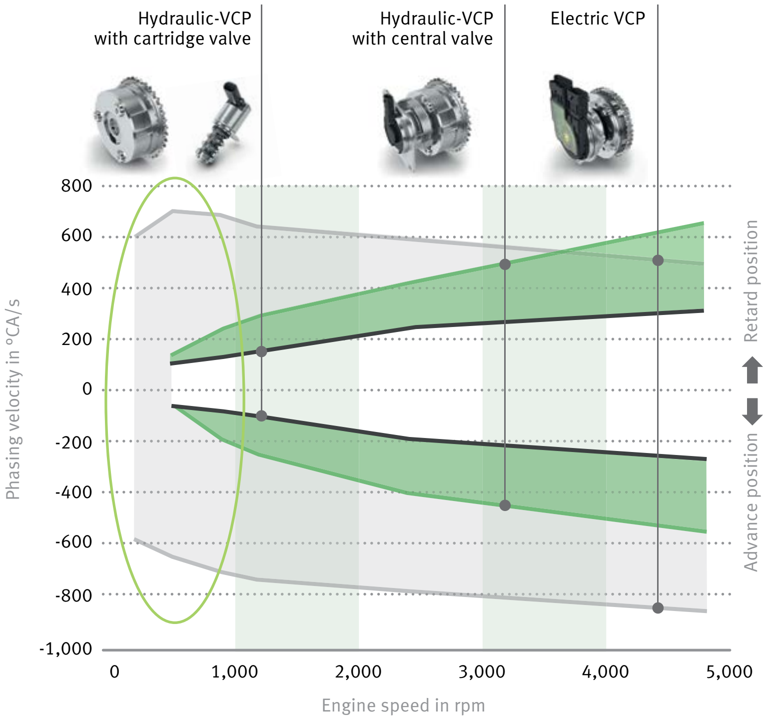
Electronic camshaft phasing systems enable higher phasing speeds than hydraulic systems. This allows a more aggressive calibration of the valve timing to minimize active intervention in the ignition and fuel injection sequences. Very early advance timing makes it possible for the combustion engine to build up torque more quickly during acceleration, which means that an electromechanical camshaft phaser not only helps to achieve high operating efficiency, but also good driving performance [3].

The phasing speed of the electromechanical cam phaser system is almost entirely independent of the engine speed and engine oil temperature. This also ensures actuation during a cold start and when the engine is off, Figure 6 shows the angular velocity of two current hydraulic camshaft phasing systems and the electric cam phaser as a function of the engine speed You can clearly see that the phasing speed of the electromechanical system in the relevant engine speed range and in particular at low operating speeds is significantly higher than that of the hydraulic actuators.

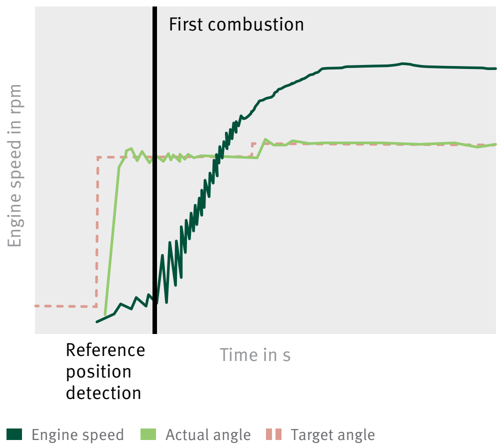
To analyze the performance capabilities of electromechanical camshaft phasing units in a realistic operating environment, Schaeffler carried out tests using a vehicle whose engine was converted from a hydraulic to an electromechanical camshaft phaser system. In the process, focus was placed on determining just how quickly the camshafts can be adjusted from their park position to the optimal position for starting up the engine. As Figure 8 shows, the actuator already reaches the phase angle required prior to the first ignition fire of the combustion engine.

Any timing angle can be set when the engine is started. You can see that the actual value of the timing angle is aligned almost instantaneously with its target value as defined by the engine control unit and that this angle is maintained very precisely.

Schaeffler has developed a specialized assist function for the start/stop strategies used in internal combustion engines. The electronic control system of the cam phaser remains active when



7 **Phasing speed of different camshaft phasing systems in relation to the engine speed.**



8 **Timing adjustment with an electromechanical camshaft phasing unit at engine start-up**

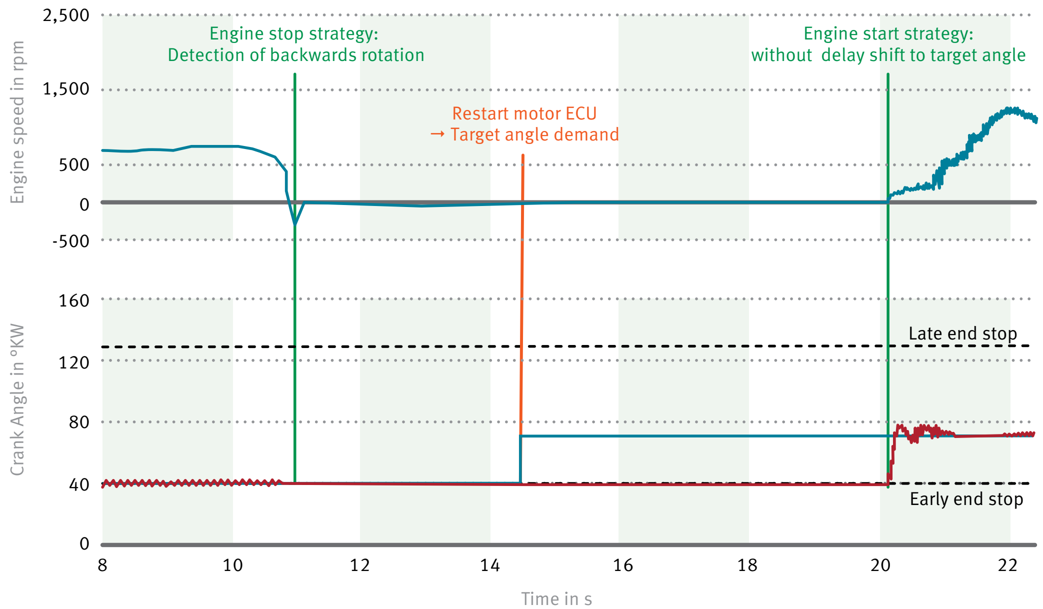
the engine is off, analyzes the data from the position sensors and synchronizes the camshaft and crankshaft positions. In the process, the timing sequences are first held at a predefined angle at engine standstill and – depending on the application –are then very quickly brought into position either before the engine is started or at the precise moment the engine is started. Figure 9 illustrates how the function works based on the readings taken from vehicle testing.

Plotted at the top of the graph is the engine speed (blue). When the start/stop function is activated, the engine speed drops from idle speed to zero, and the engine stops. The timing angle (red line at the bottom of the graph) of the camshaft is initially maintained in this situation. This is an actively regulated adjustment process defined by the controller that also accounts for minimal rollback of the crankshaft and adapts timing accordingly. In the example at hand, the engine control unit computes a new target angle for the camshafts while the engine is off. The ECP control unit was calibrated to trigger adjustment toward the target angle as soon as a defined crankshaft speed threshold is overshot. The typical time it takes to assume the target angle is below 100 ms.

Timing sequences adapted to operating conditions considerably reduce emissions. Measurements taken on a V6 engine with two electromechanical camshaft phasing units on the intake side resulted in a reduction in HC emissions of 16.7 % during the first 15 seconds of cycled engine operation [3].

Better comfort levels with the electromechanical cam phasing

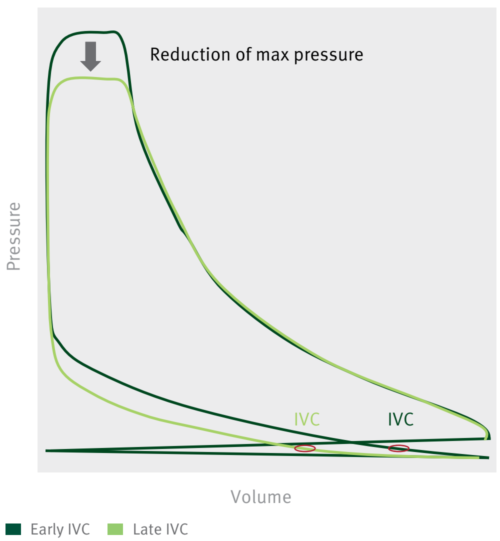
Adjusting the timing sequences before starting the engine not only reduces emissions, but can also improve comfort levels by ensuring that en-



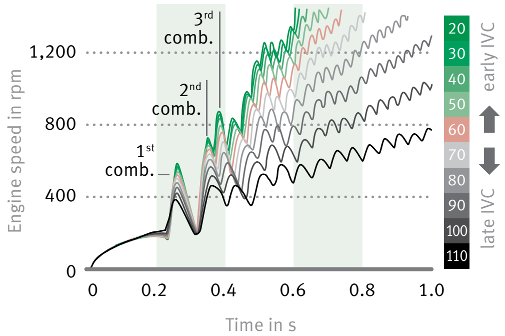
9 **Adjusting the camshaft at engine start-up**

gine starts more smoothly. This is very important for implementing start/stop functions, and when it comes to hybridized configurations that allow the internal combustion engine to be switched off entirely during periods of operation, since customer acceptance of these technology packages depends on whether the engine will restart with acceptable NVH. The theoretical background can be explained using the PV diagram in Figure 10.

The red circular marking at the right in Figure 10 designates the standard closing time of the intake valves. When adjustment of the inlet camshaft is actively retarded, this point shifts to the left on the compression curve, and the air drawn in through the (still) open intake valves is initially pushed back during the compression stroke. When the intake valves close, the effective portion of the compression stroke begins. The effective compression ratio of the internal combustion engine is reduced and as a result the engine starts smoothly.



10 The charge cycle as depicted in the schematic PV diagram: When the intake valves execute a delayed close, part of the charge is ejected during compression such that less air is compressed



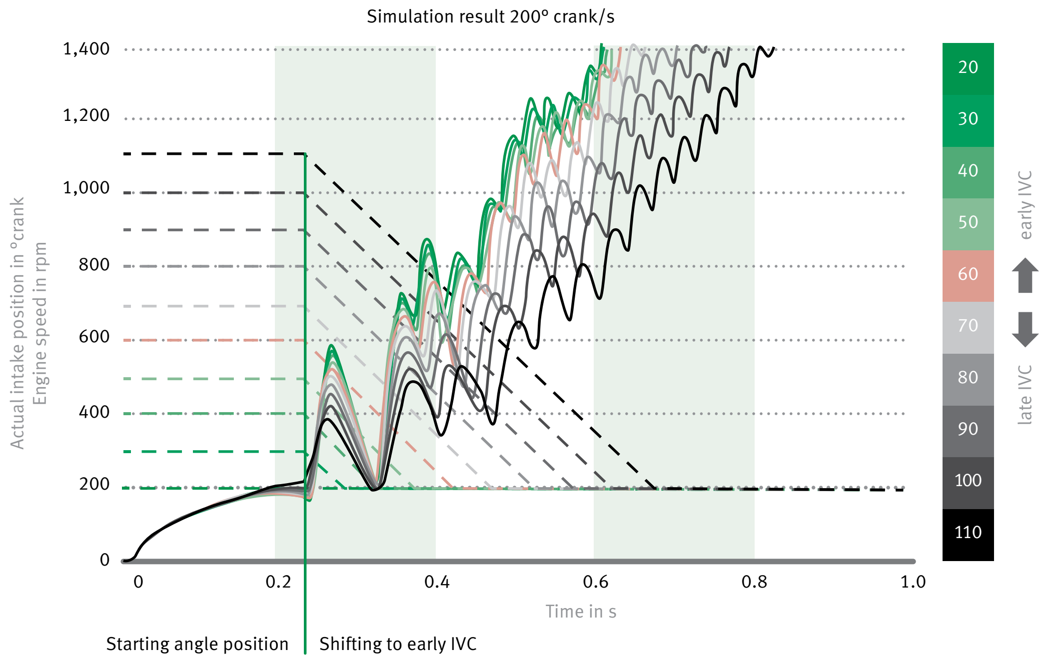
11 Engine speed ramp-up when starting at different phase adjustment angles

Simulations were carried out to analyze the effects of different intake valve timing angles during engine start (Figure 11)

The diagram plots engine speed ramp-up curves of the internal combustion engine for different intake valve timing sequences. The orange curve represents an intake valve timing sequence with IVC at 60° CA. This is a typical value that is used as the default setting by today's hydraulic phasing systems during the start-up phase of the engine. The ramp-up curve for an intake valve timing sequence with IVC at 110° CA appears as a black line. It is easy to see that the peak operating speeds caused by the combustion process and the resulting air and structure-borne noise emissions are softened. By contrast, very early timing sequences (green) considerably increase ignition pressure and engine speed amplitudes; this strategy is not practical for a comfort-oriented application and can even damage the dual-mass flywheel.

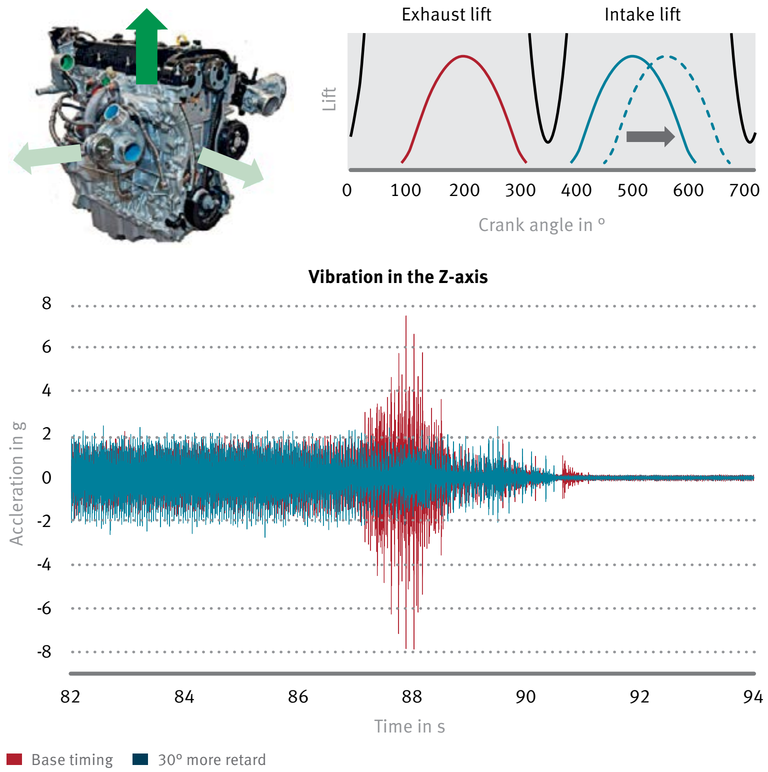
The disadvantages of using retarded intake timing when starting an engine are that the starting procedure takes longer because the momentary torque available prevents idle speed from being reached immediately. As Figure 11 illustrates, this effect becomes more pronounced the more aggressive the delayed timing sequence for the intake camshaft becomes. For an NVH-optimized timing sequence with IVC at 110° CA (black line), it would take upwards of 1 second for the engine to be ready for operation. This can cause the start-up procedure for start/stop applications to literally "drag on". Figure 12 shows a way to start the engine quickly and smoothly.

Here, the actuator system moves the camshaft to a retard (late IVC) position before the engine is started as for the approach outlined in Figure 11. Unlike in the first simulation, the camshaft does not stay at retard position, but continually reverts to the "advance" position during the start-up process. In Figure 12, the momentary timing is plotted on the ordinate next to the engine speed in relation to the crank angle. As the black line reveals, the original position with IVC at 110° CA changes to 90° for the second ignition and 70° for the third ignition. The time to the idle speed level of 800 rpm thereby drops from 1.0 seconds to around 0.6 seconds. The engine continues to start smoothly as the initial ignitions that are particularly critical from an NVH perspective occur when the camshaft position is in "retarded" positions. Schaeffler investigated different phasing speeds up to 800° CA/s during the simulation exercises. In the process, it was determined that a phasing rate of 200° CA/s is ideal. This value can be found in the plotted curves in Figure 12.



12 Comparison of different start angles for „advance" position

Not only is engine start-up a comfort-oriented aspect, but also engine shutdown during start/stop operation. As Schaeffler has proven during testing, the shutdown phase of the engine can also be optimized with the aforementioned procedure. Figure 13 charts the results of test bench measurements taken on an unfired engine with retarded intake valve timing. While the original application with unmodified timing generates considerable acceleration amplitudes (red), these amplitudes disappear completely when retarded timing is initiated (blue).



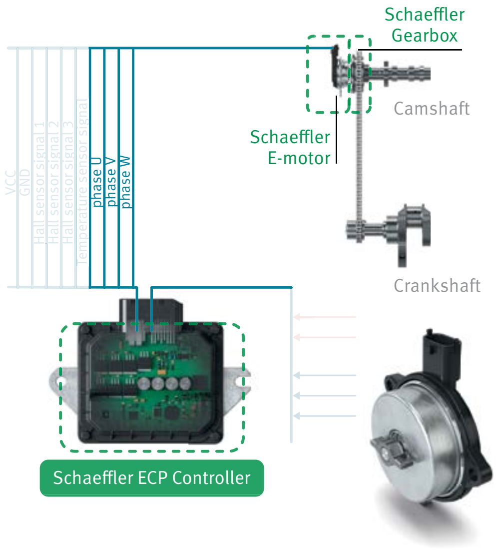
13 **An intake camshaft adjustment reduces the acceleration amplitudes as the internal combustion engine enters shutdown**

Innovation

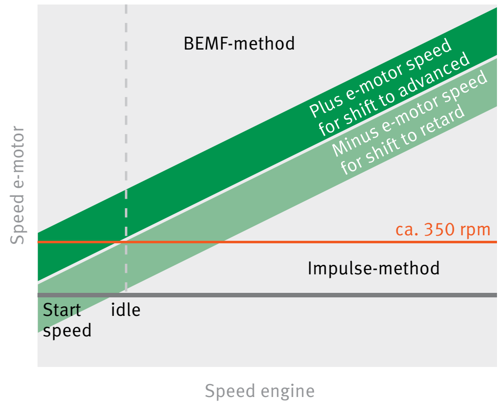
Schaeffler has been manufacturing the electromechanical camshaft phasing system since 2015 and continues to improve on its design. One promising approach to this end is what is known as sensorless BLDC engine operation. This makes it possible to reduce the number of components in the ECP electric motor as well as the wiring harness.

This approach further minimizes the packaging space requirements of the ECP electric motor while expanding its permissible ambient temperature range as heat-sensitive components can be omitted. Integrating the ECP in the cylinder head is also simplified. Technical implementation of the design concept involves replacing the Hall sensor system required for the rotor position with voltage and current measurements taken for the individual phases of the electric motor. A well-known approach to this end is utilized by the counter-electromotive force constant (BEMF, or back EMF): As soon as the motor rotates, it induces a quasi-sinusoidal voltage. The zero-crossing point of the voltage is then used to determine the actual rotor position. The downside of this method is that it only works reliably for motor speeds of 350 rpm and higher. At lower speeds, Schaeffler leverages the impulse method, which entails sending a current pulse to the phase. Depending on the position of the rotor, the motor inductance changes, which in turn impacts the current rise initiated by the current pulse. This value then serves as a basis for determining the rotor position in the motor. Figure 15 shows how both methods are combined for the electromechanical phase adjuster.

The speed of the internal combustion engine is plotted on the abscissa in Figure 15. Since the combustion engine mechanically drives the camshaft using a fixed ratio of 1:2, this is also a reference point for the camshaft speed. The ordinate shows the operating speed of the ECP motor. The vertical gray line marks the idle speed of the internal combustion engine. The white diagonal in the diagram reflects the constant operating state in which the electric motor and camshaft rotate at the same speed. The dark green bar on top shows the timing range, where the electric motor rotates faster than the camshaft. The bright green bar below the wite line symbolizes “retarded” phase



14 **Simplified electromechanical camshaft phasing unit concept without engine-integrated sensor system**



15 **The BEMF and pulse method together cover the entire operating speed range of the actuator**

adjustment, when the electric motor operates at a slower speed than the camshaft. You can see that a large portion of the start phase of the combustion engine is covered by the pulse method and extends up to idle speed. At higher speeds, a transition is made to the BEMF method. The pulse method is not favorable in these operating ranges as it takes too long to determine the rotor position in order to obtain exact results at high speeds.

Summary and outlook

Camshaft phasing units are being used in more and more gasoline engines – be it on the intake side only, or also on the exhaust side – to boost rated output and torque as well as reduce untreated emissions. Oil pressure levels, which continue to get lower, as well as heightened requirements for active adjustment, quickly reveal the physical limitations of the established camshaft phasing unit concept based on hydraulic actuators. Since 2015, Schaeffler has had an electromechanical cam phaser system in its portfolio that considerably expands the technical capabilities of this fundamental design concept. Not only is the phasing speed of the electromechanical camshaft phasing unit faster than that of a conventional hydraulic actuator, the system also operates almost entirely independently of the engine speed and engine oil temperature. This also ensures actuation during a cold start and when the engine is off, and the valve timing can be adjusted before the internal combustion engine starts. The result is fewer emissions and increased comfort levels thanks to smoother engine start-ups. This is very important when implementing start/stop functions and for hybridized configurations that allow the internal combustion engine to be switched off entirely during operation.

As development activities continue for the electromechanical cam phaser from Schaeffler, plans are in place to eliminate the Hall sensors that determine the position of the rotor along with the associated electronics, wiring and connectors.

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4

VCR

The Last Big Step for Greater Efficiency

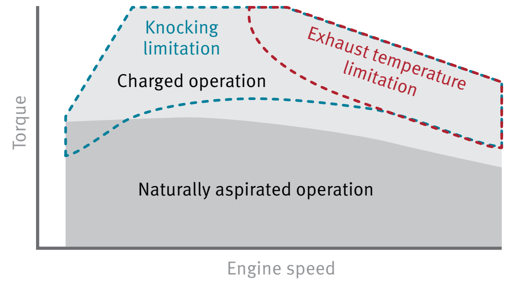
Dr. Peter Solfrank
Joachim Dietz

Introduction

The last few years have seen the trend of downsizing – the process of reducing the displacement of an engine – as a measure frequently used to reduce the fuel consumption and CO_2 emissions of gasoline engines. To maintain driving dynamics at a respective level or even improve them, downsized engines typically have air charging in one or the other form. As time went on, the level of boosting continued to be increased so that manufacturers could further reduce displacement without sacrificing engine output. The phenomenon of self-ignition induced by high pressures and temperatures (knocking), however, set limits to increasing the charging level. In Figure 1, this range is marked in the top left part of the engine load map.

This phenomenon can be counteracted by two measures: The first approach would be by retarding the ignition timing in this high-load range. While this delays the combustion process and subsequent rise in temperature and pressure in the combustion chamber, it also reduces engine efficiency, and fuel consumption rises quite significantly. The second measure is a mechanical engineering one and involves reducing the geometric compression of the turbocharged engine as compared to a naturally aspirated engine. The turbo engine can then also run under high loads without the ECU actively delaying the ignition timing, or can at least run with minimal intervention to keep the additional fuel consumption to a minimum. Ultimately, this modification shifts part of the compression work to the charging system, whereby the passive or active intercooling setup lowers the compression end temperature in the cylinder and, consequently, the gasoline engine's inherent tendency to knock.

In the partial-load range, however, the reduced geometric compression is synonymous with a lower operating efficiency. When engineers set



1 **The ranges of knocking combustion and wide-open throttle enrichment in the engine performance map**

out to design an engine, they typically aim for a compromise by slightly reducing the geometric compression and actively intervening in the ignition sequence and/or running a momentarily rich fuel-air mixture to cool down the cylinders at the remaining high-load operating points at risk of knocking events (Figure 1, upper right corner of the performance map).

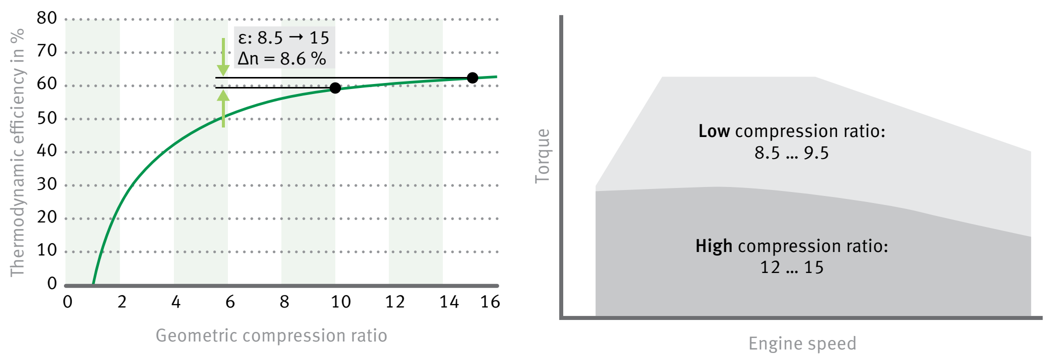
This type of engine design was more or less promoted by the testing strategy of the New European Driving Cycle (NEDC), which was the standard measure of officially rating the emissions and fuel consumption levels of a vehicle in the EU up to 2017: it included a disproportionately high share of engine operating points in the low-load and partial-load range and completely disregarded performance under wide-open throttle conditions. As such, the NEDC opened up opportunities for engine designers to fine-tune the compression ratio in favor of the "all-important" partial-load fuel consumption performance. With the advent of the Worldwide Harmonized Light Vehicles Test Cycle (WLTC), which took effect in 2017, as well as the Real Driving Emissions (RDE) testing procedure, these opportunities can no longer be leveraged as before. The reason for this is that the WLTC covers a much broader speed and load range of the engine, while the RDE takes things even one step further by making the pollutant emissions of a vehicle tested under all conceivable real-world conditions a binding requirement for becoming certified for road use. Translated into engine terms, this means that the boost level used in new engines needs to be reduced again and displacement increased which would actually increase fuel consumption and CO_2 emissions.

High efficiency under partial-load and full-load conditions

One possibility of managing the inherent conflict in optimizing an engine for partial-load and full-load performance are mechanisms that enable a variable compression ratio, or VCR. This technology allows the compression to be adapted in real time to the momentary operating point of the engine. In such an engine, as soon as an operating point where knocking can occur is approached, the compression ratio is reduced, while a higher compression ratio is run at operating points that do not run the risk of a knocking event. As such, this technology is well suited to future powertrain concepts that involve highly turbocharged, downsized engines.

A VCR system also eliminates the need for wide-open throttle enrichment – a necessary evil frequently used in turbocharged gasoline engines operated under high-load conditions to protect the components from becoming overstressed (Figure 1): In this condition, so much fuel is blended into the combustion air that the available amount of oxygen in the cylinders is not sufficient to fully burn the mixture. The vaporization energy of the increased fuel quantity lowers combustion temperatures, thereby also reducing peak pressure and material loads; the three-way catalytic converter, however, no longer operates as efficiently as it could.

Figure 2 (right) charts an example performance map for an engine with a two-stage compression ratio in relation to engine mean effective pressure and operating speed. This engine is toggled between a high compression ratio (12:1 to 15:1) under low loads and a low compression ratio (8.5:1 to 9.5:1) under high loads. Figure 2 (left) plots the resulting effect on the theoretically attainable thermal efficiency of the engine based on an isovolumetric model process. When run at a high compression ratio of 15:1, efficiency increases to a value of 66.1% compared to base value of 57.5% at a lower 8.5:1 ratio. These are the theoretical limits of the potential that can be achieved when using a VCR setup in order to increase compression ratio in the partial-load range as compared to an engine optimized for full-load performance.



2 **Influence of compression ratio on thermal engine efficiency (left) and performance map of a two-stage compression ratio (right)**

Charge cycles that involve aggressive early intake valve closing (EIVC) or late intake valve closing (LIVC) are currently gaining quite some popularity and are implemented via camshaft phasing or variable valve trains. They represent one way of increasing the expansion stroke over the compression stroke in order to further leverage the available enthalpy of the combusted gas. These methods do, in fact, lead to the effective decompression of an engine with a high geometric compression ratio. To ensure a sufficient supply of air under full load conditions, however, these concepts necessitate a higher boost pressure to assure a sufficiently high cylinder charge while meeting target conditions for pressure and temperature.

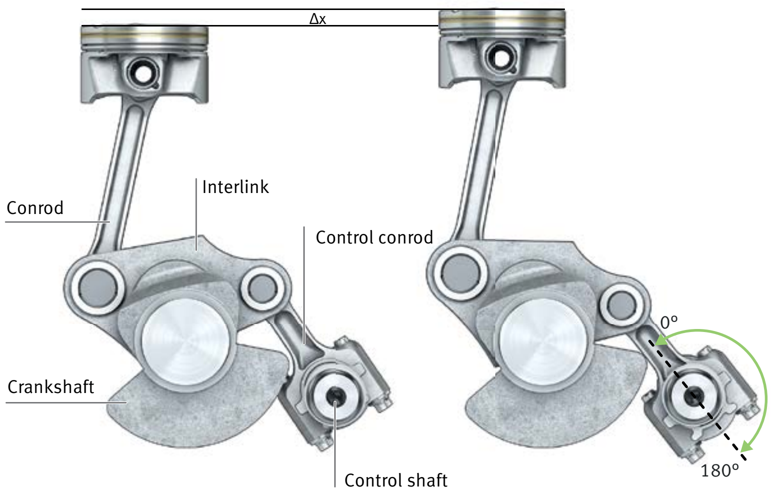
Multi-link concept

The concept of a variable geometric compression ratio is not really new. As far back as in the 1920s, test engines were built using variable compression. Although various automakers and suppliers have since experimented with the technology, none of the developed designs reached the level of maturity required for mass production. The main challenge to overcome was to engineer an adjustment mechanism that would not only be robust and reliable enough to last the life of the engine, but could also be controlled quickly and precisely as well as be cost efficient to manufacture. Due to the ever more present CO_2 debate, the aforementioned synergy effects with downsizing and the positive experience gained with other variable engine systems (such as in the valve train), manufacturers, suppliers and development service providers alike have overhauled and ramped up their VCR development programs over the last few years.

The systems currently in discussion can be subdivided as follows:

- Folding or vertically displaceable cylinder head
- Variable length conrod
- Eccentrically positioned crankshaft
- Alternative cranktrain kinematics

Schaeffler has been involved in VCR concepts for a long time and compared the different proposals in a benchmark investigation. As a result, the cranktrain kinematics of the multi-link system (Figure 3) proved to be particularly beneficial with respect to operative function and robustness. Schaeffler then began pursuing this concept further, including an actuator system architecture that resembles the one currently used in electric cam phasers.



3 **VCR system based on multi-link mechanicals**

The engine continues to have conrods and a crankshaft. The big conrod eye and crankshaft pin, however, are not connected directly, but instead by an intermediate rocker arm (interlink). This arm is guided by a secondary conrod (control conrod) via a third joint. The control conrod in turn is supported by an eccentric section on a control shaft. As Figure 3 shows, the control shaft's rotational position controls the piston top dead center position via the mechanism described above. It is thereby capable of adjusting different compression ratios. Specifically, any compression ratio in between the minimum and maximum values can be permanently adjusted by affixing the control shaft in the respective position. In addition to this, the multi-link concept also offers a host of benefits that result from the altered kinematics as compared with the conventional crankshaft drive.

The infinitely adjusting compression ratio also gives the multi-link concept a host of benefits that result from the altered kinematics as compared with the conventional crankshaft drive. When the motion conditions are analyzed in detail, it becomes apparent that a suitable geometric configuration can considerably reduce the second order inertia forces associated with a conventional crankshaft drive. For a four-cylinder engine, it is even possible to eliminate the balancer shaft system altogether [1].

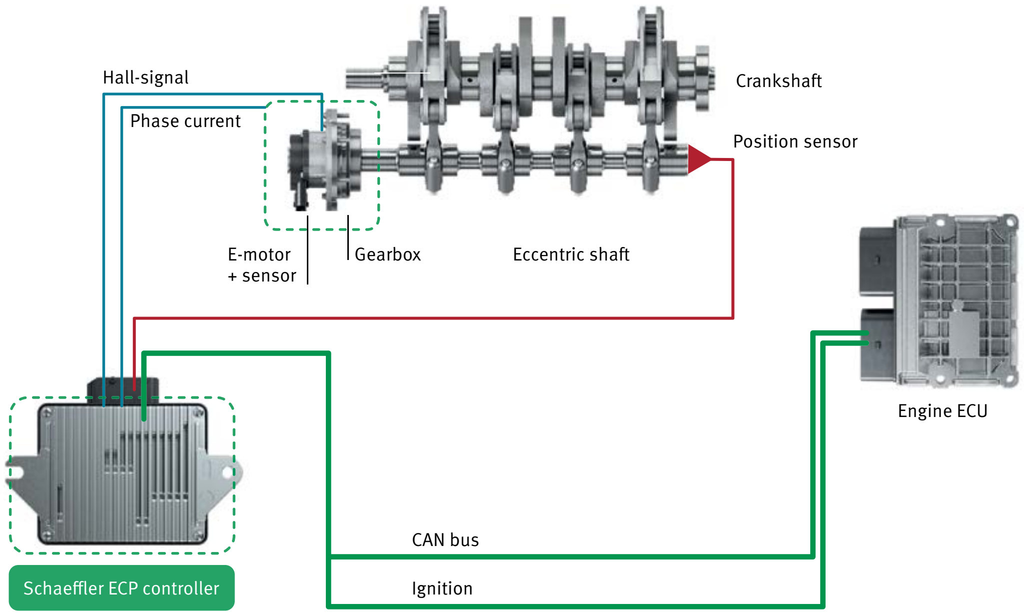
Also, based on the more even and consistent acceleration values of the pistons in the vicinity of top and bottom dead center, the kinematic fluctuation of the crankshaft torque is reduced. In [1], the authors assert that this effect on a four-cylinder inline engine equipped with a multi-link crankshaft drive can result in the engine approaching the smoothness of a conventional six-cylinder engine in a V arrangement. The reduced piston acceleration at top dead center and the subsequently slower piston speed results in less variation of the burn rate. As a result, the impact of fluctuations in heat release on thermal efficiency is minimized. This facilitates e.g. higher exhaust gas recirculation rates and potentially more degrees of freedom for lean combustion systems, thus making it easier to implement combustion strategies associated with lower fuel consumption.

Due to the larger number of moving components and bearing points involved as compared with a conventional cranktrain, it stands to reason that the total friction loss seems to be greater with the multi-link concept. A detailed examination of the kinematic configuration, however, reveals that the conrod is practically in the vertical position when high cylinder pressures are experienced. This greatly reduces lateral piston forces and the associated friction between the piston and cylinder wall during the compression phase and the all-important expansion phase. In summary, it can be stated that this effect compensates the increased losses attributed to the additional bearing points to the extent that the total friction loss of a multi-link basic engine is on par with or even better than that of a conventional configuration. This specifically holds true when considering the possible additional friction reduction by the omission of a balancer shaft system in a 4 cylinder engine [1].

Actuator system

The different compression ratios are set by rotating the control shaft, as described above. Due to the very dynamic and high torque levels that this control shaft is exposed to, the adjustment is done by an electric motor connected to a high-ratio gearbox. A separate electronic control/driver unit is used to assure fast and precise control based on:

- The target angle information provided by the engine control unit
- Information from rotor position sensors of the electric motor

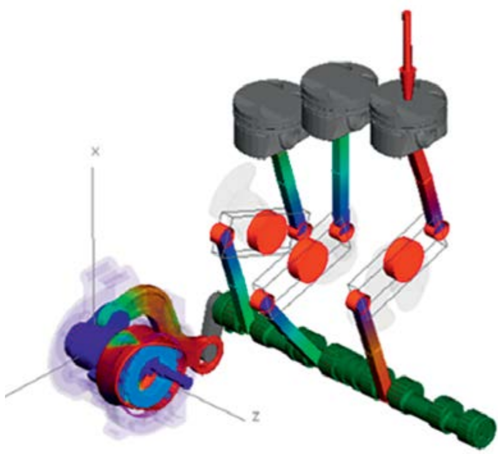


4 **Topology of the multi-link multi-link phasing system**

- Information from an external rotational position sensor of the rotated control shaft. Additionally, the control unit performs diagnostic and safety functions and feeds the corresponding information back to the engine control unit.

The system topology (Figure 4) thus largely coincides with the concept of an electronic camshaft phasing system. Based on the comprehensive experience gained with these mechatronic systems, Schaeffler decided to develop the actuator system to mass production level.

The basis for the initial design and further development activities of the actuation system described are customer requirements as well as in-house analyses. This investigative work is carried out by conducting a multi-body dynamic simulation exercise (MBS) of the entire cranktrain considering predefined combustion pressure information that is applied to the cylinders of the engine (Figure 5). The result of these efforts is an in-depth understanding of the overall multilink drivetrain in terms of nominal as well as dynamic and vibrational behavior. This allows to evaluate the nominal, transient and vibrational loads upon the actuator components to be developed.



5 **MBS model for quantifying the loads on the actuator**

Transmission concept

The key requirements for the rotation of the control shaft include the very high dynamic torques of several hundred Nm originating from combustion engine forces – which are applied in holding condition as well as during the phasing process –and an operation range of the shaft of up to 180 degrees. A high transfer-ratio gearbox is used to adapt the control shaft torque to the capabilities of the electric motor. Another important requirement is avoiding audible noises that may be caused by load reversal within the actuator system. Depending on the type of high-ratio gearbox used, this can make it necessary to implement additional measures.

To determine the gearbox concept most suitable for this task, Schaeffler conducted a comprehensive concept study back in 2014 [2, 3]. The basis for the study comprised gearbox types that are frequently used for actuators with a high transfer ratio. These include eccentric gearboxes with a rigid eccentric wheel, flexible wave type gearboxes or Wolfrom gearsets. The original selection was furthermore expanded to include common standard gearbox types from the epicyclic, stationary and coupling gear families.

The evaluation criteria Schaeffler defined for the benchmark test were as follows:

- Power density (ratio of permissible output torque to required packaging space)
- Dynamic response (acceleration capability of the gearbox during phasing)

6 **Components of the flex pot gearbox**

- Sensitivity to clearance (susceptibility to backlash and, thus, audible noise at load reversal)
- Efficiency of the overall gearbox concept
- Susceptibility to wear (wear tendency of gear pairs)
- Parts complexity (as an indication for the design and manufacturing efforts)
- Parts count (in reference to the main functional components: gear wheels, shafts, bearings)
- Experience with technology (in-house know-how at Schaeffler from an engineering and manufacturing standpoint)

As a result of the benchmark investigation, the flex pot gearbox emerged as the best choice when all characteristics are taken into account. It offers the highest power density and the lowest possible clearance. The concept description is quite similar to the one found in the respective article about electric cam phasing.

In a flex pot gearbox, the driving element is the oval inner ring of an oval ball bearing that is positioned in the interior of the open end of a flexible pot. The oval contour is transferred to the flexible pot outer contour which carries a ring set of gear teeth. Due to the oval form, the set of gear teeth meshes at two opposing points with a hollow gear wheel that is rotationally fixed. Once the oval inner ring of the ball bearing starts rotating the two points of meshing gears between flex pot and hollow gears start traveling along the inner contour of the hollow gear wheel. Due to the fact, however, that the number of teeth between flex pot outer contour and the hollow wheel interior toothing is slightly different a slow rotation of the flex pot occurs. Figure 6 shows an exploded view of the gearbox that has been designed based on this principle. Figure 6 shows an exploded view of the transmission.

In the design process, the engineers leveraged the experience acquired from developing similar gearboxes for electromechanical camshaft phasers and made specific adaptions to the needs of the VCR application. Key technological properties of the exemplary solution developed by Schaeffler include:

- Transmission ratio of 100:1
- High gearbox efficiency of approx. 60 %
- Torque capacity › 350 Nm
- Compact dimensions measuring 100 mm in diameter and 40 mm in width
- Extremely low gear backlash
- Lightweight design weighing just 970 g.

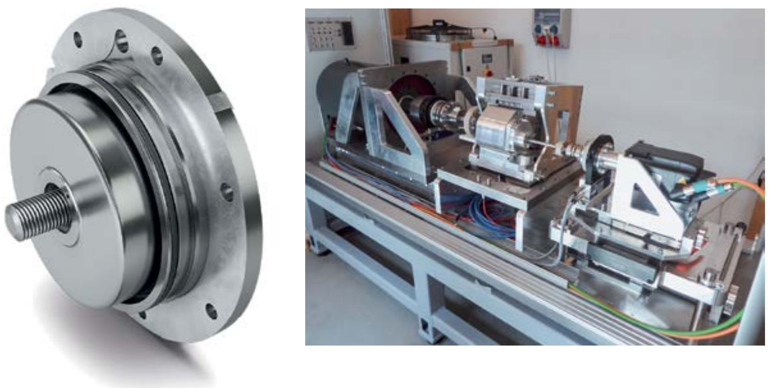
Figure 7 shows a hardware sample for the gearbox that meets the requirements described above. In order to verify the operative function and durability of the assembly, a specific test stand was developed that has sufficient reserve capacity to also accommodate higher-load components of this kind. The development time required to adapt this technology to different combustion engines can therefore be minimized.

Brushless direct-current motor

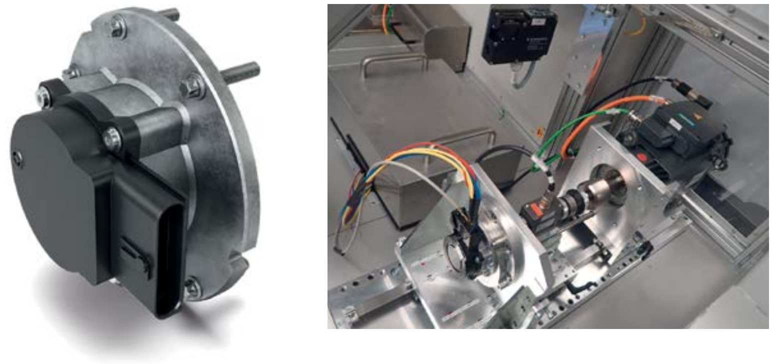
Schaeffler developed a brushless direct-current motor (brushless DC, BLDC) to drive the VCR actuators. Compared to conventional brush motors, BLDC drives offer a longer service life, higher efficiency and produce less heat. The optimized design with respect to packaging space and magnetic, thermal and rigidity requirements as well as vibration resistance was verified in targeted tests conducted on respective test stands, partly specifically developed for this type of components.

The flex pot gearbox already offers a high transmission ratio of 100:1. To further reduce the loads from the crankshaft drive for the electric motor and to minimize the torques as well as the electrical power consumption for holding the VCR in intermediate positions, a spur gear reduction with a ratio of ~2:1 can be integrated in the actuator housing to increase the total ratio of the VCR actuator to e.g. 200:1. The torque reduction by the transmission allows a compact and lightweight electric motor design. Key elements of the operation of the VCR system are precise position detection and a high positioning accuracy of the actuators. The electric motor is therefore equipped with a sensor-controlled block commutation facility, where highly precise Hall sensors determine the rotor position. Based on the required system state the control unit identifies when the BLDC motor phase windings need to be energized. Technical specifications of the electric motor are (Figure 8): Technical specifications of the motor (Figure 8):

- Load torque of 1.7 Nm
- Speed under no load 2,800 rpm
- Compact dimensions measuring 80 mm in diameter and 40 mm in width
- Integrated high precision sensor system



7 **Example of a development sample for the gearbox for the multi-link phasing system and gearbox test stand for component testing**



8 **VCR BLDC motor prototype and test stand for component testing**

During final assembly, the electric motor and flex pot transmission are assembled to form a compact, maintenance-free actuator unit (Figure 9) that Schaeffler can then deliver ready to bolt-on. Depending on customer requirements and the application scenario, the unit can also be integrated in a housing and supplemented with additional components such as an absolute angle sensor.

9 **Motor and transmission are axially interlocked to form a complete unit**

Electronic control unit

The electronic control unit processes the information from the engine management system and realizes the commutation and active regulation of the electric motor for the VCR system. By following a time and cost saving standardized parts concept, Schaeffler currently integrates housing and connectors that have already proven their suitability in similar applications. The specific requirements of the VCR control system made it necessary to design an all-new printed circuit board that could also fit in the available space offered by an existing series production housing (Figure 10).

10 **Combination of a new printed circuit board and a proven housing concept**

A standard CAN bus interface is used for communication with the engine control unit. The overall electronics concept – with a control and sensor system – also incorporates a comprehensive set of monitoring and diagnostic functions. Engineers at Schaeffler also placed special focus on realizing an adequate fail-safe function.

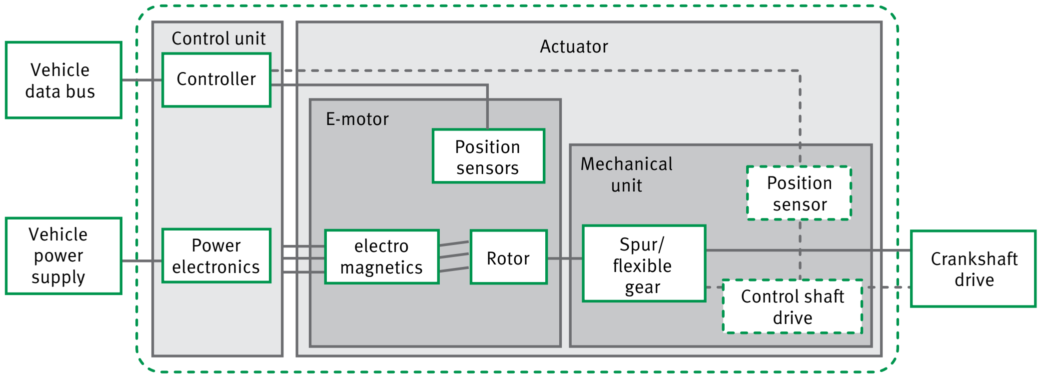
System-based approach for expedited engine integration

Over the years, Schaeffler has achieved a great deal of experience in the development and mass production of complex mechatronic systems in a wide variety of application areas. The company uses this experience to continuously improve its development processes in line with the latest standards that apply in this environment. Figure 11 again summarizes all system components that make up the current development scope and that as a system can be supplied by Schaeffler.

Engine manufacturers thus receive a well-tuned and tested functional unit that can be quickly and easily integrated into an existing engine architecture. By using development tools and methods tailored to the specifics of VCR systems, Schaeffler not only assists customers with mechanical integration with the engine, but also with the software-based application of the VCR functions.

Summary and outlook

In the last few years several engine functions have been "variable-ized" by applying new mechatronic systems. One of the last untouched parameters of engine operation is the geometric compression ratio, which offers significant potential for improving engine efficiency when varied. The compression ratio as it applies to classical gasoline engine designs leads to a conflict of interests as the engine must be trimmed for good performance in either the part-load range or full-load range. The result is typically a compromised overall engine design. This "middle of the road" approach will in many cases no longer be sufficient in light of the new emission and fuel consumption regulations. In future, RDE testing in the complete engine operating map will be required to obtain certification.



11 **Supply scope of the complete system from Schaeffler (green)**

A VCR system enables optimal efficiency in all engine load ranges and at the same time allows adaption to critical combustion or component load conditions. Based on comprehensive experience with other mechatronic systems, Schaeffler has chosen the multi-link VCR concept to be particularly beneficial. Based on the close synergy with electric cam phasing systems Schaeffler is able to offer a comprehensive actuator system. It comprises a very compact, robust and energy-saving actuator with a flex pot type gearbox and BLDC electric motor along with the control system and accompanying sensor elements. Furthermore, Schaeffler is able to strongly support engine manufacturers in integration of the VCR actuation system into their respective engines.

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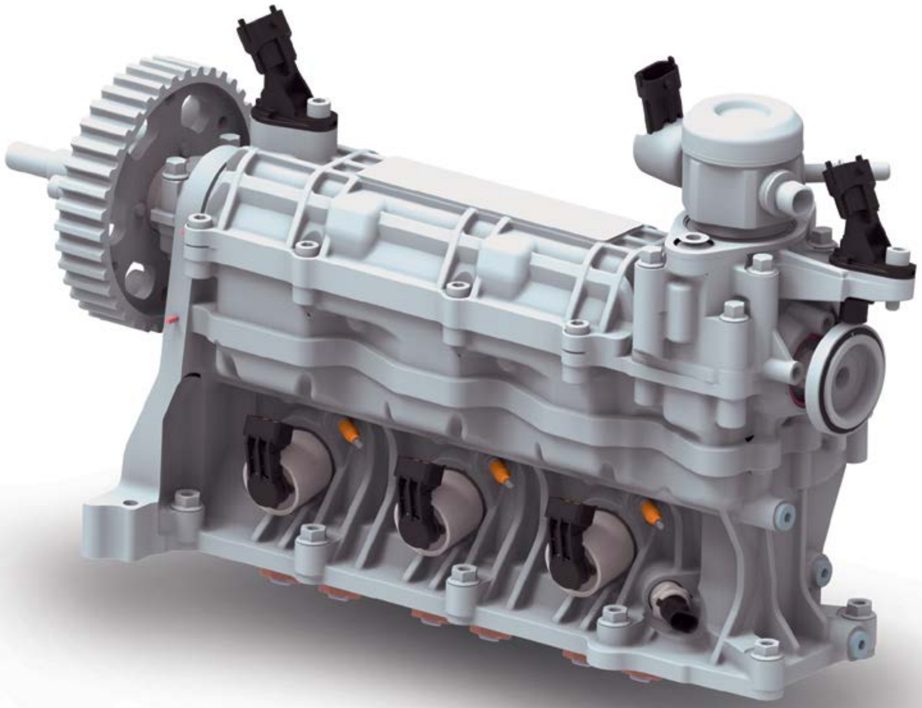


5

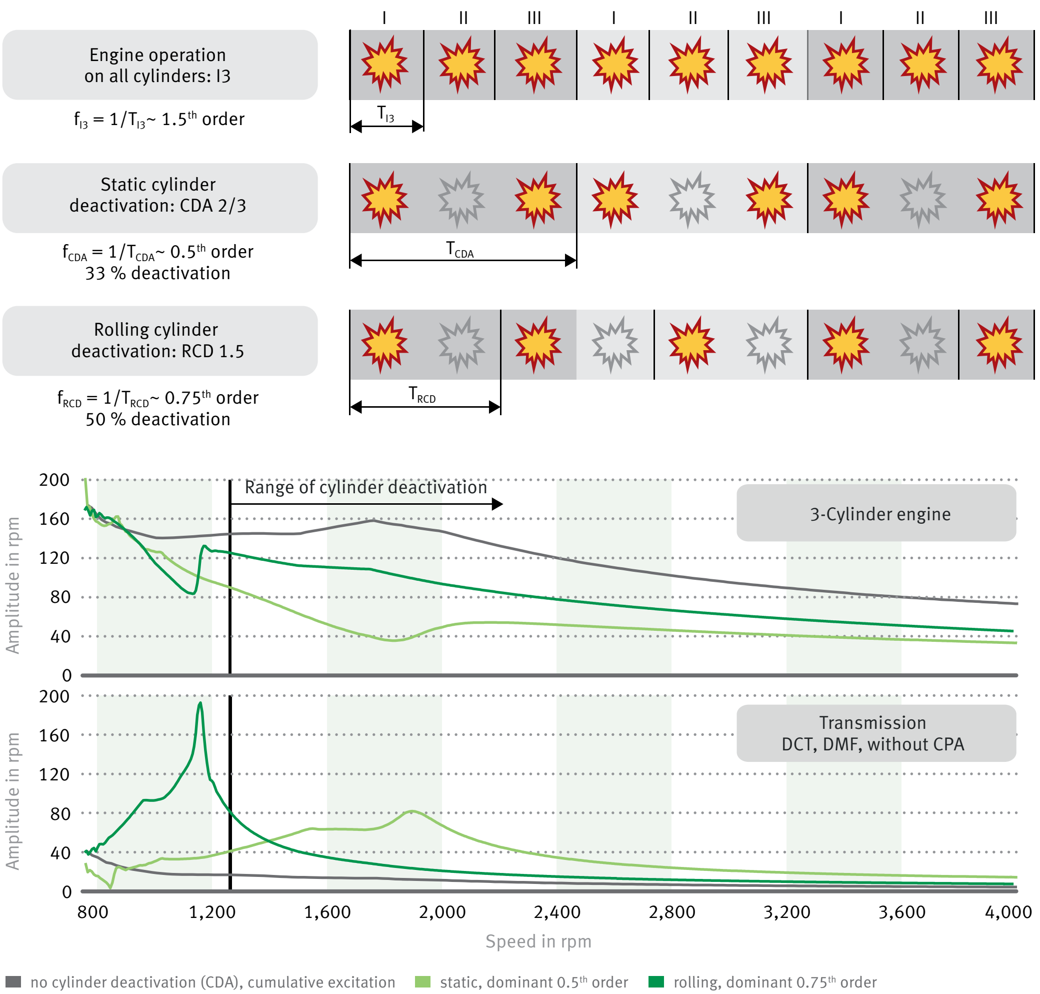
3 Cylinder as an 8 Stroke

Rolling CDA

Dr. Michael Elicker

Introduction

Cylinder deactivation has been discussed ever since the 1980s as a measure for reducing the fuel consumption and, in so doing, CO_2 emissions of gasoline engines. When operated under low-load conditions, the engine management system interrupts the supply of fuel to several cylinders of the engine and deactivates their ignition. The affected cylinders will then only be towed. With systems that realize variable valve timing such as switching or switch-off elements (switchable tappets, switchable roller finger followers, collapsing pivot elements and cam shifting systems) as well as fully variable systems such as the UniAir electrohydraulic system from Schaeffler, the valves of the deactivated cylinders also remain closed to minimize charge cycle losses and to prevent air from being pumped through. The fuel-saving effect of cylinder deactivation is achieved by shifting the load point to the remaining cylinders that are still actively firing. This results in a larger filling and a corresponding de-throttling of the intake system. The increased charge reduces wall heat losses during the high-pressure process, and the de-throttling effect minimizes charge exchange work. Both factors considerably improve the effective operating efficiency of the engine at constant torque output. Until just a few years ago, cylinder deactivation concepts were typically implemented for large engines with six or eight cylinders. Depending on the system used, cylinder deactivation involved either switching off an entire bank of cylinders or individual cylinders of both banks on a V-engine configuration. Due to ever stricter CO_2 emissions limits, cylinder deactivation systems are also being increasingly used on smaller inline engines with four or, in some cases, only three cylinders [1, 2].



1 Deactivation level and torsional vibration excitation of different deactivation strategies for three-cylinder engines

Vibration analysis for a three-cylinder engine

Vibration excitation is of critical concern, however, particularly for cylinder deactivation on three-cylinder engines. As a result of the irregular firing interval, statically deactivating a cylinder results in a dominant 0.5th engine order as the ignition sequence only repeats after two rotations of the engine. By integrating corresponding damping concepts – such as Schaeffler's combination of a dual-mass flywheel with specially adapted spring characteristics as well as centrifugal pendulums – it is possible to reduce the excitation of the drivetrain to an acceptable level, even at very low engine speeds. By contrast, a regular firing interval of 480° CA as is associated with an alternating cylinder deactivation system produces a more manageable dominant 0.75th engine order, whereby integrating a combination of dual-mass flywheel and centrifugal pendulum also extends the potential range of application down to lower engine speeds. To evaluate these potentials for the three-cylinder engine, Schaeffler analyzed the concept of rolling cylinder deactivation (RCD) by conducting simulation calculations and experiments on a fired engine. In the process, the individual cylinders of the engine are deactivated at periodic intervals.

This, then leads to "1.5-cylinder operation" with an overall deactivation percentage of 50 %. Due to the alternating sequence of fired and unfired cycles in a cylinder, the excitation repeats after two cylinders are rolled through. This leads to the aforementioned doubled firing interval of 480° CA compared with engine operation on all available cylinders. The periodic vibration excitation repeats after two-thirds of a full camshaft revolution instead of one complete revolution as would be the case if a permanently designated cylinder were to be deactivated in a static system.

In addition to the improved vibration excitation performance, the even greater potential for reducing fuel consumption represents the main benefit of rolling cylinder deactivation as is afforded by the higher deactivation percentage of 50 % as opposed to 33 % with static cylinder deactivation (Figure 1). This higher percentage, however, also means that the limiting threshold for torque output, or the mean effective pressure in the engine map for cylinder deactivation, is lower.

Operating strategy for a rolling cylinder deactivation system

In the context of the aforementioned benefits offered by these systems, Schaeffler has been working for some time on rolling cylinder deactivation concepts that can be applied to a three-cylinder engine [2]. In the process, it was also investigated which operating strategy leads to the biggest fuel savings. Current series cylinder deactivation systems typically confine fresh air in the deactivated cylinder, compress it and expand it without combusting. An alternative approach would be to confine the residual gas or deactivate an already evacuated cylinder. Since the intake and exhaust valves are closed on a deactivated cylinder, there is also no charge cycle present, meaning that the cylinder passes through the compression and expansion phases twice – without combusting – during one camshaft revolution. By contrast, an actively fired cylinder completes the conventional four-cycle process by inducting, compressing, igniting and working, and ejecting the combusted gas. The excitation of a deactivated cylinder therefore occurs twice per camshaft revolution, whereas that of a fired cylinder occurs once only [1].

In order to estimate the potentials of the three feasible operating strategies for rolling cylinder deactivation on a three-cylinder engine, Schaeffler subjected each strategy to a comparative simulation study. Figure 2 charts the calculated effects on fuel consumption at a steady-state operating point of 2,000 rpm with a mean effective pressure of 2 bar.

Coming in at a 12.5 % reduction in fuel consumption, the RCD strategy with evacuated cylinders (Figure 2, right) offers additional savings potential as compared to the roughly 10 % reduction achieved with static cylinder deactivation (CDA) of the second cylinder, both with reference to full engine operation on all cylinders. This can be explained by the distribution of efficiency loss across the individual operating modes in the top section of the graph. A conventional cylinder deactivation system is already capable of considerably reducing the pumping losses that occur during operation on all cylinders. This is the main reason for the reduced consumption. The quantity of gas confined in the cylinder, however, which results from a gradual pressure equalization over the blow-by effect, has an additional, negative impact on efficiency due to the continuous compression and expansion phases and the ensuing wall heat loss. In a rolling cylinder deactivation setup with confined residual gas, the temperature and pressure of the confined gas are very high. This, then, dramatically increases wall heat loss as the elevated internal cylinder pressure produces a high-volume blow-by mass flow. The result is an even higher fuel consumption, at +12 %, than during operation on all cylinders, which means that the residual gas confinement strategy is not at all suited to implementing a rolling cylinder deactivation system. When operating with a fresh air fill level, fuel consumption can only be reduced by approximately 4 % compared to operation on all cylinders despite the fact that losses are considerably lower during deactivation. The problem here is the slow combustion that results from the complete break-up of the tumble flow due to the two intermediate compressions of the fresh gas mixture. This reduces high-pressure efficiency by around 3 %.

2 **Fuel-saving potential of different operating strategies for cylinder deactivation in steady-state operation**

During rolling deactivation with cylinders evacuated by a suitable valve train, the losses accumulated during the unfired working cycles are reduced to a minimum. This approach decreases fuel consumption over a conventional cylinder deactivation setup by an additional 2 to 3 % at the operating point in question. In the test engine, the residual exhaust gas of the fired working cycle, combined with the deactivated and evacuated cylinders, was under 10 %. As such, the total residual exhaust gas was only half as much as during operation on all cylinders or with static cylinder deactivation. The reason for this is that the intake and exhaust valves are not simultaneously open at any time, greatly minimizing the possibility of actively controlling the internal residual gas fraction. This points to the idea that improving the control of the residual gas fraction in combination with the rolling cylinder deactivation system can lower fuel consumption still further.

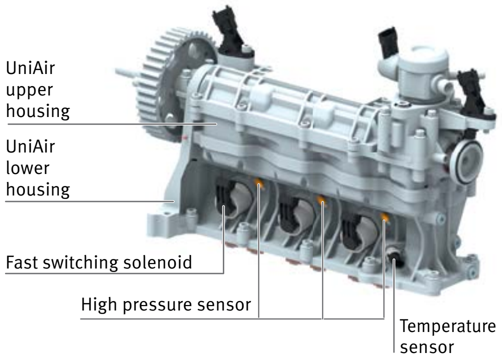
Design of a test engine

The promising results of the simulation study motivated Schaeffler to analyze the potential of a rolling cylinder deactivation system on a physical test engine subjected to a comprehensive series of fired dyno tests. The base engine used for the test is a 1.0-liter three-cylinder gasoline engine from Ford. This engine has a factory-installed direct-injection system, a turbocharger and hydraulic phase adjusters for camshaft phasing on the intake and exhaust sides. The valves are actuated by bucket tappets in the engine generation used for the tests. In the meantime, a new, revised version of the engine is also available that uses finger followers in the valve train.

To integrate a rolling cylinder deactivation system in the test object as physical hardware, the valves of all three cylinders in the engine must be able to be controlled independently of each other. The minimum requirement for this is implementing a zero stroke in all valve actuation sequences by way of switching elements. Greater flexibility in realizing a fuel-saving, low-emissions operating strategy is available in the form of a continuously variable valve train on the intake side that can be configured with almost no restrictions for the valve lift and valve timing sequences. Back in 2009, Schaeffler launched its electrohydraulic UniAir system, which offered a very versatile approach to implementing a wide range of valve timing strategies for series-production applications. Not only can the system be used to deactivate cylinders, but it can also de-throttle the engine via early intake valve closing (EIVC) or late intake valve closing (LIVC) strategies as well as via variations in valve lift height such that the overall benefits of the system outweigh the on-cost associated with the additional hardware. Basic mechanical systems on the exhaust side that switch between full lift and zero lift are sufficient for the rolling cylinder deactivation system. These can take the form of switching roller-finger followers or switching bucket tappets.

To convert over to the rolling cylinder deactivation system, the test engine was fitted with a UniAir system on the intake side. Switching bucket tappets on the exhaust side replace the standard series-production parts. The camshaft phase adjusters on the intake and exhaust camshafts of the base engine were set to an optimized but fixed valve timing. Realizing the design of the new valve actuating system for the prototype required major modifications to the base engine. These modifications affect the cylinder head, belt drive and engine periphery (Figure 3).

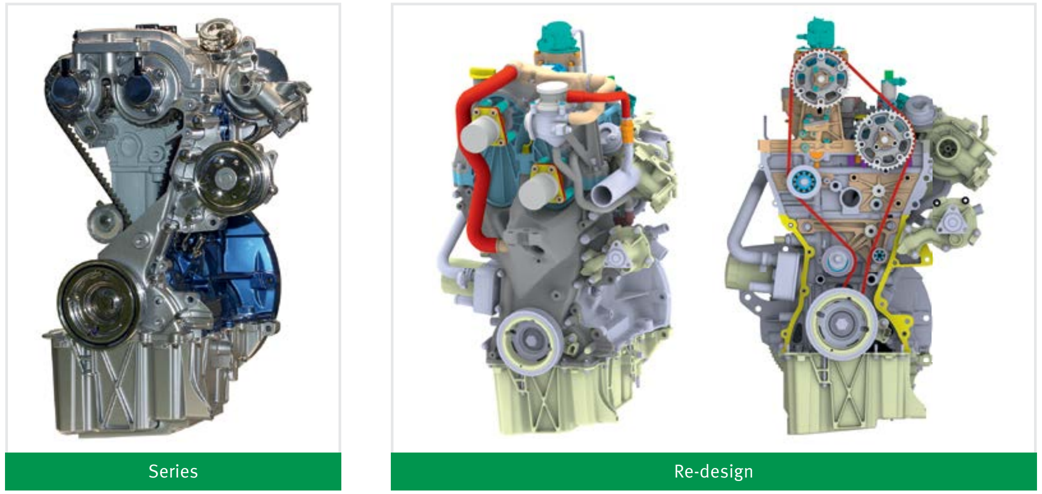
The complex redesign of the cylinder head was carried out in collaboration with Ford and a deve-



4 UniAir cylinder head module

lopment service provider. In the process, the entire factory-fitted valve train was removed from the intake side and replaced with the aluminum UniAir module. The module encompasses all components for actuating the intake valves – including the UniAir actuator system, camshaft and associated bearing mounts – in a single unit (Figure 4). Oil ducts are used to connect the module to the engine's lubrication system.

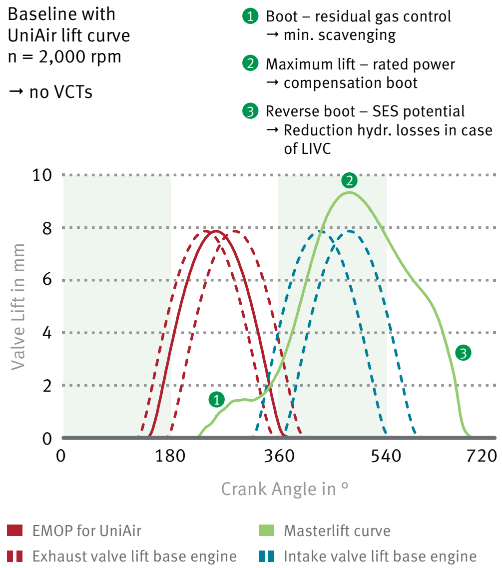
The relatively tall design concept on the intake side as compared with a series-production solu-



3 Comparison between series production and test engine

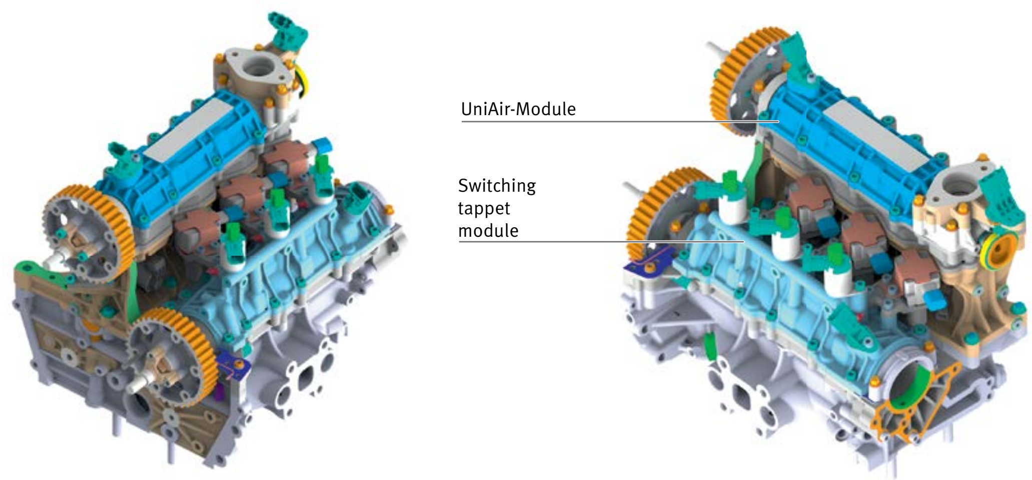
tion was mainly chosen for the prototype because it allows the UniAir module to be integrated into the cylinder head without having to make any modifications to the intake duct geometry or positional arrangement of the valves, spark plugs and injectors. This is also the reason why the standard bucket tappets were merely replaced with switchable ones on the exhaust side. Additional ducts milled into the camshaft carrier connect these switching tappets with the hydraulic circuit of the cylinder deactivation system, whereby the tappets are actuated independently for each cylinder via separate electromagnetic oil flow control valves. Figure 5 shows the modified cylinder head with both valve train modules.

Figure 6 plots the valve lift curves that were realized with the new valve train system. The dotted lines represent the lift curves of the series production engine, while the solid lines show the valve lift after the conversion. On the production engine lift curves, you can see the possibility for shifting the opening and closing times of the valves via cam phasers. The exhaust timing of the test engine, on the other hand, is predefined, or fixed, while the UniAir system ensures a high de-



6 Comparison of valve lift curves for the base engine and converted RCD test engine

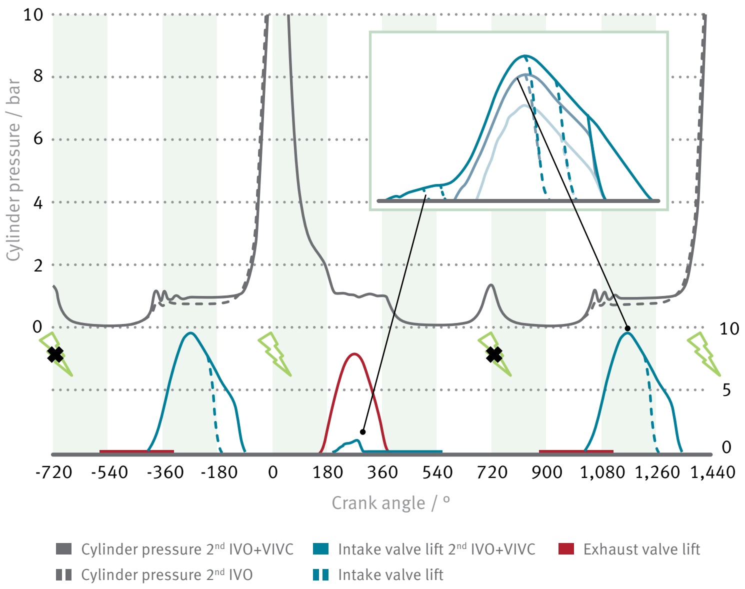
gree of variability on the intake side as the green line in Figure 6 indicates. This line delineates the maximum valve lift that is possible with the UniAir system, meaning that the valve lift can assume almost any pattern below this line.



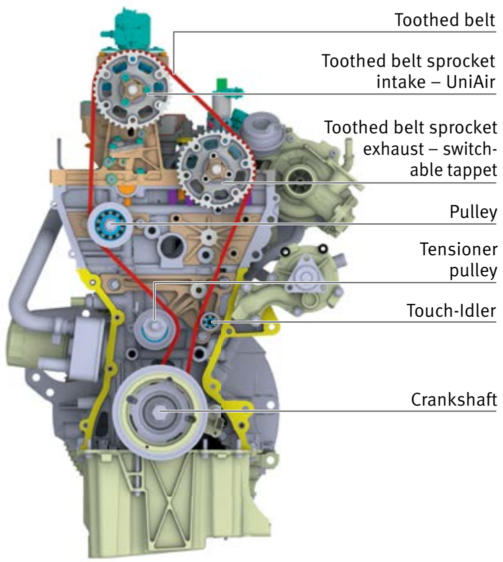
5 Cylinder head with valve train modules for the intake and exhaust sides

This flexibility lays the foundation for de-throttling the charge cycle by way of EIVC or LIVC strategies. The intake valves can also be quickly opened with minimal lift at point 1 to control the residual gas – relative to the engine load – by realizing a small advance boot cam elevation. To prepare for cylinder deactivation, the residual gas in the target cylinder is pushed into the intake manifold and held there for the next fired working cycle. During the next intake valve lift, this gas enters the combustion chamber together with the fresh gas to condition the air-fuel mixture. At the same time, the fired cylinder is further de-throttled using an EIVC or LIVC strategy as shown in Figure 7.

By establishing optimal charge mixture conditions, the stability of the combustion process and the raw emissions in the first working cycle following a deactivation are positively impacted. Since the intake camshaft of the test engine rests higher than that of the production engine due to the retrofitted UniAir module, the entire belt drive had to be redesigned along with all idlers and tensioner pulleys (Figure 8). As the new layout was being engineered, Schaeffler leveraged its vast in-house experience in developing and applying customer-specific drive solutions for valve trains with matching timing belt drive units.



7 **Strategies for controlling the residual gas and intake valve lift with the UniAir system for rolling cylinder deactivation**



8 **Redesigned timing belt drive**

Additional modifications focused on the ventilation system of the engine and the fuel injection and ignition components. An additional aluminum front engine cover provides shielding protection of the timing drive.

The modified cylinder head of the test engine was placed on a motored test stand to laser-measure the valve lift. This test forms part of the established procedure that Schaeffler uses when integrating the UniAir system for a new engine. To this end, the solenoid valves of the UniAir actuator system are actuated as the camshaft rotates, and the valve lift is recorded relative to the crank angle. Using a laser to take measurements ensures the highest measurement resolution. Determining the correlation between the actuation of the electrohydraulic solenoid valves and the resulting valve lift in the context of the target application and engine conditions lays the groundwork for the calibration of the UniAir system – designed as a smart actuator – which is provided by Schaeffler along with the corresponding control software.

The tests on the engine should make the best possible use of the potential of variable intake control by the UniAir system. This includes de-throttling the charge cycle by realizing smaller valve lifts during part load operation. If the combustion chamber is not modified accordingly, however, there is the risk of the air-fuel mixture no longer being conditioned sufficiently due to a drop in charge motion. In a simulation-assisted design process, masking contours were developed for the combustion chamber near the intake valves and tested by taking measurements on a flow test stand. The additional material added in the combustion chamber minimally increases the compression ratio of the engine from 10.1 to 10.4. Subsequent engine tests confirm the positive effects of the masking measure.

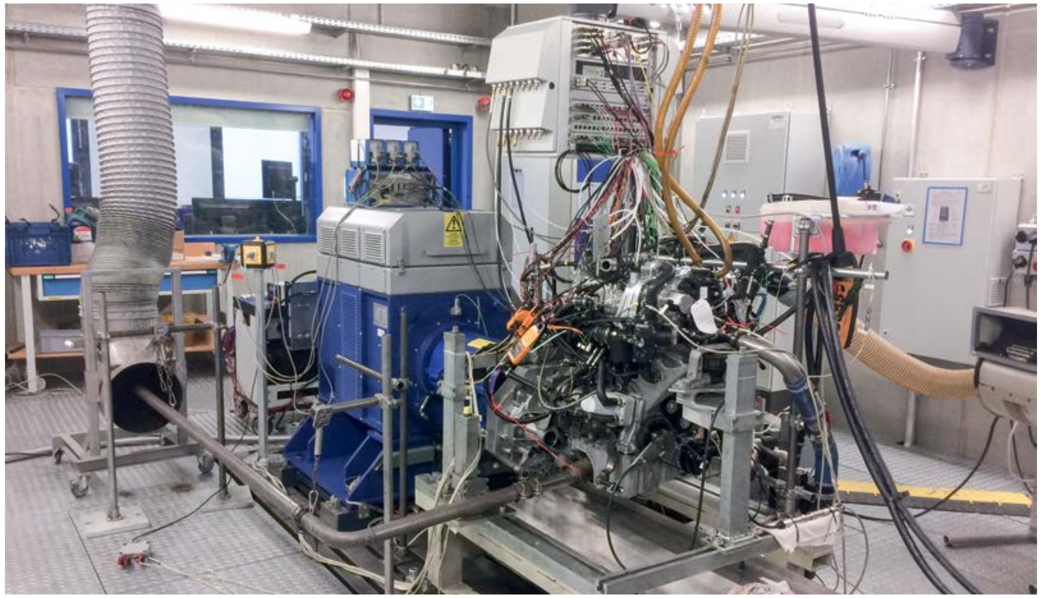
Fired engine test

The test stand trials with the fired prototype engine (Figure 9) were carried out by Schaeffler together with a development service provider as were the preceding thermodynamic simulations.

As the test stand was being designed, special focus was placed on ensuring the use of comprehensive measuring equipment so that the internal engine processes and their impact on emissions could be quantified during rolling cylinder deactivation. The following measuring equipment was used:

- AVL Indiset system (high-pressure indexing)
- AVL Mexa emissions measurement
- AVL 489 particulate meter
- Fuel consumption measurement system
- IAV KIS knocking detection system
- Sensitec valve lift measurement system from Dewetron
- Various pressure and temperature sensors (including low-pressure indexing)
- Oxygen sensor and scanner
- Speed sensor for the exhaust turbocharger
- AVL Sensyflow flow meter for fresh air

A special highlight is the additional measurement of the valve lift heights during fired engine operation through the use of inductive sensors. The harmonious interaction of all modules along the measurement chain produced reliable, highly reproducible results of all combustion-relevant parameters throughout the duration of the test bench campaign. This, in turn, made the processes in the combustion chamber transparent directly at the first ignition point after deactivating the cylinder. This phase is regarded as the neuralgic point of rolling cylinder deactivation as unlike a conventional deactivation system, the cylinders must fully contribute to producing the engine's rated torque at every alternating firing point without misfiring. What helps is that the cylinders in a rolling deactivation system do not cool down as



9 **Set-up of the test bench for the fired engine tests**

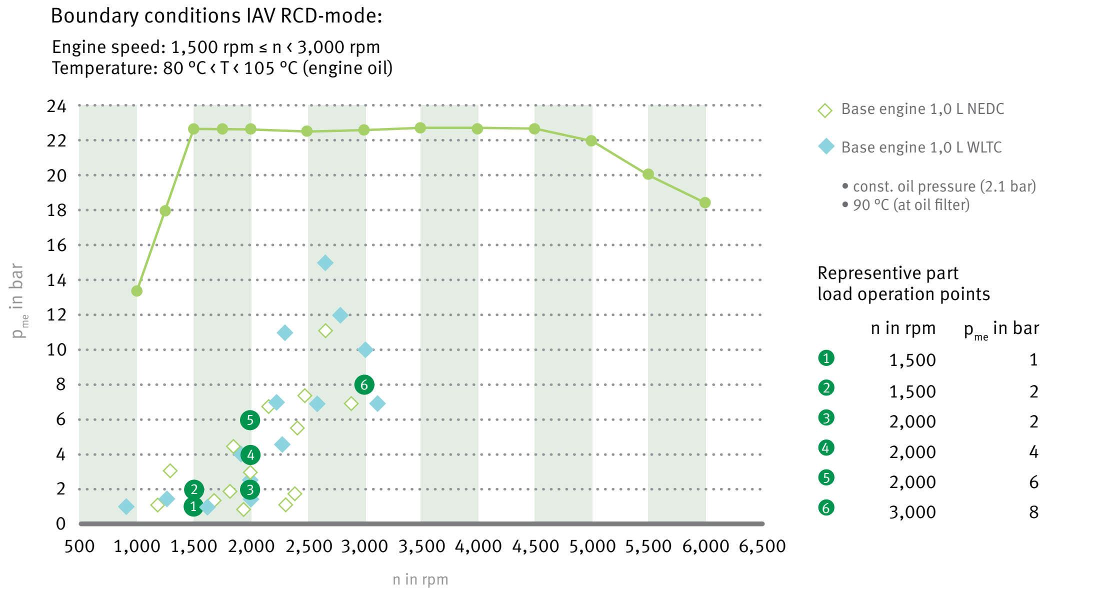
much because they will very soon fire again during the next camshaft revolution.

Measurement results

To obtain comparable results, the engine was run on the test stand with conventional and rolling cylinder deactivation. Figure 10 depicts the test program for the measurement series, which included six engine map points ranging from the low to medium load/speed range. These map points were run at steady state conditions. The conditions for the individual tests – such as the ambient temperature, engine temperature and oil pressure – were identical in each case.

The question as to which cylinder should preferably be deactivated during static cylinder deactivation depends on many engine-specific factors so that it is not possible to provide a definitive answer that covers all applications. With our starting point being Ford's series production three-cylinder engine, it was discovered that Schaeffler's test engine was best operated by always switching off cylinder one [3].

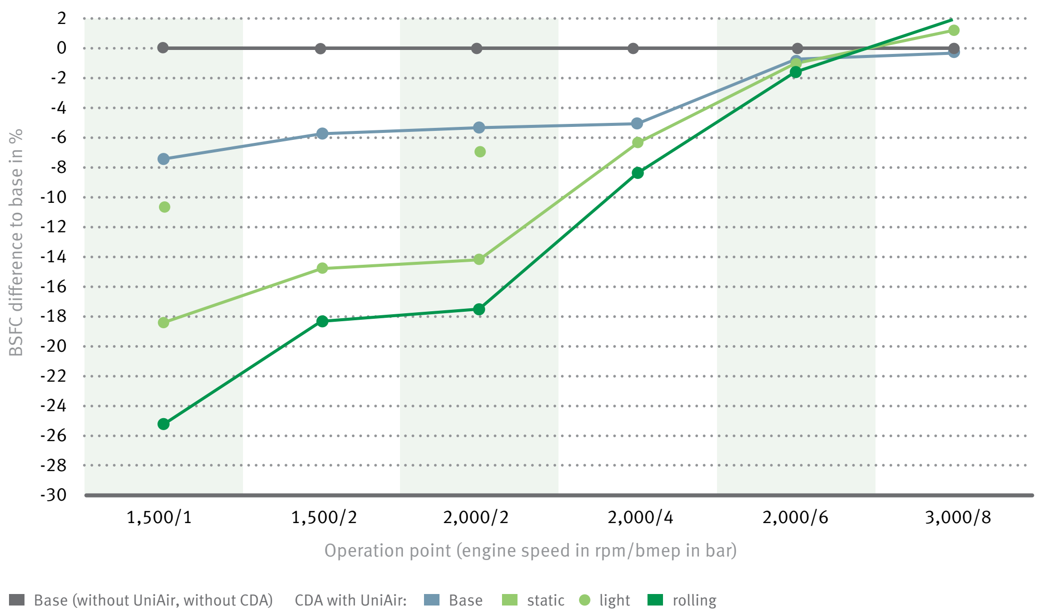
Figure 11 shows the relative reduction in fuel consumption measured at the tested engine map points taken from Figure 10. Depicted are the percentage-based differences in specific fuel consumption compared to the base configuration (in black), which is characterized by full-cylinder operation without the use of the de-throttling potentials afforded by the UniAir system on the intake side in the form of an EIVC or LIVC operating mode. In the application examined here, this corresponds to the factory-tuned operating state of the test engine, which has a conventional charge exchange strategy with two camshaft phasers. The line plotted in blue in Figure 11 charts the optimized fuel consumption running an EIVC configuration via the UniAir system on the intake side,



10 **Operating points of the testing program across the engine performance map**

which further de-throttles the engine in the part-load range. This strategy can reduce fuel consumption by approximately 5 % without cylinder deactivation. Shown in bright green is the progression for the application involving static cylinder deactivation and EIVC. At low loads, considerable additional fuel-saving potential was realized compared to the base configuration as well as the configuration involving the EIVC strategy only. The two round bright green dots in Figure 11 indicate the results of an additionally examined "low-cost" approach to cylinder deactivation. In this configuration – referred to here as the "light" version of static cylinder deactivation – the UniAir system continued to close the engine intake valves at the cylinder deactivation point, whereas the exhaust valves opened in their standard sequence (i.e. were not switched to the zero lift position). Despite the resulting friction and charge cycle losses, the measured reduction in fuel consumption compared to an EIVC strategy without cylinder deactivation was approximately 2.5 %.

This strategy can be implemented in engines already fitted with a UniAir system on the intake side and a rigid valve train on the exhaust side without significant additional design changes required, whereby part of the potential of a static cylinder deactivation system can be achieved without oncosts of the valve train. As the green line illustrates, a rolling cylinder deactivation system tuned in line with the design concept outlined here can lead to extensive additional fuel savings. The measurement results largely confirm the possibilities for reducing the fuel consumption by RCD that were predicted during the simulation exercises. When the engine is run in RCD mode at low to mid operating speeds/loads, substantial savings of 15 to 20 % can be realized, which then taper off as the engine transitions to higher loads. The ideal application range for deactivating cylinders on the test engine is below approximately 2,200 rpm and 5 to 6 bar mean effective pressure. When the engine is run at higher load levels, the fired cylinders are almost entirely



11 **Impact of EIVC and cylinder deactivation on specific fuel consumption**

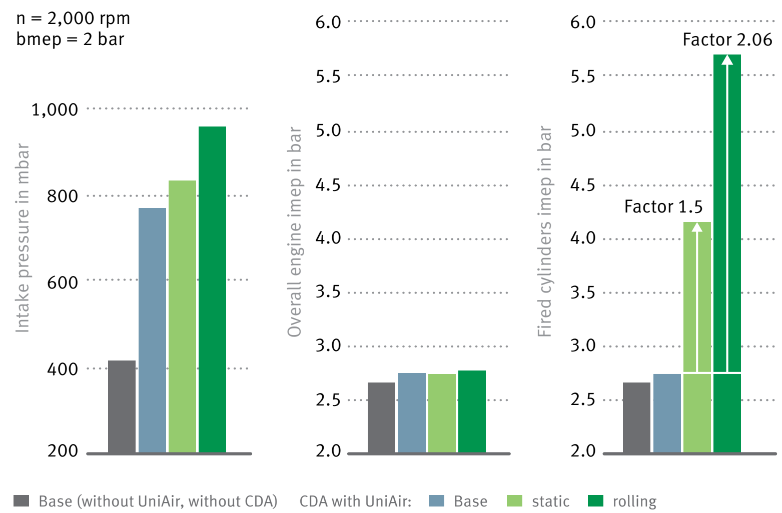
de-throttled; as such, shifting the load point further by CDA/RCD will not continue to reduce fuel consumption.

Figure 12 (left) compares the charge cycle losses of the different test configurations. While the standard application without EIVC and cylinder deactivation involves heavy throttling with approximately 400 mbar of intake pressure, the other applications reveal a substantially lower difference to ambient pressure (approximately 1,000 mbar) and, thus, lower throttling losses. As expected, the rolling cylinder deactivation system offers the most pronounced de-throttling effect. As you can see in the center bar chart in Figure 12, the indexed mean effective pressure was at a comparable level for all concepts tested. Since all cylinders are permanently activated in the base and EIVC application, the mean effective pressure level from Figure 12 (center) also corresponds to the mean effective pressure of the fired cylinders in Figure 12 (right). With static cylinder deactivation, two cylinders are always actively running (equates to a deactivation percentage of 33 %), while an engine with rolling cylinder deactivation has one or two cylinders deactivated, depending on the cycle (equates to a deactivation percentage of 50 %). Averaged out, this is consistent with operating the engine on 1.5 cylinders. Load point shifting is considerably more effective in a rolling cylinder deactivation system, which also becomes apparent by the 200 % higher mean effective pressure of the fired cylinders as measured.

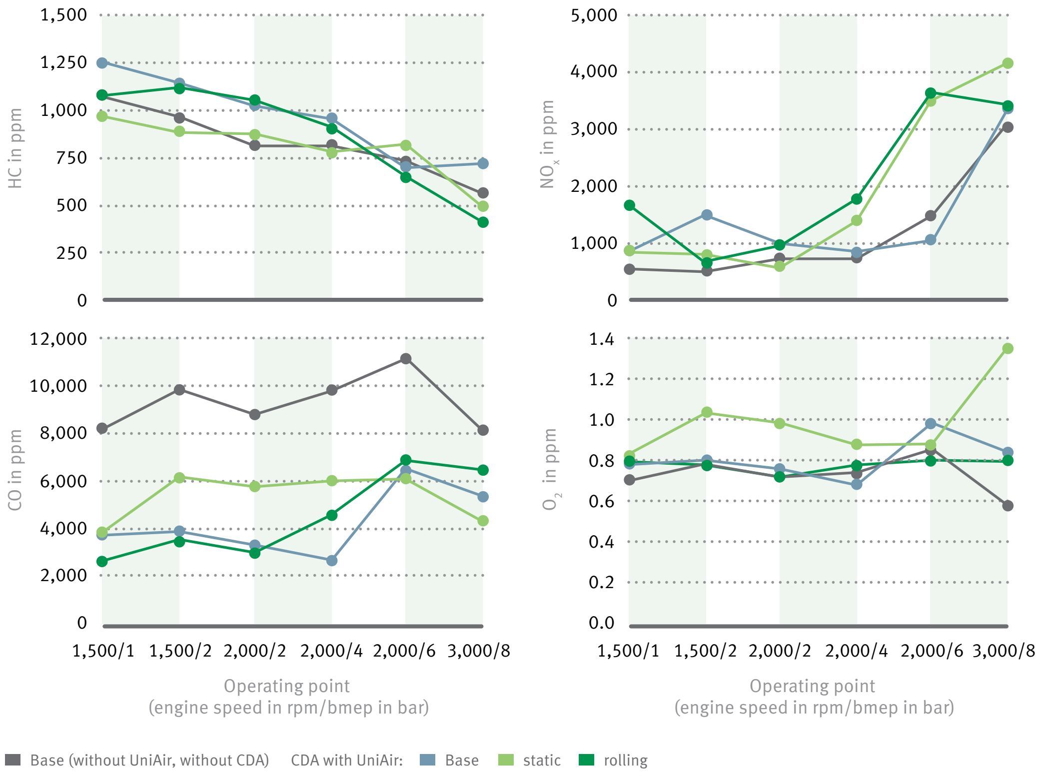
The effects of the engine modifications and the different operating strategies on the gaseous raw emissions at the engine map points from Figure 10 are depicted in Figure 13. Here, too, the basis for comparison is the production engine in factory configuration, without any modifications to the valve train or combustion chamber via masking

(in black). The blue line represents the mechanically modified engine without cylinder deactivation but with an EIVC strategy for partial de-throttling. The raw engine-out emissions measured for static cylinder deactivation on cylinder 3 are plotted in bright green in Figure 13, whereas emission levels with rolling deactivation are plotted in green. You can clearly see that untreated HC, NO_x and CO emissions and the O_2 percentage in the exhaust gas are at a comparable level with both static and rolling cylinder deactivation. The deactivation strategy used, then, does not have a major impact on pollutant emissions. When it comes to HC emissions and O_2 concentrations, this also applies to an operating strategy that does not incorporate cylinder deactivation, regardless whether the production engine or modified engine is used. Particularly noteworthy are the relatively high CO emissions of the base engine as compared to the modified test engine. Since this effect is independent of the engine operating strategy, it stands to reason that the differences can be attributed to the combustion chamber masking that takes effect via early intake valve closing in conjunction with a de-throttling strategy. Both cylinder deactivation strategies revealed slightly higher, yet uncritical, untreated NO_x emission levels –particularly at engine map points 4 and 5. This is apparently the result of shifting the load points of the remaining operative cylinders during cylinder deactivation.

Figure 14 shows the results of the raw engine-out particulate matter emissions during steady-state deactivation mode. At the majority of the engine map points analyzed, the particulate matter generated with cylinder deactivation is either lower than or at the same level as that of the full-cylinder engine, regardless whether static or rolling deactivation is used. Only under very low-load conditions or engine speeds slightly higher particulate counts were measured, which could be reduced within the engine by adapting the injection strategy. Overall, the particulate count can be regarded as uncritical, especially in light of the upcoming particulate filters that will be fitted to gasoline engines in the wake of the new RDE (Real Driving Emissions) emissions regulations. During the transient switchover back to operation on all cylinders, which was not measured here, a static cylinder deactivation system is likely to produce a higher amount of particulate matter as the cooled down and unscavenged cylinder is reactivated – an effect that is not expected to be observed with rolling cylinder deactivation due to the merits of the continuously alternating operation sequence of the cylinders.



12 **Measurement results: Intake manifold pressure and indicated mean effective pressure for different operating strategies**

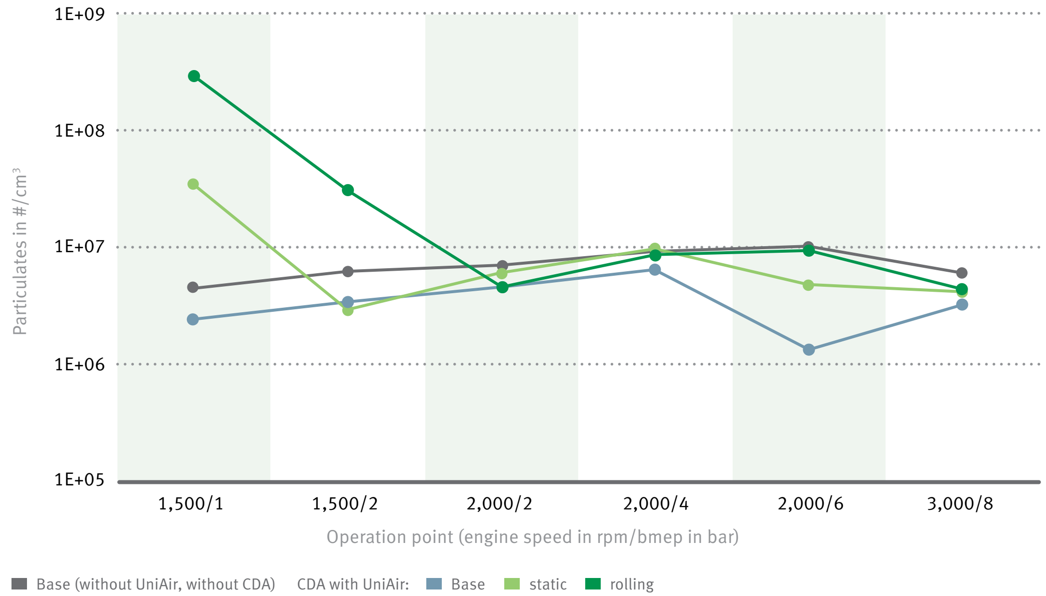


13 **Measurement results of gaseous untreated emissions and O_2 concentration levels**

Summary and outlook

Cylinder deactivation has emerged as an effective way to optimize the fuel consumption of an engine and therefore reducing CO_2 emissions during cycle-based measurements as well as measurements taken when driving in real driving operation. To this end, one or more cylinders are deactivated while the engine is running at low load levels by interrupting the ignition and injection sequence as well as deactivating the intake and exhaust valves. The savings potential is yielded by shifting the load points of the remaining active cylinders at the point of deactivation. As initial product launches have already proven successful, the market will be seeing an increased number of three-cylinder engines fitted with this technology as time goes on. These powertrains do have a weak point, however, which is the pronounced vibration excitation of the dominant 0.5th engine order. One suitable measure for isolating this critical engine order from the rest of the drivetrain is Schaeffler's proposed combination dual-mass flywheel and centrifugal pendulum.

By integrating the concept of rolling cylinder deactivation as discussed in this paper, which involves deactivating individual cylinders at periodic intervals, it is possible to shift the base excitation frequency from the 0.5th order to a more manageable 0.75th order. This excitation can also be further minimized by adding a suitable damping package comprising a dual-mass flywheel and centrifugal pendulum. The rolling mode likewise increases the deactivation percentage of the individual cylinders to 50 as opposed to 33 with static cylinder deactivation. Although this higher deactivation percentage shifts the load range in the engine performance map at which cylinder deactivation is practical to correspondingly lower torque and mean pressure levels, the potential for reducing fuel consumption still further is accentuated with rolling cylinder deactivation.



14 **Measurement results of raw engine-out particulate emissions in steady-state deactivation mode**

To implement the concept in a practical application, Schaeffler used simulation calculations to analyze three different operating strategies for rolling cylinder deactivation, which involved fresh air and residual gas confinement as well as an evacuated cylinder. Calculations revealed that the latter is the most favorable and formed the basis for a comprehensive range of fired engine investigations. The engine used started out as a mass-produced three-cylinder engine that was then modified considerably to accommodate the rolling cylinder deactivation function. These modifications included integrating a UniAir valve control system on the intake side, a switching tappet system on the exhaust side and a new timing belt drive. The operating parameters of the engine were quantified during the test using comprehensive measuring equipment. When the test results obtained are compared with the simulation calculations made for the rolling cylinder deactivation system, the potential estimated fuel savings are validated from a qualitative and – within the limits anticipated – also from a quantitative perspective [2]. The higher cylinder deactivation percentage of the rolling cylinder deactivation system likewise shifts the load range points of the fired cylinders to a higher output level in practical engine operation so that substantial fuel savings of 15 to 20 % can be achieved at low – to mid-range operating speeds and loads. When it comes to reducing emissions, the initial firing of a cylinder after deactivation is particularly critical. Schaeffler has counteracted this with what is known as boot cam elevation. In the process, residual gas is retained in the inlet duct before cylinder deactivation, then enters into the combustion chamber together with the fresh gas during the intake stroke of the subsequent fired working cycle. This approach also facilitates on-demand control of the residual gas (including further de-throttling via variable intake valve closing) in rolling cylinder deactivation mode to optimize the mixture formation and stabilize combustion. The end result is a level of performance under all-cylinder engine operation that offers comparable raw engine-out emissions which, overall, can be regarded as inconspicuous.

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6

Schaeffler eRocker System

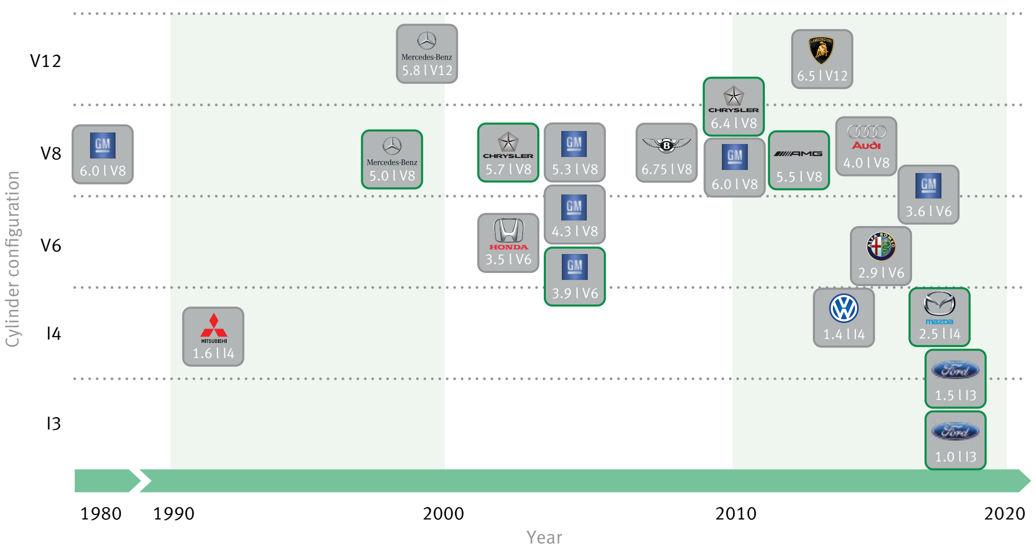
New Concepts
for Switchable Roller

Dr. Frank Himsel

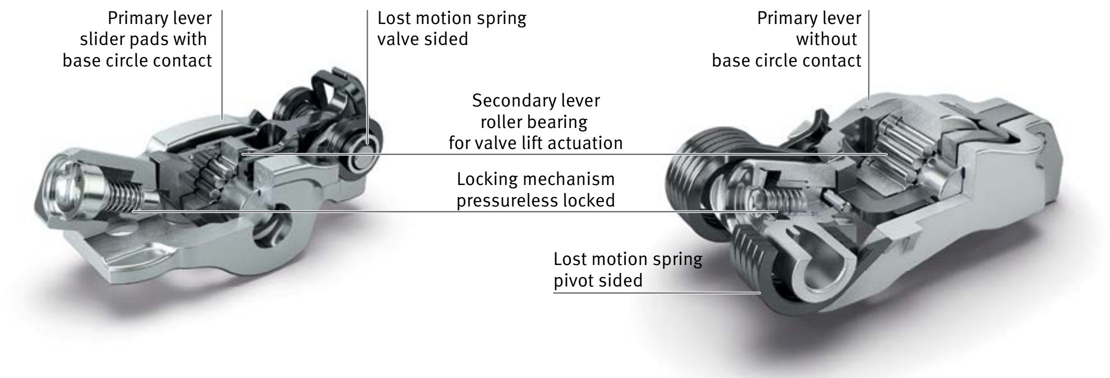
Introduction

For emissions and fuel consumption reasons, switchable valve train components have enjoyed widespread application in gasoline engines for many years. Whereas the first applications were limited to cylinder deactivation on large displacement gasoline engines, the application range of this technology has been expanded in recent years [1]. On the one side, focus is being placed on intake de-throttling via cam profile switching – such as for early intake valve closing (EIVC) or late intake valve closing (LIVC) strategies. On the other side, the concept of cylinder deactivation is now also being applied to engines with less cylinders and smaller displacement (Figure 1). Early 2018 saw the first three-cylinder engine with cylinder deactivation enter series production [2], more applications are under development. Switchable roller finger followers (SRFFs) make it possible to switch between two valve lifts (cam profile switching) or – for a cylinder deactivation – from full lift to zero lift. This makes these elements applicable for the tasks listed above. Switchable roller finger followers also produce less frictional loss compared to other systems.

Diesel-engine concepts involving switchable valve train components have only been implemented in very recent times. By using switchable roller finger followers on the exhaust side of the diesel engine, engineers can implement an internal exhaust gas recirculation function that improves combustion stability during a cold start. It also promotes initial heat build-up in the exhaust system to minimize the time until the "light-off" point is reached for the exhaust gas aftertreatment components. As internal exhaust gas recirculation also quickly responds to parameter changes in transient engine operation, it improves compliance with emissions limits in dynamic situations as well. This paper examines the latest developments in the field of switchable roller finger followers and their application for gasoline and diesel engines.



1 **Product launch of gasoline engines with cylinder deactivation**



2 **Switchable roller finger followers on an early generation (left) and on the current generation (right) for gasoline engines**

Development of switchable roller finger followers for the gasoline engine

In the last 20 years, the trends for variable valve train systems on a gasoline engine have continued to change. This has primarily been in response to economical, environmental and legal requirements but also pertains to the technical development of the systems over time, particularly concerning controllability and switching speeds. Schaeffler has continued to push the concept of switchable roller finger followers for gasoline engines in years past and adapted it to meet current requirements. In the process, three main requirements have been identified:

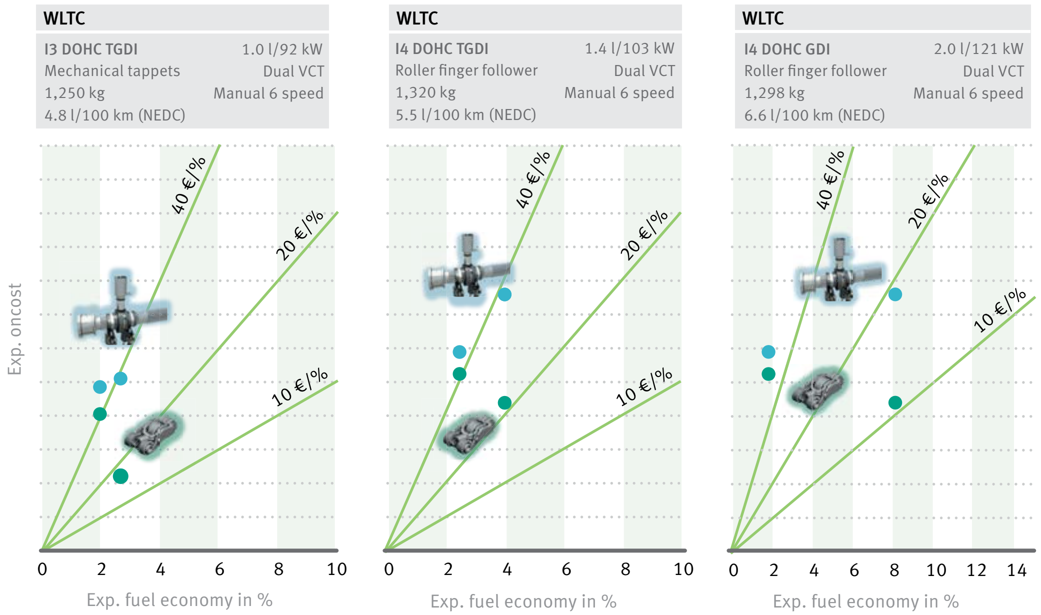
- Low mass moment of inertia
- High level of integration of all functions to minimize packaging space requirements as far as possible
- Low friction levels

As the development process continued, the design of the switchable roller finger followers was further refined and optimized to achieve these targets. Figure 2 shows the design of an early generation compared to the current one.

While the basic principle of hydraulic actuation had proven itself and was carried over, the drawbacks of the earlier design were compensated. This was achieved by implementing the following measures:

- The lost-motion spring was moved from a valve-side position to the pivot point side. This considerably reduces the mass moment of inertia about the pivot point. The design also increases the level of force acting on the spring such that the system no longer needs support on the cam base circle when switched off. As a result, the camshaft does not require the previous special-purpose solution and a lower-cost standard camshaft can be used instead.
- The friction on the sliding surface could be reduced by omitting the base circle contact,
- and the primary lever is now produced in a metal injection molding (MIM) process instead of investment casting. This improved the degree of functional integration while at the same time minimizing machining effort.
- By optimizing the design, the secondary lever could be produced using low-cost sheet-metal forming technology.

As Schaeffler has many years of experience with conventional, non-switchable roller finger follo-

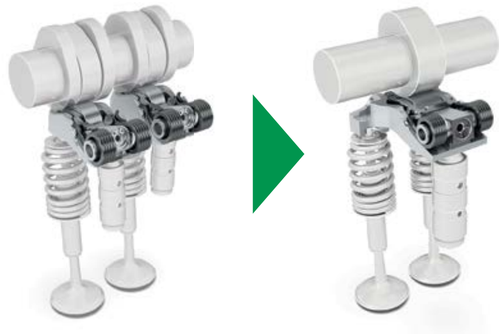


3 **Relationship between additional expense and fuel savings of switchable finger followers (turquoise) and cam shifting system (light blue) for different vehicle segments**

wers, it makes sense to draw a comparison with their switchable counterparts. The comparison reveals that although the switchable roller finger follower has a higher mass moment of inertia and requires more space to install, it is a worthy competitor to the conventional finger follower due to its greater functional scope. After all, in the majority of engine applications, the mass moment of inertia and length of the finger follower are determined by the basic constraints inside the cylinder head and not by the finger follower itself – a fact that applies equally to conventional and switchable roller finger followers. Schaeffler is pursuing several different development projects to make the switchable finger followers still smaller and lighter.

On gasoline engines with small to medium displacements, cylinder deactivation systems that deactivate the intake and exhaust valves during non-fired operation compete with familiar EIVC and LIVC de-throttling concepts. As such, deciding in favor of or against a concept is not only determined by the respective fuel-saving potential, but also the cost of the components. Likewise, the system must be versatile enough to be implemented in many different target vehicle classes while still meeting all relevant emissions regulations. This is why Schaeffler collaborated with a development service provider to analyze the cost-benefit ratio of various switching strategies with variable valve train systems for mass production. In this study, the fuel savings were simulated for systems with switchable roller finger followers and cam shifting system, and the additional expense associated with their integration into an engine was calculated (Figure 3).

From this study, it can be stated that compared to concepts that switch between two valve lift pro-



4 **The innovative twin-pallet design allows two valves to be actuated using just one switchable finger follower**

files, a cylinder deactivation system will save more fuel and cost less to integrate. Two main reasons why dual-lift systems are more expensive are the higher effort for manufacturing the camshafts and the coating of the sliding contacts. Switchable roller finger followers offer considerable cost advantages for cylinder deactivation, whereas they are at roughly the same oncost level as cam shifting systems for dual-lift switching concepts.

As already mentioned, cylinder deactivation will also enjoy more widespread use in future downsized engines with low displacement and few cylinders due to the capability of the system to reduce fuel consumption and CO_2 emissions. Schaeffler is currently developing a so-called twin-pallet design for these applications, which is less costly and requires less packaging space (Figure 4).

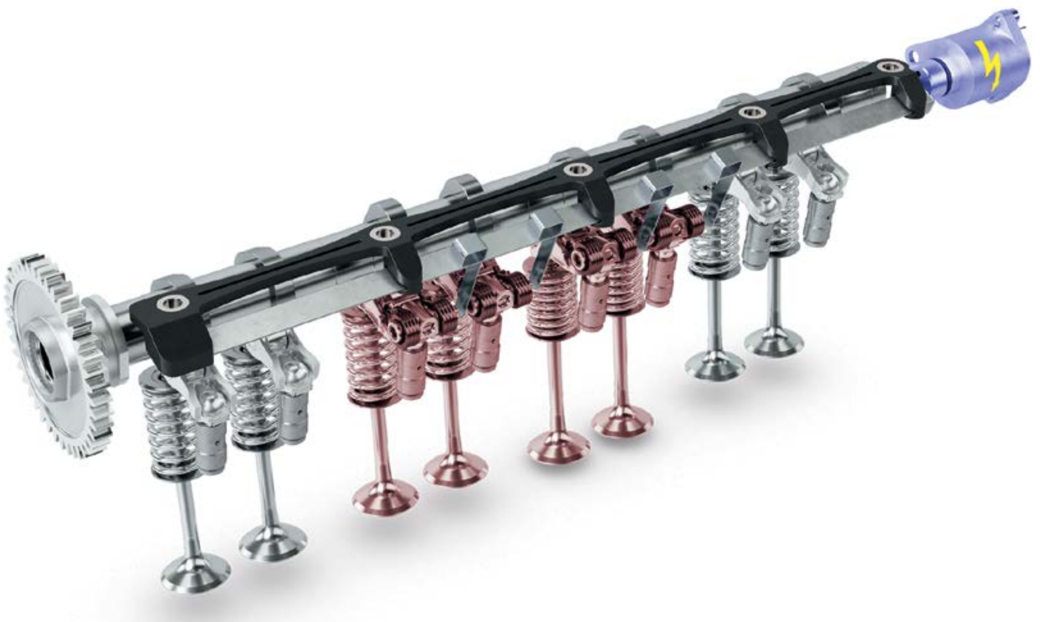
The design is based on a Y-shaped lever structure with two valve pallets that synchronously actuates the two intake or exhaust valves of the respective cylinder. The switchable finger follower itself has just one deactivation element, which means that only one cam is needed on the camshaft to open or close both valves at the same time. Due to its compact packaging, the switchable roller finger follower with twin-pallet is very slim and can minimize the longitudinal installation space required, thus making it much easier to integrate into the cylinder head. As initial estimates from Schaeffler indicate, the costs for a cylinder deactivation system that utilizes the twin-pallet design can be reduced by approximately 25 % while still retaining the same level of functionality.

eRocker System from Schaeffler

Although cam shifting systems are disadvantageous from a cost perspective, their electromechanical actuation does give them one benefit: While the oil circuit must be adapted to the requirements of the switching components for hydraulically switchable systems, electromechanically switchable systems can be implemented without any major changes to the oil system. Current engine concepts call for the oil pressure to be reduced as far as possible to minimize friction, which conflicts with the functional performance of hydraulic switching systems. The development effort for electro-mechanic systems also drops considerably because the amount of tests and calculations needed to optimize the overall system can be reduced considerably.

Schaeffler therefore made it an objective to develop a roller finger follower system with electromechanical actuation while at the same time retaining the cost advantage of the proven hydraulic system as far as technically possible. To achieve this, it makes perfect sense to use a central actuator system and avoid individual actuators as well as elaborate transfer elements such as control shafts.

The overall system design of the switchable roller finger follower system with electromechanical actuation is illustrated in Figure 5. The image shows the typical arrangement of an intake or exhaust



5 **Overall design of the eRocker System with roller finger followers switched by electromechanical actuation**

valve train as used on an inline four-cylinder engine with cylinder deactivation at cylinders 2 and 3.

The basic integration of the electromechanical roller finger followers in the cylinder head does not differ from that of the hydraulic systems. The actuation occurs mechanically via a slider bar made of sheet metal. The spring arms transfer the motion of the slider bar to the roller finger follower by actuating a shift pin in the outer lever. The pin switches the locking mechanism and disengages the connection between the inner and outer lever via an internal shuttle pin mechanism; the valve lift then is deactivated. The actuator – a linear stroke solenoid (blue, right) – is fitted outside the camshaft module or cylinder head.

A special challenge in developing such a system is that the locking pin of a switchable finger follower cannot be moved freely at any given moment, e.g. if the lever is in a differential stroke position or the mechanism is currently transferring a valve stroke. During hydraulic actuation, the motion is simply buffered, or stored in oil, by the pressure exerted. As soon as the locking pin is free to be moved again, it is displaced by the oil pressure and the locking mechanism engages or disengages.

In an electromechanincal system, the energy required for actuation must either be buffered in the solenoid or in a mechanical component. Since buffering the energy in the solenoid automatically necessitates an elaborate and costly single actuator system, Schaeffler has investigated the alternative of using a spring as an energy storage device. This spring has a leaf-spring design and also transfers the motion of the actuator via the slider bar to the switchable finger followers. Additional components are not required. In the event that the actuating pin is blocked while the slider bar is being moved, the energy is buffered in the spring arm (Figure 6, left). As soon as the mechanism is released, the actuating pin is moved accordingly and the lever deactivated (Figure 6, right).

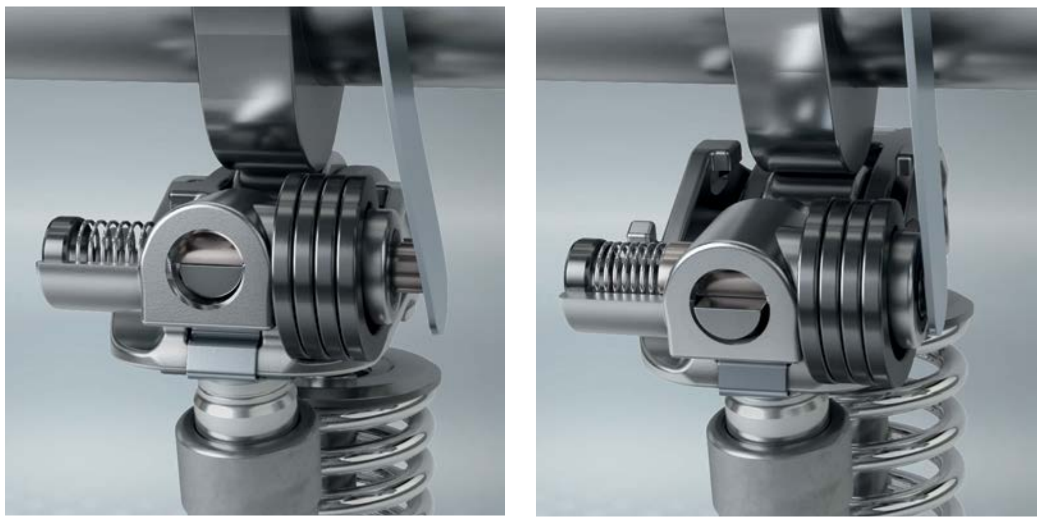
Another benefit of the system is that the slider bar can be flexibly positioned in the cylinder head as a result of the motion being executed by spring arms. By designing the leaf springs differently in length and cross section, it is possible to assume a position either closer to or farther away from the switchable finger follower in relation to the individual packaging constraints and cylinder head architecture.

The eRocker System technology can also be adapted to the aforementioned twin-pallet lever. The resulting cost savings outweigh the additional expense associated with replacing oil as a transfer medium with a mechanical component.

Switchable finger followers for the diesel engine

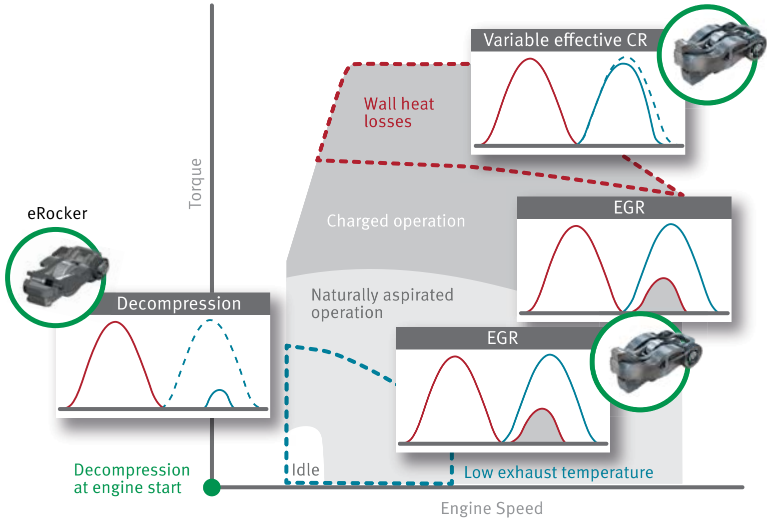
Variable valve train systems started finding their way into diesel engines at a much later point than with gasoline engines, on which they are basically state-of-the-art. As diesel engines by definition operate with an excess air ratio, there is no potential for reducing CO_2 emissions and fuel consumption by de-throttling the charge cycle. The focus rather is on exhaust temperature management and optimized effective compression ratio. One of the first applications in diesel engines was implemented by Mitsubishi [3], which used a switchable rocker arm system that was then followed by the very first switchable roller finger follower fitted to a mass produced diesel engine in Mazda's Skyactive diesel engine [4]. This concept realizes internal exhaust gas recirculation by executing a secondary exhaust valve lift (SEVL). As the engine has a very low compression ratio for a diesel engine at 14:1, this system is key to increasing combustion chamber temperatures under cold conditions and to achieve a stable combustion.

Additional motivation for developing variable valve train concepts for diesel engines was provided by the new emissions tests. Not only does the Real Driving Emissions (RDE) test now also quantify vehicle emissions when driving on pub-



6 **Actuator/slider bar motion with blocked locking pin via spring arm (left) and position after displacement (valve deactivated, right); locking mechanism partially cut away**

lic roads; the relevant operating range for becoming certified for road use has likewise significantly increased with the new Worldwide Harmonized Light Vehicle Test Cycle (WLTC). The introduction of RDE and WLTC regulations also heightened the requirements for quickly adapting vehicle emissions to changing load conditions in transient engine operation. It is this context that a host of new options for optimizing combustion can be realized variable valve train systems. It is important to note, however, that such systems do not optimize an engine all by themselves and only work effectively when integrated into the overall system. Variable valve train systems also lay the groundwork for increasing the usage rate of other measures like post injection, which then can be applied at lower operating temperatures, for example.



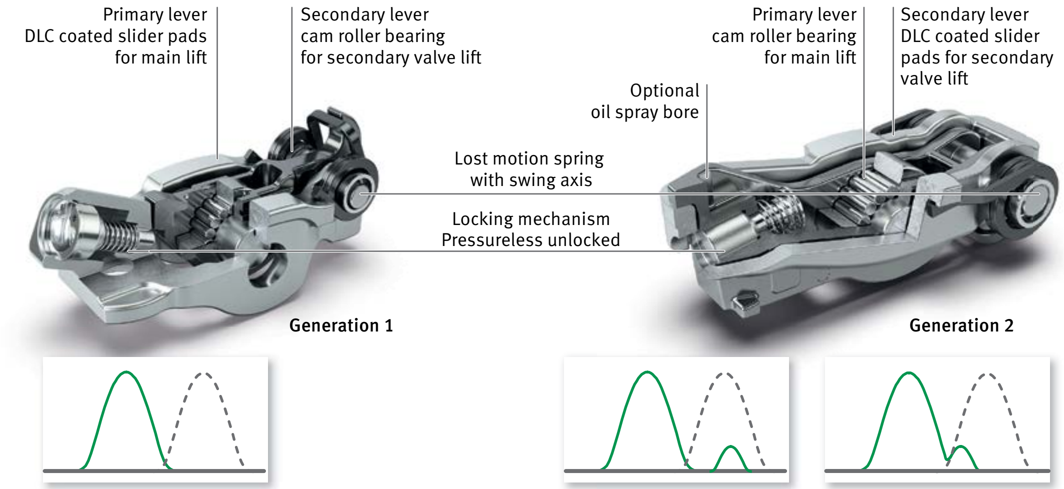
7 **Switching strategies for the variable valve train in a diesel engine**

Figure 7 shows the application range and potential switching strategies of the variable valve train for the diesel engine via switchable roller finger followers at different engine operating points.

The main strategy pursued is to use an additional exhaust valve lift to enable internal EGR, thereby controllably elevating exhaust temperatures to accelerate the light-off time for the exhaust aftertreatment components during a cold start. Unlike the established external EGR systems, which route the exhaust gas through a piping back into the intake manifold, the internal EGR system rebreathes a part of the exhaust gas from the exhaust tract into the combustion chamber through the exhaust valves. In addition to the combustion stabilization and warming-up effect this causes, the internal EGR system can respond much more quickly to changes in the EGR rate as experienced during transient operating conditions.

The first generation of the switchable roller finger follower from Schaeffler for diesel engines integrated the primary lever with a sliding surface at the cam contact point, while the secondary lever came with a rolling bearing (Figure 8).

In this configuration, the main exhaust valve lift and cam base circle phase were realized over the sliding surface and the secondary lift over the rolling bearing. The result was unfavorable friction during the main lift sequence which, unlike the secondary lift, is always mechanically transferred during engine operation and therefore has a much greater impact on the total friction loss of the system. In the second generation of the switchable roller finger follower for diesel engines, Schaeffler reversed the principle. Now, the

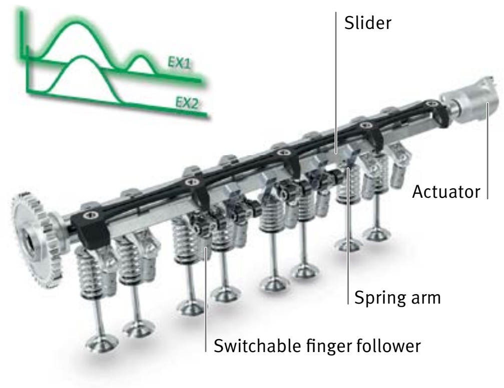


8 **Improved switchable roller finger followers for diesel engines from Schaeffler**

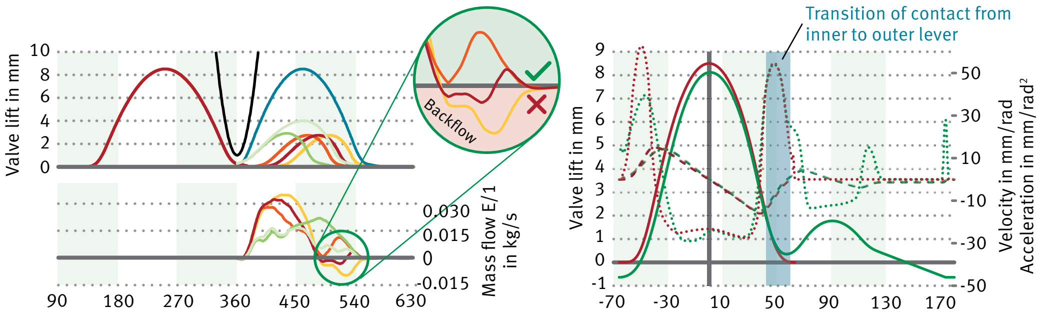
primary lever used to transfer the main lift is on the inside and has a roller bearing, while the secondary lever for the secondary lift sits on the outside and has sliding surfaces. The latter is also made from cold-formed sheet metal to reduce costs. Schaeffler is also pursuing cost-reducing concepts that do not require a DLC (diamond-like carbon) coating for the sliding surfaces as is currently the case.

While developing the eRocker system, this roller finger follower concept was also adapted to accommodate the electromechanical actuation (Figure 9). In this design, it was possible to avoid redirecting the actuation motion within the finger follower which was necessary in the cylinder deactivation system described above. The other system components, including the slider bar with spring arms and the actuator, were carried over.

An additional degree of freedom for applying the secondary lift was also realized. Previously, the secondary lift could only be executed after the main lift was already completed. Now, a "handover lift" makes it possible to add intermediate phases in which the main and secondary lift sequences overlap each other. The switching operation without handover lift made it necessary to delay the secondary lift, thus limiting the valve stroke height. As Figure 10 shows, a delayed second exhaust valve opening leads to pulsations



9 **eRocker System for secondary exhaust valve lift (SEVL) application in a diesel engine**



10 Intake backflow (left) and transfer of contact from main to secondary lift (right)

during the charge cycle that cause exhaust to flow back into the induction tract.

This should be avoided to prevent the intake side from becoming contaminated with exhaust residue. The pulsation-induced excitations in the intake manifold can also cause acoustic problems. When switchable roller type finger followers with handover lift are used, the outer finger follower for the secondary lift is actuated while the engine valves are still closing during the main lift phase. The valve motion contact is therefore transferred from the cam roller to the sliding surface during the main lift closing sequence. To ensure that no components are damaged, the contact must be transferred from the main to the secondary lift lever at a defined speed (Figure 10, right). This transfer speed must also remain constant in different tolerance situations over engine lifetime.

At the handover lift switch point, the engine exhaust valves are nearly completely closed when the piston reaches TDC to avoid valve to piston contact and are re-opened as soon as TDC has passed. This allows the valves to execute a very early secondary lift. As Figure 10 shows, not only do pressure fluctuations not occur, but air mass flow across the intake and exhaust valves is extremely consistent throughout the intake phase. Even the valve lift curve with 4 mm of amplitude depicted in Figure 10 (left), which closes even later than the standard secondary lift curve without handover lift and should therefore definitely lead to problems, does not cause backflow into the intake manifold. This curve does, however, facilitate a higher target EGR rate and resulting lower NO_x emissions for the given conditions.

The differential pressure, which greatly affects the EGR rate, varies among different engines. It mainly depends on the overall engine configuration (e.g. four or six cylinders), the design of the engine ducting, the turbocharging system used and the exhaust tract, for example. It can be varied during operation by changing the angle of the guide vanes on turbochargers with variable turbine geometry or via flaps in the exhaust tract as are frequently used to control external EGR systems.

If just one secondary lift curve is employed, however, this can lead to a major compromise between a high EGR rate at low loads (to achieve a high increase in temperature) and a limited rate in the higher load range when a greater quantity of fresh air is required. Due to this trade-off, the secondary lift may therefore be deactivated at a relatively low load and the EGR must be taken over by the conventional external EGR systems. This compromise can be avoided in a wider range by using fully variable or a multi-stage secondary lift switching [5].

The latter can be realized with two independent switchable finger followers for each cylinder. As different secondary valve lift profiles can be used for both switching elements, multiple secondary lift air mass flows and EGR return rates are possible.

While the one exhaust valve (EX2) offers a high secondary lift amplitude, a small amplitude is realized with the second valve (EX1). If only the EX1 valve is opened, the EGR rate is low (Figure 11, stage 2). Actuating the EX2 valve leads to a medium return rate (Figure 11, stage 3). When both secondary lift curves are activated at the same time, the maximum exhaust gas volume in the secondary lift phase can be pulled back into the combustion chamber (Figure 11, stage 4).

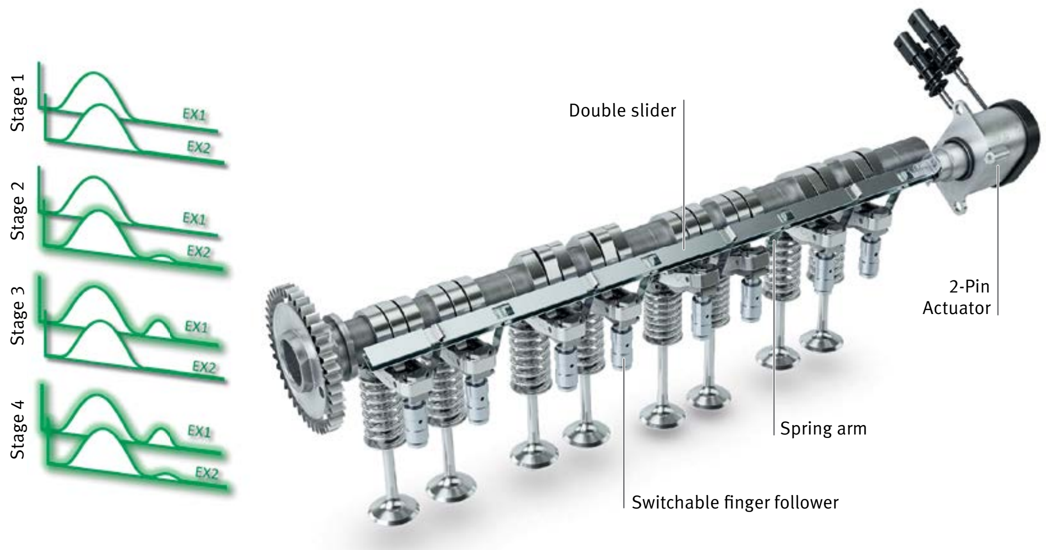
The independent switching sequence for both valves is achieved using two parallel slider bars that connect to the first or second exhaust valve of all cylinders. These slider bars are then actuated separately from one another by a dual-pin actuator.

Schaeffler also proposed a similar function for a hydraulic actuation system [6]. The effort required to integrate it into the cylinder head was relatively high by comparison, however, not to mention the fact that it could only be used with special cylinder head architectures.

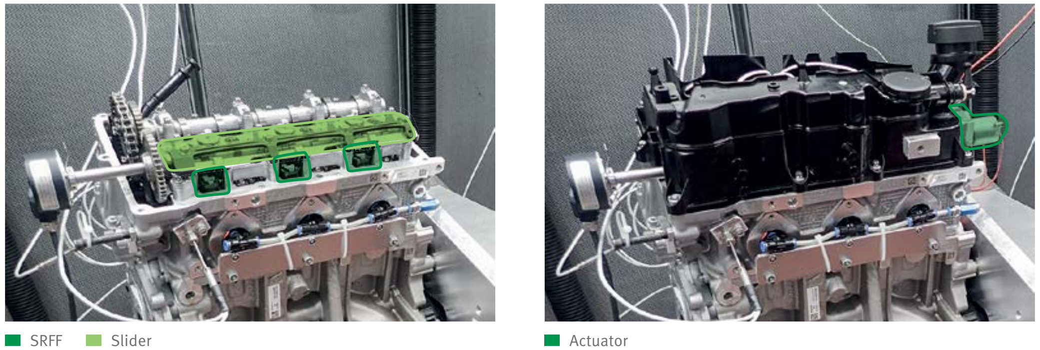
Testing of the eRocker System – Initial results

Following initial testing of the design on a test stand, the eRocker System was applied to a mass production three-cylinder engine as a single-stage variant (Figure 12). By adapting a guide to the existing exhaust camshaft journals, engineers were able to integrate the slider bar into the available space. The actuator was fitted outside the cylinder head.

Figure 13 shows the results of the switching time measurements for locking of the eRocker System compared to two hydraulically switchable systems.



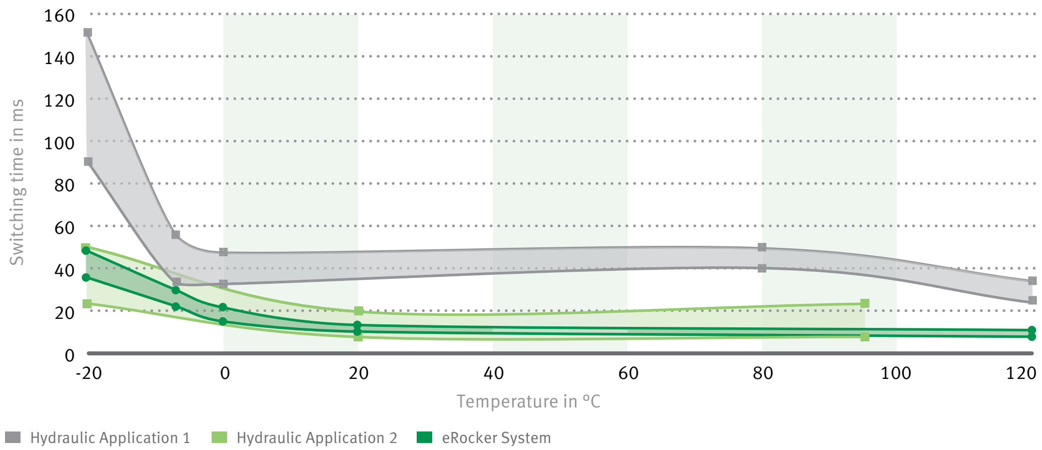
11 Multi-mode switching concept via the e-rocker system



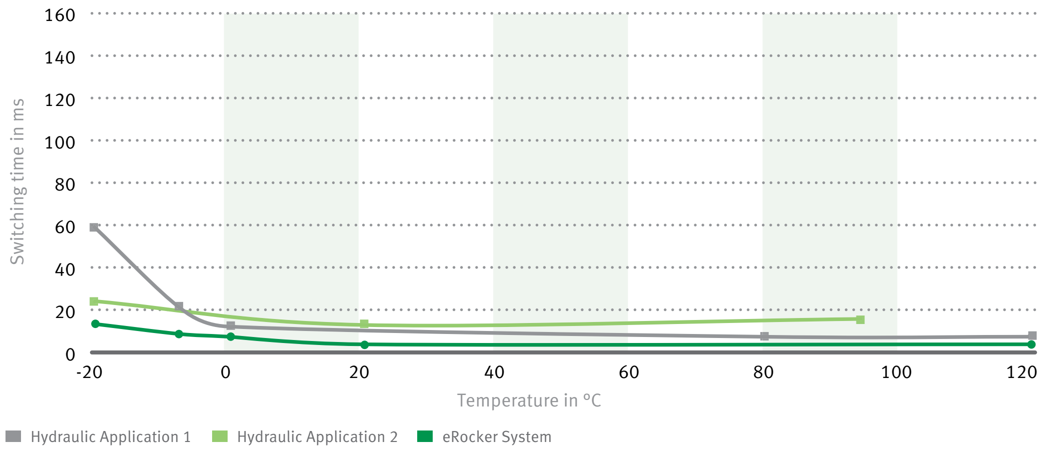
12 Multi-mode switching concept

On the hydraulic systems, you can clearly see that the switching times increase in the sub-zero range, which is to be expected. As temperatures drop, the oil becomes more viscous and the switching mechanism takes longer to actuate as the pressure increases more slowly. Depending on the conditions of the overall system (e.g. available pressure level and length/diameter of the control galleries), this decrease in performance can be lower or higher as becomes apparent when comparing applications 1 and 2 in Figure 13. In the example, application 2 is an inline six-cylinder engine. Even with a comprehensive set of system optimization measures, it would not be possible to greatly improve the switching time due to the prevailing design constraints, e.g. gallery lengths.

The eRocker System also has longer switching times at low temperatures as the mechanical components of the system are likewise in contact with oil, and the actuating forces in the assembly increase as a result. In theory, this drawback can be compensated by increasing the level of actuating force applied.



13 Switching time for interlocking SEVL systems in relation to oil temperature and internal engine constraints



14 Variation in switching time

The inherent problem here is not the switching time itself, since this could be compensated by an earlier switching signal by the engine control unit, but rather the inconsistent and varying switching times, which are worse in a hydraulic system compared to an electromechanical one (Figure 14). This effect then leads to situations in which the ECU cannot predict whether or not the roller finger follower system will switch for individual combustion cycles. This is referred to as cycle-selective or non-cycle-selective switching. Due to the minimal variation in the switching performance of the eRocker system, the speed and temperature range can be greatly expanded for cycle-selective switching and unpredictable switching situations can even be avoided altogether. With the target maximum switching time variation of 12 ms, cyclical switching can theoretically be achieved until speeds of up to 2,500 rpm on a four-cylinder engine with secondary lift (90°cam ignition spacing), which fulfills common requirement specifications for secondary lift applications. Only at speeds above 2,500 rpm is it possible that the switching state may not be known for single combustion events.

Overall, it is safe to say that the switchable roller finger follower can also be realized with acceptable switching times when actuated by means of hydraulics. Due to oil pressure and oil foaming characteristics, however, as well as the conflict between the switching and valve lash adjustment functions, the hydraulic system used inherently becomes very complex and difficult to optimize. These conflicts can be avoided with the eRocker system.

Summary and outlook

Schaeffler has been developing and manufacturing components for variable valve trains for around 20 years by now. In recent years, switchable roller finger followers have proved to be a functional and cost-effective solution, particularly as used for cylinder deactivation systems in gasoline engines. By continuous improvements in design, Schaeffler has managed to advance and optimize the system over time. As tests show, switchable roller finger followers offer very good performance when compared directly to more expensive alternative systems. With the Y-shaped, twin-pallet switchable finger follower, which can actuate two intake or two exhaust valves at the same time, Schaeffler offers a very compact and cost-effective approach to cylinder deactivation. Schaeffler develops hydraulically actuated switching components as well as the eRocker System for cylinder deactivation and cam profile switching as used for secondary exhaust valve lift on diesel engines. This electromechanical system is designed as a “plug-and-play” solution that does not impact the oil circuit of the engine and is therefore easy and convenient to integrate, whereas the hydraulic system continues to be a cost-effective concept for high-volume applications.

Ever stricter emissions requirements also increase the need for variability in the valve train as used in diesel engines. In this context, combustion systems with a high internal EGR rate that are implemented by switching the exhaust valves via a secondary valve lift are of particular interest. By integrating the innovative function of a handover lift, the timing of the secondary lift can be adapted to the specific engine application environment such that further emission and NVH benefits can also be attained. Better control over exhaust gas recirculation is made possible by approaches that use different secondary lift heights at both exhaust valves that can be combined at will. This arrangement is considerably easier to implement than a hydraulically switching system thanks to the eRocker System and its minimal implications for engine peripherals.

In a three-cylinder production diesel engine with secondary exhaust valve lift, the eRocker System successfully proved its basic operative function and the ease of application. Further activities include adaptation for cylinder deactivation on a gasoline engine in the first half of 2018, followed by the independent switching of both exhaust valves on a diesel engine in the second half of the year.

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7

Breakthrough of Rollerized Crank Shafts

Dr. Frank Schlerege

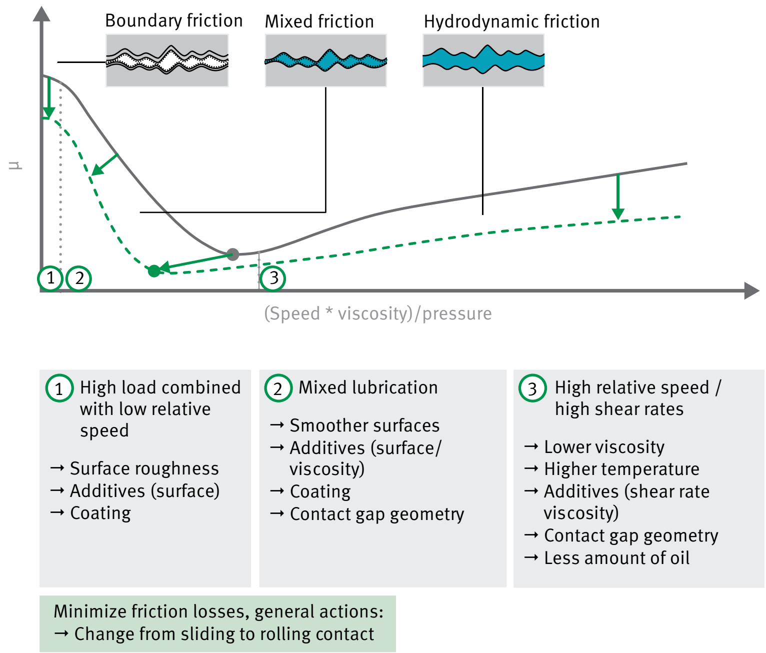
Introduction

The internal combustion engine continues to be a key player for the future, accordingly challenges currently faced by this engine will not be going away anytime soon. More specifically, in order to reduce the CO_2 emissions, it is essential that frictional loss be minimized as far as possible, and doing just that remains to be an important measure for optimizing the combustion engine. One design measure that has been pursued to do this is the concept of downsizing, which involves reducing the displacement of an engine to improve its operating efficiency. This measure ultimately places higher loads on the bearings, however. Start/stop functions that automatically cut out the internal combustion engine when it is no longer required (e.g. at a stoplight) and restart it when the driver wants to set off also increase friction levels in the plain bearings of engines. In P0 hybridized setups, a starter-generator is integrated in the belt drive of the auxiliary assemblies. This assembly assists the internal combustion engine e.g. when starting and “sailing”. The resulting belt forces, which can be substantially higher, combined with frequent starts, further increase the load for the first main bearing. This leads to higher contact friction and possible wear risks at the plain bearings used in engines today.

These conditions are making it more attractive to use rolling bearings instead of plain bearings in the engine [2] – an approach that Schaeffler has already laid the groundwork for by carrying out comprehensive advance development work and conducting tests in numerous applications. One development direction involves new approaches that call for rolling bearings for the crankshafts of passenger car engines. To this end, Schaeffler collaborated with Ford on a development project to investigate in detail the requirements for crankshaft rolling bearings and the resulting benefits for a 1.0-liter three-cylinder gasoline engine.

Tribology in the internal combustion engine

Hydrodynamic bearings are found in many places in internal combustion engines. Their friction coefficient and, thus, the overall friction level, depend on the relative speed, load and viscosity in the lubricated contact zone. As the Stribeck curve in Figure 1 shows, a hydrodynamic bearing passes through the three windows of static or boundary friction, mixed friction and viscous friction in operation. These three friction states are characterized by different physical laws and can all be optimized in individual ways.



1 **Stribeck curve with static friction, mixed friction and viscous friction.**

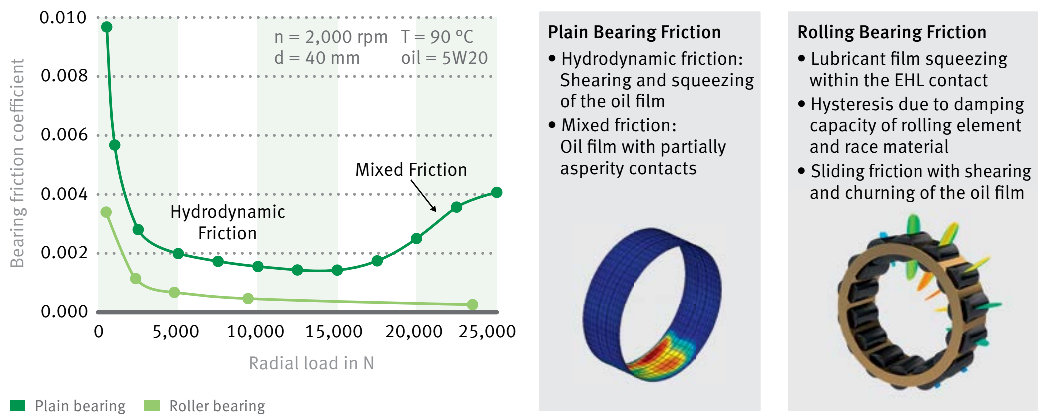
The key influencing parameters are:
- Static/boundary friction: Surface roughness, surface additives and coatings
- Mixed friction: Surface roughness, additives (surface and viscosity), coatings and lubricated contact zone geometry
- Viscous friction: Oil viscosity, operating temperature, additives (shear rate or viscosity), lubricated contact zone geometry and oil volume.

One way to reduce friction loss is therefore to use lubricants with low viscosity. Today even 0W16, 0W12 and 0W8 oil grades (SAE J300) [1] are available. Although these extremely low viscosity oils reduce shear losses and, thus, power loss, mixed friction levels increase during engine operation. Tribological developments focus on this effect by counteracting wear and addressing the higher mixed friction levels to minimize friction loss. Potential corrective actions include applying surface coatings that reduce wear and friction, preconditioning run-in (contours, roughness) and using high-additive lubricants.

More generic design measures for reducing friction are to optimize moving mass while lowering acting forces and changing the type of contact from plain bearings to rolling bearing mounts. Figure 2 illustrates the basic differences between a cylindrical roller bearing and a plain bearing as used in the engine block of the three-cylinder test engine. The friction coefficient of the respective bearing is plotted above the radial load. You can clearly see that the friction coefficient of the rolling bearing is lower than that of the plain bearing at all load points and drops further as the load increases.

Rolling bearings in an internal combustion engine

Today, combustion engine applications of rolling bearings can be found in the belt and tension pul-



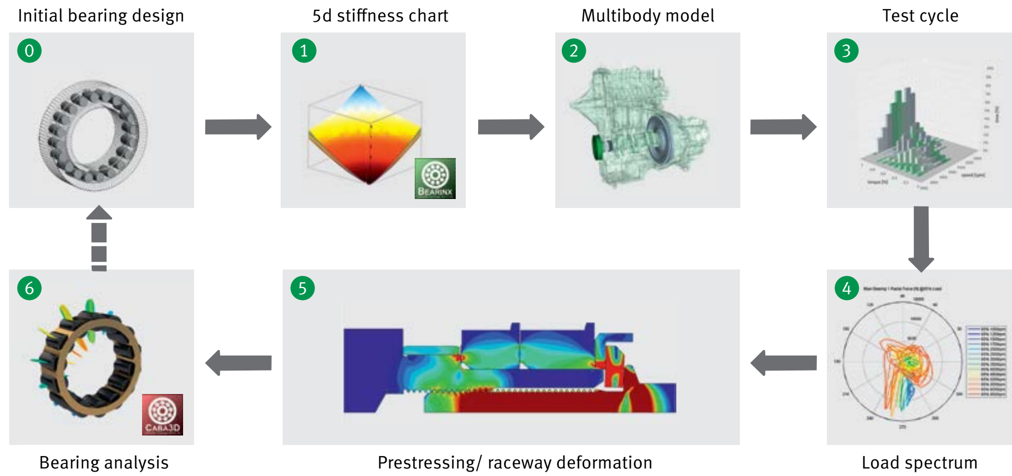
2 **Comparison of friction between a cylindrical roller bearing and a plain bearing.**

leys of the accessory drives, water pump bearings, cam and balance shafts, turbochargers and roller finger followers (Figure 3). Plain bearings continue to be commonplace at the crankshaft of passenger car and commercial vehicle engines despite the fact that rolling bearings have been successfully used in the engines of motorcycles, snowmobiles, outboard motors and jet skis. Crankshafts with rolling bearings were actually used in passenger car engines, aero engines and tanks up to the 1950s, after which plain bearings took over. The reasons for this change are that plain bearing assemblies for crankshafts are extremely cost-effective, easy to mount and robust. OEMs and plain bearing manufacturers have a huge amount of experience in the behavior of the bearings during dynamic operation as concerns friction, wear and oil supply. Rolling bearings are also more difficult to mount on the crankshaft. In years past, for example, built crankshafts were sometimes used. This concept is too costly from a manufacturing and assembly standpoint, however, due to today's large series production applications. Schaeffler has spent the last few years working intensely to determine the benefits and drawbacks of the different constructions and devised engineering solutions such as split outer rings and cages combined with rolling bearing raceways mounted directly on the crankshaft.

The potential for reducing CO_2 emissions is one reason for Schaeffler to thoroughly investigate integrating crankshaft rolling bearings into passenger car engines. Another reason has to do with their inherently better performance during start/stop sequences and under low speed and high load conditions. As research results from OEMs and Schaeffler point out, it is not sufficient to simply replace the well established plain bearing constructions with rolling bearings without making design changes to the engine. Rather, plain and rolling bearings must be viewed and optimized in the context of the engine as a whole to maximize the potential of rolling bearings for reducing CO_2 emissions without introducing any drawbacks such as increased noise levels.

This is why Schaeffler collaborated with Ford in a research project to analyze the properties and characteristics of using a crankshaft rolling bearing mount assembly on a 1.0-liter three-cylinder gasoline engine [3]. The objectives included not only quantifying NVH performance, but also determining what savings could actually be achieved at different operating points in the engine characteristic map. Generally speaking, a three-cylinder engine is a very challenging starting point for using rolling bearings. The unit has just four crankshaft main bearings, which evokes the question of why rolling bearings should be used there in the first place. Right from the beginning, all possible combinations were observed and evaluated and included a full rolling bearing assembly through to hybrid configurations integrating both rolling and plain bearings. Also factored into the evaluation were criteria such as the assembly effort and oil supply at the conrod and remaining main plain bearings.

3 **The use of rolling bearings instead of plain bearings has proven successful in different applications.**



4 **Self-contained development process for designing crankshaft bearings.**

Development method

To determine which approach leads to the best solution, parameters for reducing friction and wear at the respective bearing position, complex simulation and testing procedures must be run through during the development period. The high simulation outlay required and resulting long calculation times give rise to the conflicting targets of achieving a meaningful and practical level of detail while at the same time defining necessary system limits. One the one hand, the correlations at the bearing point must be understood as accurately as possible to obtain usable results. On the other, the constraints of the overall system (operation, periphery, electrification, start/stop system, fuel) greatly impact the outcome and had to be taken into account. Adding to this is the fact that not only design, but also manufacturing aspects play a role (e.g. precision machining the bearing shape to precondition the run-in pattern). The development method applied to analyze the rolling bearing mounts on the crankshaft is coordinated with the research-specific purposes (Figure 4). When designing the crankshaft rolling bearing positions, all relevant parameters are taken into account and determined by coupling the multi-body simulation model (MBS model), calculating the elastohydrodynamic lubrication properties and correlating the results obtained using the Schaeffler-developed BEARINX bearing simulation software. These include the following:

- First and second-order inertia and rotational forces
- Fully elastic crankshaft dynamics
- Realistic reaction torques and forces of the rolling and plain bearings
- Reciprocal effect between crankshaft deflection and bearing reaction torques
- Reciprocal effect between rolling and plain bearings

BEARINX can also conduct a thorough analysis of the bearing itself. The software can quantify results for the following:

- Realistic load distribution inside the bearing
- Offset and tilt
- Profile of the roller and inner ring
- Edge stress
- Estimated service life i.a.w. ISO/TS 16281

The development process starts by choosing a rolling bearing whose technical properties make it generally well suited to the application. BEARINX is then used to create a non-linear stiffness map for the bearing that serves as the basis for defining the reaction of the bearing assembly to offset and deflection forces. BEARINX can also be used to perform a detailed analysis of rolling bearings as the contact pressure at every rolling element is accounted for in the simulation. The stiffness map of the bearing is integrated in the fully elastic MBS model of the engine created by Schaeffler. For this purpose, the project partner provided all relevant engine components as CAD (computer-aided design) data and assisted in modeling realistic constraints such as tolerances, material data and operating conditions by defining the respective gas pressure curves.

During the simulation exercise, the engine is operated virtually through several test cycles in line with the gas pressure curve. The test cycles represent the load, speed and temperature operating states, which are relevant when it comes to service life, consumption and NVH performance. The resulting load spectra of the bearing indicate how frequently and for how long loads have been applied. As Schaeffler has a great deal of experience in designing rolling bearings, it can account for influential forces – such as deformation – that occur during assembly and operation. Within the calculation sequence, the MBS software application CABA3D from Schaeffler allows a detailed analysis to be conducted of the dynamic processes that take place inside the rolling bearing. This, in turn, makes it possible to determine the dynamic movements of the bearing components, the forces that act between them and the resulting friction. If after evaluating the test results, the engineers concluded that an additional iterative loop is required in the optimization process, they recalculate the revised design using updated input data. Friction loss, service life and the noise level of the system are factored into this evaluation.

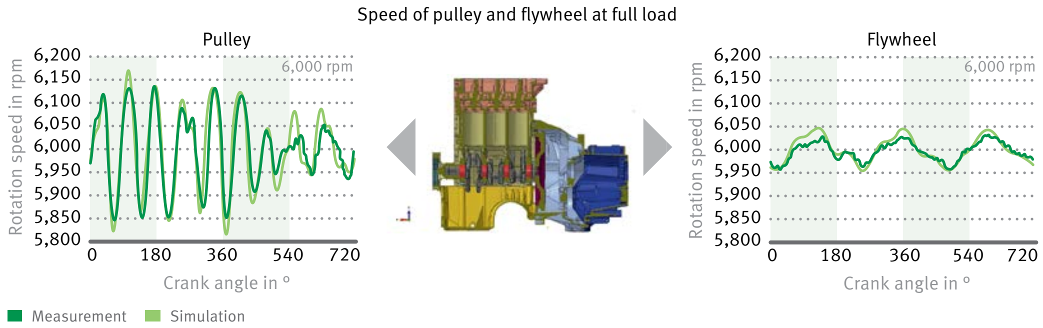
Validation

A validation process is required to develop a functioning design method. In this method, the previous production configuration of the engine was validated in line with the measured data available on the dynamic response of the crankshaft, friction loss percentages and structure-borne noise. Validated part models for the rolling bearings used by Schaeffler were also integrated. The combination of a validated base and part model makes it possible to design – in a virtual environment – a system that has not been built yet. Comparing the virtual and actual results helps predict the friction potentials and effects of changes to the system on such variables as the service life and structure-borne noise.

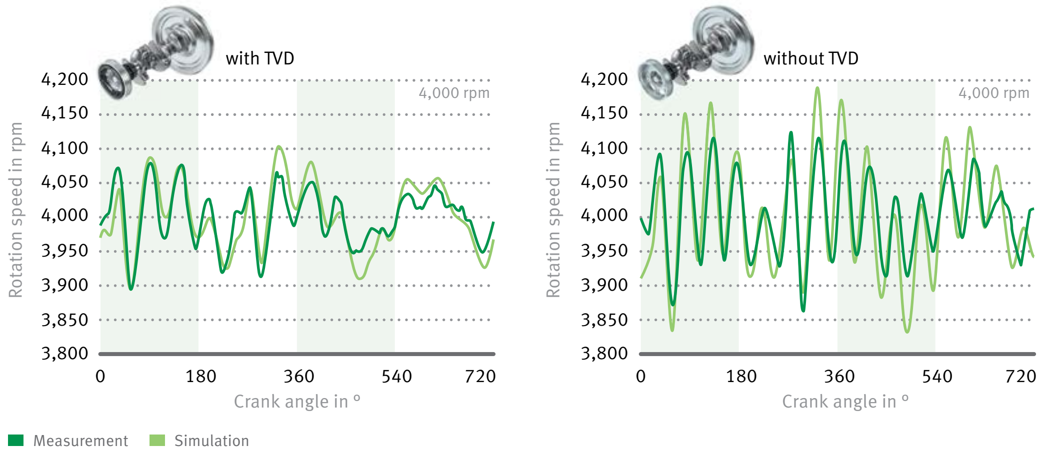
In order to validate the simulation method with respect to the crankshaft dynamics, engines with conventional plain bearings were taken out of current series production and analyzed on the engine test bench operating at different engine speeds. Schaeffler also ran a series of calculations applying the simulation method depicted in Figure 4.

During the application of the method, identical geometric dimensions and constraints were defined for both analytical approaches. Generally speaking, this method is equally well suited to computing plain and rolling bearings such that the plain bearings of the production engine could be modeled without having to make any fundamental changes. The cyclic irregularity of the crankshaft on the flywheel and the accessory side acted as a reference parameter for comparing the measurement and simulation results. As Figure 5 indicates for full-load conditions at 6,000 rpm, the results obtained on the test stand coincide almost perfectly with the virtual results over the complete speed range. The crank angle resolved speed is predicted very well by the simulation model on both sides of the crankshaft. This applies to the curve as a function of crank angle as well as to the amplitude of the curve.

The simulation model was further validated by conducting tests under full-load conditions, whereby the belt pulley of the accessory drive was equipped with and without a torsion vibration damper on separate test runs (e.g. at 4000 rpm in



5 **The comparison between the physical engine and the engine model reveals close conformity with the crank angle resolved speed.**

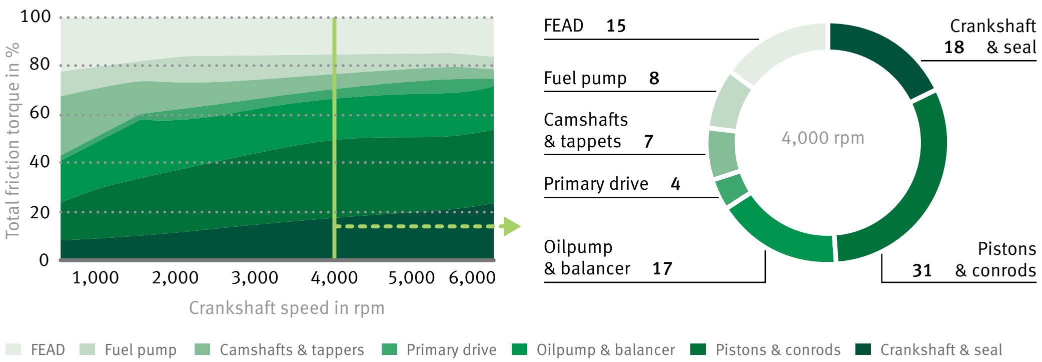


6 **The measurements and engine model calculations showed a high level of congruence both with and without the vibration damper.**

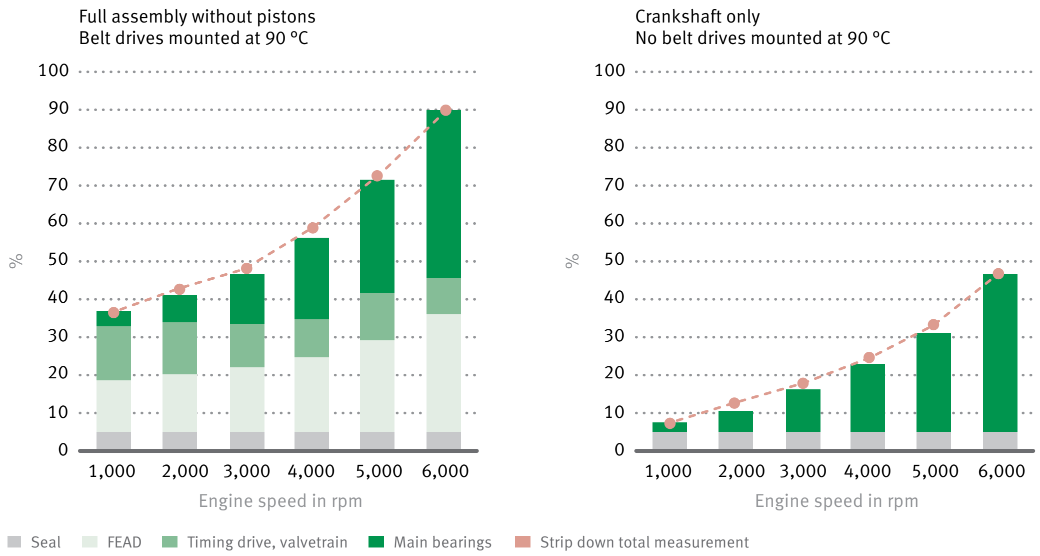
Figure 6). In these scenarios as well, the simulation model was able to prove its accuracy in calculating the crankshaft dynamics to the extent that it can be viewed as formally validated for quantifying crankshaft dynamic response. Figure 6 also plots the effects of the torsion vibration damper in the engine system as an additional validation result. In the diagram at the left – with the damper fitted – you can see significantly reduced cyclic irregularity of the crankshaft as compared to the diagram at the right – without the damper.

The main validation aspect is the frictional loss of the engine. One method that has been established for conducting this type of analysis is the so-called strip-down method, which involves dismounting the individual assemblies of the engine step by step. The comparative measurement with and without the corresponding components reveals their frictional impact but does not fully account for the reciprocal interaction among the components. As Figure 7 indicates, the friction percentages of the individual assemblies greatly vary, depending on the engine's operating speed and load. In general, however, the significance of the piston and conrod dominates at all speeds, followed by the crankshaft bearings and seals, and the oil pump and balance shaft. Figure 7 (right) shows detailed friction percentages at 4,000 rpm. The system comprising crankshaft bearings and seals accounts for 18 % of the total friction at this speed. This value fluctuates between 10 and 20 % across the engine speed range of 1,000 to 6,000 rpm. Contributing to this friction are the two crankshaft seals at 0.25 Nm each, which is a relatively low friction level that remains constant across all engine speeds.

Figure 8 charts the results of the strip-down measurement and calculation. The bar values of the individual assemblies were taken from the simulation exercise for the crankshaft main bearing as well as the measured percentages and, when combined, yield a value that coincides almost exactly with the friction level measured (red dotted line in Figure 8). This applies to the motored overall engine (left) and the separate analysis of the crankshaft drive (right). The validation sequence for the development model is then complete, and optimization measures for the crankshaft bearings of the test engine can be efficiently examined at the virtual level.



7 Breakdown of friction percentages using a strip-down measurement.



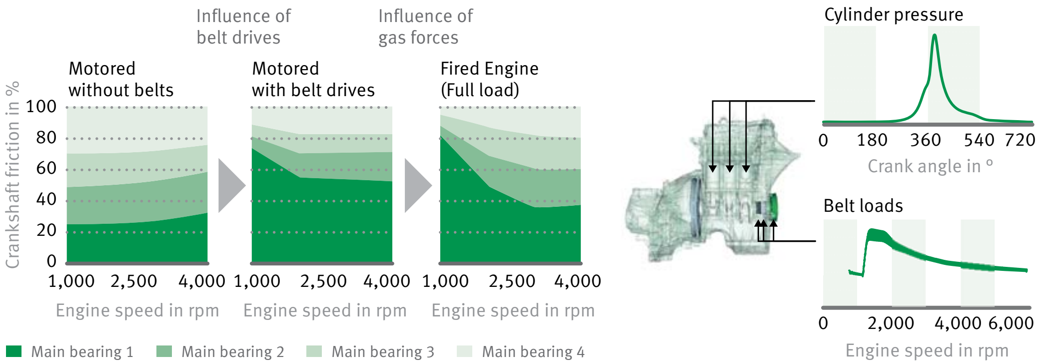
8 Comparison of measured and calculated friction values.

Friction-reducing potential

Applying a validated computation method for the overall engine and its subsystems enables qualified statements to be made that would not be possible only by taking measurements. The simulation can be seen as an analytical tool to interpret the measurement results and to find the interactions within and between the investigated systems. The three diagrams in Figure 9 (left) graph the friction percentages of the four crankshaft plain bearings in the base engine with respect to the total crankshaft friction. The first diagram shows the results of the test stand measurements without the accessory drive. All four bearings then roughly account for the same percentage of overall friction. This homogeneous spread shifts when the load exerted by the accessory drive is also factored in. The second diagram summarizes the results for the virtual analyses of the engine model. You can easily see that the friction percentage for the first crankshaft bearing greatly increases compared to the other three bearings. This phenomenon can be attributed to the bending loads that act on the crankshaft by way of the timing belt drive and accessory drives. The bearing loads shift once again when the gas forces during fired engine operation are factored in (third diagram). This diagram reveals a pronounced dependency on the engine speed.

Figure 10 plots the potential for reducing friction loss as the differential between the plain bearing engine (blue line) and the system with a single rolling bearing as the first main bearing. The first crankshaft bearing now also has the lowest percentage of the total crankshaft bearing friction during fired engine operation under full-load conditions. The rolling bearing likewise shows dynamic benefits, particularly under low-speed and high-load conditions.

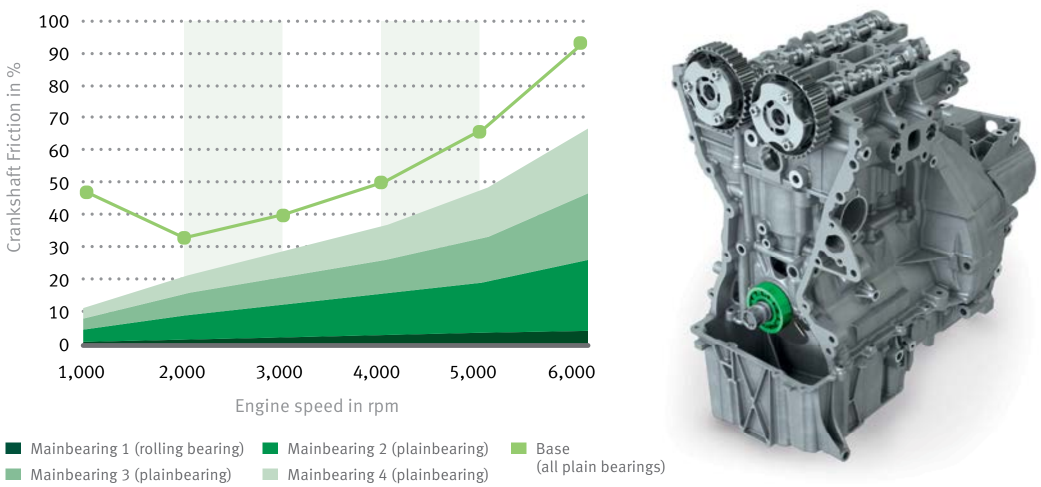
These results served as the basis for constructing a prototype engine that has a rolling bearing in place of a plain bearing for the first crankshaft bearing only. To this end, an optimization mea-



9 **Friction percentages of the four crankshaft main bearings and virtual reciprocal effects.**

sure was carried out in multiple iteration steps by applying the aforementioned development method. The initial rolling bearing with an outside and inside diameter of 72 mm and 35 mm, was optimized and adopted into the specific engine application (Figure 11). The resulting rolling bearing requires 7 percent less space in the crankcase and has a 14 percent larger crankshaft diameter. Adding to this are a substantially longer service life and lower friction loss than the larger bearing assembly.

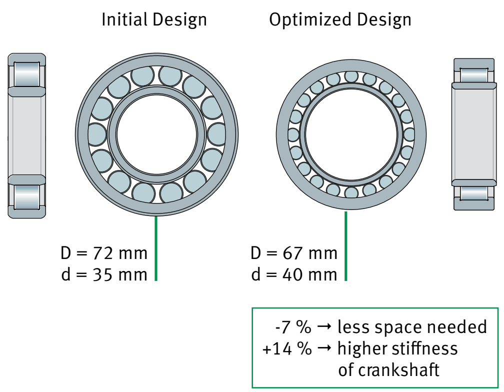
This was also verified by comparison of overall friction losses measured at both the base (all plain) and prototype (1st rolling) engine. For this purpose, the engine was measured at different temperatures and speeds and showed lower losses. Benefits calculated in the simulation can



10 **Potential for reducing friction using rolling bearings.**

also be verified by extensive measurements. Across the entire engine performance map analyzed, the rolling bearing revealed substantial benefits over the plain bearing. This effect is especially well pronounced under conditions involving high loads and low speeds.

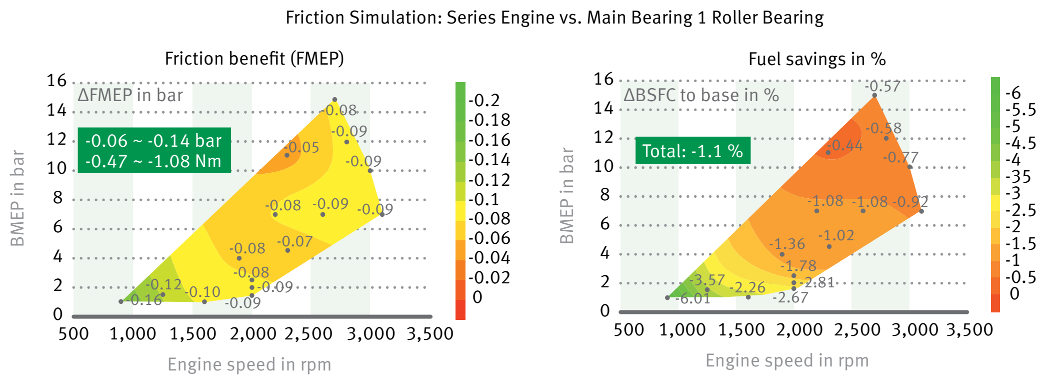
Figure 12 shows the potential of the first crankshaft bearing as a rolling bearing instead of a plain bearing across the consumption map when fitted to a Ford Focus with the 1.0-liter EcoBoost test engine. In the Worldwide Harmonized Light Vehicles Test Cycle (WLTC), the lower friction levels correspond to a reduction in fuel consumption of 1.1 %. When this concept is applied to other applications, however, it must be noted that the fuel saving potential greatly depends on the accessory drive and gas pressure loads that act on the crankshaft bearing. This predicted fuel consumption reduction was confirmed by a test conducted by Ford on the fired engine running minimum and maximum running resistance curves, with of 0.9 to 1.2 % lower fuel consumption.



11 **Optimally designed rolling bearing for the first crankshaft main bearing in the 1.0-liter three-cylinder engine.**

NVH behavior

Not only reduced friction levels and an adequate lifetime are key design criteria of the crankshaft rolling bearing, but also NVH performance, which was also thoroughly investigated. The most important criterion for ensuring good NVH performance are the surface velocities of the engine, which are root causes for air and structure-borne noise. To determine whether the excitation frequencies of the oscillation system have changed and if so, to what extent, the air and structure-borne noise was measured on both engines (i.e. with plain and then



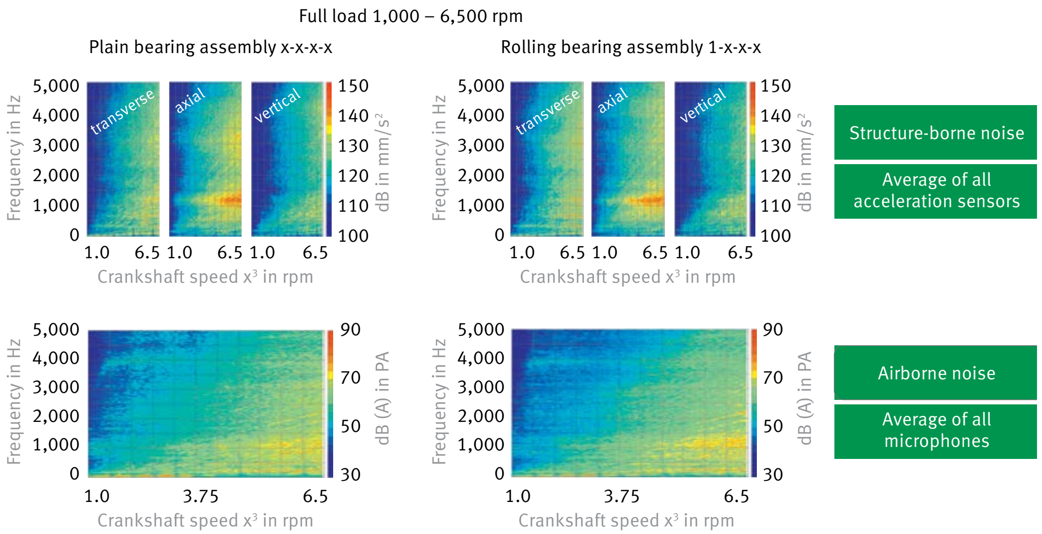
12 **The reduced friction attributed to the first crankshaft bearing fitted as a rolling bearing averages out to 1.1 % less fuel consumption in the WLTC.**

with rolling bearings). Figure 13 (top) summarizes the results for the structure-borne noise. No significant deviations could be determined when comparing the results. Even when it comes to airborne sound (Figure 13, bottom), the values for each engine are more or less the same. The outcome of the measurements is then that the NVH performance of the engine with rolling bearings is comparable with that of the engine with plain bearings. This outcome also coincides with the noise levels as subjectively perceived by the testers who were asked for their acoustic impressions as part of a final testing sequence.

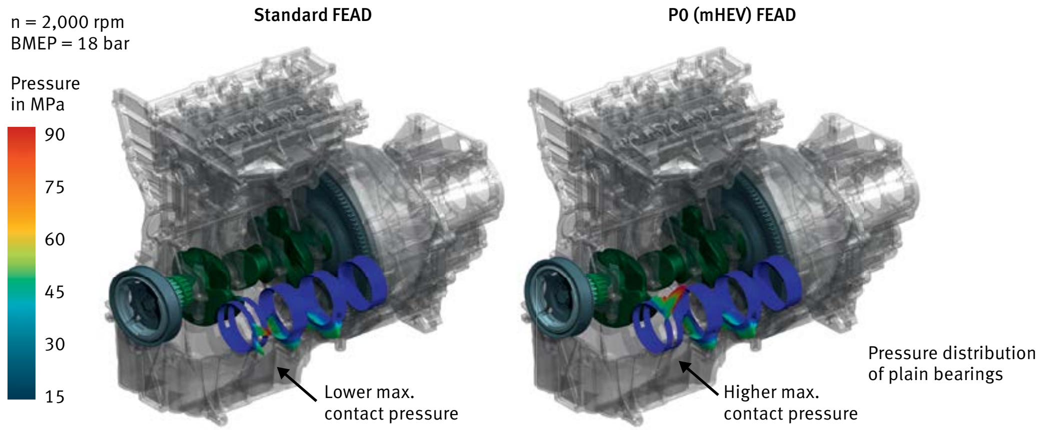
P0 hybridization

In an additional step, Schaeffler investigated the effects of a P0 hybridization setup as a starter-generator assembly on crankshaft bearing loads. The higher tensile forces and resulting change in the load direction result in an additional bending stress on the crankshaft and, thus, increased bearing loads. Figure 13 (right) shows the resulting main bearing load of a standard FEAD (front-end accessory drive) compared with the increased bearing load on the first main bearing in a P0-hybridized internal combustion engine. The substantially higher edge pressure – largely solid-body pressure – points to an elevated wear risk of the plain bearings. This application is therefore predestined for a rolling bearing on the crankshaft to improve robustness and reduce friction levels.

Another benefit is realized when the engine is started under a high belt load as Figure 15 depicts for a start sequence at up to 1,000 rpm and at 90 °C. The torque that the starter motor must output at this time – approximately 30 Nm – is greatly influenced by the mass inertia. A significant benefit can be obtained, however, when the necessary breakaway torque is reduced by a factor of 10. This benefit becomes even more pronounced at lower temperatures.



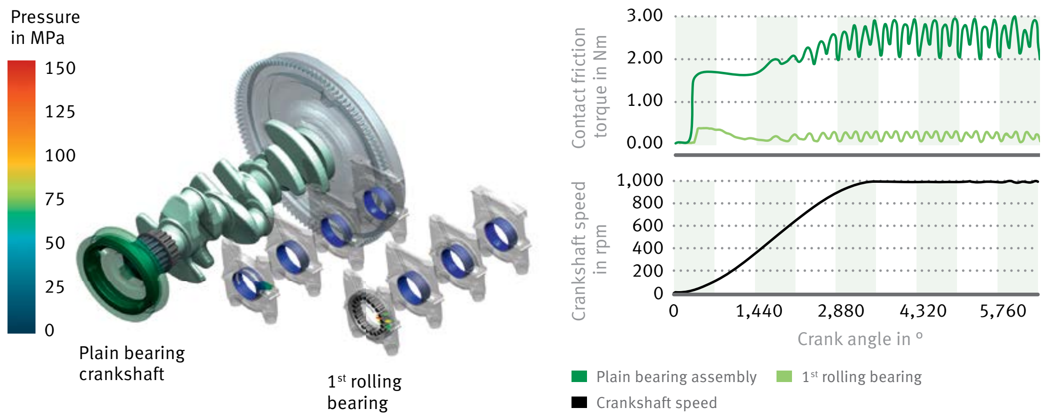
13 **The subjective analyses revealed that there was no perceptible difference in NVH performance between the engine when fitted with rolling and with plain bearings.**



14 **The load exerted on the first crankshaft main bearing can be very high, particularly on a P0 hybrid, which means that a rolling bearing mount can measurably reduce fuel consumption.**

Conclusion and outlook

Schaeffler cooperated with Ford to investigate a 1.0-liter EcoBoost engine whose first crankshaft bearing was switched to a rolling bearing from a plain bearing. A validated method using simulation models and measurements devised by Schaeffler was applied. As it became apparent during the course of the research project, the rolling bearing needed to be customized in line with the operating conditions of the respective engine in order to utilize its full optimization potential. The rolling bearing engine was predicted to consume 1.1 % less fuel, which was verified by measurements, without degrading the initial NVH performance. To counteract any NVH abnormalities that could occur in a future development project, Schaeffler is currently working on a series of ac-



15 **The start sequence of a P0 hybrid application as compared with a plain bearing and a rolling bearing assembly.**

tive and passive measures that will address the problem sufficiently.

Application of rolling bearings at crankshafts of internal combustion engines – particularly running a P0 hybridized systems – has huge potential for reducing friction levels and improving the durability of the crankshaft bearings. Schaeffler is furthermore intensively investigating the benefits of crankshaft rolling bearings in other engine concepts, like those with four cylinders.

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Transmission

8

The Transmission

Now and in Future the Most Efficient Connection Between the Powertrain and the Road

Dr. Hartmut Faust

Introduction

2030 transmission systems vision

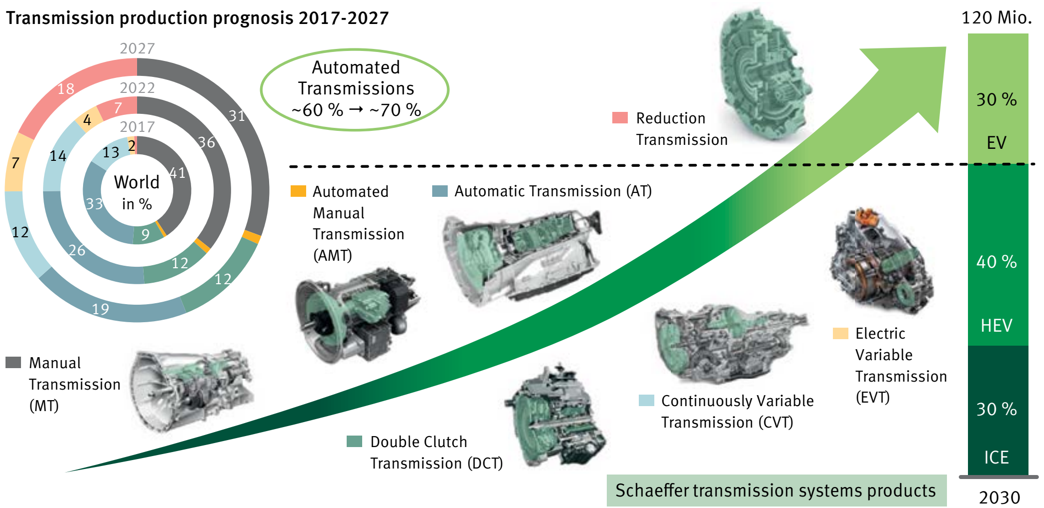
Over the next few years, the number of powertrain concept variants for motor vehicles will continue to grow. But regardless of whether they are driven by internal combustion engines, hybridized powertrains or electrical-only vehicles, as shown in Figure 1: The goal of all of these concepts is to use as little primary energy as possible for driving motor vehicles, thus reducing CO_2 emissions to the lowest amounts technology can achieve.

For the transmission, this results in the requirement to generate as few losses as possible when transmitting the power and converting the speed ratio and torques – in particular, this includes losses through friction as well as auxiliary energy for actuation – to supply as much mechanical energy as possible to the wheels. The trend of automating clutches and shifting operations helps implement optimized operating strategies, in particular with regard to the increasing electrification of powertrains.

Challenges

With regard to internal combustion engines, there are additional requirements for the powertrain due to engine-based measures to reduce consumption and emissions [1] such as downsizing and cylinder deactivation, which is now also used for three-cylinder engines [2]. The resulting torsional vibrations on the crankshaft must be damped effectively and isolated from the rest of the powertrain to meet drivers' comfort requirements.

Besides the minimization of friction and auxiliary energy losses [3], a further weight reduction of transmission components can also contribute to CO_2 savings. At the same time, lighter and smaller



1 **Automation trend in transmission systems and percentage of powertrain concepts in the overall market in the Schaeffler scenario for 2030**

Transmission Trends	Product innovation fields
Reduction of losses: all types of transmissions for ICE, hybrid, EV	• Friction optimized bearings • Increased ratio spread and # of gears
Cylinder deactivation on ICE: CDA, RCD, DSF	• Damper, CPA, double CPA, coupled CPA, iso-radial pendulum
Automatization: AT, CVT, DCT, MT	• Launch: TC/iTC/4TC, WDC/DDC • Inside TM: Planetary gearset, SAX, clutch packs, CVT chain • Actuation: PoD (HCA, MCA, GA), E-Clutch (MT*plus*, CbW, ECM)
Hybridization/Electrification: P0, P1, P2, P3, P4, DHT, EV	• PYD, PYD-S • P2HM, 3K, short synchronizer • PoD actuators: ECA, EPA, EAA, IPS, PRND • DHT damper w/ torque limiter • DH-ST 6+2, DH-CVT, serial/parallel (e. g. Twin Drive) • E-Axle, high speed planetary sets, high speed bearings

2 **Transmission system trends and products and system solutions developed by Schaeffler**

components can help meet the ever more challenging design space specifications, particularly in hybrid vehicles. In addition, a systematic search for CO_2 reduction potential must also consider the large number of small consumers of auxiliary energy in the powertrain. This primarily includes the hydraulic pumps and electric motors used to operate the actuators for automation that actuate clutches and other transmission components. Every single watt counts: With real power-on-demand concepts it is now possible to change the average power consumption of actuators from the three-digit watt range to the low two-digit watt range.

No matter how good technical solutions are, though: They will not prevail on the market unless they are available at marketable costs. This requires components that are designed precisely for specific requirements – which means that over-engineering must be avoided – as well as highly efficient manufacturing processes. However, cost evaluations must include all costs – not only the direct costs of the components but also the indirect costs incurred from power loss and related CO_2 emissions for which legal regulations are becoming increasingly strict worldwide.

Solutions

The following chapters show how Schaeffler develops its components, subsystems and systems for all conventional transmission designs – from automatic planetary gears with individual market preference in the US to the CVT, primarily in Japan, and the double clutch transmission, primarily on the European market, to the classic manual transmission, mainly in Europe and BRIC countries – as well as for new hybrid transmissions and electric axle reduction gears, with these requirements in mind, Figure 2. This includes low-friction rolling bearings, efficient actuators with appropriate power consumption and effective vibration damping with new centrifugal pendulum-type absorber concepts. It is what makes the use of even more efficient engines possible. One requirement for customized transmission systems that are designed precisely for specific requirements is the wide use of simulation technologies and CAE methods that takes aspects such as future load spectra into consideration at an early stage in the design phase.

Another trend in transmission design is the increasing automation. Automated transmissions can contribute to the reduction of CO_2 emissions because they permit new shifting strategies with optimized consumption and more complex driving strategies such as coasting and are able to fully utilize the potential of hybridized powertrain concepts through appropriate recuperation actions. In addition, they take the increasing comfort requirements of many drivers into account.

Based on market assessments by Schaeffler, the percentage of automated transmissions will increase from approximately 60 % to around 70 % worldwide in the next ten years. At the same time, the percentage of hybrid and electric vehicles on the world market will increase to 70 % by 2030. Since the percentage of hybrid vehicles alone will be at 40 %, the percentage of new vehicles sold on the market with an internal combustion engine will still be 70 % by the end of the next decade. Besides the well-known transmission types that involve the insertion of a P2 hybrid module with 48 volt or high-voltage technology in automatic transmissions, CVT and dual clutch transmissions, there are additional transmission designs with growing market shares. The transformation of transmission systems will result in a new diversity, such as the electric variable transmission (EVT) and general dedicated hybrid transmissions (DHT), in which the full functional capability is ensured by the integration of an additional electric motor as a second power source. At the same time, simpler reducing gears for P4 hybrids as well as electric vehicles will gain higher market shares [3].

For all transmission types whose job it is to transmit drive power to the vehicle's wheels, the reduction of losses occurring during torque conversion remains an essential development goal in order to minimize the use of primary energy and increase the driving range of the given energy accumulator.

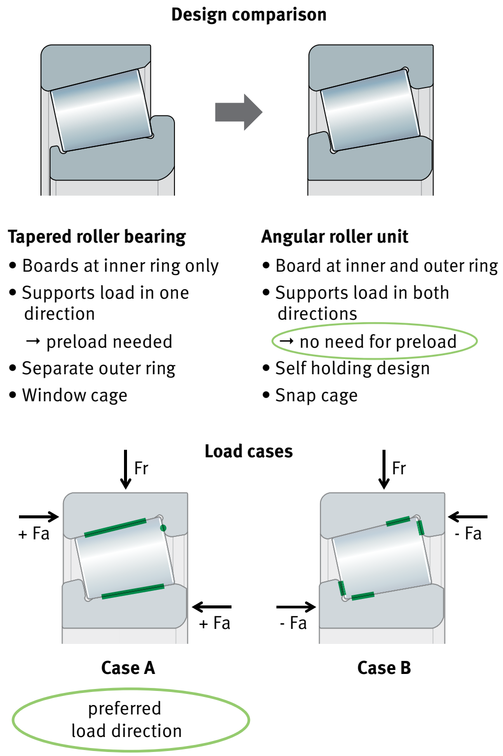
New concepts for low-friction rolling bearings in all transmission types

Task

In the world of internal combustion engines, it was an important task for developers of transmission bearings to reduce friction in order to decrease fuel consumption. This task is not going to change in the future: It is important to reduce CO_2 emissions and increase the range of electrified drives. Depending on the load conditions, this goal can be achieved by using tapered roller bearings with friction-optimized geometries that have been developed with the help of new CAE methods, the replacement of tapered roller bearings with double-row angular contact ball bearings as well as by locating/non-locating bearing concepts that do not require axial preload. In addition, friction power can also be reduced and high load capacity maintained by completely new bearing concepts such as the angular roller unit (ARU) which is similar to a tapered roller bearing but, unlike the tapered roller bearing, is able to support axial forces in both directions with the help of an innovative arrangement of lips on the inner and outer ring [5].

Angular roller unit (ARU) as a new bearing design

Locating/non-locating bearing supports cannot always be implemented without modifying the design space due to the relatively low load rating of deep groove ball bearings because the required load ratings must be achieved by means of a larger bearing diameter if necessary. This raises the question of which alternative bearing design can be used as a locating bearing instead of a deep groove ball bearing.



3 **Comparison of the design principle of a tapered roller bearing and an angular roller unit (ARU) (left) as well as loading conditions for an ARU with preferred direction (right)**

Figure 3 shows Schaeffler's approach with a new locating bearing – the "angular roller unit" (ARU). It has a higher load capacity than a deep groove ball bearing but operates with less friction under axial loads than a cylindrical roller bearing. The ARU can support axial forces in both directions as a self-retaining single bearing. However, it should be mounted in the preferred direction so that the higher axial forces can be transmitted via the raceways, similar to tapered roller bearings.

The locating/non-locating bearing supports using ARU and cylindrical roller bearings perform similarly well in terms of friction as the solution with deep groove ball bearings. With the ARU as a locating bearing, the required rating life is maintained at the same time. This allows the changeover from adjusted tapered roller bearing supports to locating/non-locating bearing supports without modifications to the design space.

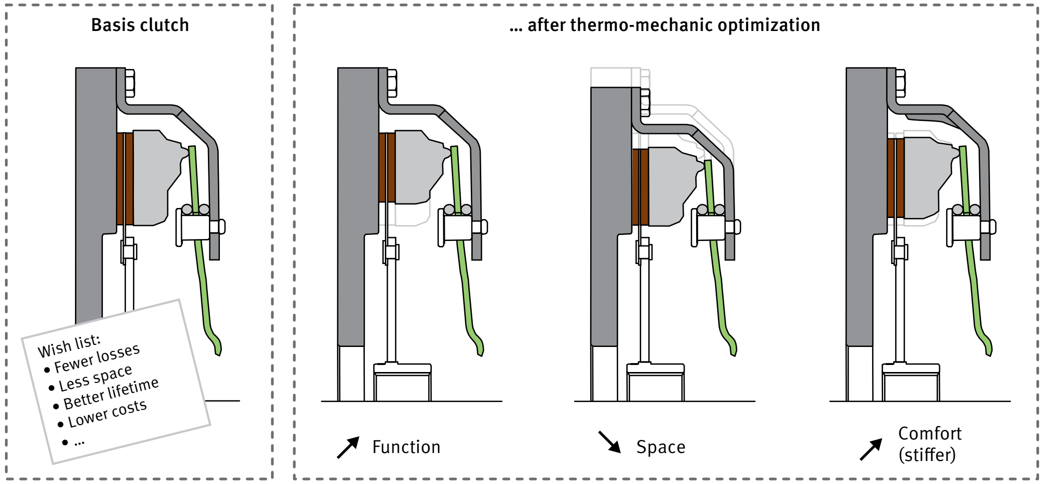
Use of CAE tools

Task

Simulation methods are able to make a reliable prediction of varied phenomena such as vibrations in the powertrain, supply valid information on sources of losses and consumption benefits and, if required, to map the complete powertrain system with a very high level of detail. Schaeffler uses CAE tools such as the BEARINX calculation software to optimize transmission bearings. In addition, simulation methods are also used for virtual testing and for the design of other subsystems such as clutches. Here they allow the design of components to be customized for their intended use with the help of load spectra, i.e. neither too big nor too small. If attempts to secure operation close to the design limit by using CAE methods are successful, transmission components can be designed as small as possible. Not only does this reduce costs, weight and design space, it also reduces friction losses, which helps achieve the broader goal of reducing CO_2 emissions [6].

Application example: Thermo-mechanically optimized clutches

In the thermo-mechanical optimization of clutches, the interaction of all relevant factors in the complete system must be analyzed. That is why Schaeffler has combined the previously separate



4 **Optimization variants reached in comparison with a basic clutch**

models for thermal behavior, mechanical deformation and friction coefficients in the CLUSYS (Clutch Systems) software. The optimized thermo-mechanical clutch design takes the friction coefficient behavior, component geometries, cushion deflection and system rigidities into account. The software permits rating life calculations for various utilization profiles, the calculation of thermal damage due to misuse and the calculation of clutch capacity for transmitting torques.

The optimized approach is shown using the example of a self-adjusting clutch. The conicity behaviour of the previous clutch is not ideal; however, this could not be taken into account adequately in the classic design due to extremely long calculation times, but it is now fully possible with the new thermo-mechanical model. The optimization results in a significant increase in torque capacity while also reducing wear by preventing excessive local stress. The improved torque capacity can be utilized to reduce contact pressure and thus pedal force and to increase actuation comfort. The excess gained in rating life requirements can be used elsewhere.

Figure 4 shows potential variants. This allows the inside diameter of the clutch disk and the pressure plate to be increased, making lower contact pressure and reduced pedal forces possible. At the same time, space is freed up inside the clutch for installing a centrifugal pendulum-type absorber on the clutch disk with a greater damping effect, for instance. Another variant consists of reducing the outside diameter of the entire system, thus decreasing the required space for the clutch system as such. A combination of small changes to the diameters of the friction partners creates the exact space needed to optimize the components with regard to rigidity, thus further reducing pedal forces.

Isolation of torsional vibrations

Task

Internal combustion engine-based measures to reduce fuel consumption such as downsizing, cylinder activation and down-speeding – i.e. driving with a long ratio just above the idle speed – make

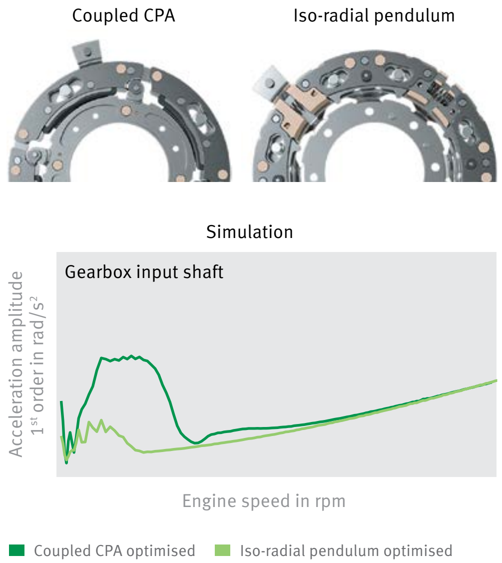
high requirements on the isolation of torsional vibrations of the crankshaft. In order to prevent undesirable NVH phenomena such as gear rattle, body boom and other noises and vibrations, Schaeffler has developed specific solutions such as the dual mass flywheel for manual transmissions and double clutch transmissions as well as damping systems for torque converters in automatic transmissions and CVTs. Here centrifugal pendulum-type absorbers (CPA) are increasingly being used which are suitable for applications in systems with a dual mass flywheel or hydrodynamic torque converter as well as directly on the clutch disk.

Further development of the centrifugal pendulum-type absorber

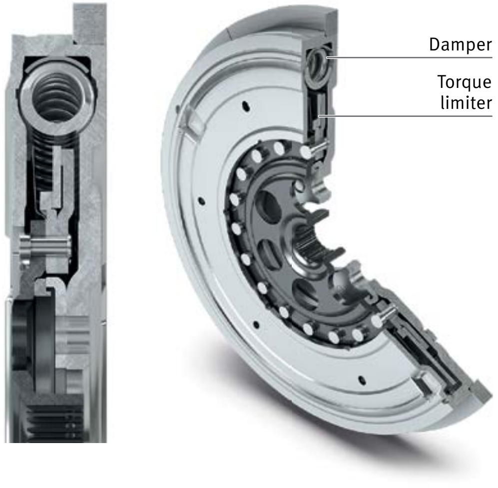
The great market success of the CPA can be attributed, among other things, to the fact that the physical principle automatically results in a balance between excitation and pendulum vibration with the right frequency or excitation order. In this process, the vibration amplitude keeps increasing until the exciting mass no longer vibrates. This means that the CPA can compensate different phase positions that can occur at higher speeds or when coasting. The natural frequency of the damper changes over the engine speed in relation to the centrifugal force which itself is increasing quadratically with the speed – just like the firing frequency of the internal combustion engine that is the main excitation force, so that, given the relevant adjustment, the main excitation order of the engine is minimized.

However, undesirable, noticeable inherent noises may occur in a CPA. This is because gravity is dominant over centrifugal force starting at a certain point when the engine is turned off at decreasing speeds. This causes the pendulums to lose contact with the rollers. As a consequence, the rollers and pendulums may hit the flange or each other.

Figure 5 shows two solutions by Schaeffler. With the couple pendulum (left), which is already in volume production, the pendulum masses support each other through springs in a circumferential direction. Here the spring preload is selected in such a way that the pendulum remains in the guide track even if the engine is at a complete standstill. The effect of the almost constant spring forces overlapping with the speed-based centrifugal forces is largely compensated by correcting the order of the track. This type of spring arrangement is particularly helpful for first order pendulums, such as those needed for cylinder deactivation from four to two active cylinders. This is because gravity also generates a first order excitation in a rotating pendulum, which is another interference factor in addition to the excitation from cylinder deactivation.



5 **Couple pendulum and iso-radial pendulum (top) with a diagram of vibration amplitudes in relation to the speed (bottom)**



6 **DTH damper with integrated slip clutch as a torque limiter**

The approach using the iso-radial pendulum (right) is entirely different, permitting even lower vibration amplitudes to be achieved (bottom). In this approach, the individual pendulums are connected in one point by a ring not located in the torque flow, which means that the pendulum masses are now synchronized. One of the usual two spherical rollers is eliminated, causing the pendulum to carry out a swiveling motion rather than a purely radial motion. This design eliminates the first order excitation from gravity on the individual pendulum masses. Noise is controlled by means of stop elements to counteract contact loss at low speeds during stops.

In applications with dedicated hybrid transmissions (DHT) without a start-up clutch, it may be necessary to protect the entire powertrain from the occurrence of impermissible high peak torques. Special DHT torsional dampers have been developed for this purpose that include an additional slip clutch as a torque limiter, Figure 6.

Real power-on-demand actuators

Task

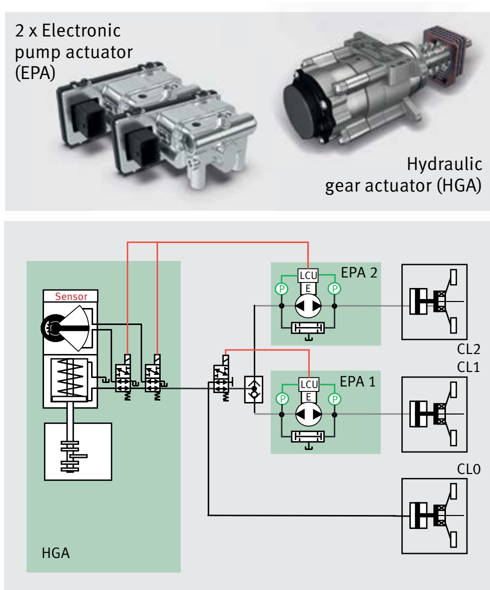
Automotive development focuses on energy efficiency. All energy consumers must be taken into account in order to utilize the full potential. These include actuators that actuate the components in the automated powertrain. If real power-on-demand actuators are used, the energy supplied to the electric motor must be converted appropriately and as directly as possible into adequate forces and pressures with accurately fitting travel and volumes. Another aspect is maintaining positions. Theoretically, there is no active energy involved, but in reality, a lot of energy is spent on maintaining a condition [8].

One important approach consists of designing actuators in a way that several consumers can be supplied and modulated. P2 hybrid structure concepts require a third clutch for double clutch transmissions so that another clutch actuator including electronic components must be added. In this case, hydraulic solutions are a better choice than electro-mechanical solutions because they are easier to expand by adding valves and scaling the pump-accumulator module. However, to turn precisely these hybridized double clutch transmissions into a milestone of efficiency requires appropriate actuators for such P2 double clutch transmissions with three clutches. A new approach here is the electrical pump actuator (EPA).

Application of the electrical pump actuator (EPA)
In the EPA, the sequential pressure build-up on the two working ports is achieved with the help of a passive two-pressure valve. Regardless of the EPA's conveying direction, the two-pressure valve connects the lower pressure with a reservoir. This means that the EPA can be applied to a consumer, such as a clutch, in a forward direction and also modulate it by turning it back and forth. In reverse operation, the clutch pressure can be reduced completely. If the EPA continues to run in reverse operation, pressure builds up in the second working port to shift to a different gear selection. Two independent consumers can be operated sequentially by using an EPA with a two-pressure valve.

Figure 7 shows an EPA application in the double clutch transmission of a hybrid vehicle in P2 arrangement. The transmission can be operated with just two EPA and one hydraulic gear actuator (HGA). The control of the separating clutch (K0) can be mapped with another valve and another pressure sensor, similar to a conventional hydraulic control unit.

On closer inspection, the electrical system architecture varies considerably. All that is left of the active electric motors of the gear actuator is two simple switching valves that can be controlled



7 **Actuator system for double clutch transmissions with two EPA and one HGA**

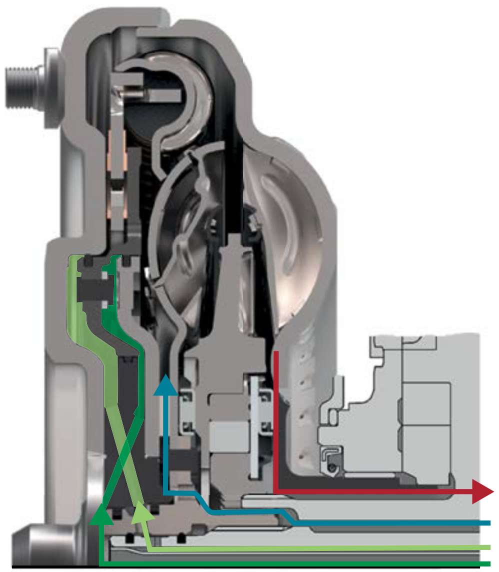
with simple valve output stages for the operation of electric motors. This makes it possible to question the entire transmission control unit and to replace it with the two EPA. As a consequence, expensive components can be eliminated, however, the intelligence of the transmission control unit must be mapped by the EPA. This requires a system and software structure that is oriented towards the independent actuation of both sub-transmissions. Only the shifting coordinator with the gear selections and the coordination of the overlapping gearshifts must be doubled, distributed or transferred to a superordinate control unit for the powertrain which is usually available in hybrid drives. The elimination of the transmission control unit makes this architecture a very cost-efficient variant.

More efficiency in individual transmission designs

Torque converter for automatic transmissions and CVT

The good controllability of the lock-up clutch in the torque converter of automatic transmissions and CVT is essential for the efficiency and isolation behavior of the system because it allows the torque converter to be locked as early as possible or to be operated with very little and finely controllable slip. The converter lock-up is controlled by applying pressure to the lock-up clutch piston. However, interference factors include fluctuations in the converter charging pressure and differences in the centrifugal oil pressure on both sides of the piston [9].

Figure 8 shows the design of the four-channel torque converter that allows the lock-up clutch to be controlled very precisely and independently of dynamic operating conditions. Two of the chan-



8 **Design of the four-channel torque converter with four hydraulic connections for compensating centrifugal oil pressure**

nels are used for the flow through the converter. The third channel serves to control the clutch, and the additional fourth channel serves as a pressure compensation chamber. This fourth channel ensures identical fluid conditions on both sides of the piston. The dynamic centrifugal force oil pressure is identical on both sides of the piston because the diameters of the actuation and compensation chamber seals are identical and the oil rotates at engine speed on both sides independently of the slip speed. In addition, the pressure chambers of the clutch are shielded from variations in converter charging pressure.

Double clutch transmission

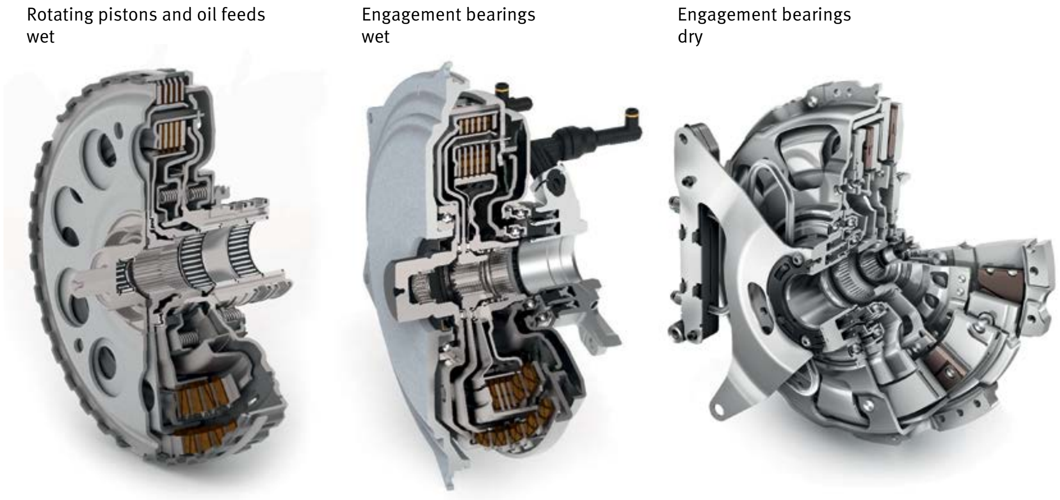
In transmission development so far, the focus has been on the direct costs of components and systems. Against the background of increasingly strict CO_2 legislation, however, the indirect costs of consumption and power losses that lead to increases in CO_2 emissions due to drag losses and power consumption of actuators and cooling systems must also be reviewed [10].

Figure 9 shows three designs by Schaeffler with varying direct system costs and indirect operating costs. With regard to power loss, the systems with engagement bearings offer an advantage (center and right). The actuation system consists of a clutch slave cylinder and is activated by a hydrostatic clutch actuator (HCA). It uses a leak-free hydrostatic section to transmit the actuation energy to the clutch with little loss. Due to a travel sensor, the HCA does not need any additional return force gradient in relation to the engagement travel. This allows the power losses in the clutch to be minimized.

A more cost-driven approach is characterized by the use of a mechanically driven hydraulic pump (left), and, if necessary, supported by an additional electric pump to cover volume flow peak demands The transmission of clutch actuation energy is not achieved with the help of bearings but rather with rotary oil feeds, and the actuation of the transmission shift system is achieved hydraulically or electro-mechanically. The system does have its disadvantages because of the continuously running pump, but does well overall in terms of system costs.

Manual transmission

In spite of an increasing variety of transmission concepts, the manual transmission continues to be one of the most important transmission designs with regard to volumes. This means that efficiency gains have a particularly strong effect globally. In order to utilize any additional potential offered by new technologies to reduce fuel



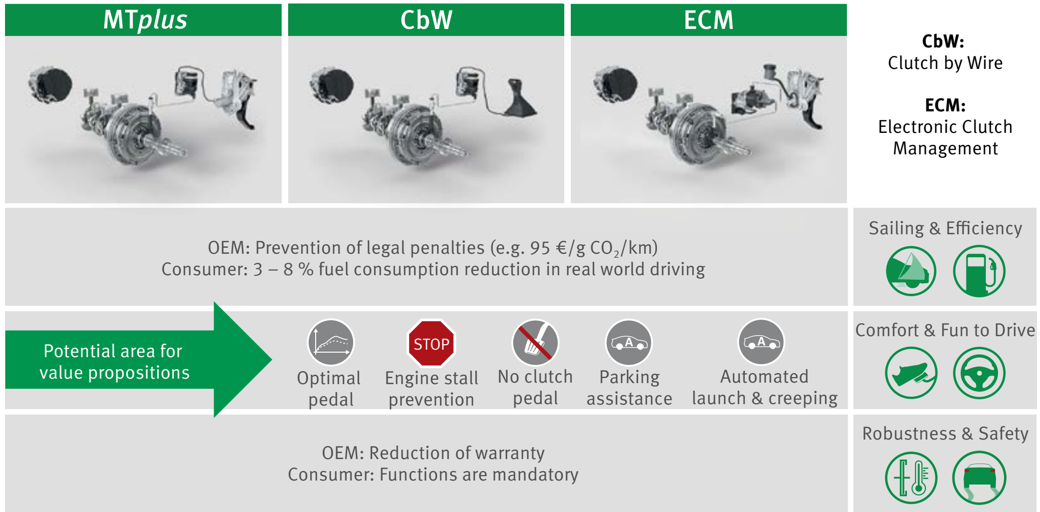
9 **Wet double clutch with rotary oil feeds and rotating cylinders (left) as well as wet and dry double clutch with engagement bearings (center and right).**

consumption and CO_2 emissions, however, it is advantageous to automate the clutch in manual transmissions. Based on measurements by Schaeffler, using a sailing only strategy on an RDE-compliant test track allows fuel and CO_2 savings of between 3 % and 5 %. In addition, the expanded recuperation of brake energy enables improvements of around 5 % with P0 mild hybrids and, when used in combination, of around 8 % [11].

Figure 10 shows three variants of how clutches of manual transmissions can be automated. MT*plus* (left) is an entry-level system for the partial automation of the clutch that allows driving strategies such as sailing without the driver's involvement, even for manual transmissions. An additional actuator with a suitably small design is used in parallel with the existing master cylinder on the clutch pedal. Clutch-by-wire systems (CbW) are also perceived as normal manual transmissions with a clutch pedal by the driver. The actual actuation of the clutch, however, is always carried out by an actuator. This allows additional functions such as the start assist, traffic jam assist and slip control. As the name "by wire" suggests, there is neither a mechanical nor a hydraulic connection between the clutch pedal and the slave cylinder of the clutch. This also applies for ECM systems (right) that do without the clutch pedal altogether. Instead, a sensor on the gear selector recognizes the intent to shift gears and controls the opening of the clutch via the actuator. Besides its efficiency potential and new functions, the automated clutch has the added benefit of being able to protect the manual transmission from excessive loads, for instance in cases of misuse.

Summary

With the growing number of drive concepts, the variety of transmission designs and components also continues to increase. This paper has shown



10 Variants for clutch automation: MT*plus*, clutch-by-wire (CbW) and electronic clutch management (ECM)

numerous examples of the ways that this new variety of individual solutions advances the development of transmissions – including with regard to hybridization in all its complexity and electrical-only mobility. General trends such as automation share the goal of using as little primary energy as possible for driving motor vehicles, thus reducing CO_2 to the lowest amounts technology can achieve as well as increasing the driving range for a given energy accumulator.

Here, all relevant components in the transmission system must be included, from the further optimized transmission bearing to effective damping concepts that make drive strategies for reducing consumption and emissions possible to actuators with minimal power consumption. The implementation of customized concepts requires sophisticated CAE tools that put subsystems and systems in context. The result is efficient solutions for individual transmission designs that only involve minimal losses when transmitting power and converting torques.

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9

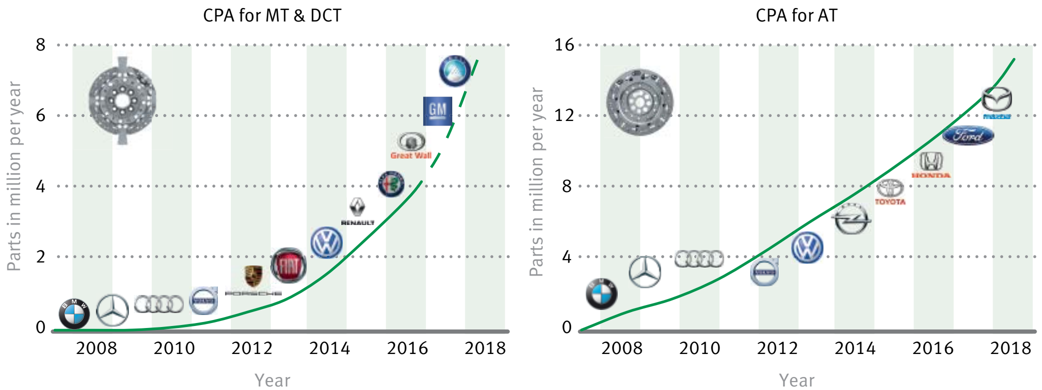
Best-in-Class Dampers for Every Driveline Concept

Dr. Ad Kooy
Roland Seebacher

Introduction

As a consequence of the increasing variety of drive concepts, an ever greater range of vibrations are generated in the drive train which must be damped between the engine and the transmission. Schaeffler believes that classic internal combustion engines will continue to play an important role for a long time to come [1]. In the future, numerous concepts are expected to help further reduce fuel consumption. Driving with a long ratio, for instance, reduces internal combustion engine losses due to lower speeds. If, however, the internal combustion engine is operated at just above the idle speed range, it is in a particularly economical range but this also produces strong, low-frequency excitations with high vibration amplitudes. In addition, the torque must be increased to prevent loss of performance. Other vibration-related challenges arise from cylinder deactivation, downsizing and the increasing number of hybrid vehicles with various arrangements of the internal combustion engine, electric motor and transmission. The on-demand switching between electric motor drives and internal combustion engine drives that is typical for hybrids and that is intended to go widely unnoticed by the driver leads to additional tasks for the dampers in the starts and stops of the internal combustion engine.

The trends for the drive train of the future result in high requirements for damping systems between the engine and the transmission that are increased even further by the growing expectations drivers have regarding comfort and driving dynamics. In order manage the ever growing number of variants, transmission manufacturers rely on the increased use of CAE tools with the aim of including the requirements for vibration damping and noise behavior (noise vibration harshness, NVH) as early as possible in the development process and reduce the number of tests with real prototypes This requires high development quality, something that Schaeffler ensures with actions such as the simulation of vehicle influences for tolerance evaluations and damper optimization [2]. Beyond that, it is important to develop effective and customized damping systems for what tend to be smaller and more complex design spaces. This paper aims to show some of the ways that Schaeffler looks for even better operational principles and keeps driving damper technology with innovative combinations.



1 Schaeffler production of centrifugal pendulum-type absorbers per year for manual transmissions and double-clutch transmissions (left) as well as automatic transmissions (right)

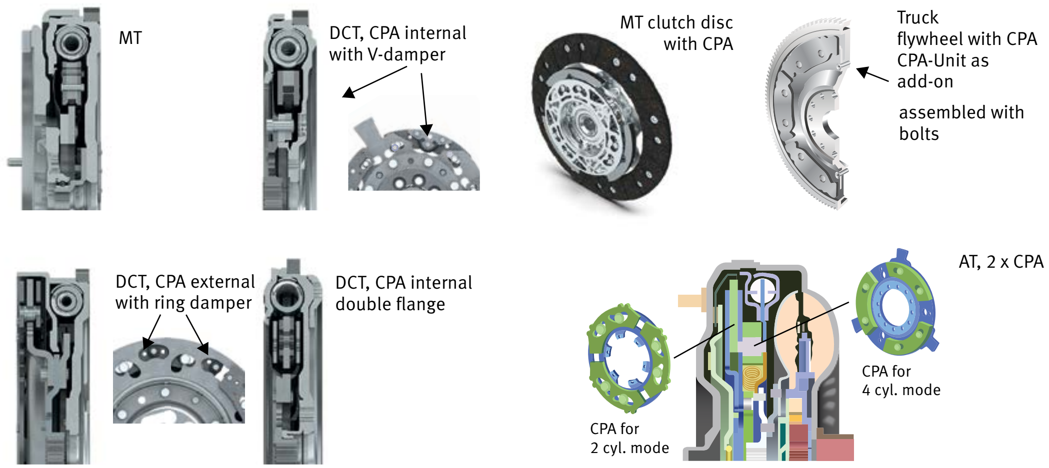
Refinement of the centrifugal pendulum-type absorber

Initial situation

Centrifugal pendulum-type absorbers (CPA) have been used since 2008 to isolate the torsional vibrations between the engine and the transmission and are now widely used. As shown in Figure 1, around 20 million CPA are produced every year for transmission applications. Around two-thirds of those are installed in the converter in automatic transmissions and around one-third in manual transmissions and double-clutch transmissions. For arrangements in the dual mass flywheel (DMF), a distinction is made between internal CPA – which are arranged in a space-saving fashion under the arc spring – and external CPA which are placed next to the arc spring and achieve an even better isolation of the rotational irregularities due to the higher moment of inertia.

The great market success of the CPA can be attributed, among other things, to the fact that the physical principle inevitably results in a balance between excitation and pendulum vibration. In this process, the vibration amplitude keeps increasing until the exciting mass no longer vibrates. This means that the CPA can compensate different phase positions that can occur at higher speeds or when sailing. The natural frequency of the damper changes just as much via the centrifugal force increasing quadratically with speed as the main excitation firing frequency of the internal combustion engine. The most important disturbance is the guide track accuracy of the rotating masses in conjunction with the flange, pendulum and roller geometry that Schaeffler is able to guarantee with the highest precision due to its state-of-the-art manufacturing processes on an industrial scale.

Pendulum mass guide friction may constitute another disturbance. That is why very rigid rollers are mostly used here that are virtually friction free. Hardly any heat is lost because the CPA stores the engine's pulsating mechanical energy temporarily in an efficient manner. Finally, it releases the energy and damps the powertrain.



2 Applications for the centrifugal pendulum-type absorber in the DMF (left) and other applications (right) on the clutch disk, in the single mass flywheel and as a double CPA in an automatic transmission converter

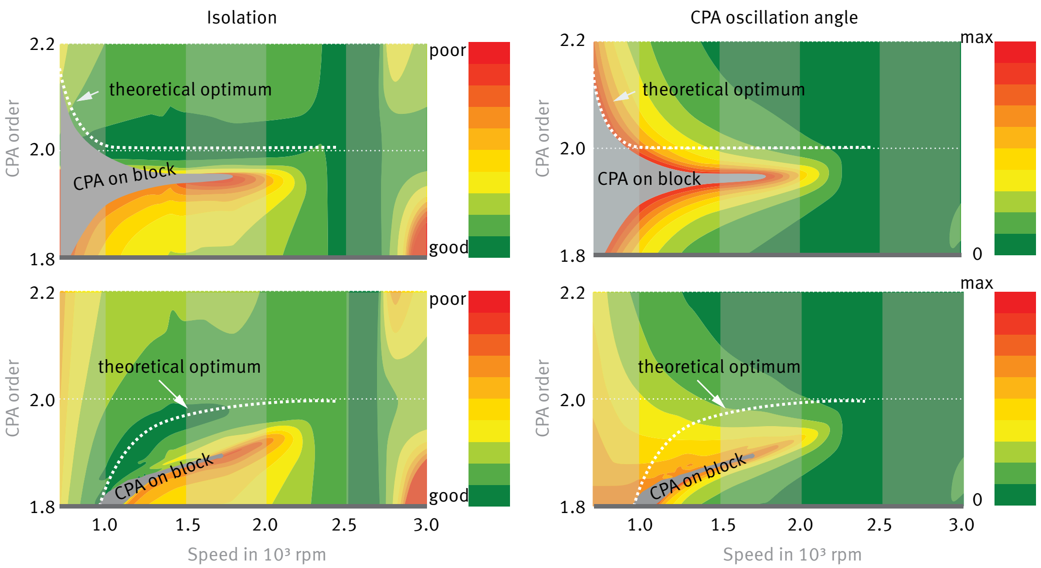
Under tolerances, a slight offset of the pendulum masses may also occur or the flange may be in a tilted position. Both may cause the pendulum to touch the flange. For this reason, ribs on the roller keep the components apart. In addition, sliding elements are used between the flange and the pendulum to minimize wear and friction [3].

This wide range of applications has led to the use of CPA in many powertrain positions, Figure 2. It was initially developed to improve isolation in the DMF and in automatic transmission converters and, in a subsequent step, designed for the clutch disk. Meanwhile, some automatic transmission converters also use two CPA at the same time. In the double flange CPA, the pendulum mass is arranged between two flanges to facilitate the arrangement of the damping elements that are necessary for what have become very frequent starts and stops.

The CPA in the single-mass flywheel of a commercial vehicle drive is another development. A special feature that must be mentioned here is the flange on which the pendulums are arranged and that represents the geometry of the guide tracks. It is screwed to the flywheel, allowing a modular design.

Design of centrifugal pendulum-type absorbers from idle speed

The design of CPA in the range starting at idle speed has three objectives: First, it is important to achieve good isolation even at low speeds. This requires as much mass as possible and a vibration angle that is as wide as possible. Second, it is important to prevent the pendulum from lifting off from the guide track at low speeds and wide vibration angels to prevent wear Third, the goal is to cause as few damper stops at the



3 **Basic simulation for optimization of the guide track curvature of the pendulum and the flange in combination with the roller diameter using the example of an CPA without (top) and with (bottom) spring elements between the pendulums (couple pendulum).**

end of the guide track as possible at low speeds in order to preserve the rubber stops that absorb part of the impact energy to prevent inherent noise.

The basic simulations based on a vehicle model with a four-cylinder engine in a P2 hybrid drive that are shown as color charts in Figure 3 indicate the most important dependencies of the selected order. The relevant order is achieved in the CPA via the guide track curvature of the pendulum and the flange in combination with the roller diameter. The isolations reached and the vibration angles required for this are shown in relation to the speed.

The change in the guide track curvature in relation to the vibration angle in the CPA results in a change of the order. The effect of this change can be seen in the color diagram. If an order different from the second order is used, isolation deteriorates, but the vibration angle is also reduced, preventing the pendulum from stopping. That is why relatively strong imbalances are used at the end of the guide track.

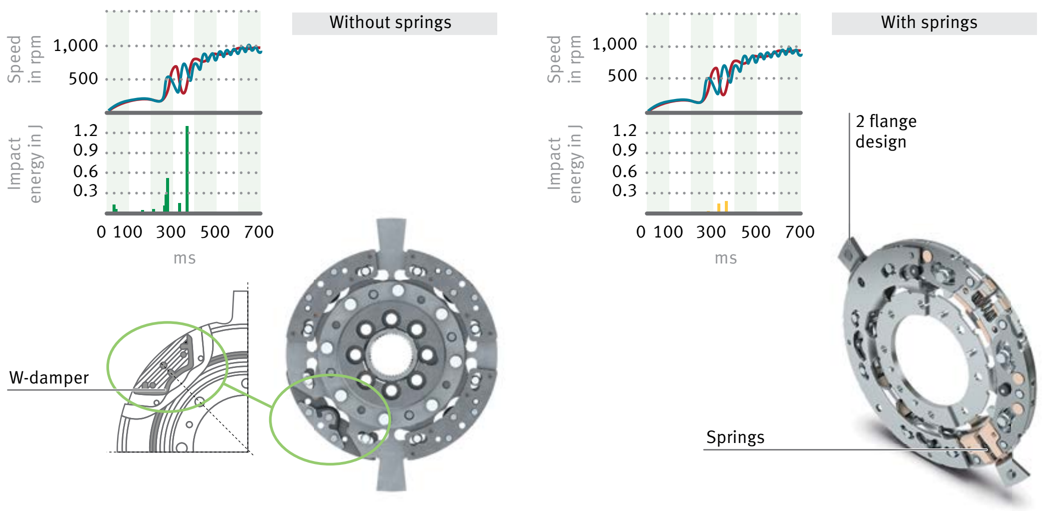
In the simulation of the isolation effect of an CPA it must be remembered that, with large vibration angles, the guide track areas for small vibration angles are also over-swept which affects the resulting integral order. For this reason, the basic simulation is performed initially at full load. At part load with its smaller vibration angles, the curve of the order and thus the isolation effect is different even though the guide track is identical. Since the engine excitations are also lower at part load, however, no compromise is generally required when designing the guide track geometry. As a result, very good isolation can be achieved with the CPA. It is only in the range just above the idle speed where it is not quite ideal. This is a starting point for improvements. For internal CPA, the use of significantly greater masses is a solution that can be considered if there is sufficient design space. Another option is the use of CPA positioned next to the arc spring and located further outside radially, if there is enough axial design space. The close connection between design space specifications and potential design solutions for damping the torsional vibrations underlines the fact that validation is required at an early stage through valid simulations.

Design in the start-and-stop range below idle speed

Speeds go down quickly when the engine is turned off. Starting at a certain point, gravity dominates centrifugal force, and the pendulums lose contact with the rollers. This may lead to noticeable, undesirable noise, particularly for external CPA whose pendulum masses are positioned far outside radially and that are not encapsulated as they would be in the internal CPA. Compared to this, the engine start is characterized by relatively strong excitations, so that stopping at the end of the guide track dominates here.

It is possible to calculate the impact energy of the vibrating components at the start, but this is more difficult when the components moving in relation to each other rebound because this leads to rotary motion that are dependent on the impact point and guide track tolerances. Figure 4 shows a potential solution for this problem. Undesirable intrinsic rotary motions can be largely prevented with the help of a so-called W stop damper via a two-point guide of the pendulum. The elastomer damper used here has helped achieve a significant reduction in these rotary motions. As with the V stop damper, the impact on a wide rubber surface significantly increases the impact energy that can be absorbed compared to previous, ring-shaped stop elements.

For internal CPA integrated into the flange, the noise described when turning off the engine can



4 Design of an internal CPA with W stop damper (left) as well as tangential connecting elements on the pendulums (couple pendulum, right)

generally be reduced to a reasonable level. This is more difficult for external pendulums. One solution is the so-called couple pendulum in which the pendulums support each other in a circumferential direction through springs.

Here the spring preload is selected in such a way that the pendulum remains in the guide track even if the engine is at a complete standstill, thus not losing contact between the pendulum, the roller and the flange. The effect of the spring forces overlapping with the centrifugal forces is largely compensated by correcting the order of the track. This type of spring arrangement is particularly helpful for first order pendulums, such as those needed for cylinder deactivation from four to two active cylinders. This is because gravity also generates a first order excitation in a rotating pendulum, which increases the excitation from cylinder deactivation.

The principle can also be applied to an internal pendulum. Here too the so-called two-flange design is used, Figure 4 (right), which allows an easy integration of the springs for pendulum preload. For cost reasons, however, this principle is only used for very high requirements and comfort demands such as hybrid vehicles with frequent restarts that occur automatically and without the driver's involvement.

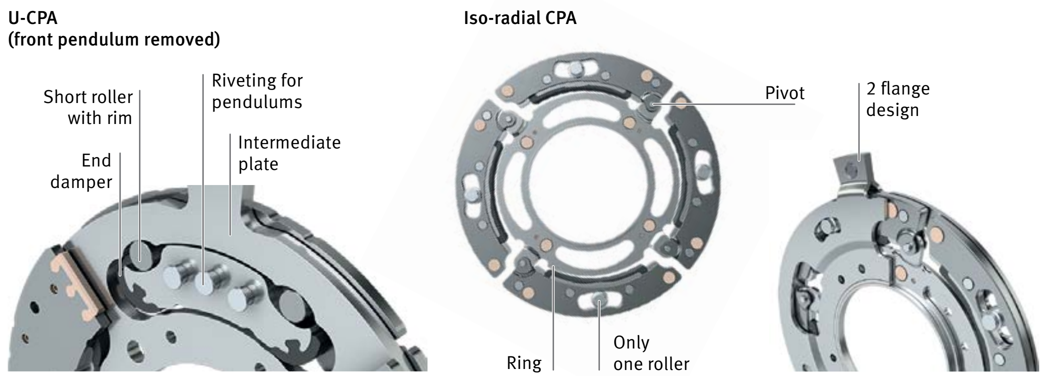
Another potential solution is an CPA design principle with a U pendulum, Figure 5 left. Here the CPA is supported by the rollers right below the flange via a hardened intermediate piece with sheets installed on the side and located in a flange cavity. This way, the rollers can be designed much shorter and lighter which significantly reduces noise developing when stopping. Noise in the CPA is generated by the pendulum hitting the flange, which can be largely compensated by appropriate rubber dampers, as well as by the rollers hitting the pendulum and the flange, which is more difficult to manage. The sheets and the connecting piece together form a U-shaped cross section. This arrangement also allows slightly higher pendulum masses for improved isolation. The narrow rollers require a guide which is usually secured through the roller lips.

The new so-called iso-radial pendulum solution takes an entirely different approach, Figure 5 right. In this approach, the individual pendulums are connected in one point by a ring not located in the torque flow, which means that the pendulum masses are now synchronized. One of the usual two spherical rollers is eliminated, causing the pendulum to carry out a swiveling motion rather than a purely radial motion. This design eliminates the first order excitation from grvity on the individual pendulum masses. However, this principle does not help against contact loss at low speeds while stopping; here too noise has to be controlled by stop elements. One advantage though is the fact that coupling via springs can be eliminated. This way, improved isolation can be achieved, particularly at low speeds.

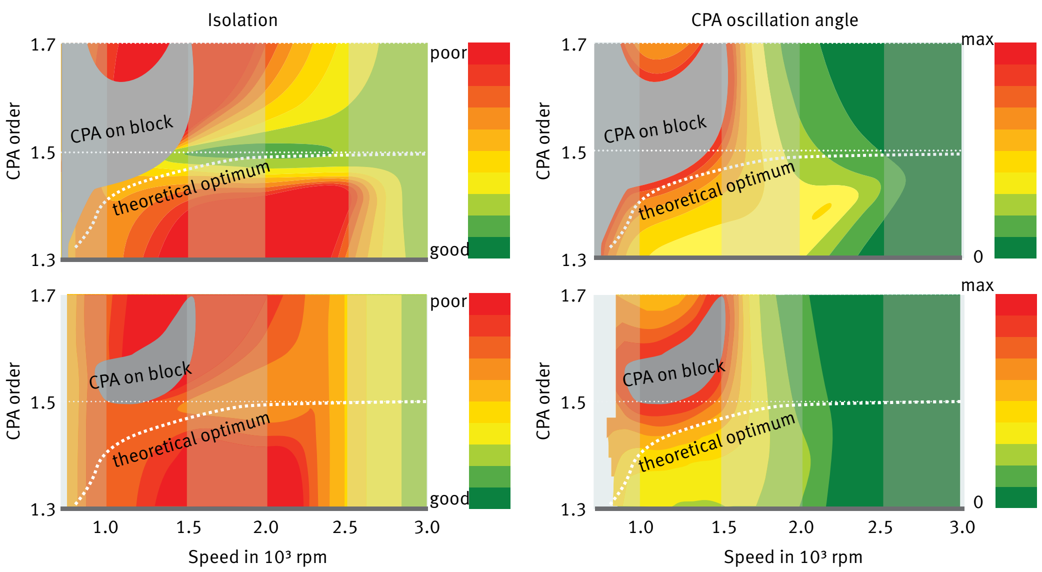
Centrifugal pendulum-type absorber on clutch disk

The CPA cannot only be arranged in a DMF but also directly on the clutch disk at the transmission input shaft. This design and basic functions were already presented at the 2014 Schaeffler symposium [3]. It has successfully passed serial testing and will soon be available on the market. The design was basically carried out in the way shown above. Figure 6 shows analog the isolation and the vibration angle similar to Figure 3, in this case for a three-cylinder engine without (top) and with friction elements on the CPA (bottom). For small vibration angles and high speeds, a guide track arrangement was chosen that is basically identical to the main excitation 1.5 order but a much lower order for larger angles in order to avoid resonances with the powertrain. Being thoroughly familiar with the resonances in the powertrain and their dependency on the relevant gear is an essential requirement for optimizing the isolation behaviour.

In addition, sufficient contact forces at low speeds and large vibration angles must be ensured to prevent wear caused by slipping of the rollers. For this purpose, slipping was analyzed experimentally with the help of Hall sensors right on the CPA. This made it possible to measure the relative motion between the two rollers, tilting and the



5 Design of an CPA with U pendulum (left) and an iso-radial pendulum (right)



6 **Basic simulation for optimization of the guide track curvature of the pendulum and the flange in combination with the roller diameter using the example of an CPA on the clutch disk without (top) and with (bottom) friction elements**

rollers' axial motion. Figure 7 left shows that multiple pendulum motions cause one roller each to slip at the end of the guide track while the motion of the rollers going back is linear, i.e. with a constant distance. In the picture, this leads to what appears to be a hysteresis. An analysis based on simulation showed that the contact forces were significantly lower for the slipping roller. Based on this, a parameter for the minimal contact forces can be established.

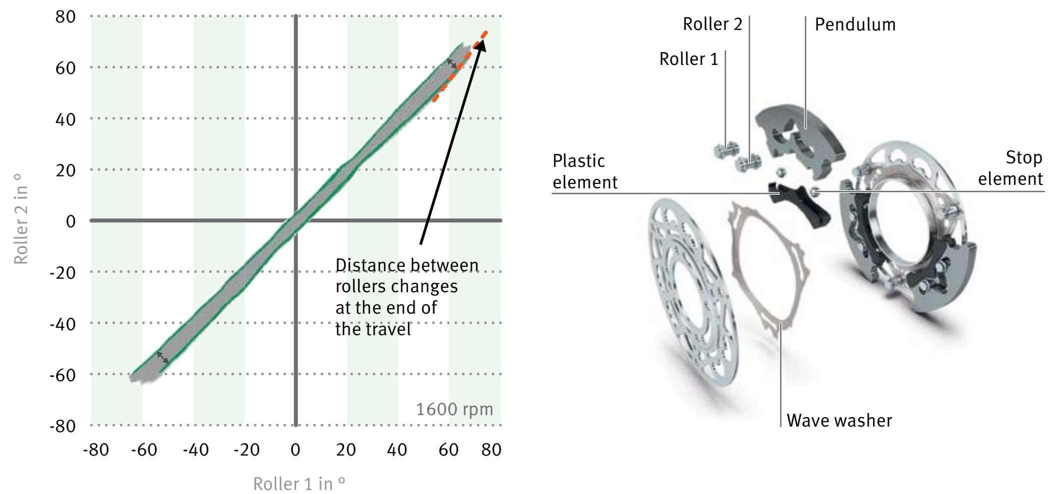
The best compromise between necessary contact forces and isolation behavior can be found by inserting friction elements on the CPA, the normal force coming either from a waved disk or a diaphragm spring between the pendulum and the flange, Figure 7 right. These do not have a noticeable effect on isolation at higher speeds, but they do help optimize the vibration angle of the CPA at lower speeds. In addition, they ensure the axial stabilization of the pendulum and reduce stopping noises. The friction elements are made from plastic, have very high wear resistance and achieve a sufficient life even for the very high number of vibrations. This design principle can be used for both three-cylinder engines and four-cylinder engines.

Centrifugal pendulum-type absorbers in hybrid vehicles

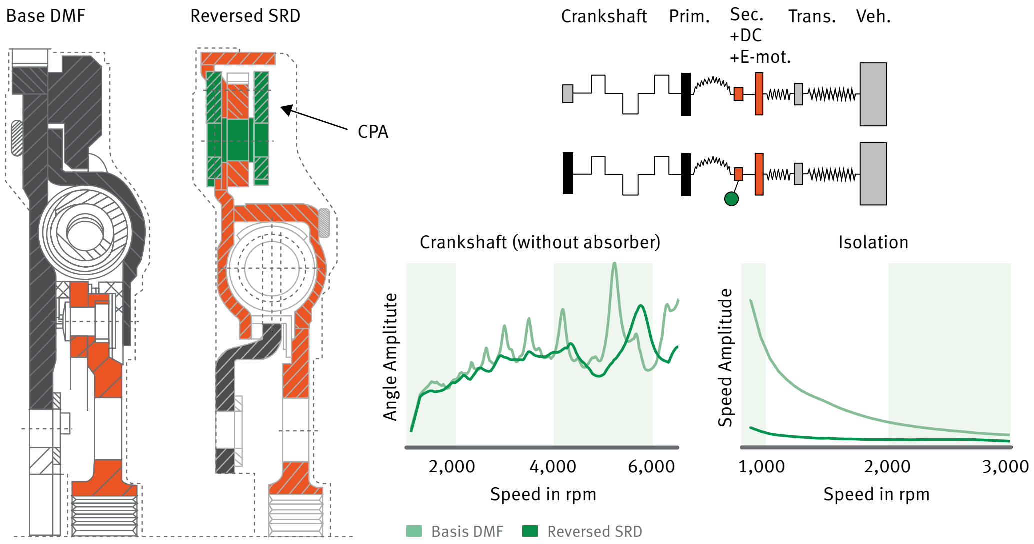
With plug-in hybrids, the goal is for perceptible vibration-related differences between electric motor-only driving and driving with an internal combustion engine to be as small as possible. This results in complex requirements for isolation that are further complicated by the fact that the electric machine often limits the design space for the damper.

The arrangement of the electric motor also plays a significant role. In P0 and P1 arrangements, the additional torque of the electric motor must be transmitted by the damper; this is not the case in P2 to P4 arrangements. In particular for P2 arrangements, the additional inertia of the electric motor helps with the damping of torsional vibrations. The optimum solution depends greatly on the design space. Various operational principles such as masses, spring dampers and CPA can be combined here. Due to the drag start to re-start after sailing with the internal combustion engine turned off, there is another critical operating point in addition to regular starting. The restart should go unnoticed by the driver if possible. If an CPA is used, a couple pendulum, as described above, can help prevent stopping noises.

An interesting approach for a particularly slim damper is an CPA arranged on the secondary side towards the transmission as a so-called "reversed small radial damper" (RSRD), Figure 8. In this concept, the flange that actuates the arc spring is bolted right on to the engine crankshaft. Due to the transfer of the primary inertia to the front end of the engine, damping of the crankshaft's natural frequency of 300 to 500 Hz has to occur at the rear end of the crankshaft. Simulations have shown that the arc spring can dampen vibrations in the crankshaft given the appropriate design. This allows a conventional, separate crankshaft damper to be eliminated. Another advantage: The arc spring is arranged radially under the CPA, the only component installed outside of this is the actual CPA with burst protection. This creates additional design space for the electric machine, along with significantly improved isolation.



7 **The distance between the rollers with multiple pendulum motions (left) and inserted friction elements for roller stabilization (right)**



8 **Reversed small radial damper (RSRD) with CPA and flywheel on front end of the crankshaft**

New approaches to vibration damping

Analyses at the operating principle level

In Schaeffler's search for new damping concepts, one approach consists of using simple models to analyze different operating principles at the theoretical level. These models focus on operational parameters and are compared on a physical basis while maintaining the same inertia and spring energies. Limiting factors such as design space, costs and feasibility are not included until the second step. If an operating principle proves to be promising, the function of the new concept is analyzed with regard to expanded operating situations such as an engine start. In a simulated vehicle model, this is carried out with all relevant non-linearities.

Generally, there are three types of vibration damper systems:

- Active systems: Energy is supplied to and extracted from the system externally. This can be carried out, for instance, by an electric motor in the powertrain to produce a counter-excitation. Another possibility are semi-active systems in which a parameter – such as a spring rate – is simulated specifically.
- Passive systems: Energy is stored temporarily, alternating between kinetic and potential energy, such as in a classic spring-mass damper such as the DMF or the CPA which temporarily stores potential energy in the centrifugal force field.
- Slipping systems: Energy is removed from the system. This can be achieved hydraulically, such as in a torque converter, or through friction, such as in a clutch [6].

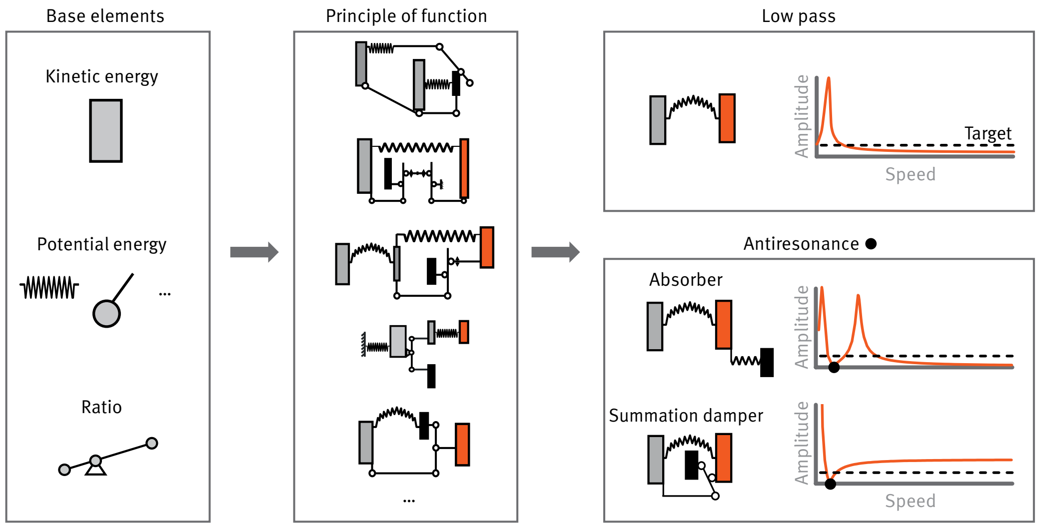
Good results can also be achieved with an efficient combination of all of these systems.

The section below deals exclusively with passive systems. As shown in Figure 9, basic elements for kinetic energy, potential energy and ratio can be combined arbitrarily and thus describe a nearly unlimited number of operating principles [7].

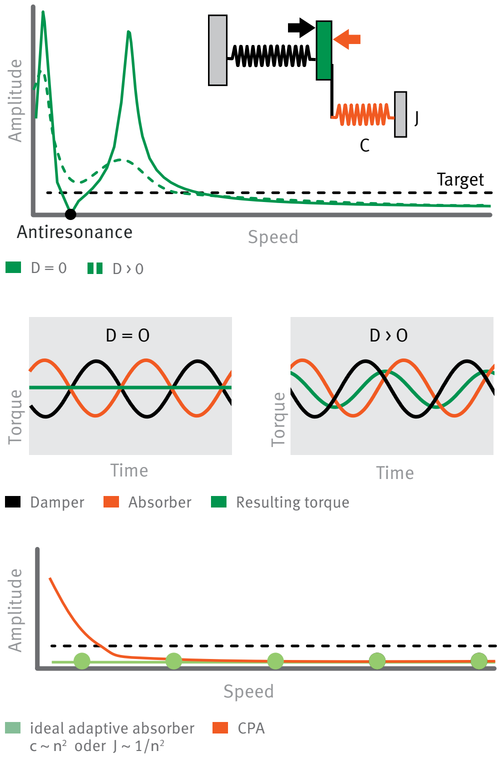
The analysis of these operating principles shows two basic characteristics of all systems capable of vibrations: Each degree of freedom contains a resonance and an anti-resonance. Essentially, the goal is to manage the resonance while also using the anti-resonance effectively.

The anti-resonance principle is particularly promising when it comes to isolation at low speeds. That is because the vibration movement for a particular frequency is erased entirely. The torque fluctuations add up on a rotating mass so that the sum of all torques equals zero. Figure 10 shows the anti-resonance principle can be achieved by a simple damper. The rotating mass is set in motion on one side by torques acting on them in a certain frequency. A damper, in turn, is supported on the vibrating mass by a spring and causes a torque that is phase shifted by 180°. The torque fluctuation of the damper grow with increasing frequency until it is equal to that of the mass that is acting on it and erases its motion entirely. Theoretically, designing the frequency of the anti-resonance to an arbitrary frequency. However, this creates an additional natural frequency that can manifest itself in a disturbing resonance magnification. It must be remembered that damping in the system has a severe effect on anti-resonance.

Anti-resonance can also be achieved by means of a so-called summation damper or power splitting. Here too, torque amplitudes are added in such a way as to ensure that they completely erase each other for one frequency. These systems have a ratio as an additional parameter. The advantage of this is that no additional resonance is generated because there is no additional degree of freedom from an additional spring.



9 **Various operating principles for damping vibrations**



10 Implementation of the anti-resonance principle using the example of a damper

It is true for all anti-resonance systems that the amplitude of the counter excitation should be exactly inverse. It is possible to select the anti-resonance freely by carefully selecting the parameters. However, if the parameters remain constant, the anti-resonance also remains on a fixed frequency. The excitation frequency though changes in the vehicle in proportion to the engine speed. The problem can be solved by changing one of the relevant parameters in proportion to the excitation frequency. The CPA is one example. It uses the centrifugal force as an energy accumulator and thus, in a certain way, has a spring rate depending on the engine speed.

Limiting factors here include the limitation of the vibration angle by the design space and the non-linear curve of the tangential return force that decreases as the pendulum angle increases. This causes the CPA to lose potential with decreasing speed.

Application of operating principles on the centrifugal pendulum-type absorber

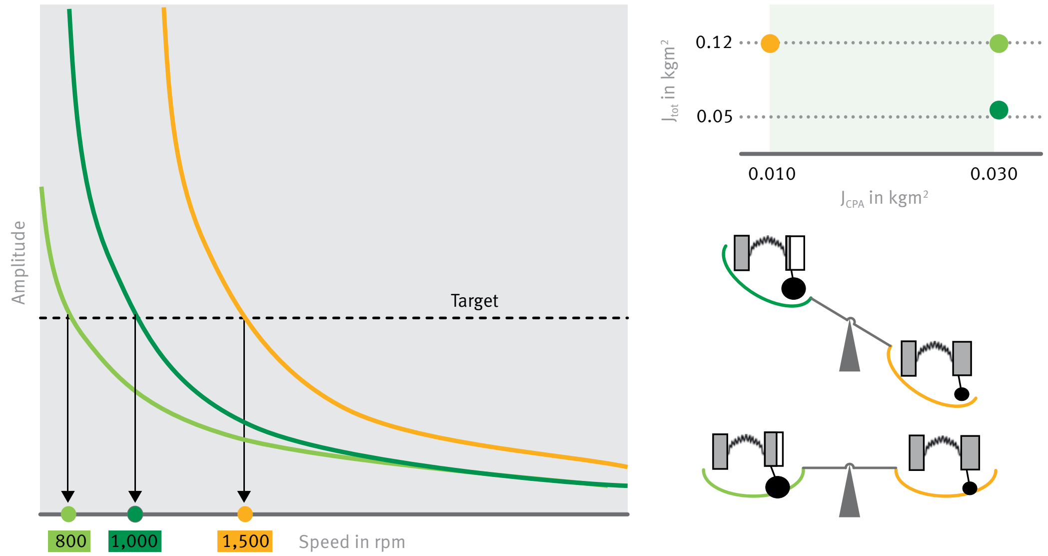
The application of the operating principles described above can open up new opportunities for the CPA. Insufficient design space, for instance, can be counteracted by redistributing the mass of the damper system in the direction of the CPA, Figure 11. This helps improve isolation with the same total mass; a reduction in the overall mass is also a possibility.

Application of operating principles for dampers on intermediate flange

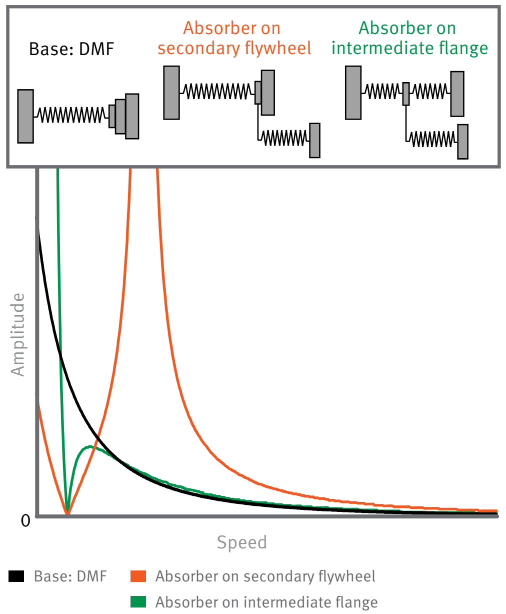
However, design spaces are not always ideal in terms of offering sufficient CPA mass. That is why analyzing the interaction of resonance and anti-resonance is well worth the effort. Operating principles can be helpful here too: If the damper is not arranged right on the secondary mass to be damped, but rather symmetrically between the primary and secondary mass, anti-resonance may result – without the interfering resonance in the higher speed range.

Implementation: Damper on intermediate flange

The symmetrical arrangement can be achieved by placing the damper between the two masses on a so-called intermediate flange. This is a potential option in combination with an CPA if an CPA that



11 Theoretical potential of the CPA in the redistribution of the mass down to a reduction of the total mass of the damper system



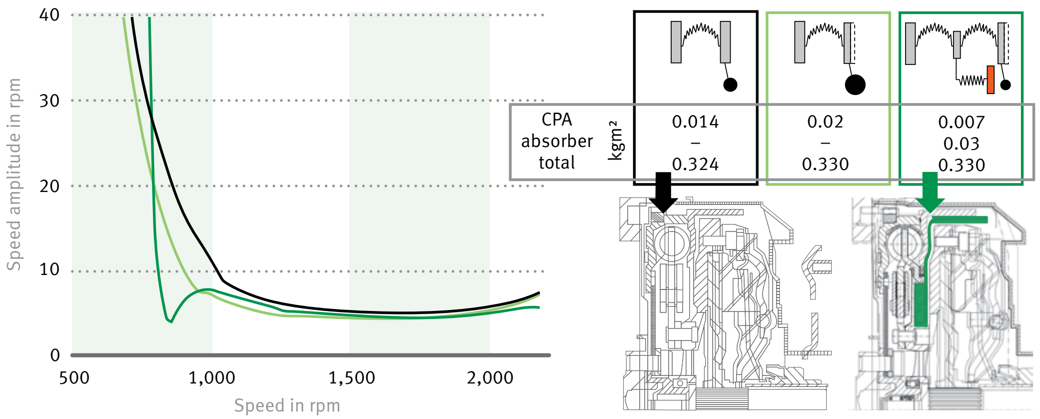
12 Operating principle of the damper on the intermediate flange: Achievement of an anti-resonance without interfering resonance in a higher speed range

is too heavy leads to strength problems, if possible limitation stops cause noise or if the available design space leaves too little clearance for radial deflection.

Figure 13 shows an example with an awkward design space. The available design space does not have enough room for the radial deflection of the CPA. That is why the damper is placed on an intermediate flange. In combination with the CPA already in place, engine speeds can be decreased by around 300 rpm without any additional mass, which makes fuel consumption savings of approx. 6 % possible.

Summary

This paper has shown some of the ways in which Schaeffler reacts to the growing variety of drive concepts and the resulting requirements for vi-



13 **Placing a damper on an intermediate flange to utilize anti-resonance**

bration damping. These requirements are very diverse and include changeovers from drives based on electric motors to those using internal combustion engines and the resulting start and stop challenges, cylinder deactivation, reduced design spaces, driving at low engine speed and increased NVH requirements. The refinement of the centrifugal pendulum-type absorber principle and the combination of various damper principles lead to function and cost optimized designs. In addition, the approaches presented in this paper allow another decrease in speed for driving in order to reduce consumption and emissions. The design methods for centrifugal pendulum-type absorbers have also been described.

In the search for new damping concepts, the operating principles they are based on play an important role. The resonance and anti-resonance principle in particular is suitable for developing matching damping concepts for complex design spaces.

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10

The Manual Transmission Has a Future

E-Clutch and Hybridization

Dr. Roland Welter
Markus Kneißler

Introduction

In view of an increasing variety of transmission concepts, the manual transmission continues to be one of the most important transmission designs. Worldwide, 43 % of all car buyers order their new car with a manual transmission. Most manual transmissions are sold on the Indian subcontinent, but they are also very popular in Europe and China. Although the percentage of manual transmissions will decrease slightly over the next few years on a global scale, the upward trend will continue in a growing market when measured in absolute production quantities. According to market observations by Schaeffler, the number of manual transmissions produced worldwide, which amounted to 40 million units in 2016, will continue at its current high level.

There are distinct regional differences. While, based on these assessments, the market share in India will be at 90 % in 2025, it will reach 57 % in Europe, 51 % in Southeast Asia and 45 % in China [1]. The major reasons for the popularity of manual transmissions in these markets are a comparatively low price in combination with very high efficiency. For a long time, manual transmissions were superior to automatic transmissions in terms of fuel consumption. Then, however, automatic transmissions used optimized gearshift curves and were able to achieve advantages in driving cycles compared to the predefined shifting points for manual transmissions. These consumption benefits were primarily made possible by down-speeding, supported by continuously improved torsional vibration dampers and, in torque converter transmissions, by closing the lock-up clutch at a much earlier stage [2, 3]. Under real driving conditions and with carefully selected shifting points, manual transmissions can further demonstrate their efficiency benefits.

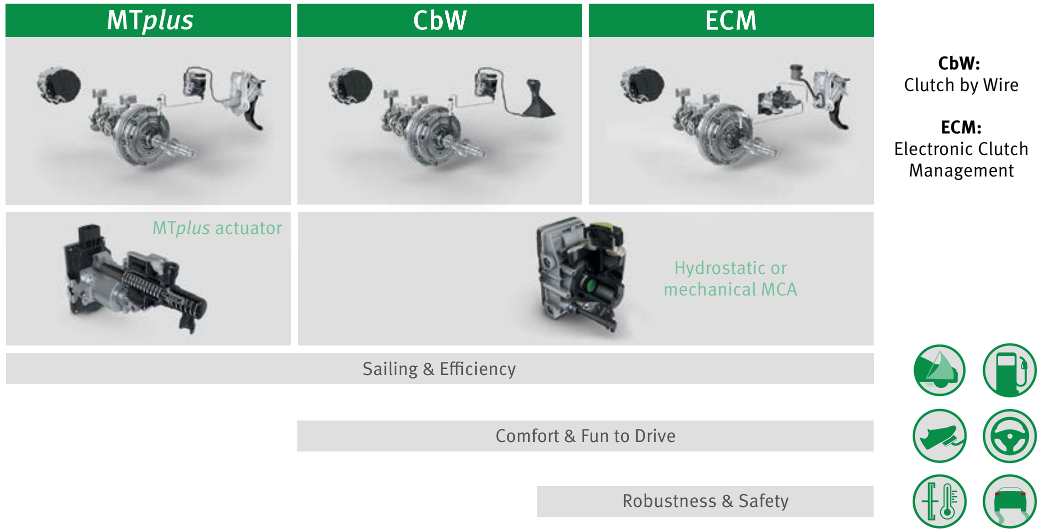
In order to use the opportunities offered by new technologies to reduce fuel consumption and CO_2 emissions, it is necessary to automate the clutch in manual transmissions. For instance, achieving fuel-saving driving strategies such as coasting with the engine turned off or the recuperation of braking energy with 48 V hybrid systems in P0 or P1 arrangement is neither comfortable nor feasible without the driver's assistance unless an automated clutch is used.

In view of global climate goals, the utilization of this potential is extremely effective because, as shown above, the worldwide usage of manual transmissions will continue. Using a coasting-only strategy can achieve fuel and CO_2 savings of 3 % to 5 %, and the expanded recuperation of braking energy with mild hybrids can reach around 5 % and, when combined, around 8 %. This paper aims to show solutions designed by Schaeffler that follow this approach and enable the automation of manual transmissions for more efficiency and reduced emissions.

Automated clutch actuation

Initial situation

Coasting with the engine turned off and the recuperation of brake energy with 48 V hybrid systems in a P0 or P1 arrangement require optimized clutch actuation. Since the driver usually is not able to perform this operation, automation of the clutch becomes mandatory. When coasting, the primary objective is to open the clutch at the right time so that the engine continues to idle at low speed or, even better, can be turned off altogether. In hybridization, the automated clutch is useful for disconnecting the internal combustion engine and the electric motor for coasting, according to the current traffic situation and driving strategy, or for connecting a fuel overrun cut-off to support recuperation.



1 **Variants for clutch automation:**
MT*plus*, Clutch-by-Wire (CbW) and Electronic Clutch Management (ECM)

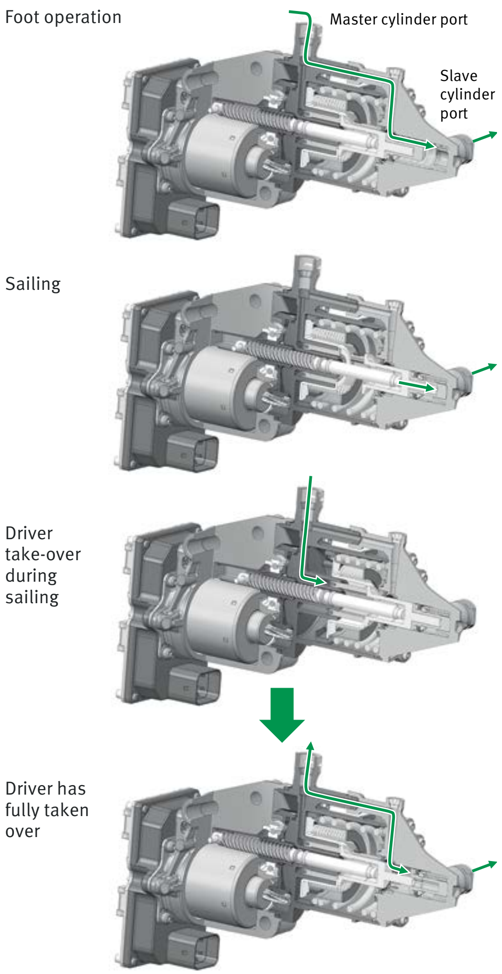
An automated clutch can also be used to implement additional functions in a simple way. For instance, it permits extended assistance functions such as the traffic jam assist that carries out starts and stops in slow-moving traffic on the interstate. Other functions are the protection of the clutch and the powertrain from excessive loads, such as in case of misuse, and the increased safety of vehicles and passengers. One example here is the automatic opening of the clutch when the assistance system initiates an emergency brake application to prevent the engine from stalling.

Implementation variants of the automated clutch

Three implementation variants of the E-Clutch are described below, each of them different with regard to available or missing clutch pedals as well as with regard to the degree of automation, Figure 1. This includes the partly automated and low-cost MT*plus* solution, Clutch-by-Wire (CbW) systems with a clutch pedal emulator as well as electronic clutch management (ECM) systems that can do entirely without a clutch pedal.

MT*plus* is an entry-level system for the partial automation of the clutch that is below the technical level required for CbW and ECM systems [3]. An actuator with a suitable design is used in addition to the existing master cylinder on the clutch pedal. This actuator, whose functional principle is described in more detail in the following chapter, has a time interval of 300 ms to open the clutch, providing sufficient dynamics for the basic function of entering and exiting the coasting mode. One challenge was to find an actuator concept that would permit conventional and automated clutch actuation at the same time. This is because on the one hand, the actuator must not interfere with normal foot actuation and on the other hand, the driver must be in control at all times.

CbW are perceived as normal manual transmissions with a clutch pedal by the driver. The strength of the mechanical pedal resistance, however, is "simulated" by means of a force emulator. A travel indicator recognizes the pedal position and reports it to a control unit. The actual actuation of the clutch is carried out by an actuator. As the name "by wire" suggests, there is neither a mechanical nor a hydraulic connection between the clutch pedal and the slave cylinder of the clutch. That is why the performance of the system depends on the actuator. The new hydrostatic or mechanical Modular Clutch Actuator (MCA) developed by Schaeffler reaches time intervals of just 150 ms for opening the clutch, allowing additional functions such as the start assist, the traffic jam assist and driving dynamics features such as slip control. This actuator is described briefly in the following chapter.



2 **Operating conditions of the actuator: Clutch actuation by driver and pedal, actuation by actuator only when coasting and return to actuation by the driver**

ECM systems do entirely without a clutch pedal. Instead, a sensor on the gear selector recognizes the intent to shift gears. In the past, LuK already volume produced automated clutches as two-pedal systems, such as the BMW Alpina (1993) and the Mercedes-Benz A Class (1997). At the time, however, the efficiency potential of electrified vehicles and modern driving strategies was not available. That is why the systems were not successful on the market.

MT*plus* actuator function

With the functions described above, the MT*plus* system offers an entry-level product for the use of automated clutches and is able to upgrade a manual transmission significantly. The core of the system is an actuator that ensures that the driver's commands override the automated functions at any time. Figure 2 shows the actuator in conventional driving operation (driver and clutch pedal), in coasting without driver involvement and during the transition from coasting back to conventional driving (driver takes control).

The cross-section shows the electric motor with the spindle drive and the connections to the master and slave cylinder. In normal operation, the master cylinder is actuated hydraulically (yellow arrow in figure) through the open ventilation hole with a connection to the clutch pedal, and the actuator is disabled. When coasting, the actuator moves a split piston (depicted in blue in the figure) forward via the screw drive and the spindle, interrupts the hydraulic flow from the pedal and actuates the slave cylinder directly to open the clutch. However, if the actuator is now in coasting mode and the driver wants to take control, the idea is for the driver to feel a pedal curve similar to the one in normal operation, even though the clutch has already been opened by the actuator. That is why the hydraulic fluid is transferred to a cache piston in the actuator and pressure is built up when the clutch pedal is actuated. Simultaneously, a sensor on the pedal informs the actuator that the driver wants to take control. When stepping on the pedal, the driver feels a counterforce that is generated by a spring intended for this purpose (depicted in red in the figure). Starting at a specified point, the pressure in the cache piston and the pressure on the clutch side are compensated. The rear part of the piston now moves back and frees up the channel with the hydraulic fluid which takes control of the front part of the piston. The clutch remains open throughout this process. This change in slave cylinder control implemented in this way goes unnoticed by the driver.

3 **Schaeffler's modular clutch actuator (MCA)**

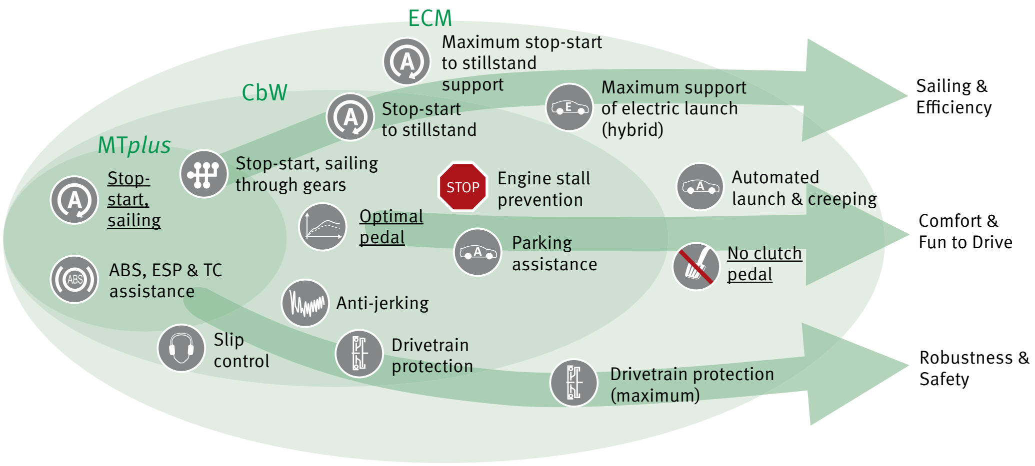
Clutch actuator

Schaeffler has developed a hydrostatic actuator called Modular Clutch Actuator (MCA) for the actuation of CbW and ECM systems, Figure 3. It is briefly introduced here, with a more detailed description provided elsewhere [4]. As the name suggests, development focused on creating a modular design. The brushless electric motor, the controller and the compact planetary screw make up the core of the actuator. The mechanical system is connected via a hydrostatic cylinder, and the data bus is connected via CAN, CAN FD or Flexray.

The drive unit of the actuator can be designed in three variants: Hydraulic with or without an integrated reservoir or mechanical with a tappet that actuates the clutch lever directly. The integrated power and electronic control unit contains five sensor inputs for additional sensors, and the driving strategy is represented by the software. The compact and robust design allows the actuator to be installed right on the transmission in the engine compartment, making it a plug-and-play solution at the plant that is manufacturing the vehicle. Schaeffler has equipped several test vehicles with this system, and start of production is scheduled for the end of 2018.

Functions of automated clutches

Figure 4 shows the most important functions of the implementation variants for automated clutches. Generally, the functions are divided into three groups that are of varying relevance to drivers.



4 **Functions of the automated clutch in the MT*plus*, CbW and ECM variant**

The functions affecting start/stop and coasting are especially important to automobile manufacturers because they can be instrumental in reaching CO_2 fleet goals. In the figure, they are referred to as “coasting and efficiency”. The second group, “comfort and driving pleasure”, by contrast, are primarily important from the customer’s point of view because of the functions shown here, such as a park assist system that customers can experience firsthand – and will buy. The third group refers to “robustness and safety”. This is what car buyers take for granted.

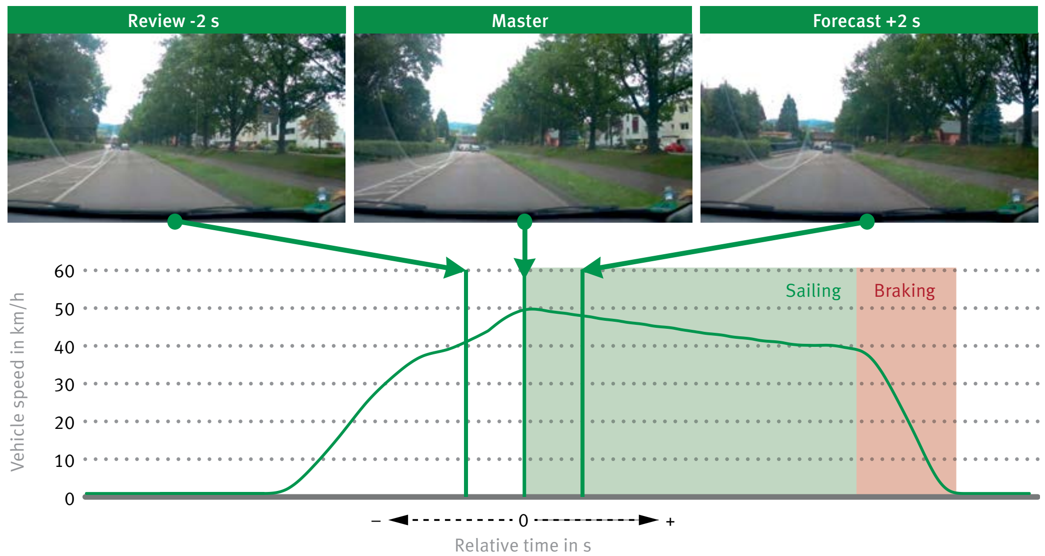
Development of driving strategies

Development and evaluation method

One of the elements used by Schaeffler to analyze potential implementation variants for the coasting functions is an RDE-compliant track around the company’s Bühl location and consistent testing conditions. The consumption measurements taken here are an important basis for the development and design of coasting strategies for automated clutches.

In a field test by Schaeffler on the design of a passive coasting function, which is described below, a total of 42 test drives were performed by four different vehicles on the RDE-compliant test track. Half of the drives (21 cases) included coasting mode, while the other half did not use coasting. The test drives were completed by a total of nine different drivers to allow a better analysis of driver influence.

Video cameras recorded all drives. Based on the measured data, Schaeffler engineers analyzed the time intervals with coasting mode and assigned potential influencing parameters to them. An engineer manually defined and assigned the actual reason for entering coasting mode with the aid of video material and measured data. Synchronization of the video recordings with the measured data on a playback panel, Figure 5, allowed the exact time of the coasting start to be



5 **Field test on coasting mode: Video material synchronization with measured data**

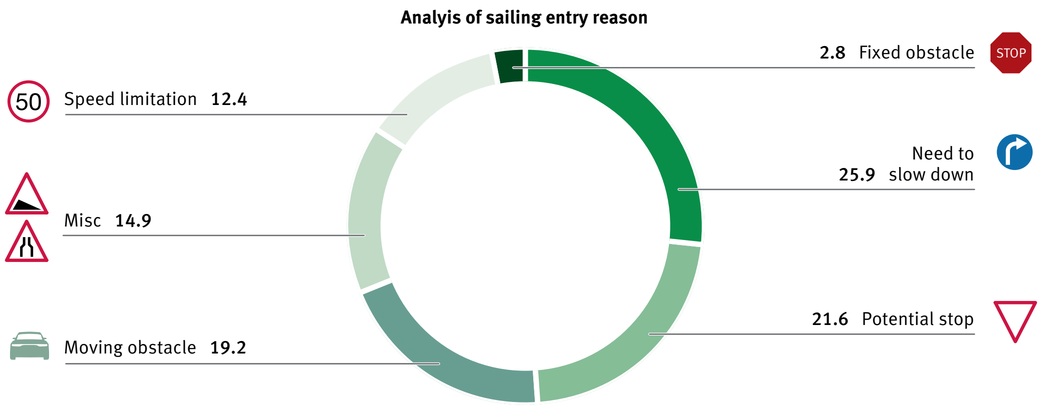
detected automatically. Each event was documented by a photo and additional images were played from the review (2 s before coasting start) and a preview (2 s after coasting start). The reason was selected from a specified list, then the reasons for entering the coasting mode were classified.

Coasting mode analysis

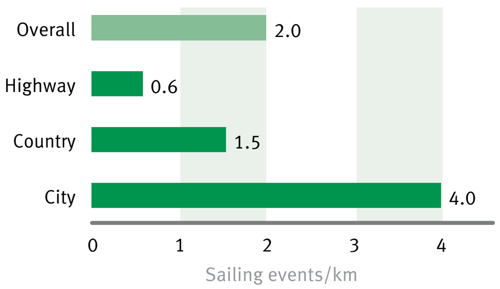
Figure 6 shows the six major reasons for entering coasting mode defined in the analysis. Various other reasons were summarized in a category and the image was summed up to 100 percent in a pie chart in order to obtain a better assessment. For the sake of completeness, it must be mentioned here that there may very well be other reasons for starting the coasting mode.

The most frequent reason why a driver took his foot off the gas pedal and the car changed to coasting was upcoming deceleration due to driving dynamics, such as a curve (25.9 %), followed by deceleration due to the driving situation, for instance when approaching a yield sign (21.6 %), as well as a slower vehicle in front of the driver’s car (19.2 %). Other important reasons include inclines and various other factors (14.9 %), speed limits (12.4 %) and stop signs (2.8 %).

Another interesting question is how often coasting even occurs. Figure 7 shows the analysis of various driving situations. Overall, the vehicle began coasting twice on each kilometer of the RDE track. As expected, city driving involved considerably more coasting at four incidents per kilometer than on highways (1.5 incidents) and interstates (0.6 incidents). What was particularly remarkable was that the cars coasted over an average of 27 % of the track, more than a quarter of the overall distance. The result was similar for the amount of time: Between 18 % and 31 % of the driving time was in coasting mode. These results show the great potential of coasting in a real driving cycle.



6 **Reasons for entering coasting mode**



7 **Number of coasting operations in various driving situations**

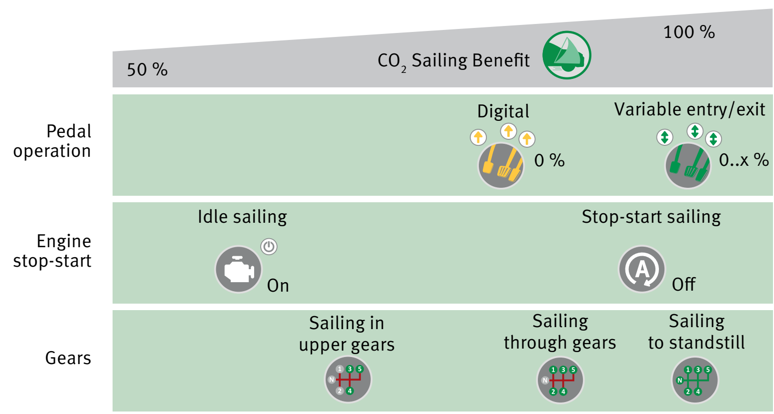
Analysis

The potential of coasting implementation variants is shown in Figure 8 . In this analysis, pedal actuation by the driver and the resulting entry and exit strategy were taken into consideration, as were engine operation (idling or start and stop mode) and gear actuation for deceleration and standstill.

The diagram shows that the full coasting potential can only be utilized if the driver's intention is recognized for a specific situation based on the pedal signals, the engine is always turned off during coasting and coasting mode is used until the vehicle stands still. With this constellation, the consumption benefit that can be achieved with coasting is at around 5 %. Without coasting in the lower gears (1, 2 and 3) for deceleration and with a simpler control strategy, the consumption benefit (and thus the benefit with regard to CO_2 emissions) achieved around 3 % based on measurements by Schaeffler.

Result and conclusion of the analyses

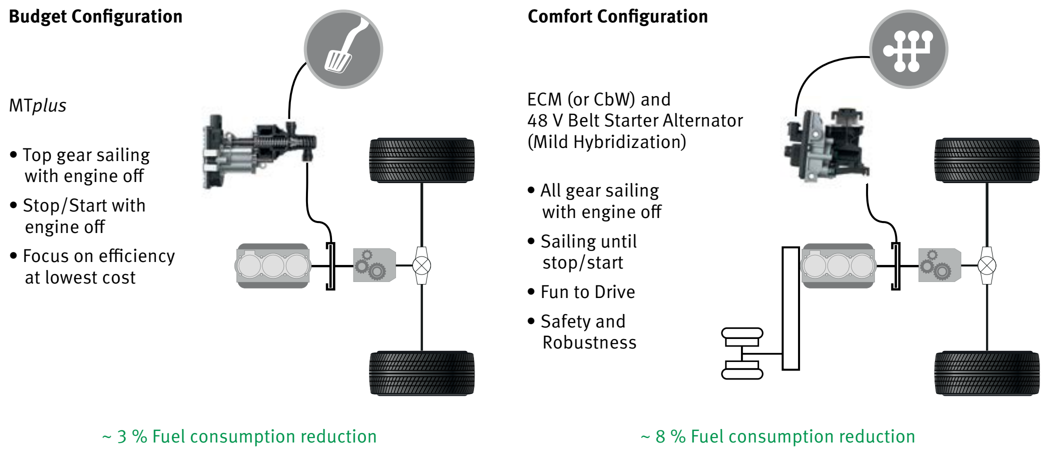
Schaeffler also performs numerous test drives with customers to analyze additional questions related to coasting. So far, more than 500 coasting test drives have been completed. One important question has been whether a dynamic start with the help of the clutch while the vehicle is rolling after exiting the coasting mode is desirable from the customer's point of view if the gas pedal is actuated while coasting. From a technical standpoint, a dynamic start is a feasible alternative for a restart with a belt-driven starter generator. The result of the test drives, however, was



8 **Evaluation of potential implementation variants for coasting mode**

that many customers rate a dynamic start as having too great a loss of comfort. Vehicle deceleration was felt to be particularly annoying below third gear. That is why a dynamic start without a belt-driven starter generator as a low-cost alternative is only suitable for simple coasting strategies in which only the higher gears are intended for coasting when the internal combustion engine is turned off.

The tests showed that a CO_2 reduction by around 3 % can be achieved, Figure 9, by using the MT*plus* concept without the belt-driven starter generator and a limited coasting strategy at relatively low cost. However, further hybridization is required to utilize the full savings potential of automating the clutch and to achieve the full benefit regarding consumption and CO_2 emissions of around 8 %. The recommendation is a P0 constellation with a



9 **Recommendations for manual transmission automation**

48 V system [6] so that the use of a dynamic start can be avoided. A potential solution that takes exactly this approach is described below.

Example of an automated clutch application

Potential solution

The technical solution consists of combining the automated clutch actuation (E-Clutch) with a P0 hybrid concept for vehicles with manual transmissions. The components of this approach include:

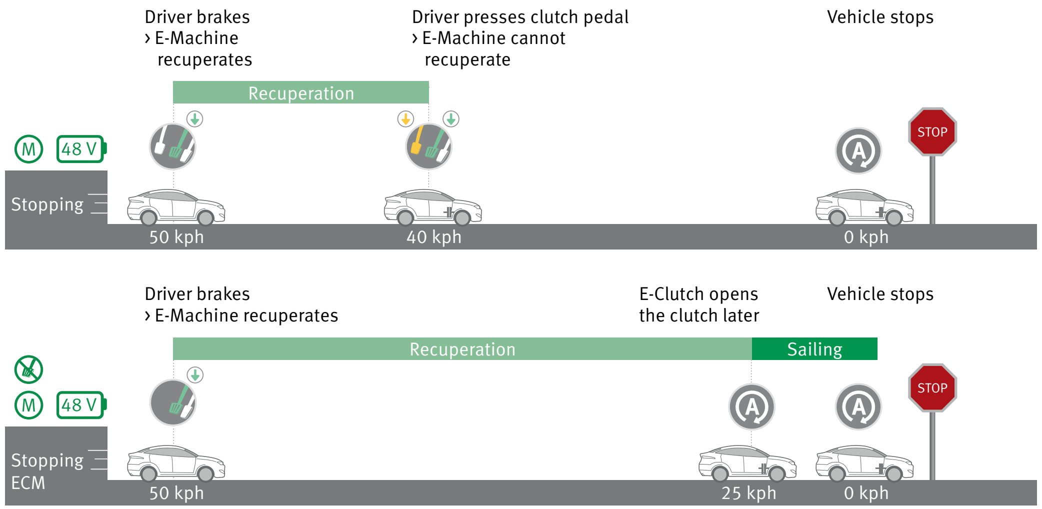
- A driving strategy to optimize consumption and emissions
- The automated actuation of the clutch based on the current driving situation
- The best possible recuperation irrespective of the driver's clutch operation
- The Modular Clutch Actuator (MCA) described above.

Advantages under real driving conditions

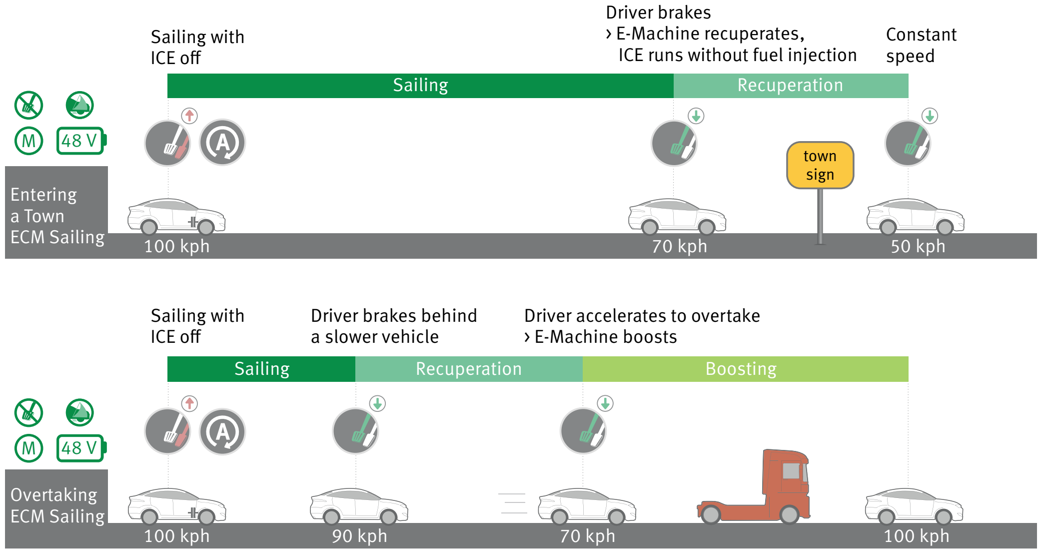
Figure 10 shows the advantages of automated clutch actuation under real driving conditions by comparing a stop without an E-Clutch (top) and with an automated clutch (below).

The figure shows vehicle deceleration with a belt-driven starter generator. The driver takes the foot off the gas pedal and starts to brake. Recuperation starts immediately but ends in the top part of the figure as soon as the driver opens the clutch. The field tests described above have shown that this happens too soon with a manual transmission because of the ignorance of many drivers; the driver thus wastes potential for recuperation through the P0 or P1 hybrid system installed here.

The bottom of the figure shows the same process with an E-Clutch: The driver starts to brake. If there is a clutch pedal, it is connected electrically to the clutch and can be overridden by the electronic control system. If the pedal is pressed too soon, the clutch will still open later to be able to recuperate as long as possible. If the driver wants to stop the maneuver to accelerate again or to shift, the requested vehicle reaction comes almost without delay.



10 **Stop from 50 kph without and with an automated clutch**



11 **Driving strategies with automated clutch on approaching a city limits sign and when passing other cars**

The automated clutch can also be used to combine coasting and recuperation, Figure 11 by the clutch initially opening and the internal combustion engine turning off (top part of figure) when the vehicle is coasting down, such as when approach a city limits sign. If the driver signals by actuating the brake that he wants more deceleration, recuperation will start through the generator by engaging the clutch and the internal combustion engine running without injection. If the driver takes his foot off the brake pedal and, possibly, briefly steps on the gas pedal, fuel is injected to continue normal driving.

The bottom of the figure shows a scenario in which coasting and recuperation are followed by passing another car. The driver takes the foot off the gas pedal and does not brake yet. Now the clutch can be opened, and the internal combustion engine turns off in the first phase in order to start coasting. When the driver brakes, the recuperation process starts through the generator while the internal combustion engine is rotating without injection. Next, when passing another car (driver steps down on the gas pedal), the electric motor can support the acceleration by the internal combustion engine by boosting during the recuperation process.

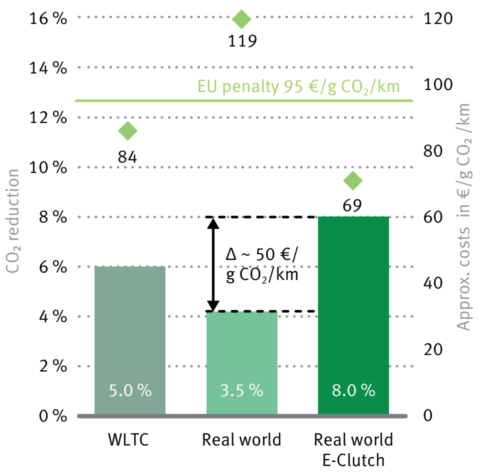
Cost evaluation

Figure 12 shows consumption benefits and system costs per gram of CO_2 saved per kilometer driven comparing systems without and with an automated clutch. In a hybrid system without an

E-Clutch, the advantage in the WLTcc cycle is 5 %, but only 3.5 % under real driving conditions. That is why system costs per gram of CO_2 saved increase from 84 to 119 euros. By contrast, a system with an automated clutch achieves a consumption benefit of 8 % in real operation, and the system costs per gram of CO_2 saved are only 69 euros. Here payback on the investment in an E-Clutch in addition to the belt-driven alternator starter is at a level of 30 €/g CO_2/km. This make it an attractive idea.

Summary

Manual transmissions are widely used all over the world and, even though the percentage of their market share is expected to decrease slightly over the next few years, their absolute numbers will continue to grow. For this reason, the further optimization of their CO_2 efficiency is an important element in reaching global climate goals. The automation and electrification of manual transmissions play an essential role.



Conditions: Base CO_2 emission: 110 g/km
A-segment
P0 48 V system costs: ~ 460 €
E-Clutch costs: ~ 145 €

12 CO_2 reduction and system costs using the example of a manual transmission in a 48 V hybrid concept in P0 arrangement

This paper has shown three approaches. The MT*plus* actuator concept maintains the mechanical or hydrostatic connection between the clutch pedal and the clutch, while the connection is purely electronically using the CbW and ECM concepts with a Modular Clutch Actuator (MCA), and, in the latter, the clutch pedal is eliminated.

Extensive field tests to analyze driving strategies with automated clutches, in particular with regard to coasting mode with the engine turned off, have produced two favorite concepts: MT*plus* as a budget solution and ECM with a 48 V hybridization in P0 or P1 arrangement to utilize the full CO_2 potential, in particular through the combination of coasting and recuperation. Various real-life driving situations were shown.

An evaluation of costs finally shows: The automation and hybridization of manual transmissions not only offers additional driving functions, comfort characteristics and operational safety. It is also reasonable with regard to forthcoming emissions legislation.

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11

Efficient Solutions for Automatic Transmissions

Torque Converters and Clutch Packs

Thomas Heck
Brian Zaugg
Thorsten Krause
Benjamin Vögtle
Martin Fuß

Introduction

In the automotive industry, the global trend to favor more efficient systems continues with the goal to further reduce CO_2 emissions. As a result, internal combustion engines continue to be optimized through the extensive use of cylinder deactivation and turbocharging. It is not just the weight of the complete vehicle – and thus of the drive train – that must be reduced, but also the mass moment of inertia of the components in order to offer improved driving dynamics and to rapidly reach the optimum-efficiency operating point of turbocharged engines.

Schaeffler offers technologies that improve the efficiency of the transmission through reduced losses while also allowing the use of new engine technology. In terms of the requirements for the torque converter lock-up clutch and damper, this means that vibration isolation becomes significantly more challenging [1]. One potential solution could be light-weight components, and another could be the intelligent integration of existing masses into the system to isolate the vibrations.

In addition, the reduction of CO_2 emissions also requires a lock-up clutch that is capable of precisely controlling slip in all operating conditions. In new-generation transmissions, there has been a tendency to increase the number of gears; resulting in additional reductions of installation space for the torque converter.

This paper aims to show technical solutions for the challenges described above.

Four-passage torque converter

Initial situation

The ability to control the lock-up clutch in the converter is essential for the efficiency and vibration isolation of the system. It is important to ensure that the clutch can be opened or closed on demand in all operating conditions and that the targeted slip speed can be maintained precisely. Allowing slip in the torque converter clutch is an effective method to reduce drivetrain vibration, however, efficiency is lost in the form of heat as a function of increased slip speed.

The torque converter clutch is controlled by applying fluid pressure to the piston. Traditionally, there are two-passage and three-passage systems. With the two-passage converter, the clutch is activated by reversing the flow of the automatic transmission fluid (ATF) through the converter. With the three-passage converter, two passages are used for ATF circulation through the torus and to cool the clutch, while the third is used independently to control the lock-up clutch. Figure 1 shows a comparison of two-pass and three-pass torque converters.

In simple terms, the torque converter clutch capacity is characterized by the following:

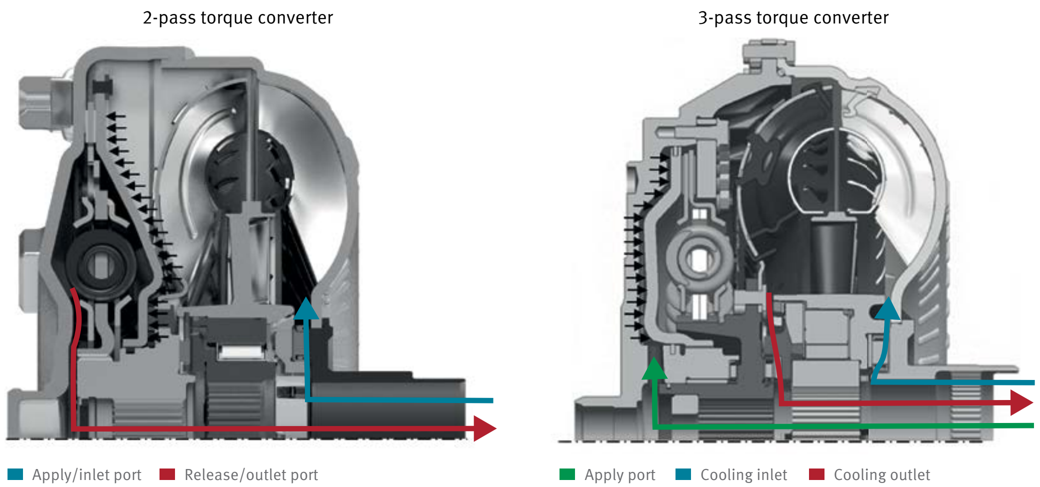
- The active surface area of the piston, the effective radius of the friction material and the number of friction surfaces
- The friction coefficient between the friction material and steel
- The pressure applied to the piston.

While the geometry and the friction coefficient of a given design are fixed, the pressure that controls the clutch can be varied by the hydraulic control system of the transmission. In an ideal system, the torque capacity of the clutch and the required slip would be controlled exclusively by varying the apply pressure.

Torque converter clutch interference factors

In reality, there is no ideal system because there are other factors that affect the transmitted torque of the clutch. The interference factors below represent the areas in which two-pass and three-pass systems have shortcomings, making it difficult to accurately control clutch slip.

1. Pressure drop across friction material: In two-pass converters, the friction material not only serves to transmit torque, but it is also a sealing component on the outside diameter of the piston. In order to cool the clutch, a groove pattern is often pressed into the friction material. When ATF flows through the grooves from the high pressure side of the piston to the low pressure side, it experiences a pressure drop. The magnitude of this pressure drop is dependent on the groove geometry, consistency of the friction faces, temperature, and slip speed.
2. Absolute system speed: After the ATF has flowed through the friction material grooves in a two-passage converter, it must be transported radially from the outside diameter of the converter to the inside towards the transmission input shaft. Since the entire system is spinning, the fluid particles are subjected to Coriolis forces on their way to the inside, leading to the formation of a spiral flow in front of the transmission input shaft. This results in back pressure that reduces the effective pressure on the piston.
3. System pressure variation: Fluctuations in converter charging pressure affect the high-pressure side of the piston in a two-pass system and the low-pressure side of the piston in a three-pass system.
4. Differential speed (slip): During open or slipping conditions, two and three-pass systems have components such as the damper, turbine or cover on either side of the piston which are rotating at different speeds. These components dominate the mean rotational speed of the ATF on either side of the piston, which results in a different centrifugal force, creating a relative pressure across the piston.

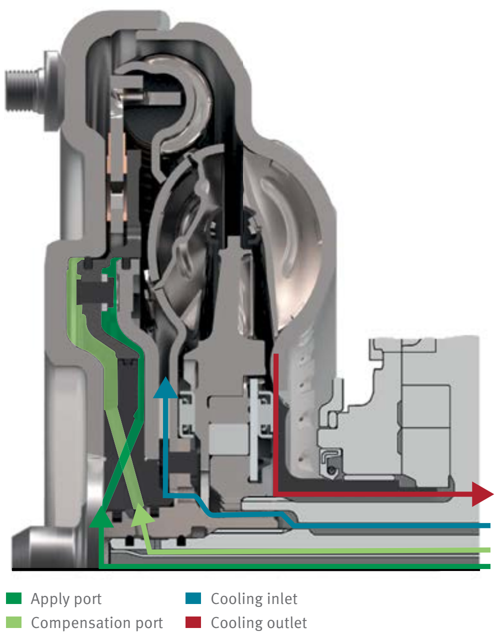


1 **Comparison of a two-pass and a three-pass converter**

Potential solution: Converter with four hydraulic passages

Interference factors 1 and 2 can be largely neutralized by a three pass-system. The remaining interference factors can also be improved significantly in a three-passage system or compensated by the calibration software in the transmission. However, in order to be able to compensate all factors entirely without additional software requirements, a different principle is needed: the four-pass torque converter. As the name suggests, this is a converter system with four hydraulic passages. A potential design as well as the functional principle are shown in Figure 2.

Like the three-pass system, two of the passages are used for the flow through the converter, and the third passage serves to control the clutch. The unique feature of the four-pass torque converter is the additional fourth passage, which feeds a pressure compensation chamber. This results in identical fluid speed conditions on both sides of the piston. The dynamic centrifugal force of the ATF is identical on both sides of the piston because the outside diameters of the activation and compensation chamber seals are the same. This means that the piston pressure is now independent of slip speed, and furthermore, the pressure chambers of the clutch are shielded from system pressure variations, i.e. from charge pressure fluctuations.



2 **Design of the four-pass torque converter**

Comparison of systems

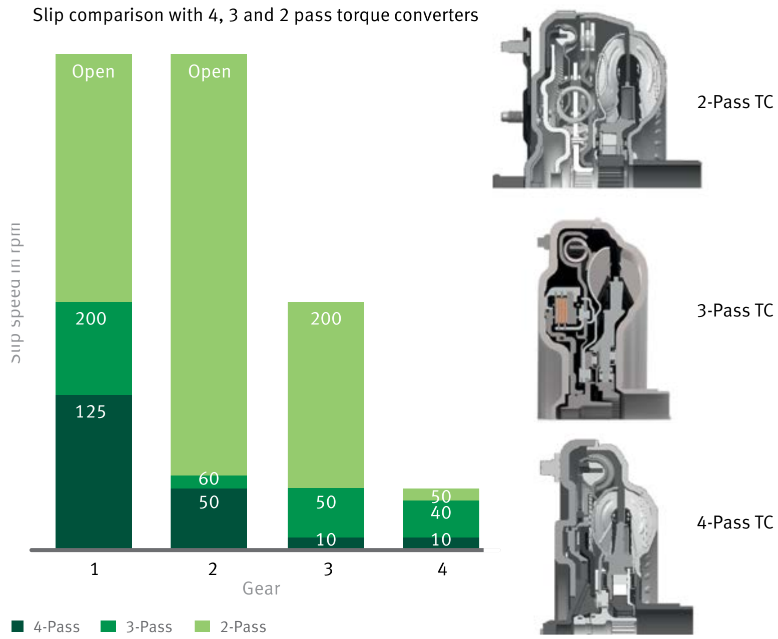
With the four-pass converter, the clutch can be controlled very precisely, independent of operating conditions. Schaeffler started volume production of the system presented in 2014 and is currently working on its implementation with other customers. A study was completed of production two-pass, three-pass and four-pass torque converters to compare the slip speed during operation. The results of this study are shown in Figure 3.

The comparison shows that in this specific four-pass application the lock-up clutch can be engaged even in first gear. Besides fuel consumption savings, this also means that the lock-up clutch can be used as a launch device in line with the torus of the converter. This allows a smaller and more lightweight design of the torus. In higher gears, the four-pass converter can be operated at a very low slip speed due to its precise controllability. As a result, the damper can be designed on a smaller scale, allowing a more space-saving design of the converter as a whole.

Integrated torque converter – iTC

Design

The integrated torque converter (iTC) is a design in which the piston for the lock-up clutch is integrated into the turbine of the hydrodynamic circuit [2]. This principle is shown in Figure 4. The integration of the piston in the turbine results in a simplified design through component elimination, and it offers further optimized space utilization as compared to a conventional converter. This allows the use of larger higher performance dampers, particularly with a centrifugal pendulum-type absorber (CPA); or it can be a solution for reduced design envelopes resulting from transmissions with more gears.



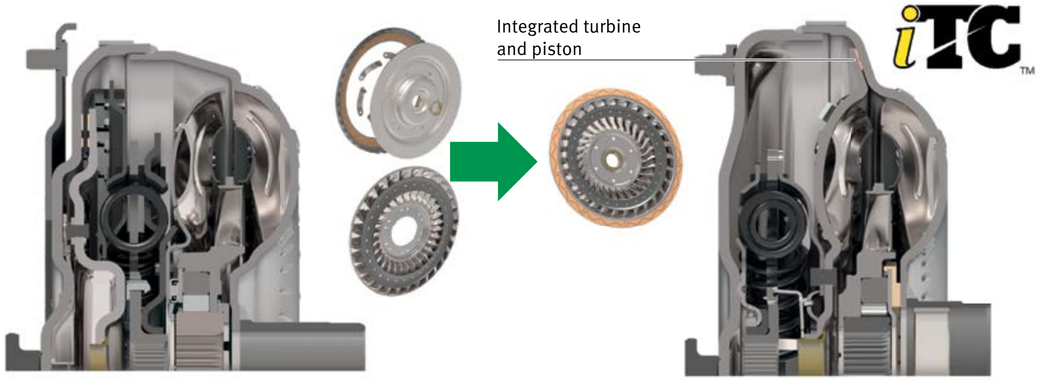
3 **Comparison of the slip speeds of two-pass, three-pass and four-pass systems.**

Since its launch in 2014, volume production has started for several applications, as shown in Figure 5. In comparison to the conventional torque converters shown below, the axial space requirements decrease by up to 17.5 mm and the mass decreases by up to 3.8 kg.

A comparison shows that many of the conventional applications have a dual-surface clutch while the iTC uses a single-surface clutch. This reduces complexity, but to ensure sufficient clutch capacity to accommodate engines of similar or higher torque, the clutch in the iTC is designed with a high taper, resulting in higher torque capacity due to the increased angle. Furthermore, the higher taper impeller surface improves stress distribution, allowing the use of thinner material, thus reducing mass. Additional savings are achieved by using Schaeffler's stamping technology that replaces heavy sintered hubs and other components.

Coast engagement diaphragm spring

In the pursuit of efficiency, the precise controllability of the torque converter clutch during coast is a fundamental part of the modern powertrain. This can present a challenge when utilizing a 2-pass clutch, such as the one in the iTC; but this is addressed through some of the unique characteristics of the design. Since the clutch function is combined with the turbine, the axial forces of the hydrodynamic circuit have an effect on the controllability of the clutch. In drive mode, the iTC principle offers an advantage because the axial

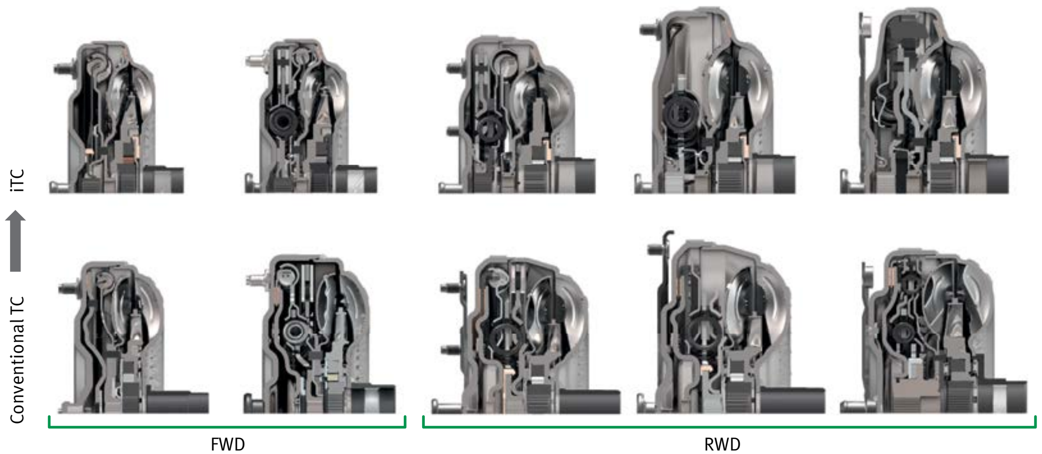


4 **Design of a conventional converter (left) and an iTC (right)**

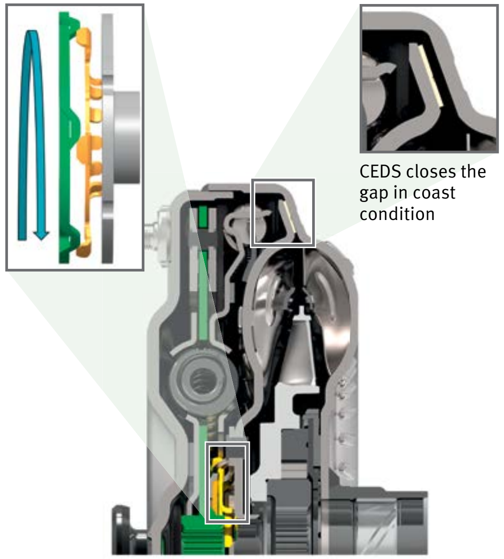
forces in the converter ensure that the clutch is biased towards engagement.

During coast, however, the opposite is true. The clutch is biased open by the hydrodynamic axial forces. In order to override this condition, ATF flow can be used to generate increased pressure, but this may cause the clutch to rapidly close causing an undesirable torque bump. An alternative to the control strategy of keeping the clutch closed at the transition to coast is to support the clutch mechanically in coast in order to achieve a seamless engagement.

This is made possible by the coast engagement diaphragm spring (CEDS), as shown in Figure 6. The spring is arranged between the damper flange and the turbine. As coast torque begins to build, the flange rotates and engages the formed ramps on the diaphragm spring. This results in an axial



5 **Integrated torque converters (iTC, top) for front-wheel (FWD, left) and rear-wheel drives (RWD, right).**



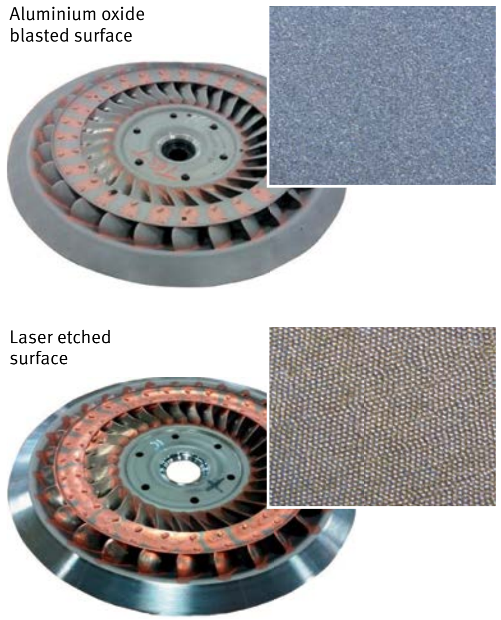
6 **Design and function of the coast engagement diaphragm spring**

force which counteracts the fluid loads on the turbine and aids in engaging the clutch during coast conditions. As a result of the success of this system, all iTCs today are designed with a CEDS.

Laser etching surfaces

With the introduction of the iTC, new processes were developed to accommodate the differences in design while still meeting all customer requirements. Integrating the clutch function into the turbine requires the friction material to be bonded to the turbine instead of to a separate plate. As an industry standard, blasting with aluminum oxide (Al2O3) has been used to prepare and roughen the steel surface for the bonding process. Due to the geometry of the turbine, with its numerous blades, there is a risk that aluminum oxide residue will remain on the component even after washing and eventually enter the transmission. This could damage the transmission valve body and would not be acceptable for meeting the increasingly stringent contamination requirements. Schaeffler has found the answer in the laser etching process. In this process, individual particles are vaporized on the steel surface by a laser, resulting in a rough and clean surface.

Figure 7 shows a comparison between traditional blasting and laser etching. During laser etching, the steel particles are completely vaporized and do not constitute contamination as per the residual contamination requirements. In fact, this solution enables compliance with much higher cleanliness classes than typically possible with conventional converters. Another advantage of laser etching lies in the fact that the surface roughness can be set numerically and can be measured by programming the laser – unlike blasting, which tends to be random.



7 **Reparation of adhesive surface by laser etching**

Torsion dampers for torque converters

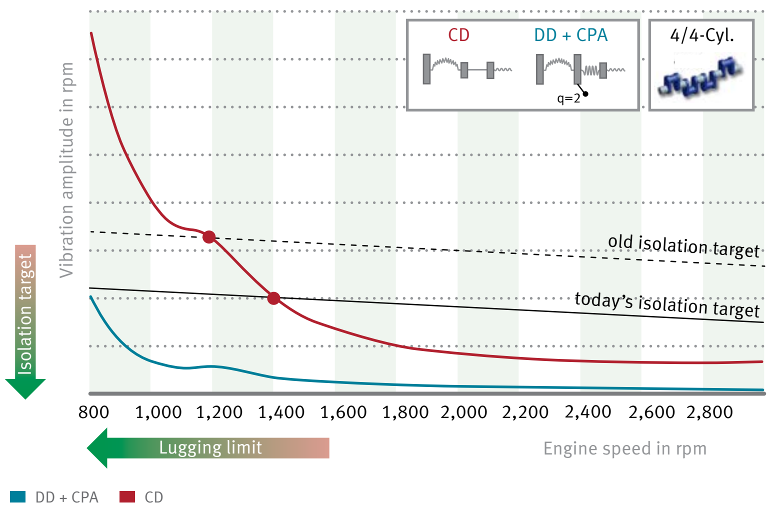
Potential of the centrifugal pendulum-type absorber

The advancement of absorber technology is critical given the ever increasing requirements for NVH isolation and efficiency. In this pursuit, consideration must be given to optimizing both the packaging space needed and the effectiveness of the absorber, specifically to provide excellent NVH at low engine speeds with a fully locked clutch.

The potential of the centrifugal pendulum-type absorber (CPA) for isolation in the drive train has been clearly recognized by the automotive industry. After Schaeffler introduced the CPA for the dual mass flywheel (DMF) in a dry environment in 2006 and started volume production in 2008 [3]; volume production for the CPA for the torque converter began in 2010. The CPA was designed to function in a wet environment and to provide excellent vibration isolation with a fully locked torque converter clutch.

Further development of the CPA focused not only on robustness and acoustics, but also on improving the isolation effect. The evolution of the torsion damper and CPA system has resulted in the ability to provide excellent NVH in four-cylinder applications down to 1,000 rpm with a fully locked clutch, as shown in Figure 8. This allows modern engines to run at higher efficiency regions of specific fuel consumption, therefore improving fuel efficiency.

In the future, even lower lock-up speeds down to 800 rpm will be required to further increase the efficiency of the drive train.



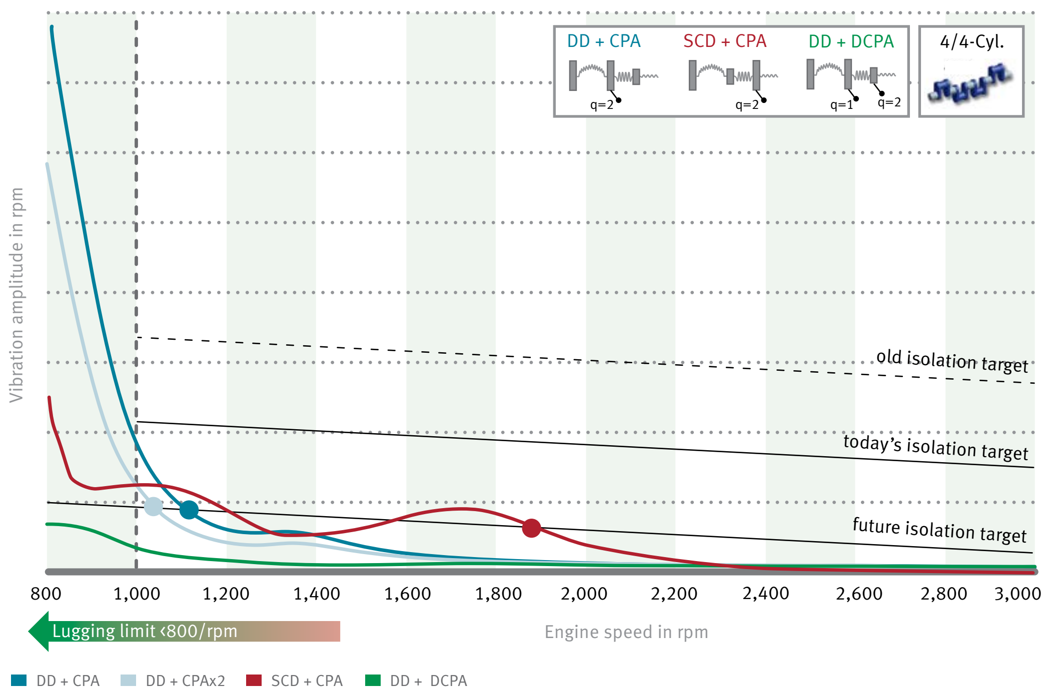
8 Isolation comparison of torsion dampers with and without a CPA

Torsion damper with double centrifugal pendulum absorber

The torsion damper (TD) and the double torsion damper (DTD) are the most frequently used damping systems with a CPA. While the DTD has been established as a CPA damping concept for applications with rear-wheel drive, the TD with a CPA is particularly suitable for front-wheel drive vehicles. This is because the axial design space required is small and the sensitivity is in the medium speed range, which is lower than that for rear-wheel drives, allows the elimination of a spring damper behind the CPA.

In the DTD, the CPA is arranged with the turbine on the intermediate flange between the springs. Its isolation capacity is excellent above 1,000 rpm. The CPA has limitations at low speeds in this configuration due to the vibration mode of the input shaft. In the specific example shown in Figure 9, even if the CPA mass were doubled, and the spring rate were reduced by 20%, it would still not be possible to reduce the lugging limit from 1,000 RPM to 800 RPM.

If the CPA were instead arranged with the turbine at the damper output, the natural frequency of the transmission input shaft mode would shift to the medium speed range. Compared to the DTD with a CPA, this series torsion damper (RTD) with a CPA offers much improved isolation below 1,000 rpm. However, this is at the expense of the resonance point at medium speeds between 1,000 and 2,000 rpm.



9 Isolation comparison of torsion dampers with CPA and double CPA

A potential solution could be the use of a DTD with a double CPA. The innovation of the double CPA is the combination of the strengths of previous damper concepts. Both CPAs are adjusted to the same order. Above 1,000 rpm, the CPA offers excellent isolation on the intermediate flange, and below that, the second CPA at the damper output offers the best possible isolation for lock-up speeds down to 800 rpm. A double CPA utilizes smaller masses than a torque converter with a single CPA, but it achieves significantly better isolation. This clearly demonstrates the importance of arranging the CPA. The concept is also suitable for applications with cylinder deactivation. Here, one of the CPAs is designed for full engine operation and the other for cylinder deactivation.

When designing the DTD with a double CPA, the primary goal was to make the overall system as compact as possible. In order to reduce additional axial space requirements for the other CPA, the "inline CPA" was developed, as shown in Figure 10.

In this design, the masses and rollers are in the same plane as the flange. The mass is centered axially by a thin cover plate on both sides of the flange, which helps increase the weight of the pendulum masses. With a comparable effective mass and inertia, the axial design space requirements of the second CPA are reduced by around 2 mm. In addition, the first CPA on the intermediate flange can be designed smaller due to the use of the second CPA. This allows the additional design space requirements of the damper system to be minimized.

Cylinder deactivation in four-cylinder engines

CPA dampers are well established solutions for the cylinder deactivation on eight-cylinder engines. In four-cylinder engines, the use of a CPA is even more beneficial. Here, isolation at low speeds is one of the greatest challenges. A damper concept must provide good isolation not only in full engine operation but also while only half of the cylinders are active. This represents a particular challenge because the engine's excitation frequency is cut in half, shifting critical natural frequencies to the driving range. The goal is to achieve driving with a fully closed lock-up clutch during both operating conditions, even at low speeds. Damper concepts without a CPA – such as the series damper shown in Figure 11 – provide sufficient isolation in full-engine operation, but unacceptable isolation if two cylinders are deactivated.

Combining a CPA on the intermediate flange and a two-stage damper, where the CPA and the first damper stage are designed for two-cylinder operation, significantly increases the isolation effect and allows driving with a fully closed lock-up clutch down to 1,200 rpm. Isolation in four-cylinder operation remains nearly constant.

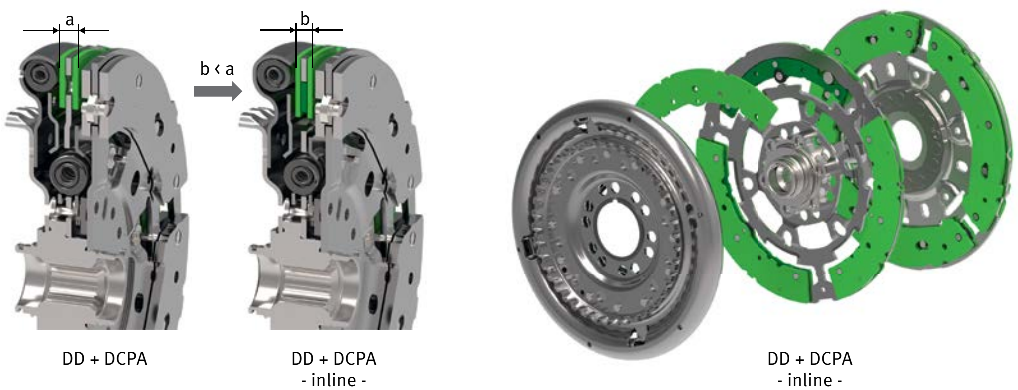
A good solution for further improving this isolation is the addition of a CPA for full engine operation, i.e. a concept with a double CPA. The second damper arranged behind the CPA is replaced by a CPA so as not to increase design space requirements. In addition, the remaining upstream damper is enlarged to ensure good pre-isolation.

The resulting TD+dCPA damper concept offers very good isolation in four-cylinder and two-cylinder operation for nearly the entire speed range.

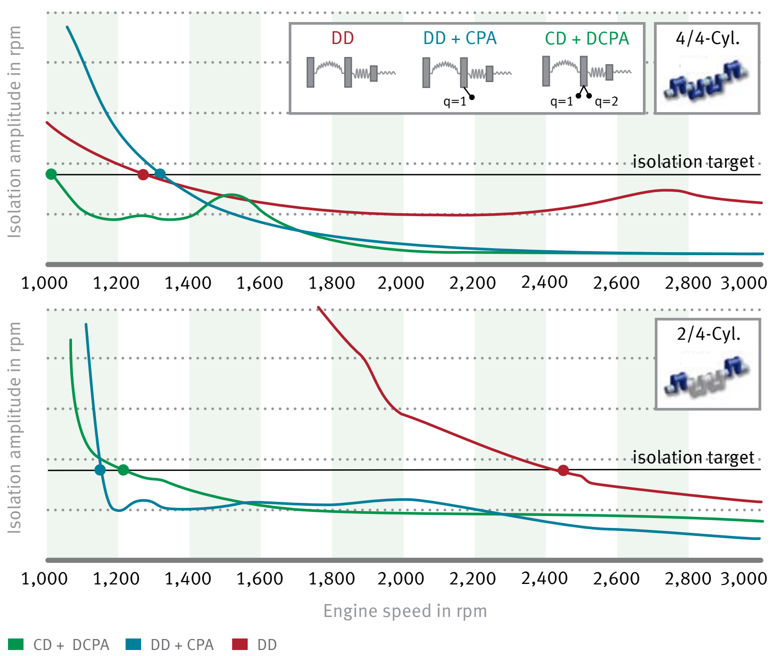
Effects of gravity on the CPA

In applications with a main excitation order of one, a solution must be found for not only drivetrain isolation, but also for the influence of gravity. These gravitational effects are typically trivial. This is not the case for a first order CPA.

Gravity affects all pendulum designs, no matter the tuning order. This is because of the way the TC is mounted in the vehicle. The spin planes of the CPAs are in-line with the gravitational field of the Earth. Gravity acts on each pendulum individually as the assembly rotates, once per revolution. This is the



10 Design of the inline CPA using a DTD with a double CPA as an example



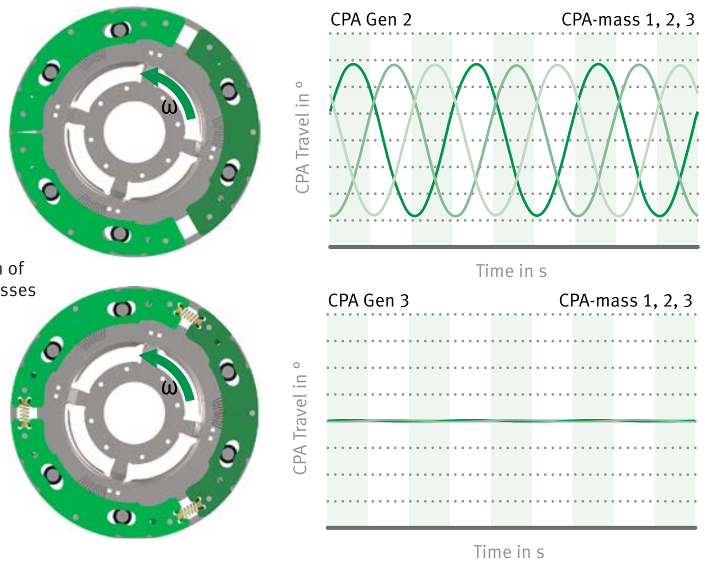
11 Isolation potential of torsion dampers with CPA and double CPA upon cylinder deactivation of two cylinders of a four-cylinder engine

key as gravity is a 1st order input to the pendulum. At speeds of more than 1,000 rpm, the centrifugal force is 100 times larger than that of gravity. Nevertheless, a 1st order pendulum will respond to the 1st order gravitational input (gravity excites the pendulums at their resonate frequency). In an arrangement with 3 pendulum masses, this gravitational input has a 120° phase shift for each pendulum mass. This will drive the masses to oscillate 120° out of phase with each other just under rotation, without any torsional excitation, as shown in Figure 12. Over a short time period, very large vibration angles can be reached. Consequently, due to the effect of gravity, the vibration angle in the CPA is not sufficient to absorb additional torsional vibrations, especially at low speeds.

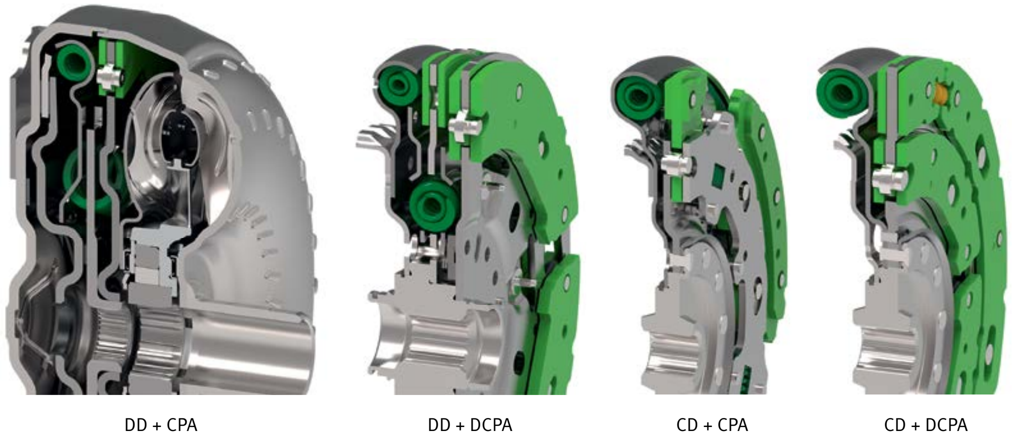
A solution for this gravitational resonance is offered by the Generation 3 CPA where the pendulum masses are coupled to each other by springs. The springs essentially link the CPA masses together, preventing the asynchronous excitation of the masses and effectively eliminating the gravitational excitation. The effect of the spring on torsional isolation is corrected by shifting the CPA tuning. This leaves a sufficient vibration angle in the CPA for appropriately absorbing the first order of the torsional vibration.

When development of a first order CPA began in 2015, it soon became clear that the gravitational excitation would play an important role, requiring the synchronization of the pendulum masses. The effectiveness of the Gen3 CPA was proven in a prototype test. Volume production of the first solution of this type started in 2017 for an application with cylinder deactivation in a four-cylinder engine.

The CPA has become established as the best solution for NVH isolation in a wide range of applications. Various damper concepts with a CPA are available for individual requirements regarding the isolation goal, drive train and design space, as shown in Figure 13. The current portfolio includes applications ranging from eight to three-cylinder engines, with and without cylinder deactivation, front and rear wheel drive, and torques between 250 and 900 Nm.



12 **Effect of gravity on the pendulum motion in a 1st order CPA**



13 **Damper concepts TD and DTD with double CPA**

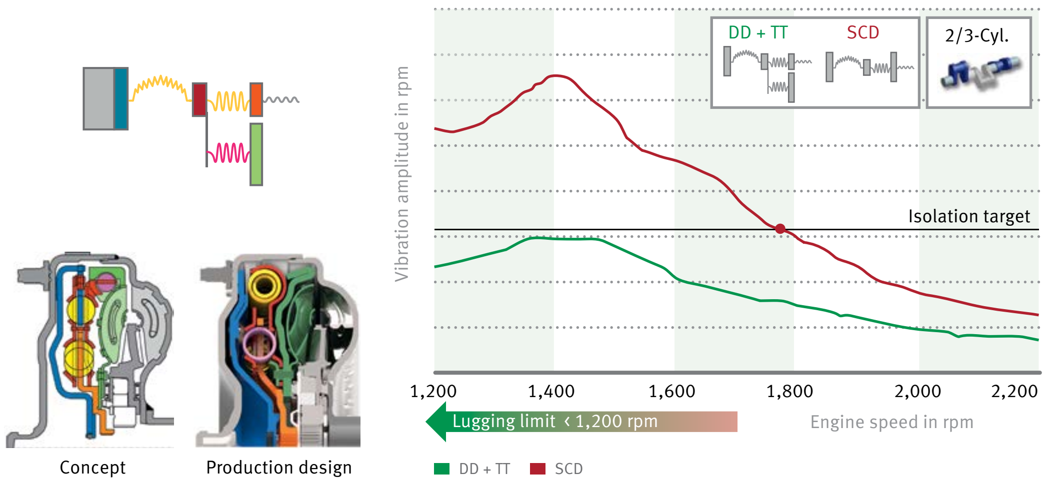
Absorbers in the torque converter

With the CPA, excellent isolation is achieved across the entire driving range due to its ability to cancel specific order content regardless of engine speed. However, for applications with excitation orders below one, CPA design solutions become increasingly challenging due to CPA envelope requirements (track length and lack of curvature).

At these extreme low orders, it is also evident that the ability to adapt the speed only rarely provides a significant advantage. A fixed frequency absorber can be an alternative to the CPA. Designed as a Tuned Mass Absorber, the Turbine Tilger (Absorber) uses the turbine inertia as an absorber mass during the closed lock-up clutch condition. Since the turbine is already an integral part of every torque converter, its mass/inertia can be used as an absorber for “free” (no additional absorber mass/inertia to the TC). Of course, additional absorber mass/inertia can be added to improve the absorber’s effectiveness. A reasonable embodiment of the Turbine Tilger concept consisting of the spring rate, mass distribution and friction can achieve an improvement in isolation in the lower speed range compared to a conventional serial damper, as shown in Figure 14.

The first damper concepts with Turbine Tilgers were developed as far back as 2007, sharing the same timeframe as the CPA. Initially, the CPA took center stage with its speed adaption advantage. Recently, cylinder deactivation applications (e.g. three-cylinder engines with static cylinder deactivation) have increased demand for sub-first order isolation solutions. Turbine Tilger optimization quickly followed.

Initial Turbine Tilger concepts had the turbine connected to the series torsion damper through an additional third torsion damper (the absorber spring). Optimization led to the integration of the absorber spring element into the main torsion damper envelope. The newly freed-up design space can then be utilized for increasing the absorber mass/inertia, if required. The stamped/formed part used for this can be easily adapted to the given design space. Schaeffler has been able to use its early experience with Turbine Tilgers for the rapid development and introduction of a volume production solution since the start of 2018.



14 **Isolation comparison of torsion dampers with and without turbine tilger**

Technology transfer to future products

The previous chapters discussed technologies for torque converters and related damping systems. These concepts can also be used for other components, assemblies and systems in the drivetrain. Examples of these technologies include the following:

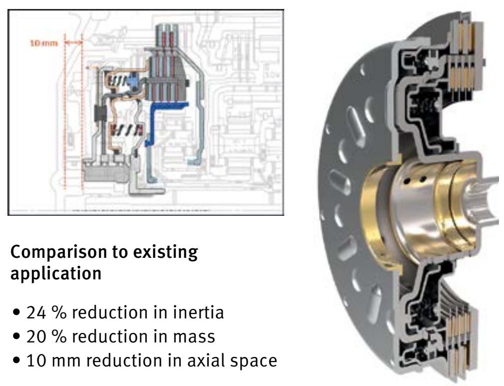
- Forming technology: Expensive, heavy sintered or cast iron parts can be replaced by stamped parts in a lightweight design or integrated into other components.
- Joining technology: Numerous innovations have been achieved over the past few years, particularly in riveting technology, included in combination with forming technology.
- Friction systems: Schaeffler has been actively involved in the development of friction material for many years. A special friction material has been developed for torque converters that is now used in almost all Schaeffler converters for the lock-up clutch. This expertise in wet friction material [4] can be easily applied to other clutch functions in the automatic transmission because the requirements and environmental conditions are similar.

Drivetrain electrification will include the automatic transmission. At Schaeffler, the technology transfer from conventional converters to hybrid solutions results in a P2 hybrid module with an integrated converter [5]. Instead of a traditional sandwich design, which places the hybrid module between the engine and the converter, the rotor of the electric motor is riveted to the converter cover. This eliminates an intermediate wall, which saves on axial design space. In addition, the wet K0 separating clutch is designed as a stamped component and uses the same friction material utilized in converter clutches. Volume production of a P2 hybrid module, similar to the one described here, will begin in 2019.

In order to take technology transfer to the next level, the insights gained from the development of the hybrid module and the double clutch will be applied to clutches in automatic transmissions. For this purpose, Schaeffler has developed designs within optimized envelopes as well as a friction system with high transmission capacity and low drag torque. The concept study of such a module is shown in Figure 15.

The use of stamping and joining methods makes it possible to reduce the axial design space by 10 mm as shown in the concept above, while the mass is ~20% less than a comparable volume production design. The study used process as well as product related methods. For instance, CFD analyses were performed to understand flow characteristics within the concept. This type of technique is also utilized in the field of torque converters. The target of these flow simulations was to optimize the friction material groove geometry to ensure that drag torques are as low as possible in addition to ideal oil flow for optimum heat transfer.

An advanced concept of a self-amplifying clutch is shown in Figure 16. Here, a leaf spring generates additional contact pressure. This makes it possible to reduce the clutch actuating pressure as well as hydraulic losses [6].



15 **Concept for clutches with an optimized design space**

16 **Concept of a clutch with self-amplifying actuation**

In a modified form, this technology is used in a volume production application for motorcycles.

Summary

This paper has presented various approaches by Schaeffler for optimizing space, mass, control and efficiency of damper and torque converter systems for automatic transmissions. For improved controllability of the lock-up clutch in all driving conditions, a four-pass torque converter is an excellent solution. The integrated torque converter (iTC) offers a simplified setup and significantly optimized design space utilization by integrating the piston into the turbine. Additional improvements with regard to design and process technology have been described for the iTC.

Vibration isolation also plays an important role in the converter. The potential of the centrifugal pendulum-type absorber (CPA) is far from being fully utilized. A new approach, shown here, is the use of a double torsion damper with a double CPA in the torque converter. Further, the paper describes a damper concept for four-cylinder engines with cylinder deactivation and the use of the turbine mass as an absorber mass to be able to offer an alternative to a CPA in certain applications.

Schaeffler's expertise in torque converters and related damping systems is also applied to electrified drivetrains. Schaeffler is always ready to support transmission manufacturers and developers in future challenges.

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12

Innovative Bearing Concepts for the Powertrains of the Future

Georg von Petery
Reinhard Rumpel

Introduction

Modern rolling bearings are indispensable when it comes to the refinement of transmissions with regard to friction and power density. New design space conditions and requirements that this key component will be confronted with by developments such as powertrain electrification may result in optimized designs as well as new bearing types and concepts.

This paper deals with a new bearing design that makes it possible to use low-friction locating/non-locating bearing supports even more frequently than before. It also shows that the consistent optimization of known bearing designs can help further reduce friction, decrease design space requirements and also reach the high speed level of electric drives at moderate temperatures. In addition, the paper describes ways in which application limits can be extended through the use of coatings.

Future requirements for rolling bearings

The increasing regulation of CO_2 emissions defines essential trends, including the development of transmissions, largely independently of the transmission design. Faster than was predicted only a few years ago, it is the new electric drives in particular that have changed the requirements for bearing technology. They are driven by hybridization, the development of dedicated hybrid transmissions (DHT) and electric axles. For this reason, the discussion of future transmission bearings also includes the electric motor as a drive, in particular since it will be combined with the transmission to form one drive unit.

This results in new requirements for bearings, including:

- Higher load capacity with even less friction
- Higher speeds: Today, the speed parameter that is the product of the speed and the mean bearing pitch circle is at around 0.2 - 0.5 million mm/min for oil-lubricated transmission bearings and will increase to up to 1.3 million mm/min for grease-lubricated bearings in electric motors.
- Noise reduction
- Greater temperature differences between the bearing rings: The temperature differences between the bearing inner and outer ring increase from 15 - 20 K to up to 70 K due to the more frequent starts and stops as well as the heat transferred from the rotor. This large temperature difference must be taken into consideration when dimensioning the internal clearance and the heat treatment of the bearing rings.
- Protection from electric current and resistance to non-operate currents
- Increased power density
- Grease lubrication: The temperature limit of available high speed greases is too low. Classic high-temperature greases are thermally very stable but they are limited to speed parameters n x dm up to 1 million mm/min. Schaeffler has resolved this conflict of goals between speed parameters and temperature stability by means of a newly developed grease
- Oil lubrication: In transmission bearings, the use of oils with ever lower viscosity leads to more frequent operation in mixed friction. The probability of surface-induced damage increases. Some modern transmission oils even promote the occurrence of white etching cracks in the bearings. The Durotect B coating developed by Schaeffler counteracts both effects.

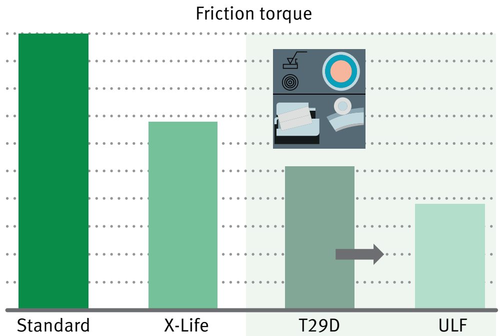
New bearing concepts

Friction-optimized tapered roller bearings and locating/non-locating bearing supports

Modern transmissions are unthinkable without high-performance rolling bearings. The bearings are designed for specific applications and with the overall system in mind. Schaeffler has developed its own calculation programs [1].

Many transmission shafts are currently supported by tapered roller bearings. The reason for this lies in the high robustness of this bearing type and in its very good ability to support loads. In addition, tapered roller bearings are characterized by a comparatively small cross-section as well as mostly simple assembly.

Tapered roller bearings are continuously refined by Schaeffler to increase their performance and reduce the friction torque level. Its latest designs combine individual, application-specific designs with the best possible manufacturing standard. These include the optimized design of the contact angle, special raceway profiling, the use of tapered rollers with optimum length and diameter, the right number of rollers for a specific load, narrow tolerances and improved surfaces as well as the use of improved materials and heat treatment processes [2].

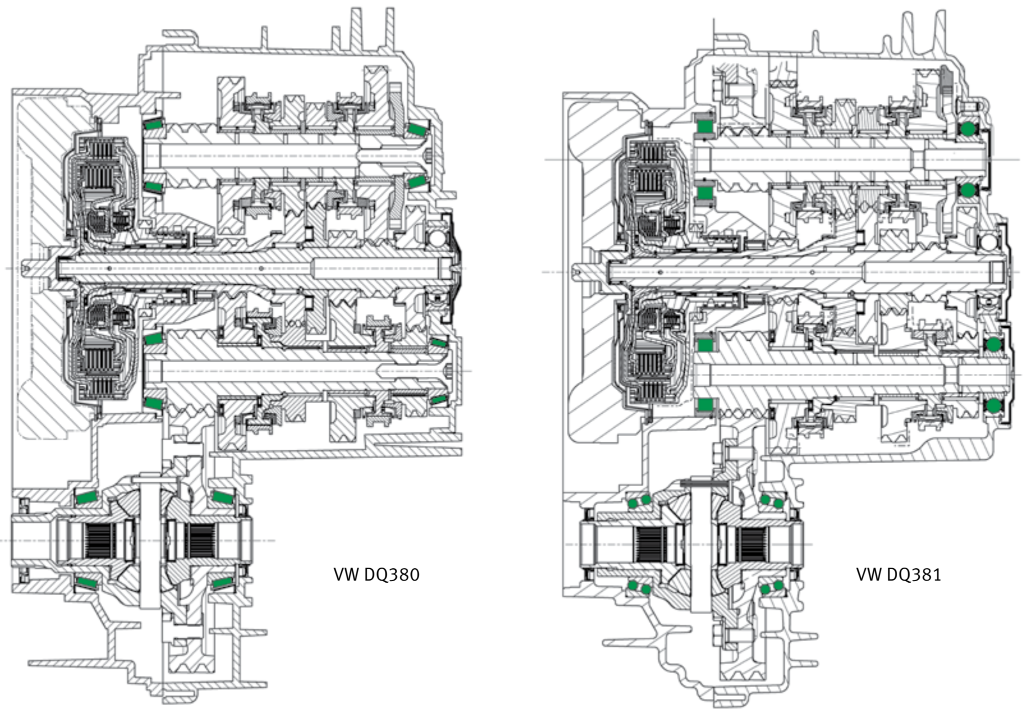


1 **Comparison of ultra low friction tapered roller bearings**

The OPTIKIT program developed at Schaeffler as an optimization tool for CAE models allows tapered roller bearings to be designed in relation to freely definable target parameters and auxiliary conditions. Due to the use of this automated optimizer that is integrated in BEARINX, much better solutions have been found than would be possible manually in the same time period [1]. The latest result of this computer-assisted optimization is the “ultra low friction (ULF)” tapered roller bearing that is based on the T29D manufacturing standard and has a very low friction torque, Figure 1.

In spite of the progress that has been achieved in the development of tapered roller bearings, changing to locating/non-locating bearing supports can further reduce the friction power because there is no axial preload of the bearing. The time and money invested in the reconfiguration of the transmissions justifies the change in the bearing concept for fundamental revisions of existing transmissions or new designs. Due to such successor transmissions, manufacturers successively change their transmissions to locating/non-locating bearing supports [3], Figure 2.

Deep groove ball bearings are used as locating bearings and cylindrical roller bearings or radially compact drawn cup cylindrical roller bearings are used as non-locating bearings. One advantage here is that the point contact of the deep groove ball bearing in partial-load operation produces less friction than the line contact of the tapered roller bearing. However, deep groove ball bearings have a comparatively low capacity for supporting loads. In addition, they have a certain axial and radial clearance. That is why their suitability for specific applications must be checked on a case-by-case basis.

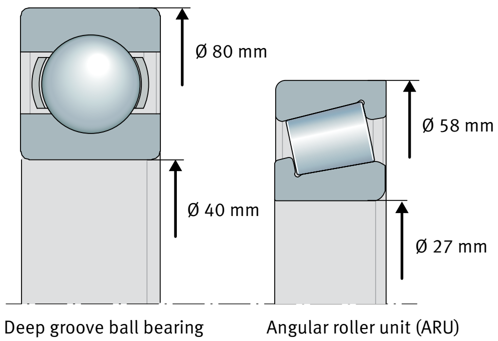


2 **Adjusted tapered roller bearing supports (left) versus locating/non-locating bearing supports (right)**

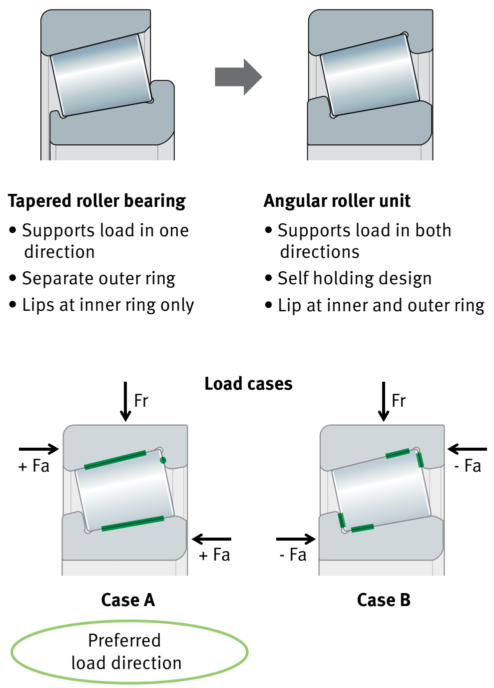
Angular roller unit (ARU): Innovative locating bearing with high load rating and low friction

In order to implement locating/non-locating bearing supports without any modifications to the design space, Schaeffler has developed a new innovative locating bearing called “angular roller unit” (ARU). It has a significantly higher load capacity than a deep groove ball bearing but also runs with low friction. Figure 3 shows a comparison of the dimensions of a deep groove ball bearing and an angular roller unit with the same load rating.

The new bearing design benefits from insights from the tapered roller bearing development described above. At first glance, that is why the ARU



3 **Comparison of the dimensions of a deep groove ball bearing and an angular roller unit with the same load rating**



4 **Comparison of the design principle of a tapered roller bearing and an angular roller unit (ARU) (left) as well as loading conditions for an ARU with preferred direction (right)**

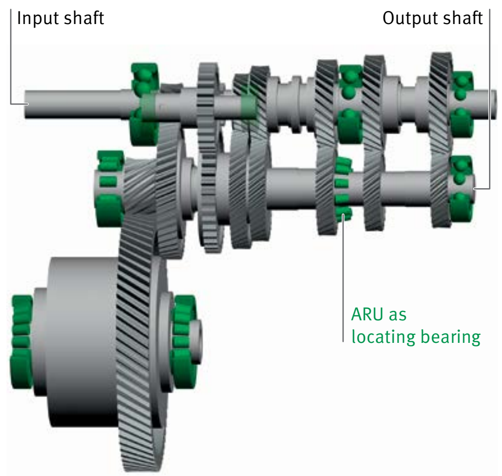
resembles a tapered roller bearing, Figure 4 left, but some of its essential design features are different. For instance, the ARU can support axial forces in both directions as a single bearing, unlike the tapered roller bearing, Figure 4 right. However, it should be mounted in the preferred direction so that the higher axial forces can be transmitted via the raceways, similar to tapered roller bearings. Lips on the inner and outer ring transmit those axial forces that work against this preferred direction. For this reason, the tapered rollers are machined on both end faces, unlike tapered roller bearings. Due to the ribs, the ARU is self-retaining on both rings, allowing it to be mounted in the same way as deep groove ball bearings, and is manufactured with a predefined internal clearance. The resulting axial clearance is also comparable to deep groove ball bearings.

The ARU is suitable for all mounting positions that require a higher load rating than a deep groove ball bearing can provide. The height of the axial forces that occur must be checked in detail and the bearing must be mounted in the preferred direction.

Simulation of various bearing concepts

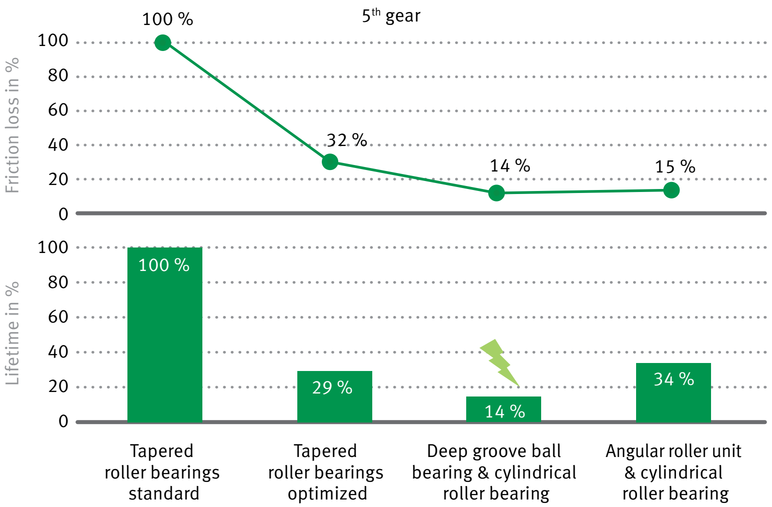
Various bearing concepts are compared below with regard to their characteristics, using a current six-speed manual transmission for front-transverse installation. Figure 5 shows the calculation model of such a manual transmission for torques of up to around 250 Nm used in compact cars with front wheel drive. This example already includes an ARU on the transmission output shaft.

For a comparison of the bearings, four different concepts are analyzed using the output shaft as an example as well as using identical design spaces.



5 **Mounting position of an angular roller unit (ARU) in a six-speed manual transmission on the transmission output shaft**

- Adjusted bearing supports with standard tapered roller bearings
- Adjusted bearing supports with friction optimized tapered roller bearings
- Locating/non-locating bearing supports with deep groove ball bearing and cylindrical roller bearing
- Locating/non-locating bearing supports with ARU and cylindrical roller bearing



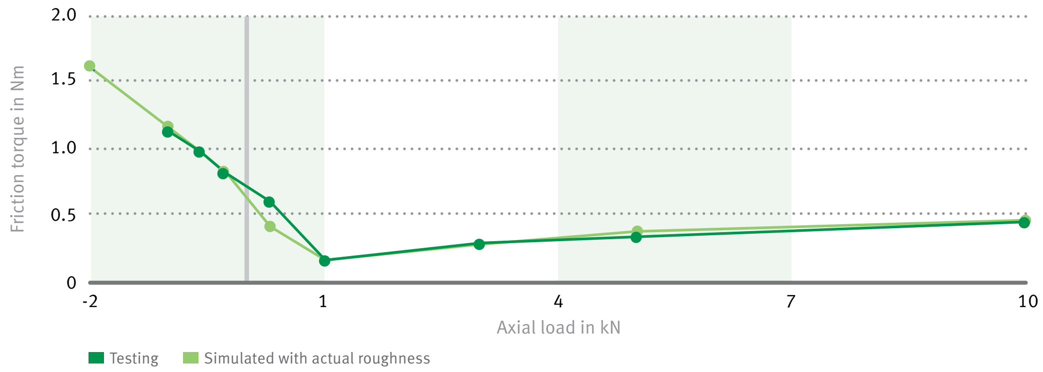
6 **Simulation of friction (top) and rating life (bottom) for the 5th gear of a manual transmission**

Figure 6 compares the calculation results for fifth gear in each bearing variant. The friction losses shown in the top part clearly show that conventional tapered roller bearings that are adjusted against each other have the highest friction. Optimized tapered roller bearings already achieve significantly lower friction torques, but they are outperformed by locating/non-locating bearing supports using deep groove ball bearings and cylindrical roller bearings. The locating/non-locating bearing supports using ARU and cylindrical roller bearings perform similarly well in terms of friction as the solution with deep groove ball bearings.

For calculated life, tapered roller bearings not specially adjusted to low friction show the best performance. However, they are over dimensioned in many applications and should not be preferred because of their high friction. Optimized tapered roller bearings continue to offer a sufficient rating life, even if it is much shorter than for bearing supports with conventional tapered roller bearings. Generally, the locating/non-locating bearing concepts should be preferred due to their excellent friction characteristics. However, the calculated life is not sufficient for the solution using a deep groove ball bearing in fifth gear. With the ARU as the locating bearing, however, a longer rating life can be achieved than with optimized tapered roller bearings, while also having lower friction. Thus the conversion from adjusted tapered roller bearing supports to locating/non-locating bearing supports without modifications to the design space would only be possible by using an ARU bearing as a locating bearing.

Validation of simulation results

Comprehensive calculations were performed for the ARU's friction behavior. The correlation of the simulation results with the tests is very high, Figure 7. Both the general curve of the friction torque and the absolute height of the measured and the calculated friction torque match well.

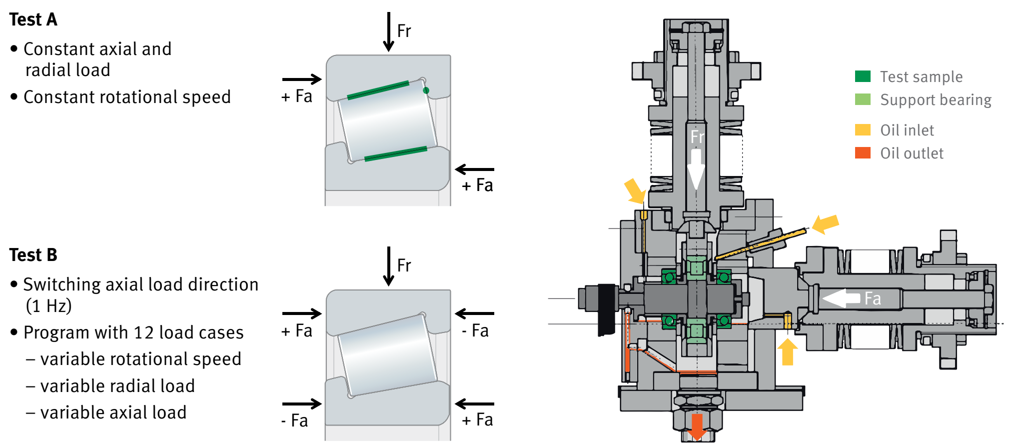


7 **Comparison of the simulated friction torque and the friction torque measured on the test stand for the new ARU bearing design**

Rating life tests were performed parallel to the friction torque and also compared to simulations. Figure 8 shows the general design of the rating life test stand used. Tests were performed under static load in the preferred direction (test A) as well as tests with changing load directions (test B).

Development of cylindrical roller bearings

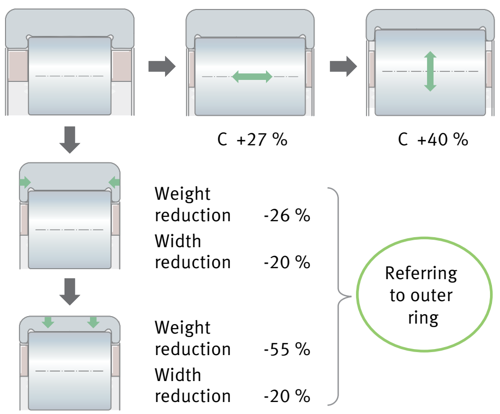
The angular roller unit is a new locating bearing that has a higher power density than conventional deep groove ball bearings. This inevitably leads to the question of what optimization poten-



8 **General design of the test stand used to validate the rating life calculations for the new ARU bearing design**

tial there is on the non-locating bearing side. Due to their design, cylindrical roller bearings are ideal non-locating bearings because they permit displacement within the bearing, thus preventing displacements between the bearing outer ring and the housing. NU and N type single-row cylindrical roller bearings with a lip-free inner or outer ring as well as NJ type cylindrical roller bearings with two fixed lips on the outer ring and one fixed lip on the inner ring are particularly suitable as non-locating bearings for compensation in the bearing [4].

Conventional cylindrical roller bearings have wide lips, even if they are used as non-locating bearings and theoretically are not subjected to axial rib loads. From the point of view of the transmission, these wide lips are not desirable because they take up too much unnecessary space. A potential optimization may be to design the bearings with lips that are as narrow as possible. Technically, the production of lips with a width of just over one millimeter is feasible. This design is interesting for two general approaches.



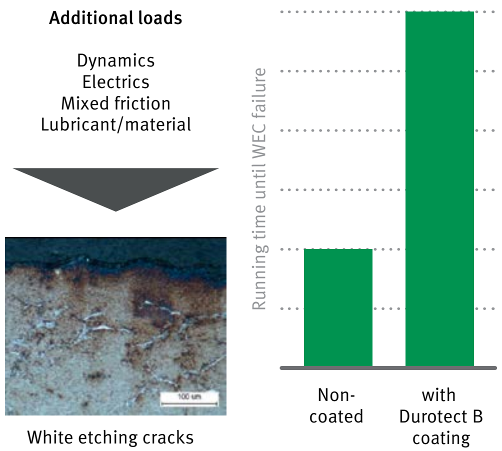
9 Increase in radial load capacity and reduction of design space requirements and weight through an improved design using an RNU308-E-XL type cylindrical roller bearing as an example

On the one hand, the bearing width and the weight can be reduced while maintaining the same radial load capacity. On the other hand, the radial load capacity can be increased significantly while maintaining the same bearing width. Figure 9 shows an RNU308-E-XL type cylindrical roller bearing as an example of the advantages with regard to load ratings, weight and design space.

Bearings for automatic transmissions and CVT

Transmissions have a large number of components with rolling bearing supports. The designs used essentially depend on the transmission type and its structural design. In manual transmissions, double-clutch transmissions and CVT, the main bearings tend to be tapered roller bearings, deep groove ball bearings and cylindrical roller bearings or drawn cup cylindrical roller bearings. The constant mesh gears are supported by needle roller and cage assemblies. By contrast, automatic transmissions with a converter have a large number of thrust needle roller bearings, radial needle roller and cage assemblies and planetary gear bearings.

Thanks to simulation-based geometry design and corresponding production processes, it has been possible to reduce the friction of thrust needle roller bearings by up to 50 %, depending on the load ratings and speed levels. The new geometry has slightly curved raceways, allowing approximate point contact at low to medium loads and thus reduced differential slip of the needles, as well as the formation of a full line contact at high loads. Initial applications have already been equipped with this bearing design, and volume production has begun. Testing has also shown considerable friction advantages in planet carrier bearing supports due to very compact thrust needle roller bearings.



10 Increase in robustness against WEC damage with Durotect B

Planetary gear bearings are subjected to extreme acceleration in stepped automatic transmissions as a result of high relative speeds, with some as high as 5,000 times the acceleration of gravity. The resulting forces lead to elastic deformation of the planetary gear bearing cages as well as to increased friction and cage wear caused by an unfavorable contact geometry. The shape optimization of the cage, in particular the cage bars between the rolling elements, has led to a significant improvement in durability. The use of an additional, directed blasting process and thus the application of residual compressive stress can increase cage strength even more. Testing on planetary gear set test stands has confirmed the increase in strength and also showed reduced cage wear. Coating with Schaeffler's Durotect M can have a positive effect on cage friction, which largely depends on the cage surface, and reduce it by up to 25 %.

So called white etching crack (WEC) damage appears more and more frequently on rolling bearings. These are cracks that form under mechanical loads and additional stress from material changes in rolling bearing steels and that can spread to the surface during the time that the load is applied, Figure 10 left. The occurrence of WEC is not application-specific or bearing-specific. The Durotect B coating developed by Schaeffler significantly increases robustness against WEC damage. Bench tests under targeted WEC conditions have demonstrated that the running time is three times longer, Figure 10 right. Transmission bearings with Durotect B have already been used successfully in some CVT transmissions.

Bearing concepts for electric vehicle drives

When bearings are used in electric drives, passage of electric current may occur and, under adverse conditions, result in melting, rehardening zones and electrical fluting [4]. There are generally two ways to prevent the passage of electric current, Figure 11. Current-insulated bearings with a ceramic plasma spray coating on the inner or outer ring are a well-known solution against the pas-

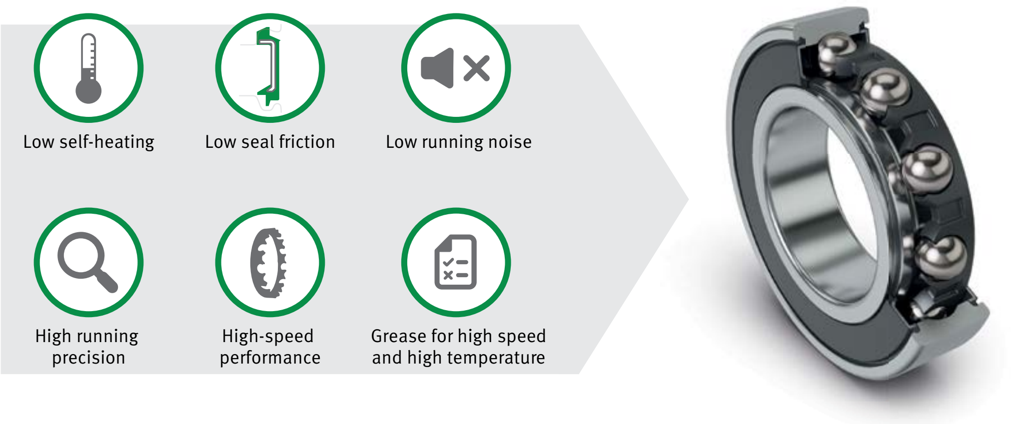
11 General measures against current damage to bearings

sage of electric current in generators and traction motors of rail vehicles, but they may shift the problem of electric current to other parts of the powertrain. That is why Schaeffler has developed bearings with integrated grounding to prevent the buildup of electric potentials between the inner and outer ring. Initial tests have already proved the effectiveness of grounding.

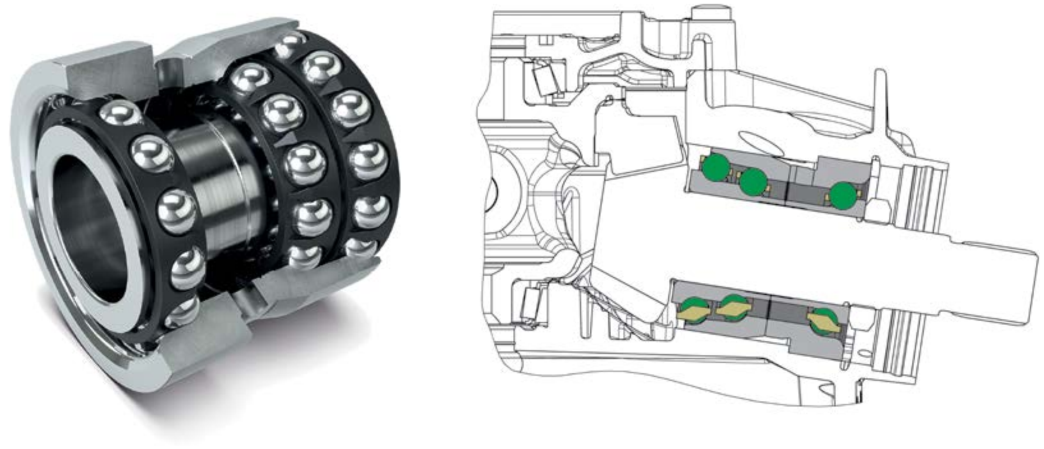
Due to the largely radial loads as well as based on cost, deep groove ball bearings [5] are used as bearing supports for electric motors. The bearing supports of electric motors must ensure the reliable guidance of the rotor to the stator, compensate misalignments and allow safe operation with low noise levels. The bearings are adjusted elastically and axially in order to compensate the increased radial internal clearance that is necessary due to the great difference in temperatures of the inner and outer ring. The bearing rings are dimensionally stabilized to prevent ring growth due to high operating temperatures.

Sealed high-speed bearings that are greased for life are required in almost all electric motors in hybrid systems or electric axles. That is why Schaeffler has developed bearings for electric drives that are combined in a modular range with preferred types and adapted specifically to these new requirements for electrified transmissions, Figure 12. The preferred types result in short sample delivery times as well as competitive pricing. High manufacturing accuracies in combination with specially developed greases, seals and cages permit the achievement of limits for n x dm (speed times the pitch circle diameter) of more than 1.3 million mm/min. This has already been proven by testing. Typical speed parameters for electric motors in an electrified powertrain today are between 0.5 and 1.0 million mm/min. However, experts are already working on increasing drive speeds in order to obtain compact drives with high power density.

In electric axles, not only the rotor shaft of the engine rotates at high speed but also the transmission input shaft that is connected to the engine via a spline. With an open design, the newly developed bearings are thus suitable for the oil-lubricated bearing supports of this transmission shaft.



12 **Bearing concept requirements for electric drives**



13 **Integration of bearing systems using a bearing cartridge for a hypoid drive as an example**

The hybridization of transmissions at the P2 position is of particular interest [6]. In P2 hybrid modules, the electric motor is arranged between the transmission and the internal combustion engine. Two clutches and their clever activation makes it possible to drive with just the internal combustion engine, with electricity only or with a combination of both drives [7]. In addition, coasting and recuperation can be implemented. That is why Schaeffler has developed hybrid modules for the P2 position. The optimized bearing supports for electrified transmissions are used in these hybrid modules as well as in Schaeffler's electric axles [8].

Integrated bearing systems

Besides optimized rolling bearings, more and more smaller assemblies have lately found their way into transmission applications. The combined use of components allows the development of integrated bearing solutions that offer advantages with regard to operating efficiency as well as mounting and installation space.

One example of this type of integration is the bearing cartridge for the hypoid drive in which the pinion flange bearing and the pinion head bearing are combined as a compact unit, Figure 13. This solution results in high rigidity for small bearing dimensions.

14 **Locating/non-locating bearing supports as a compact unit with a shaft and gear teeth for a hydraulic auxiliary drive in commercial vehicles**

15 **Bearing with integrated resolver**

Another example of bearing system integration is shown in Figure 14. As part of a hydraulic auxiliary drive for industrial vehicles such as articulated dump trucks, the locating/non-locating bearing supports form a unit with the shaft and the gear teeth. The cylindrical roller bearing can be designed with a direct raceway because the machining of the shaft is also the bearing manufacturer's responsibility. The shared outer ring of the cylindrical roller bearing and the deep groove ball bearing allows the implementation of a pre-assembled bearing system that is easy to install at the vehicle manufacturer's site.

Due to the continuing digitalization of the powertrain, the focus is on solutions that target the integration of electronic systems into the bearing. A conceivable development, for instance, might be a resolver bearing that replaces a resolver – i.e. an angular position transmitters – previously installed as separate component close to the electric motor Figure 15. Schaeffler pursues research and development activities for these intelligent bearings that can be used in applications far beyond the automotive industry [9].

Summary and outlook

Reducing friction while ensuring a high level of robustness remains the primary task of transmission bearings, irrespective of whether they are used in conventional or electrified powertrains. It has only recently become possible to optimize bearings in a very short time by using the OPTIKIT program developed by Schaeffler. Analyses of tapered roller bearings have shown that this can yield increased friction advantages of up to 20 %. Volume production of tapered roller bearings of this so called ultra low friction design recently began for axle drives.

With its innovative angular roller unit (ARU), Schaeffler has developed a locating bearing that has a 40 % higher load rating at low friction than a deep groove ball bearing with identical design space. Cylindrical roller bearings with narrow lips offer a reduction in bearing width by up to 20 % or, alternatively, an up to 27 % higher load rating with the same bearing width. This allows low-friction locating/non-locating bearing supports, such as those required for modern manual, double clutch and reduction transmissions of electric axles, to be achieved even more frequently than before.

With electric drives, there has been a trend towards increasing speeds, which means up to 30 % higher speed values for bearings. Reliable bearing supports for these high-speed drives can only be achieved with bearings that are characterized by high manufacturing accuracy in combination with specially developed greases, seals and cages. That is why Schaeffler has developed bearings for electric drives that are combined in a modular range with preferred types and adapted specifically to these new requirements. For applications in which electric current may occur, Schaeffler has developed potential solutions for integrating grounding into the bearing; these solutions have already proved to be effective in tests. Due to the shared use of components, such highly integrated bearings offer benefits in terms of efficiency as well as mounting and design space.

With these very specific bearing arrangements that are customized for the relevant application, Schaeffler will continue to contribute to efficient transmissions for conventional and electrified powertrains.

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13

Made-to-Order Double Clutch Systems

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Marco Grethel
Andreas Baumgartner
Karl-Ludwig Kimmig
Stefan Steinmetz

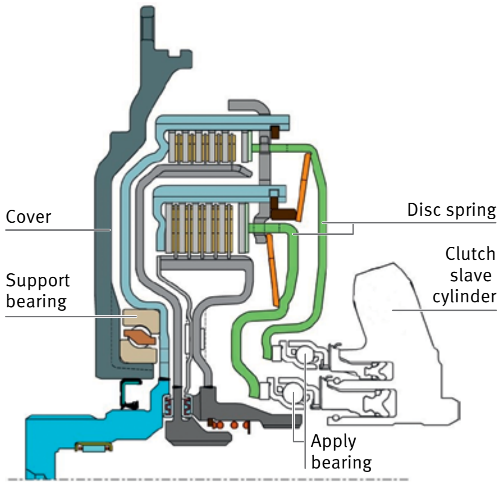
Introduction

Over the past few years, double clutch transmissions have become established as an integral part of the automatic transmission sector. They meet this sector's high requirements for driving pleasure, comfort and efficiency. Double clutch transmissions also offer great potential for further reducing CO_2levels as part of electrification.

With important transmission components such as wet and dry double clutches and a wide range of actuators, Schaeffler has maintained its leadership position on the market in the past few years, producing more than four million double clutches per year worldwide.

Schaeffler covers all vehicle segments, from compact cars to super sports cars and SUVs and develops customized solutions with the aim of improving efficiency and comfort in the power train even further.

The first part of this paper presents the various double clutch designs. The wide range of designs allow Schaeffler to offer optimum solutions for customers' specific requirements. The latest developments with regard to friction linings are introduced below. With a new wet friction lining, developed in-house and ready for volume production, Schaeffler is now able to offer a wet double clutch system with all key technologies from a single source. For dry applications, the B9000 friction lining shows improvements in terms of comfort. The final part describes the efficiency, controllability and costs of transmission actuator systems.



1 Half section of a wet double clutch with actuation bearings

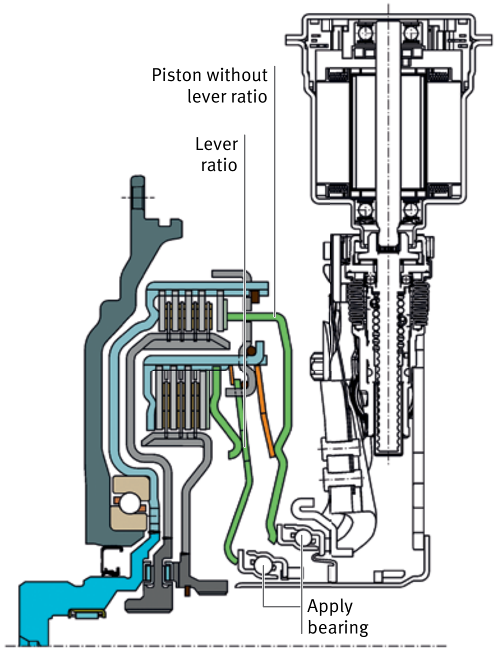
Wet double clutch product portfolio

Wet double clutch with actuation bearings

The double clutch shown in Figure 1 is nested radially and is actuated via engagement bearings. This type of actuation requires a support bearing which fixes the clutch position and supports the respective forces in the radial and axial direction. The actuation of the clutch supported by rolling bearings minimizes friction losses for rotating cylinders when compared to hydraulic unions. Here the clutch does not have an internal ratio and is actuated directly, resulting in design space advantages compared to lever-actuated clutches.

Wet double clutch with lever actuators

Figure 2 shows the wet double clutch with internal ratio. With this design principle, the efficient engagement bearings can continue to be used. The smaller friction diameter of the inner subclutch normally requires either additional friction

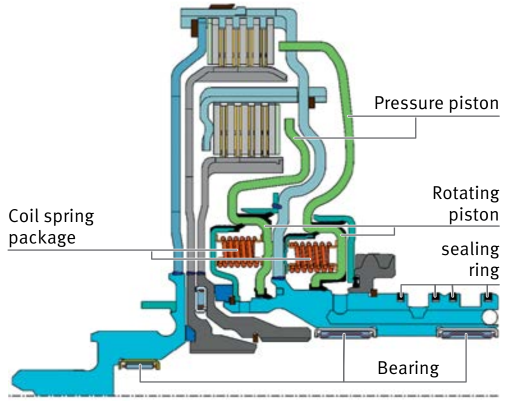


2 Half section of a wet double clutch with internal ratio on subclutch 2

plates or an increased actuation force. In order to limit the number of plates, the inner clutch requires a lever ratio to achieve a higher contact pressure by applying the same actuation forces. This makes it possible to transmit torque without CO_2disadvantages, with only a slight loss of design space. Here too the actuation force is supported by a rolling bearing bearing. In combination with the lever actuation, this clutch is an innovative solution with regard to costs and efficiency.

Wet double clutch with rotary feedthroughs

This clutch variant is particularly suitable for applications that have hydraulic transmission actuation, which allows the clutch actuation to be achieved with only little additional effort.



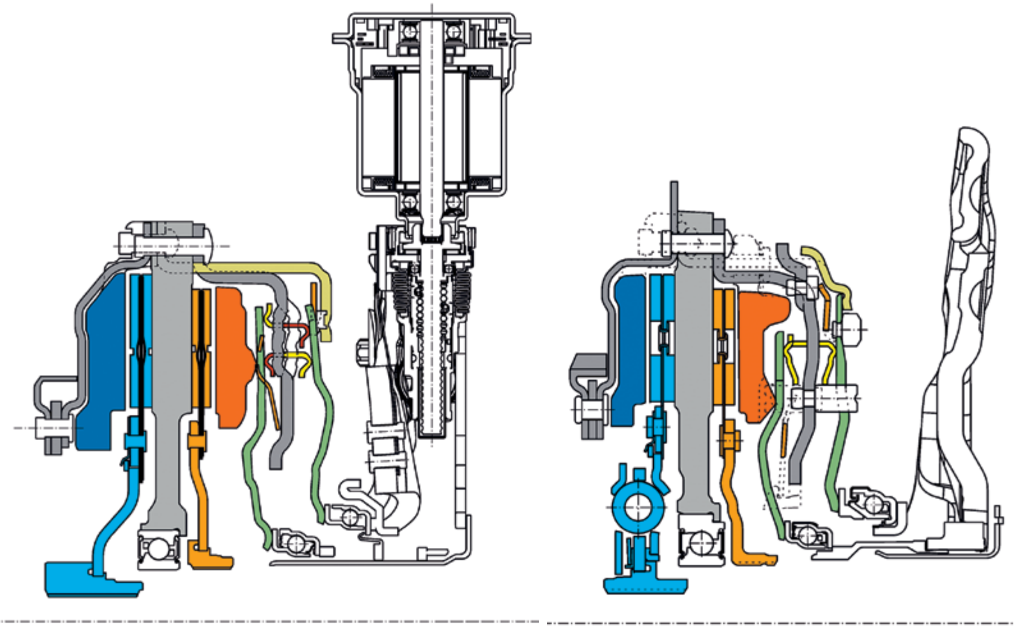
3 Half section of a wet double clutch with rotary feedthroughs and rotating cylinders

In this design, the actuation energy into the clutch is not transmitted via rolling but via a rotary feedthrough. The transmission of pressure through the sliding sealing ring is prone to leaks and also generates friction losses depending on the pressure. The direct actuation of the friction plates via the pressure piston, however, results in short actuation travel. The required return force of the clutch pistons is provided by an internal pressure spring assembly [8].

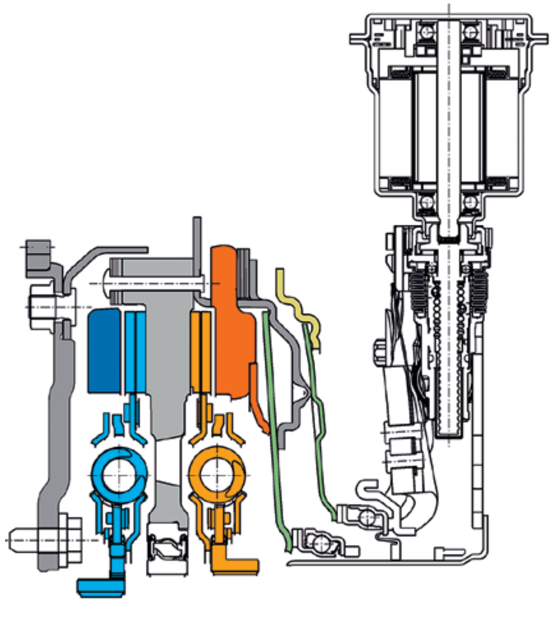
Dry double clutch product portfolio

Dry double clutches with wear adjustment

Dry double clutches will continue to play an important role in Schaeffler's portfolio. In the torque class of up to 280 Nm, this design variant represents a cost-efficient solution which also delivers CO_2 savings due to reduced drag torques.



4 **Half section of two double clutches with wear adjuster**



5 **Half section of a dry double clutch without wear adjustment**

This becomes evident in two clutches currently in production, shown in Figure 4. These clutches have wear adjustment mechanisms and are actuated in their applications by means of bearings. With regard to their thermal mass, they are designed for small and medium vehicle weights.

Since these clutches are not oil-cooled, the cast material masses required to ensure the necessary thermal capacity of the clutches lead to higher moments of inertia than the wet clutches that also serve as the secondary mass of the torsion damping system. Even though the drag torques of the sealed and grease lubricated bearings are slightly elevated, the dry double clutches are a low-cost and efficient alternative in their range of applications.

By eliminating the wear adjustment and minimizing the thermal mass, it is possible to offer

a variant for low vehicle weights and low torques of up to 150 Nm. This clutch meets customer requirements for the low cost segment.

Friction linings for double clutches

A key technology of the clutch is the tribological system; in a wet clutch, it consists of steel plates, oil and wet running friction linings. Schaeffler offers its own linings for both dry and wet running double clutch systems.

Volume production experience in wet and dry double clutches has been essential in Schaeffler's ability to provide its systems with a lining produced in-house, making it an independent supplier of complete systems. Schaeffler is thus able to customize functions to customers' specific needs. Schaeffler's friction lining is ready for volume production, and production facilities for the friction plates are in place. The manufacturing process was developed for corrugated friction plates. The steel plates have been in production since 2015.

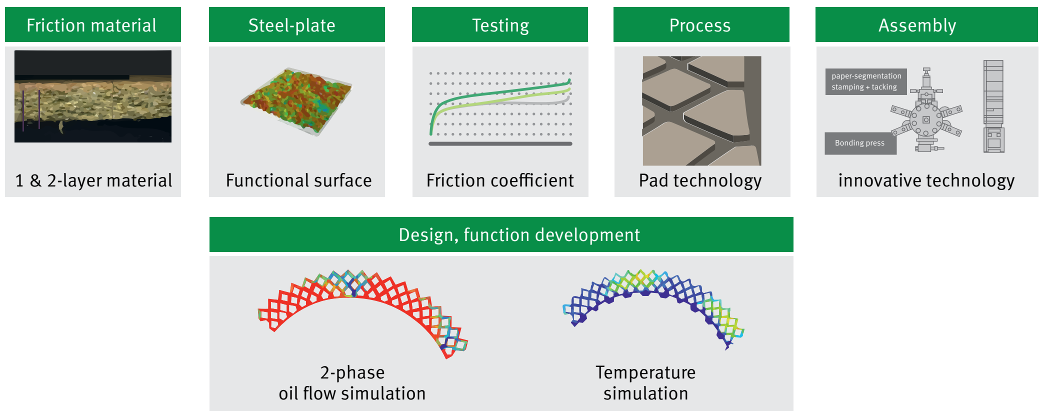
The friction system offers the following characteristics

- Good heat dissipation
- Low drag torque
- High statistical friction coefficient
- Minor torque irregularities
- Positive friction coefficient gradient (μ-v) for damping behavior during the entire life cycle.

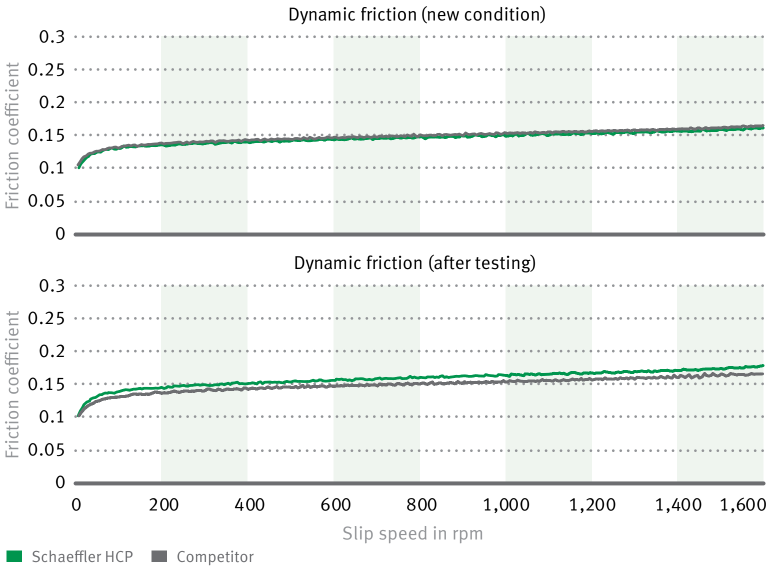
The development of this one-layer wet running friction lining followed a comprehensive approach that is shown in Figure 6 and that integrates material development, design development, surface technology, process expertise, simulation, testing and manufacturing technology. Various areas of expertise at Schaeffler were used to develop the wet friction lining with the required characteristics.

Compared with other benchmark friction linings on the market, Schaeffler's wet friction lining shows a good performance, both in new condition and throughout service life, Figure 7.

The friction coefficient characteristics of Schaeffler's lining are comparable to competitor products, which is why it can also be used in existing customer applications.

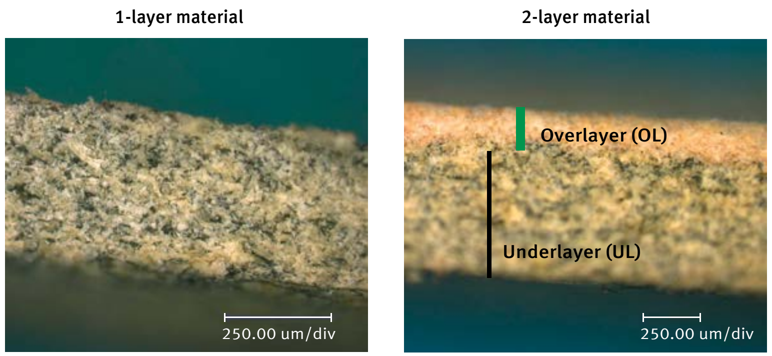


6 **Comprehensive approach to the development of Schaeffler's wet friction lining**



7 **Benchmark results for the dynamic friction coefficient of Schaeffler's wet friction lining and a competitor product (new and after endurance test)**

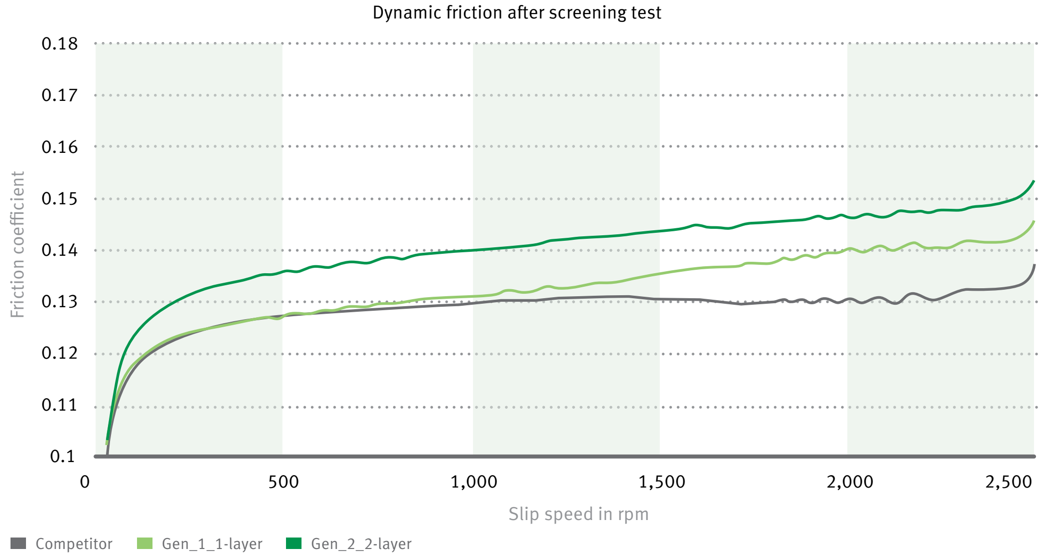
Besides the one-layer material that is already available, a two-layer wet running friction material is currently being developed in order to be able to cater more sophisticated applications as well. Figure 8 compares both material approaches. As shown in the part on the right, the two-layer friction material consists of a support layer referred to as "underlayer" in the drawing and a second layer called "overlayer".



8 **Schaeffler's wet friction lining as a one-layer design (left) and the new development approach with a two-layer wet running friction material (right)**

The friction lining can be adapted ideally to the required behavior by precisely coordinating the characteristics of the support layer and the friction contact layer. In this way, an advantage can be achieved, for instance, in terms of robustness under high stress applications – such as repeated uphill starts at full load – when compared to the one-layer friction material. As shown in Figure 9, the friction coefficient level is significantly higher and the friction coefficient gradient is more stable even after 15,000 uphill starts at full load.

Schaeffler is currently working on the development of the two-layer wet running friction material. The layer composition and the required processes, however, have already been defined. In the next few development steps, the insights gained will be applied to volume production applications. The two-layer design promises a favorable cost-benefit ratio with a definite increase in performance for extremely challenging applications.



9 **Comparison of dynamic friction coefficients of the one-layer and two-layer Schaeffler friction lining after 15,000 uphill starts at full load.**

The wet friction lining is a good example of Schaeffler's ability to expand its product expertise by applying the company's key competencies to serve our customers as a system partner.

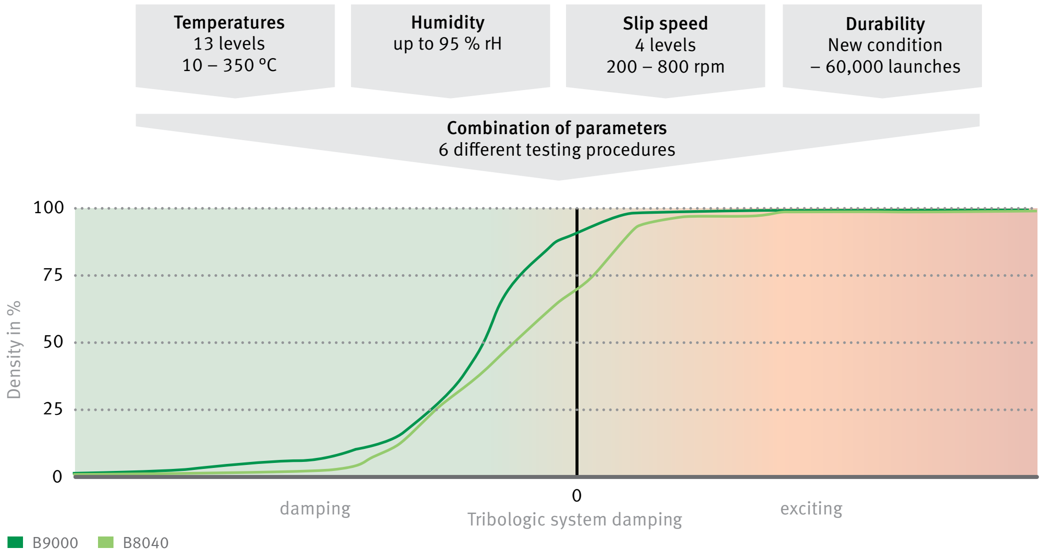
As with the wet running double clutches, the tribological system is also essential for dry clutches. Over the past few years, Schaeffler has developed dry friction linings that meet specific requirements for thermal robustness, availability and comfort in double clutch systems. Its RCF-1o and B8040 h friction linings now constitute a global benchmark.

Based on years of experience in the field and its volume production of millions of units per year, Schaeffler, has developed a new generation of dry running linings for automated applications. As announced during the last symposium in 2014, the B9000 lining is expected to improve comfort. Figure 10 shows a comparison of the tribological system damping of the B9000 and the B8040.

The damping of the tribological system is analyzed in numerous tests with varying temperatures, humidity, differential speeds and rating life, including the duration of the test. The B9000, shown in blue, offers even greater performance than the B8040, particularly in critical driving conditions with a slip speed of 200 rpm.

Actuator portfolio

Actuators and their control through intelligent software modules are essential in creating the ideal double clutch system. Figure 11 shows part of Schaeffler's portfolio of actuators that have been established successfully on the market.



10 **Comparison of tribological system damping**

11 **Portfolio of Schaeffler actuators: Hydrostatic clutch actuator, lever actuator and transmission actuator**

After providing a brief overview of Schaeffler's actuator systems, they are presented in a system context. Based on this, advanced developments are described in comparison with competitors' systems.

Hydrostatic clutch actuator (HCA)

The HCA [3] uses a leak-free hydrostatic section to transmit the actuation energy to the clutch with as little loss as possible. This makes it a solution optimized for efficiency.

Electromechanical lever actuator

Millions of lever actuators (HA) have already been produced in combination with dry running double clutches and only has to be adjusted for operation in oil for wet applications. A spring-supported lever is used to apply actuation forces on the clutch. The lever actuator is thus a solution integrated into the clutch housing that offers benefits with regard to efficiency and costs.

Electromechanical transmission actuator with "active interlock"

Over the past few years, the proven electromechanical transmission actuator has been developed further in terms of costs and functions and is now in large scale volume production. The transmission actuator is driven by two electric motors: one for the select direction and one for the gearshift direction. Due to this separation, theoretically any number of shift rails can be actuated. The parking lock, for instance, can be easily integrated. The so-called active interlock mechanically prevents two gears from being engaged inadvertently at the same time in a sub-transmission, eliminating the need for additional sensors. The use of plastic parts in many positions makes this a low-cost concept.

System expertise

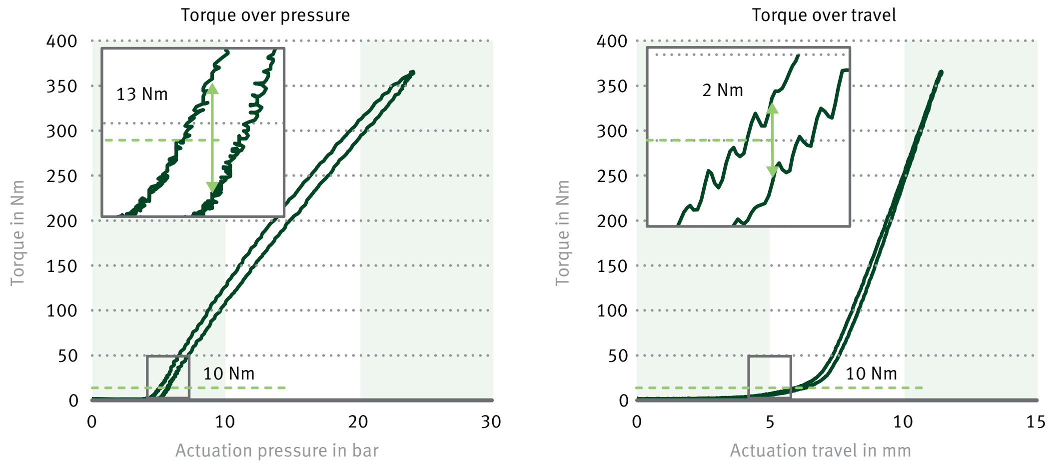
In order to make a meaningful comparison of systems that are sensible froma technical standpoint, aspects such as controllability and efficiency must be evaluated.

System controllability

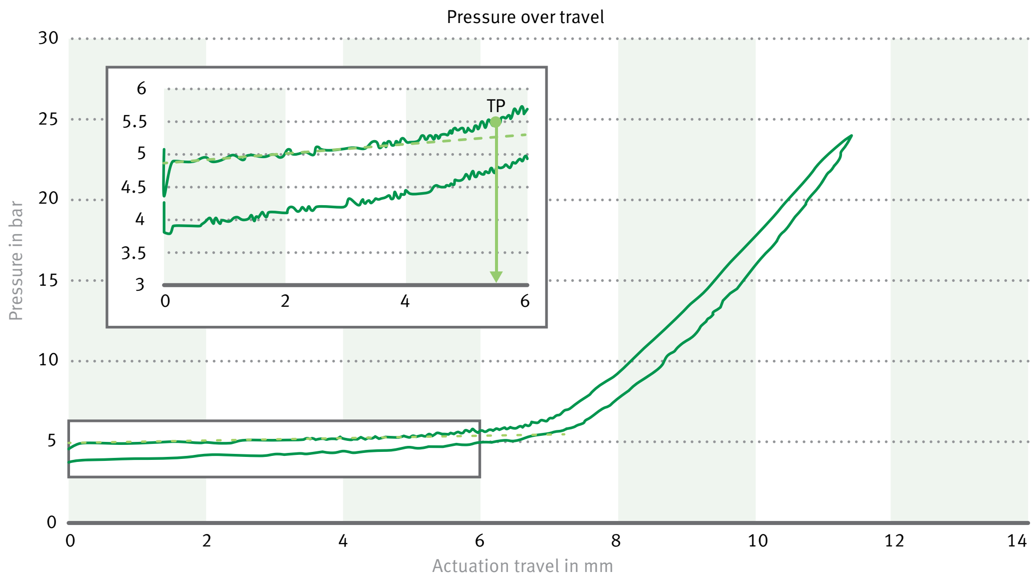
Besides minimal losses, the "Hydrostatic" system also offers advantages when it comes to control. Both the actuation travel as well as the actuation pressure are available for actuating the clutch. This is what sets this system apart from competitors' systems which typically only use pressure as a control variable.

Figure 12 shows a clutch system measurement. The diagram on the left shows the transmitted clutch torque in relation to the actuation pressure, and, on the right, in relation to the actuation travel.

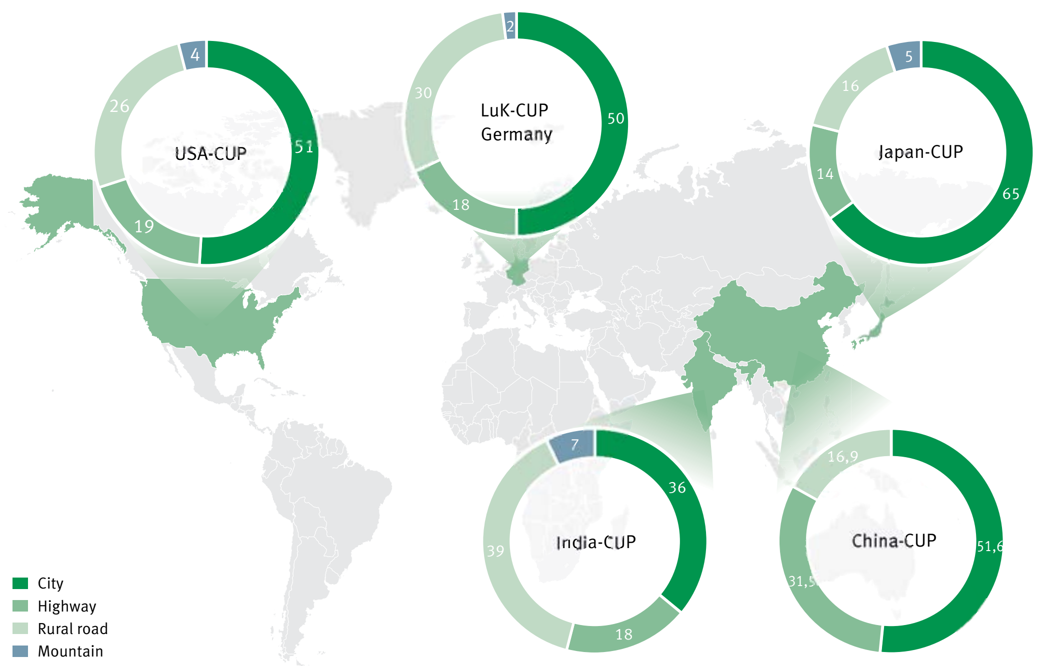
An important aspect of controllability is the accuracy with which the touch point can be controlled. This



12 **Torque characteristics of leakage-free systems with regard to controllability of the touch point**

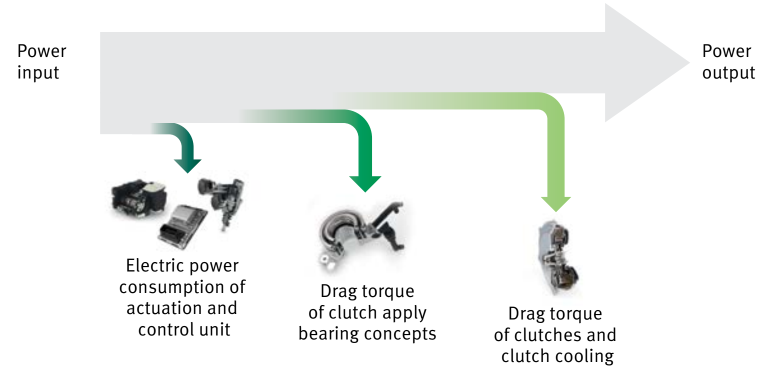


13 Determination of the touch point by means of the pressure-travel characteristic curve of a double clutch system



14 Driving profiles and load spectra worldwide, determined in a separate driving profile and broken down by city, freeway, highway and uphill starts (customer usage profile – CUP)

refers to the point at which the clutch begins to transmit the torque. When analyzing the touch point, here at 10 Nm, with regard to its hysteresis on the pressure and travel characteristic curve, we find that the travel hysteresis with 2 Nm is much lower than the pressure hysteresis with 13 Nm.



15 Components analyzed and their power losses

Since both the travel and the pressure signal may be available in a hydrostatic actuation, as shown in Figure 13, the touch point be determined exclusively by these two actuator signals. This is the system's controllability advantage: The touch point can be determined online, in the vehicle, without any clutch torque transmission. In addition, touch point point determination is independent from other input parameters such as the torque signal from the internal combustion engine. For the reasons stated above, touch point determination is relevant for systems capable of stop/start operation, and due to the increasing electrification of the powertrain.

Evaluation of system losses

For technical evaluations, Schaeffler not only relies on test stand-based consumption cycles such as NEDC or WLTP, but for years has used its own real driving cycle that addresses local differences between markets around the world [6], Figure 14. Depending on the traffic situation, this real driving cycle can meet the criteria for an RDE measurement published by the EU [10].

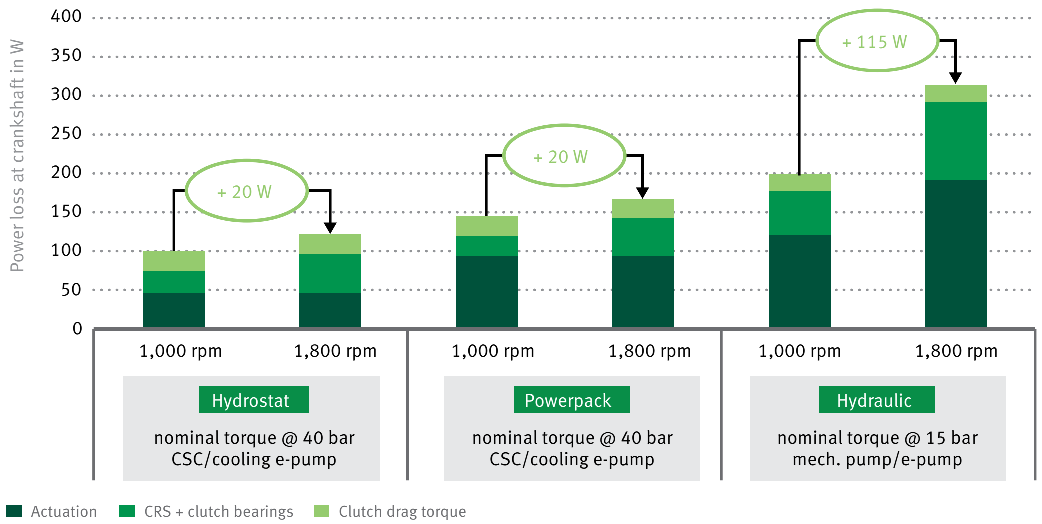
The losses evaluated for each component are shown in Figure 15.

The following systems available on the market are compared in order to compare system losses:

- "Hydraulic": wet double clutch with rotary feedthroughs, actuation of the clutch and the transmission as well as cooling the clutch via a modern hydraulic system
- "Power pack": wet double clutch with engagement bearings, actuation of the clutch and the transmission as well as cooling of the clutch via a power pack
- "Hydrostatic": wet double clutch with engagement bearings, actuation of the clutch via a hydrostatic actuator, actuation of the transmission via an electromechanical transmission actuator and cooling of the clutch via an electric pump

The "Hydrostatic" system favored by Schaeffler has the smallest power loss compared to competitors' products.

From the real driving cycle, an average engine speed of 1,800 rpm has been determined from a state-of-the-art double clutch system. The speeds in the NEDC are much lower. For this reason, two average speeds are compared to show the impact of cycle: Crankshaft speed of 1,000 rpm and 1,800 rpm

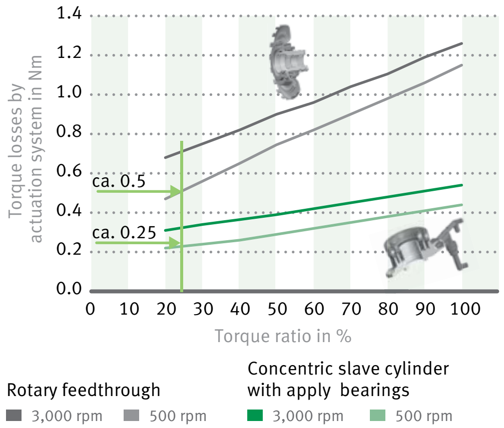


16 **Power losses compared to "Hydraulic", "Power pack" and "Hydrostatic" systems at an average speed of 1,000 rpm and 1,800 rpm in relation to the crankshaft**

Variations in speed impact on drag losses of clutch activation as follows. For clutches actuated via engagement bearings, a drag torque of 0.25 Nm is assumed, for those actuated via rotary feedthroughs, the assumed drag torque is 0.5 Nm (see Figure 17). For bearing-actuated clutches, this reduces the power loss from 50 W to 25 W and, via a rotary feedthrough, from 95 W to 50 W if the speed is assumed to be 1,000 rpm instead of 1,800 rpm.

The power consumption of the clutch or transmission actuation changes according to the speed only in those systems that obtain their actuation energy from the crankshaft, i.e. a mechanically driven pump. The "Hydraulics" actuator system is based on an 8 cm³ pump with a medium pressure level of 4 bar. This reduces the power loss for the above speed from 160 W to 90 W.

In the systems evaluated here, the clutch is cooled either via a demand-controlled electrical cooling oil pump or the mechanically driven pump installed for actuating both the clutch and the transmission. In the case of the electrical cooling oil pump, the cycle-related losses are assumed to be independent of the drive speed. Since effect of the drive speed on the mechanically driven pump has already been considered, there are no additional losses here.



17 **Drag loss measurements of engagement bearings and rotary feedthrough**

A comparison of the three "Hydraulics", "Power pack" and "Hydrostat" benchmark systems shows that the "Hydraulics" system is particularly sensitive to variations in the drive speed. The losses under real driving conditions are almost 55 % greater at approx. 315 W compared to the synthetic cycle of approx. 200 W. By contrast, the power loss of the "Hydrostatic" system only varies from 100 W to 120 W depending on the cycle.

The analysis of the current market situation is completed by the dry double clutch systems:

- "Power pack": dry double clutch with engagement bearings, actuation of the clutch and the transmission via a power pack
- "Electromechanical": dry double clutch with engagement bearings, actuation of the clutch via an electromechanical lever actuator, actuation of the transmission via an electromechanical transmission actuator

Once the boundary conditions for using dry double clutches are in place, these systems are a low-cost and efficient alternative. At 1,000 rpm, the electromechanical system has a power loss of only 75 to 90 W.

Requirements for modern powertrains

The growing use of wet double clutches and increasing electrification has resulted in additional requirements for these systems:

- Cooling of the clutch, transmission, electric motor and power electronics
- Additional shifting elements, such as K0 and parking lock
- Greater flexibility with regard to actuator positioning and design space
- Increased efficiency and reduced costs

Besides the systems introduced here, Figure 18 shows two options that meet the increased demands:

- "Electromechanical": wet double clutch with engagement bearings, actuation of the clutch via an electromechanical actuator, actuation of the transmission via an electromechanical transmission actuator and cooling of the clutch via an electric pump

Due to the use of technologies proven in volume production such as the lever and transmission ac-

	System "Hydrostatic"	System "Powerpack"	System "Hydraulic"	System "Electro-mechanic"	System "Hydraulic/ Electromechanic"
Torque	› 300 Nm	› 300 Nm	› 300 Nm	› 300 Nm	› 300 Nm
Pressure level	40 bar	40 bar	15 bar	mechanic	15 bar
E-Pumps	2x Hydrostat +1x 4 cm³/U	1x 0.4 cm³/U +1x 5 cm³/U	1x 3,5 cm³/U	1	0
Pressure accumulator	0	1	0	0	0
Mechanic pump	0	0	1x 8 cm³/U	0	1x 4.5 cm³/U
Suction jet pump	0	0	0	0	1
active valve	0	7	10	0	4
Actuator E-Motor incl. pumps	5	1	1	5	2

18 **System comparison**

tuator, a particularly low-cost and efficient solution can be created for wet double clutch transmissions.

In order to achieve greater positioning flexibility and to include the option of operating additional shifting elements, the following concept is available:

- “Hydraulic/Electromechanical”: wet double clutch with rotary feedthroughs, clutch actuation and cooling via a mechanically driven pump (4.5 cm³/rev) that is supported by an ejector pump and electromechanical clutch actuation

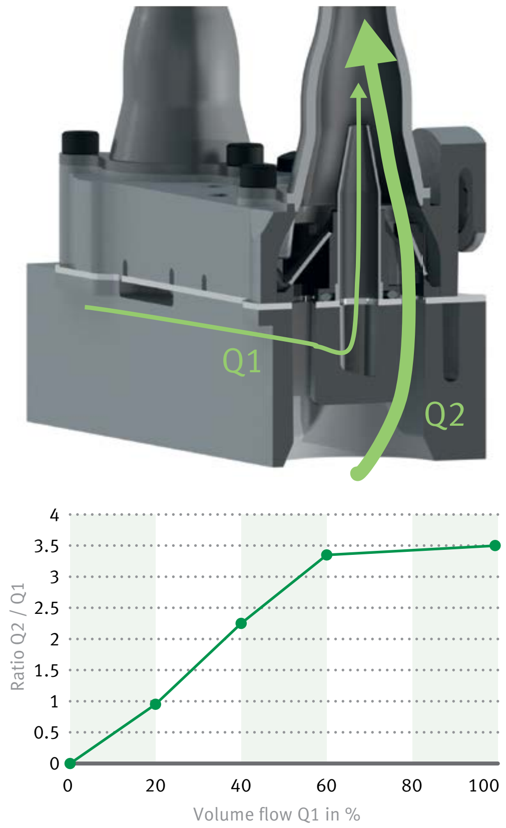
A low-cost ejector pump is used to keep the pump compact and efficient and to provide the required cooling oil volume flow.

With this concept, the clutch cooling solution deserves particular mention. The volume flow can be increased by means of an ejector pump. This allows a smaller design of the energy-producing pump.

Summary and outlook

The components developed by Schaeffler are integrated as low-cost and efficient systems according to customer requirements, offering numerous advantages including controllability. With the wet friction lining, Schaeffler has added a key technology to its portfolio. This allows Schaeffler to offer system solutions from a single source in order to meet customer requirements even better.

The new B9000 generation of dry friction linings also raises standards, in particular with regard to comfort, further narrowing the gap between dry and wet systems. The B9000 certainly represents a worldwide benchmark.



19 Ejector pump including measurement of the amplification factor of the volume flow

Its intelligent components and system expertise make Schaeffler the right partner for the development and production of new solutions for state-of-the-art powertrains.

In view of increasing electrification, the actuator system must be able to handle a growing number of consumers. In addition to the solution shown here, a hydraulic power on demand actuator is a good option. This PoD hydraulic system and clever triple clutch arrangements are described in the relevant chapters.

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14

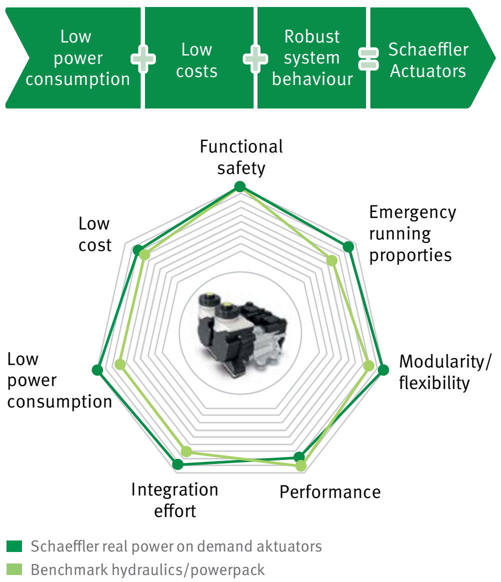
Innovative Power-on-Demand Concepts for Transmission Actuation

Bruno Müller
Marco Grethel
Mathias Göckler

Introduction

Automotive development focuses on energy efficiency In order to fully use the potential involved, all energy consumers must be considered, including actuators that actuate the components in the drive train. They can absorb average power in the three-digit watt range. Against this background, this paper discusses concepts that trace every single watt lost in the drive train. It is possible to develop actuator systems that on average only absorb 10 to 20 watts electrically. Schaeffler has demonstrated this multiple times and launched successful volume productions [1; 2].

Besides fuel savings and the reduction of CO_2 emissions, particularly in the WLTC cycle and under real driving conditions (RDE), intelligent architectures are becoming increasingly important. They ensure functional reliability in the engine and in the transmission and extended emergency running properties [3]. In addition, the number of different transmission structures is also increasing constantly. Conventional transmission structures are hybridized in various arrangements or transferred into dedicated transmission concepts for hybrid drives (Dedicated Hybrid Transmissions, DHT). The transmissions for electrical-only vehicles must also be actuated efficiently. This results in an additional requirement for modular actuators that can be used universally, can be scaled depending on the number of consumers and are compatible with various operating media. Figure 1 shows the conflicting requirements that must be considered in actuator development, from little power consumption and low costs to functionally reliable operation to optimal design for modular use in various configurations.

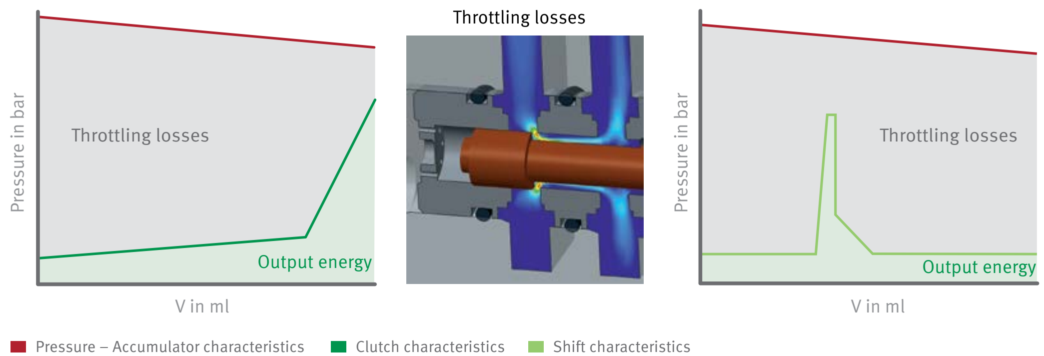


1 **Conflicting requirements for actuators in vehicle powertrains**

Real Power on Demand

Initial situation

In many transmission concepts, actuator systems are not truly or only partially demand-based. Conventional automatic transmissions, for instance, mostly use oil pumps that are permanently driven by the internal combustion engine. The disadvantage in terms of efficiency is compensated by the high number of gears and, particularly, the large ratio spread. However, the power loss of the actuators in the three-digit watt range is considerable. The use of "power on demand" actuators is difficult here because some rotating clutches can only be actuated with leaks in the supply lines and the supply to the torque converter as well as the cooling system and oil lubrication must be ensured continuously. Additional electric oil pumps or hydraulic accumulators are required to actuate the transmission when the engine is not running and to ensure quick availability after stop-start.



2 **Throttling losses on pressure reduction valves of power packs**

The situation is similar for conventional CVTs in which high pressure must be available for torque transmission in the variator. Double clutch transmissions have fewer gears but also have smaller number of clutches with drag losses. That is why it is all the more important not to lose power in the actuators of double clutch transmissions. In hybridized transmissions every watt consumed by the actuators is no longer available for the electric drive, reducing the electric range of the vehicle.

Modern transmissions with a hydraulic control system often have a mechanical pump driven by the powertrain as well as a second, electrically driven oil pump to meet the high dynamics requirements. This two-pump variant today can be regarded as the minimum standard for a hydraulically controlled transmission. In the leakage balance, hydraulic systems are designed for operating points with high temperatures and thus for low-viscosity oil and low speeds. In real operation, however, they supply more oil volume than necessary. Normal driving and sporty driving at high speeds requires the excess oil to be circulated with high losses. Actuators with permanently driven oil pumps thus waste a large amount of energy due to a volume flow that is too high in many driving situations. These types of actuation concepts generally cannot be considered real power-on-demand systems.

Power packs with an electrically driven pump and a pressure accumulator are often seen as power-on-demand systems. However, throttling losses occur when volume is removed from the pressure accumulator to reduce pressure from the accumulator on the service pressure. Pressure is converted to heat at the throttling points. In addition, part of the high-pressure fluid is transported into the tank from the pressure accumulator via the valve gap. Losses are particularly relevant if the system to be actuated has leaks. Based on this, a power pack hydraulic system is not really a true power-on-demand actuator system. Figure 2 shows potential throttling losses on the pressure reduction valves depending on the characteristic curves of the pressure accumulator and clutch and the gearshift curve.

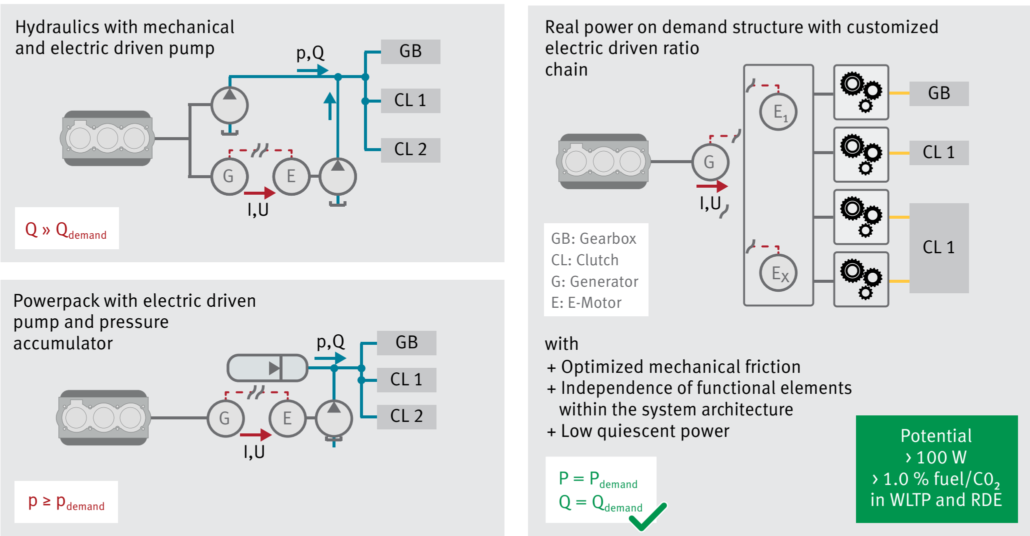
Requirements for requirement-based actuators

If real power-on-demand actuators are used, the energy supplied to the electric motor must be converted as directly as possible into adequate forces and pressures with accurately fitting travel and volumes. It is particularly important not to generate excess forces (such as pressures or torques that are too high) and no excess travel (such as too much volume), but rather precisely those forces that are required for a particular function.

Another aspect is holding positions. Theoretically, there is no active energy involved, but in reality, a lot of energy is spent on maintaining a condition.

In the actuation of transmission components, energy is often stored in the system's elasticities – particularly in the clutch. When the clutch is opened, this energy can be used to support the actuators. Not only does this enable shorter return times but also reduced power consumption by the actuators when opening the clutch. In order to minimize the adjustment and holding power, the design must find the right compromise between low and high mechanical friction. That is why, among other things, mechanical spindle drives [1, 2] or special friction elements are used. A wrap spring, for instance, can be used to adjust direction-based friction.

If the safe mode or the emergency running characteristics in the event of failure require the self-opening or self-closing of the clutch, the clutch must be actuated continuously against its return force. In this case, a motor or magnet must be supplied permanently with holding current in order to convert electrical energy directly into a holding force. Here electrical energy is lost in the form of heat in the electric motors, in the magnets and in the power electronics system, and the removal of this heat often involves a great deal of work. Such holding losses can only be prevented by using mechanically self-retaining systems and actively carrying out the opening or closing function required in the event of failure. For such self-holding systems, it is a good idea to use ar-



3 **Comparison of real power-on-demand actuators (right) with hydraulic solutions and power packs**

chitectures that, in the event of failure, can produce a safe mode with the remaining components and provide required emergency functions through functional redundancy [3].

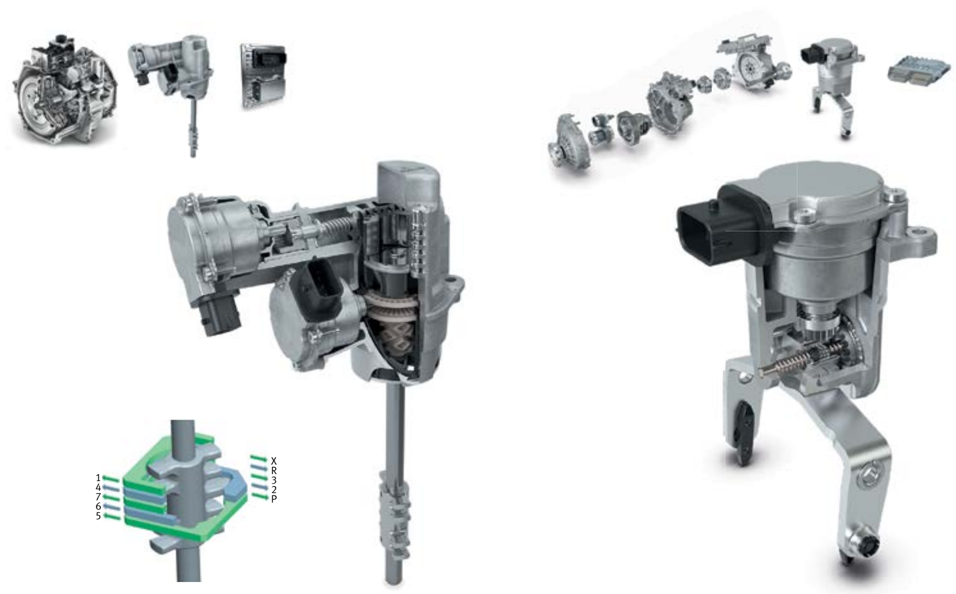
In real power-on-demand systems, all control units must also be designed for quiescent current that is as low as possible to minimize power consumption when the actuator system is not in motion. This must be taken into consideration, particularly when selecting loss-related electronics components such as microprocessors, voltage regulators and sensors. Software can also have an effect on energy absorption in static condition. Position controllers must be switched off, for instance, or parameterized to ensure that they consume as little power as possible. In addition, power absorption can be reduced significantly through holding current adaptation. Figure 3 summarizes what a real power-on-demand actuator system must include and compares it with power packs and fully hydraulic solutions.

Compared to a modern hydraulic system with two pumps, a real requirement-based actuator system has a performance benefit of more than 100 watts, which corresponds to savings of more than 1.0 % of fuel or CO_2. This illustrates how important a real power-on-demand actuator system is for transmissions. However, it must be realized at usual market costs [4, 5].

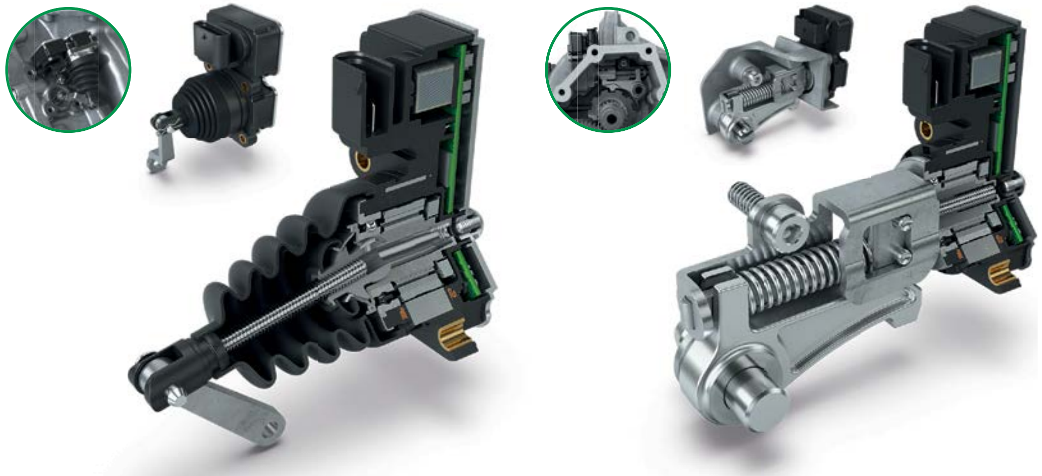
Examples of requirement-based actuators

Active Interlock transmission actuator and electric axle actuator

In the proven Active Interlock gear actuator system with integrated lock, the two adjustment motors are controlled only when gears must be engaged or released. Otherwise, the actuators are loss-free, except for a low quiescent current ab-



4 **Active Interlock transmission actuator and electric axle actuator**



5 Actuators for automated driving and for smart interior solutions (Park-by-Wire)

sorption by the control unit. The electric axle actuator works according to the sample principle, but without a selector axis. It is a good example of the way transmission actuators are also already used in volume production applications for electric vehicles – in this case, for the low-loss shifting of a two-gear electric axle [6]. Figure 4 shows both examples of the real power-on-demand actuators.

Actuators for automated driving and smart interior solutions

Schaeffler has developed two electro-mechanical actuators for applications in electric vehicles and for automated driving, Figure 5. The Schaeffler actuator for PRND selection in automatic transmissions can also enable older transmissions for automated driving, such as right at the interface of the cable tension. The integrated park lock actuator (PLA) was developed as an efficient actuation module for the park lock function, particularly in transmissions of electric vehicles, such as for electric axles or as a park lock in dedicated hybrid transmissions (DHT).

Electric Concentric Actuator

The electric concentric actuator (ECA), Figure 6, is also a purely electromechanical actuator in

6 Electric concentric slave cylinder (EZA) for activating the K0 clutch in hybrid modules

volume production applications. For the clutch, the actuator system is designed to minimize the holding currents in the operating points that are used the most. In order to find a good balance between holding currents and adjustment currents – as described as a requirement above – the clutch curve and the pitch in the ball screw drive have been matched optimally. It is essential here that this adjustment must be adapted for new applications if the frequencies of the operating points that occur shift – for instance when designing a hybrid vehicle driven by an internal combustion engine and an electric motor. Frequent operating conditions must be optimized in detail for a power-on-demand actuator system [7].

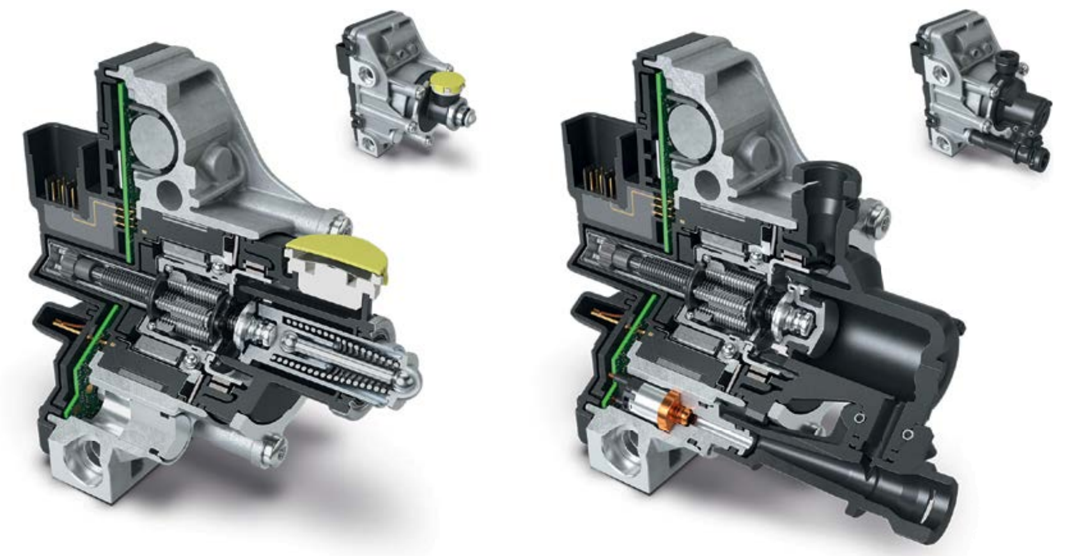
Hydrostatic clutch actuator

Another requirement-based electro-mechanical actuator is the hydrostatic clutch actuator (HCA), Figure 7. What is also special is that it uses a leak-free hydrostatic section to transmit the actuation energy to the clutch with little loss. The bibliography [1] and [3] provides detailed information about the design of this actuator, with particular consideration given to low power consumption. What must be emphasized here is the local control unit (LCU), which includes a functional redundancy when several of these actuators are used. The HCA is already in use in double clutch transmissions, in hybrid drives and for electrical clutch systems in manual transmissions. It actuates clutches in a torque range of 150 to 700 Nm and can be operated with both brake fluid and ATF [1, 2, 7, 8, 9]. It has already been demonstrated that actuators with optimized consumption can also be used in highly dynamic applications and that efficiency does not contradict high performance in actuation.

7 Hydrostatic clutch actuator (HCA)

Modular clutch actuator

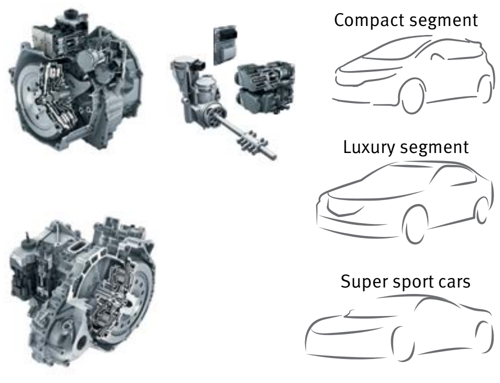
The Modular Clutch Actuator (MCA) is refinement of the HCA, which focuses on the modular design for various applications, Figure 8. A new type of the planetary spindle drive (PWG) must be mentioned in particular. It allows the elimination of the previously used, additional absolute travel sensor system because it detects the angle positions across several rotations with the help of a new multi-turn angular position sensor. The double-sided wrap spring allows the forward and backward friction to be optimized for specific applications. The MCA can be equipped with a hydrostatic or mechanical interface. The integrated local control unit (LCU) has sensor inputs and a computer capacity that make it suitable for sophisticated tasks. This allows the MCA to be used in a wide range of clutch applications [2, 8, 10, 11], such as for automated clutches in manual transmissions (E-Clutch).



8 **Modular clutch actuator with new planetary spindle drive**

Power-on-demand actuators in double clutch transmissions

The most consistent real power-on-demand actuator system for double clutch transmissions that is available on the market consists of two clutch actuators such as the HCA or MCA described above and a gear actuator such as the Active Interlock Actuator in combination with a transmission control unit, Figure 9. This system has already been used successfully for small and large, wet and dry double clutches in all vehicle classes. Due to the integrated sensors and control units, it is easy to control by travel and/or pressure control, provides the required dynamics and allows new solutions to be found with regard to reliability and emergency running.

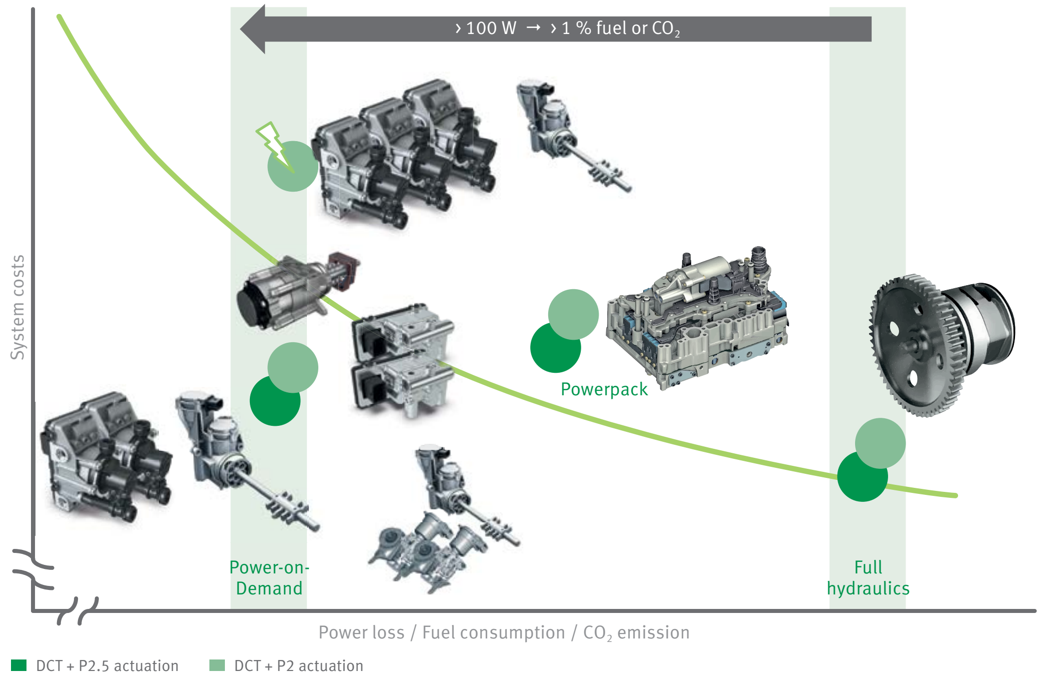


9 **Real power-on-demand actuator system for double clutch transmissions in various vehicle platforms**

However, with four electric motors and three local control units, this actuator system is relatively complex. That is why potential improvements in the continuing development of this concept have to start with the overall architecture.

Actuator systems for hybridized double clutch systems

The simplest way to hybridize double clutch transmissions is in the P2.5 structure in which a traction motor acts on one of the transmission input shafts. Here, the actuator system is comparable to that of conventional transmissions. If the customer requests additional gears, the Active Interlock gear actuator can easily be equipped with additional gear gates. In many transmissions, these are also used to actuate the parking lock. This is a positive contribution to the actuator system's total cost balance. Figure 10 shows the efficiency gains in relation to the system costs for various actuator systems. It can be seen here that a power-on-demand actuator system for P2.5 double clutch systems can be realized without significant cost increases.



10 **Efficiency gains for various actuator systems in relation to system costs for P2.5 and P2 double clutch transmissions**

P2 hybrid structure concepts usually require a third clutch so that another clutch actuator including electronic components must be added. In this case, hydraulic solutions are a better choice because they are easier to expand by adding valves and scaling the pump-accumulator module. However, turning precisely these hybridized double clutch transmissions into a milestone of efficiency requires power-on-demand actuators for such P2 double clutch transmissions with three clutches. One approach is described below using the example of an actuator system based on electric pump actuators (EPA).

Electric pump actuator

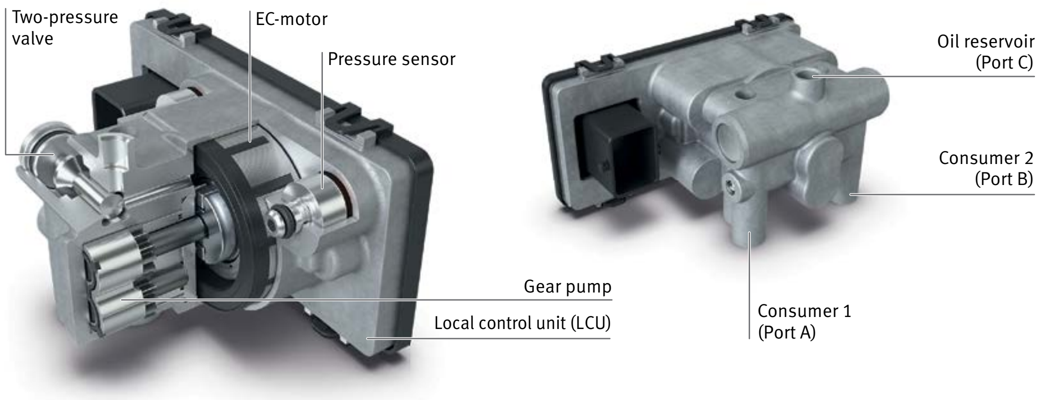
System description

The electric pump actuator (EPA) has been developed by Schaeffler as a so-called multiple consumer actuator which can be used to supply and modulate two consumers sequentially with the help of a special valve. In order to meet the high requirements for real power-on-demand systems, the EPA combines robust pump technology with the expertise gained from the development of the actuators with a local electronic unit (HCA and MCA) described above. Figure 11 shows the general design of the actuator.

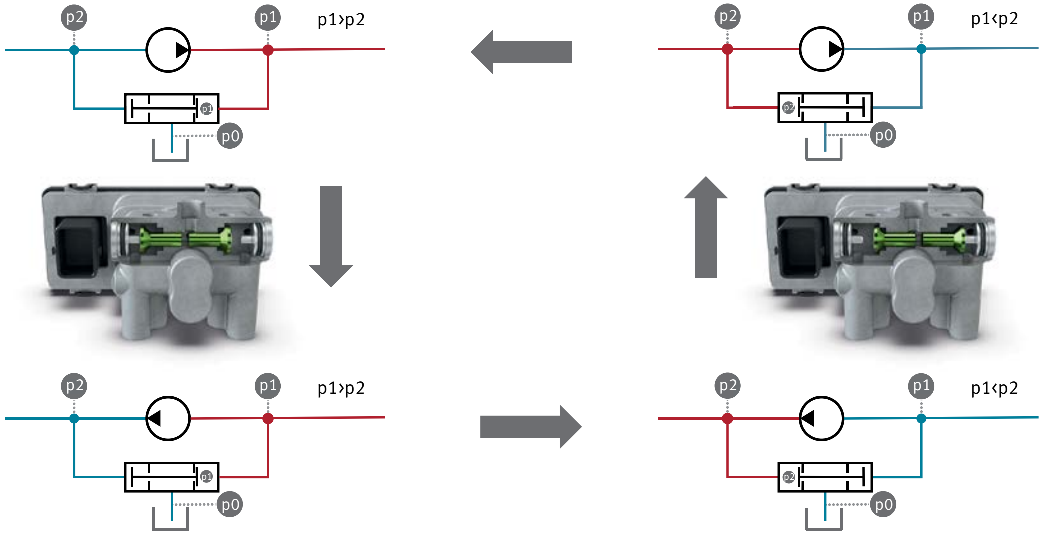
The use of a gap-compensated, low-leakage pump in combination with a high-resolution rotor position sensor allows a very precise volume adjustment. This replaces an additional travel measurement, preferably In the low-load range at low pressure - such as from the open clutch to the touch point. Integrated pressure sensors on both load outputs of the pump enable a wide range of control tasks.

The EPA can be operated in energy-saving mode as well as in dynamic mode. Under real driving conditions, the energy saving mode enables the achievement of electrical power consumption similar to the HCA and MCA. Intermittent EPA operation is achievable. Well adjusted to the clamping strategy of the clutch and the engagement system, this significantly reduces average power consumption compared to continuous operation. Dynamic mode, on the other hand, permits slip control strategies and even anti-judder control on the part of the EPA. The time for these should be limited, however, for energy-saving reasons.

The passive, so-called two-pressure valve in the EPA allows the sequential buildup of pressure on both working ports. Figure 12 shows the operating principle of this valve. Regardless of the EPA's conveying direction, the two-pressure valve connects the lower pressure with the reservoir. This means that the EPA can be applied to a consumer, such as a clutch, in a forward direction and also modulate it by turning it back and forth. In reverse operation, the clutch pressure can be reduced completely. If the EPA continues to run in reverse operation, pressure builds up in the second working port. The passive switching of the two-pressure valve prevents pressure buildup in channel 1 by shifting the pressure to the reservoir. At the same time, the connection between channel 2 and the reservoir is closed to allow pressure to build up as well as to minimize leakage in the active working channel 2. Two independent consumers can thus be operated sequentially by using an EPA with a two-pressure valve.



11 **Electric pump actuator (EPA)**



12 **Operating principle of the two-pressure valve in the EPA**

Controlling more than two consumers with the EPA requires additional valves that switch the relevant working port to the consumer to be controlled. To utilize this potential, the local control unit (LCU) of the actuator has been configured to include additional valve output stages and sensor inputs, besides the expanded processing power. This means that an EPA can be used for complex actuation systems in transmissions without having to multiply the number of actuators.

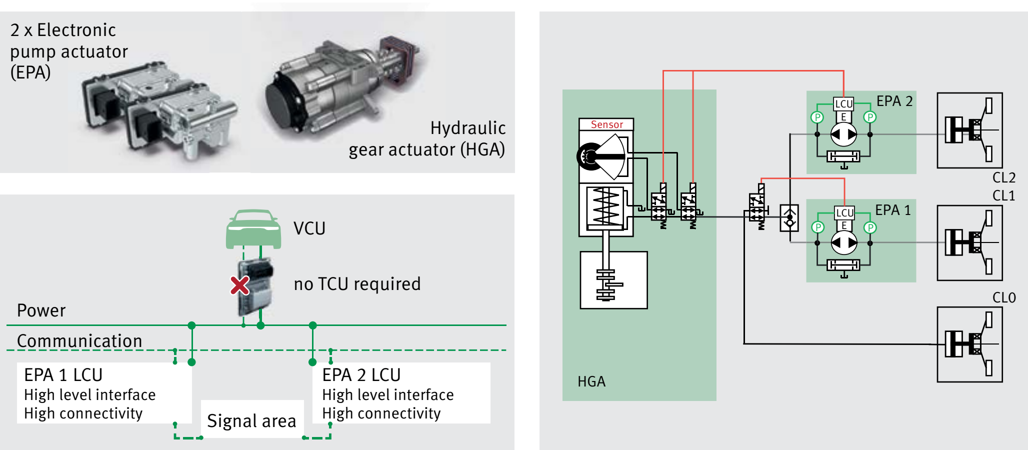
The EPA in double clutch transmissions

An efficient actuator architecture in double clutch transmissions requires two EPA units for independent clutch actuation, Figure 13. However, no additional actuator is required for actuating the transmission gears because the free EPA can carry out the gear shift and parking lock functions in its reverse direction. Typically, so-called gear slave cylinders are used here. The EPA's LCU structure makes it also suitable for four or five gear slave cylinders with the relevant number of switching valves because it not only has sufficient processing power but also three valve output stages each. Given this large number of shifting elements, the use of the well-known active interlock principle is a good idea to reduce the number of axes to two – one each for selecting and for shifting to a gear. This hydraulically driven unit including its sensors is also called a hydraulic gear actuator (HGA).

Even double clutch transmissions in P2 arrangement can be operated with just two EPA and an additional, external valve to control the separating clutch (K0). This only requires a control strategy adjusted at the overall system level to ensure comfortable and dynamic driving.

On closer inspection, the electrical system architecture varies considerably. All that is left of the active electric motors of the gear actuator is two simple switching valves that can be controlled with simple valve output stages. This makes it possible to question the entire transmission control unit and to replace it with the two EPA. As a consequence, expensive components can be eliminated, however, the intelligence of the transmission control unit must be mapped by the EPA. This requires a system and software structure that is oriented towards the independent actuation of both sub-transmissions. Only the shifting coordinator with the gear selections and the coordination of the overlapping gearshifts must be doubled, distributed or transferred to a superordinate control unit for the powertrain which is usually available in hybrid drives. The elimination of the transmission control unit makes this architecture the most cost-efficient variant.

The control of a third clutch can be easily simulated with another valve and, if required, another pressure sensor, similar to a hydraulic control unit. With this approach, the most important question is whether the control of an additional (third) separating clutch towards the engine can really be independent of the two clutches in the double clutch transmission because only two EPA are available in all. In an analysis of various vehicles with triple clutches in P2 hybrid double clutch architectures, Schaeffler found that most situations in which the separating clutch is actuated in parallel with a DCT clutch can be avoided with suitable replacement and pre-selection strategies. There are only few situations and shifting sequences in which the EPA actuator system described here has minor time limitations in the sequence of the shifting operations. Here, several circumstances have to occur at the same time: the “wrong” gear pre-selection, the driver’s change of mind and insufficient power supply from the electrical traction drive that requires the quick start of the internal combustion engine and thus the closing of the separating clutch. To cover this unlikely case, the EPA system and the software used must be adjusted, ideally in an early stage of simulation.



13 **Actuator system for double clutch transmissions with two EPA and one HGA A transmission control unit (TCU) is not required because the EPA performs this function.**

	2x EPA + HGA	Powerpack	Full hydraulics
Efficiency	+	O	O
Cost	O	–	+
Limp home capability	+	–	–
Package flexibility	+	+	O
Physical robustness	+	O	+
Controllability	+	O	O
Parallel actuation dynamics	O	+	+

O = sufficient

14 **Analysis of various actuator systems for P2 hybrid double clutch transmissions**

Comparison of the EPA system with power packs and hydraulic solutions

Compared to other potential actuator concepts for P2 double clutch transmissions, the EPA system described above has distinct advantages in terms of power consumption, in particular in the WLTC cycle and under real driving conditions, Figure 14. If this approach is implemented consistently by using the hydraulic gear actuator (HGA) and eliminating the transmission control unit, this system also offers a cost benefit. In addition, with two control units as well as distributed and redundant functions, it offers improved management of malfunctions and emergency running characteristics. In all other respects, the respective systems meet the requirements to varying degrees. Depending on the boundary conditions and weighting of these aspects, advantages and disadvantages may shift somewhat.

The EPA in other transmission applications

An EPA-HGA system can also be adjusted for other transmission applications. A hybrid system, for instance, could be based on an automated transmission and designed with just one EPA. Generally, conventional transmissions can also be actuated with several EPA, particularly because not all shifting ever have to be actuated at the same time but, similar to a double clutch transmission, usually only two to three. In rotating clutch pistons, an EPA can cover the leaks on the rotary connections. In hybridized automatic transmissions, the torque converter is often eliminated; with fewer gears, the automatic transmission can represent a dedicated hybrid transmission. Even in CVTs, it is possible to use for the clamping and adjustment control system of the pulley set. Besides, lubrication and cooling functions must be fulfilled in all transmission types, which can generally also be supported by an EPA.

Summary

When trying to further reduce consumption and CO_2 emissions, every single watt saved in the power consumption of actuators matters. Low-consumption actuators relieve electrical powertrains and increase the driving range. Real power-on-demand actuators that electrically only absorb an average of 10 to 20 watts are already feasible. Based on today’s hydraulic actuator solutions, the future potential for further reducing consumption is considerable, particularly when using the WLTC cycle and real driving conditions (RDE) as a benchmark.

This paper has shown how each watt absorbed by the actuator system can be consistently called into question and optimized. Actuators for a wide range of applications in conventional and hybridized transmissions have been discussed and presented. In addition to efficiency and cost reductions, specific actuator intelligence (LCU) can also help simulate other system requirements such as functional reliability and emergency running characteristics.

The various actuators can be integrated as a modular system or used as add-on systems. This allows the use of the actuator components in various transmission types and in particular in new hybrid transmissions (DHT) and electrically-driven vehicles.

In the development of new actuator concepts, Schaeffler can adjust the individual components in an ideal way because clutches, release systems, actuators and transmission software all come from a single source. Schaeffler's close cooperation with transmission manufacturers makes it possible to produce particularly efficient, compact and powerful actuation systems.

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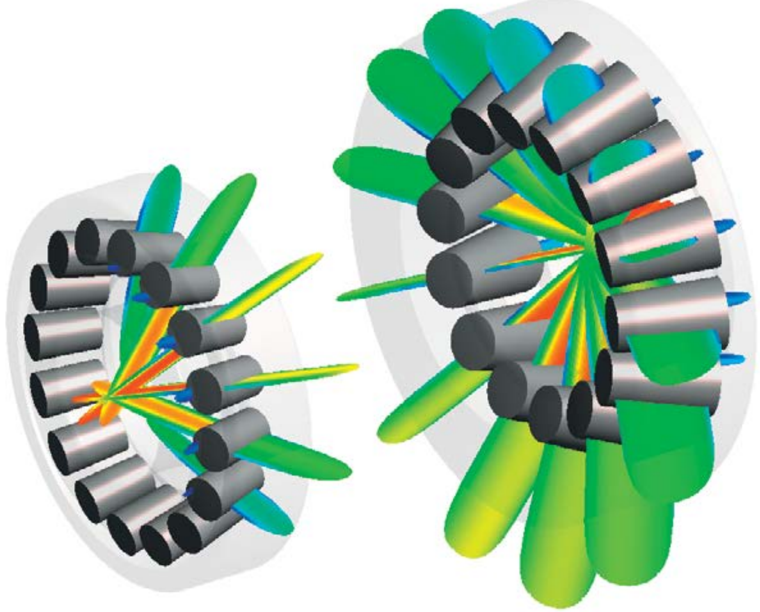
15

Innovative CAE

Optimal Layout of Transmission Components

Dr. Daniel Heinrich
Johannes Kerstiens
Michael Schneider



Introduction

In the past 25 years, software tools for computer-assisted calculation and simulation have been essential for improving the efficiency and performance of powertrains. Schaeffler has utilized and advanced this development extensively with the aim of offering its customers ideally matched products while also accelerating development processes. The essential aspect here was the development of in-house simulation tools that are different from the options available on the market because of their excellent predictive quality and their ability to speed up calculations.

Development of these Schaeffler CAE tools continues in order to meet the increasing requirements of state-of-the-art vehicles and development processes. This paper discusses four innovations as an example:

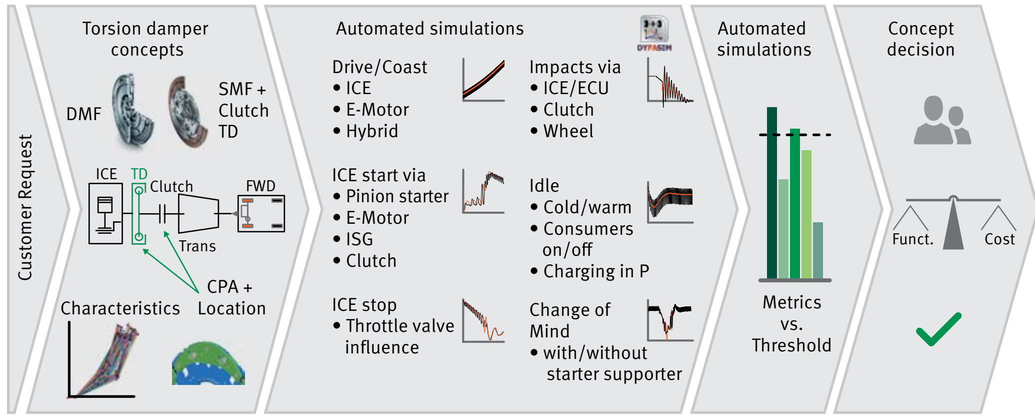
- The optimization of torsional vibration dampers for hybrid powertrains through a variety of operating points and system topologies
- The automated optimization of rolling bearings in BEARINX
- A new method for designing clutches with consideration given to mechanical and thermal interactions
- The use of virtual testing for faster product development while improving the quality of statistical information.

Simulation of powertrain vibrations for modern hybrid topologies

Schaeffler has developed the DYFASIM multibody simulation program for the calculation of vibrations in the powertrain. This efficient software includes libraries for the highly dynamic simulation of internal combustion engines, electric drives, engine control systems, clutches, torsional vibration dampers, torque converters and other transmission components and has been developed with the aim of customizing Schaeffler's products to customers' specific requirements, thus reducing the required development time, Figure 1.

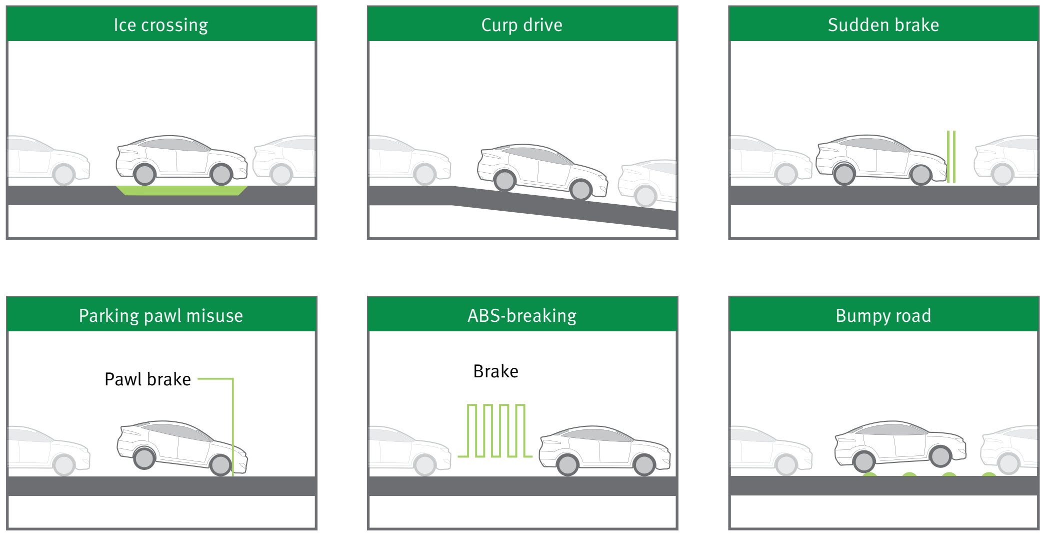
One example of the high level of integration in the development process is the optimization of modern torsional vibration dampers. For these, the whole system is simulated, including the engine

1 **Illustrations in DYFASIM with various applications**



2 **Optimization process for the design of modern torsional vibration dampers**

and bearings, starter, battery and connecting cables, transmission, drive shafts as well as tires and other components, and a wide range of design options are taken into consideration [1]. The standard process for designing a DMF involves the generation of around 2,000 possible variants of characteristic curves from which the best design for the relevant customer application is selected in a multi-stage optimization process, Figure 2. In order to enable this amount of work for every single customer inquiry and make the variant optimization described above available over-



3 **Maneuvers that must be taken into consideration in damper design**

night, Schaeffler uses its own server architecture that calculates 4.4 million simulation jobs per year. It is this process that allows the targeted design of centrifugal pendulum-type absorbers in torsional vibration dampers with the optimization quality possible today [2].

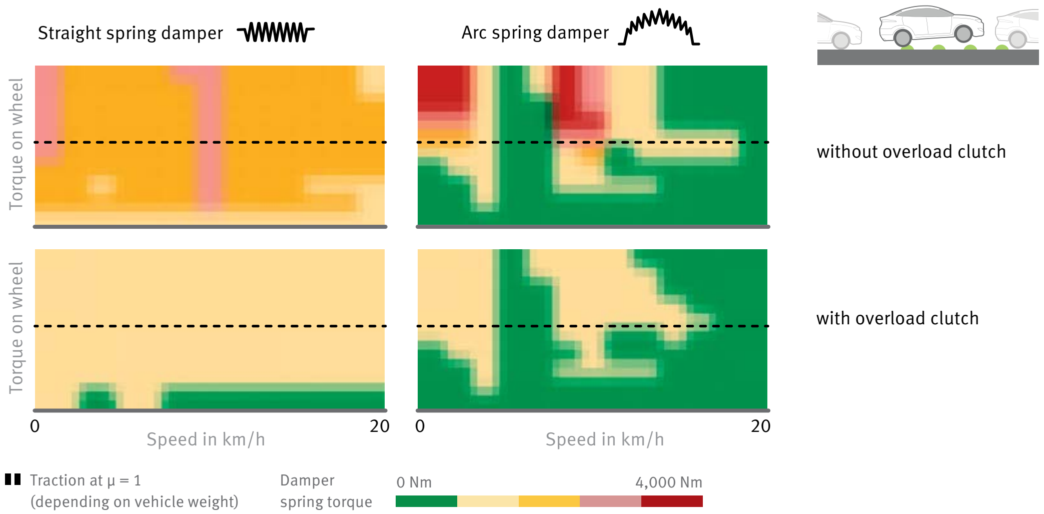
The scope of variant optimization must be expanded even further in the development of torsional vibration dampers for new hybrid powertrains. For instance, high loads on the dampers and transmission components resulting from highly dynamic wheel torques must be analyzed, Figure 3.

One example here is driving on bumpy roads. In hybrid powertrains without a separating clutch, these can lead to resonances in the powertrain due to the excitation on the wheels, causing severe damage to the transmission. The variation calculation shown in Figure 4, which considers the effect of different driving situations and damper designs, is used to perform an evaluation at an early stage. Here, for instance, a linear spring damper shows high loads over a wide range of road parameters. By comparison, an arc spring damper has better peak torques and a more robust behavior over large areas. In both damper types, an additional torque limiter clutch reduces the load throughout.

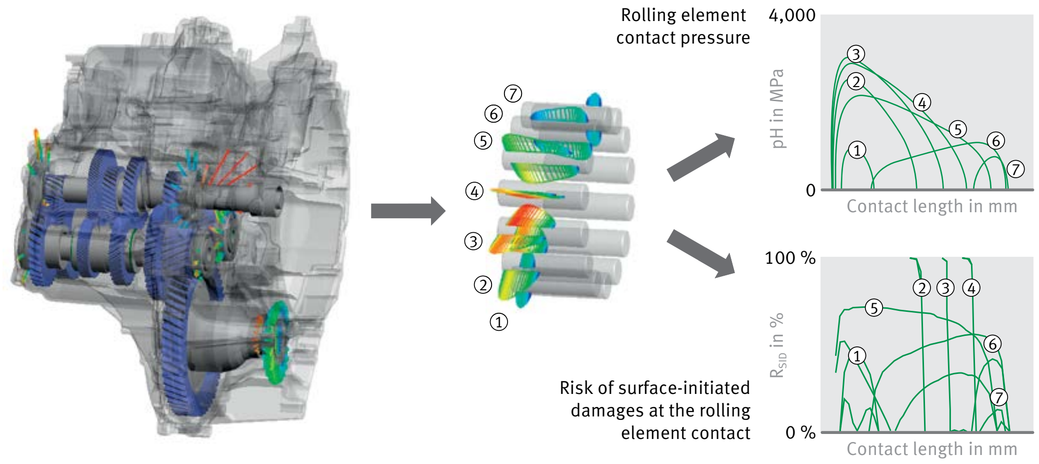
Since powertrain simulations often have to be repeated, this process has been largely automated, enabling early damper adjustments. This significantly reduces product adaptation work later on. At the same time, the early optimization of the damper and overall system can help achieve a greater total power density and thus an improved function.

Rolling bearing optimization with OptiKit

Since 1995, Schaeffler has used its Bearinx simulation software to configure, design and analyze



4 **Simulation results for bumpy roads with driving range variations for four different damper systems**



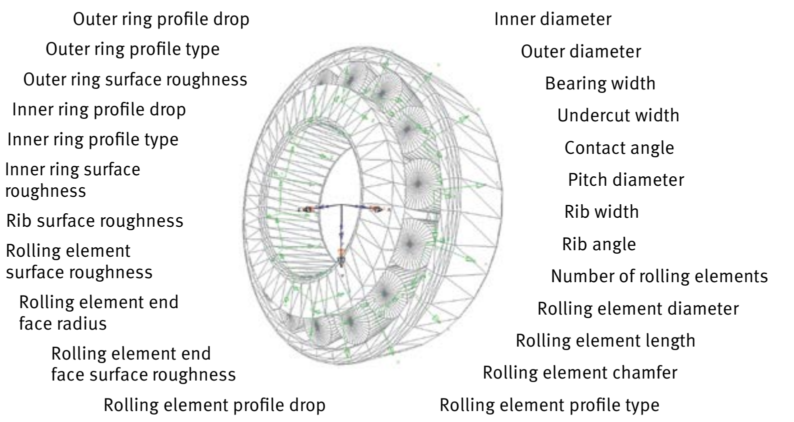
5 **Modeling depth of Bearinx from the system level of the complete transmission to an individual contact**

bearing supports in complex transmission applications. This powerful tool is expanded and improved continuously based on the latest research and is also available as a customer version. Bearinx does not only look at the bearing in isolation; it is also able to take into consideration all loads, displacements, elastic structures and gear teeth contacts that interact with the bearings in the overall system.

This modeling depth enables Bearinx to predict the behavior of both the transmission and the rolling bearings in detail, Figure 5. This includes both the overall system level, such as when forecasting power losses, and the single bearing contact level, such as when assessing the risk of surface-initiated damage to an individual rolling element [3]. This makes it possible to provide qualified advice to customers at a very early stage of development.

The high modeling depth and quality enable improved system analyses, resulting in shorter development times. However, the increase in power density and limit loads as well as the minimization of the design space and power loss require the perfect and detailed optimization of every single bearing position. Figure 6 shows the great variety of potential control variables using a tapered roller bearing as an example. These variables, whose effect is closely interrelated with other transmission components and parameters, can no longer be optimized by means of intuitive processes or manual variant calculations.

That is why Schaeffler has developed optimization algorithms that work automatically and that optimize a wide range of parameters at the bearing and system level to obtain specified targets. Generally speaking, these approaches are not new, but in the past they did not supply any satisfactory results for the goal described here. In practice, there are indeed conflicting goals due to various functional and technological requirements. An intelligent optimization algorithm is thus characterized by the fact that it can map all



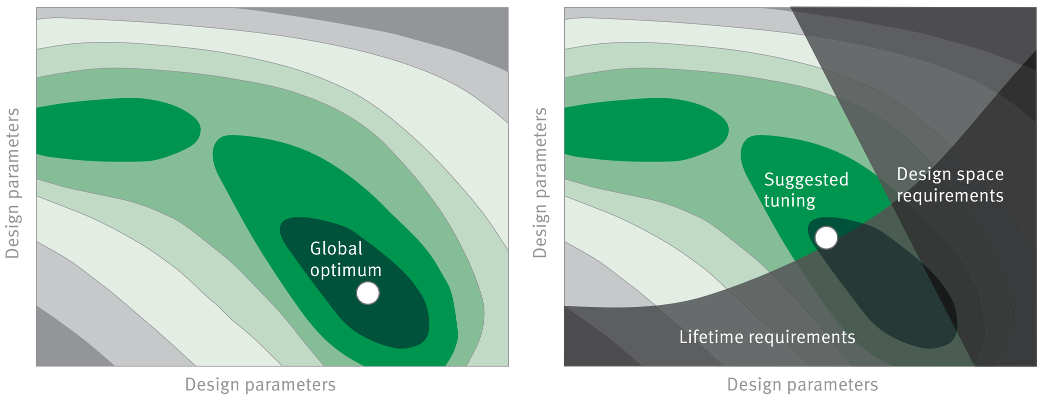
6 Variety of parameters and influencing factors in the design and optimization of transmission bearings using a tapered roller bearing as an example

customer requirements such as the design space and performance as well as take into consideration all design and manufacturing restrictions, Figure 7.

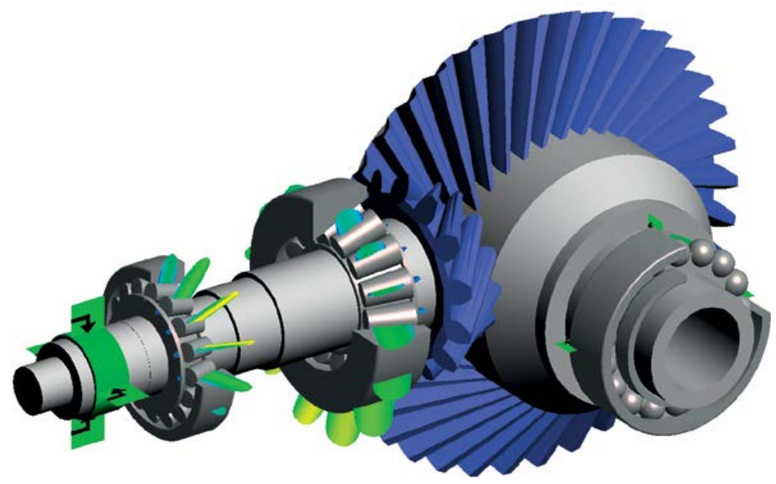
For this purpose, Schaeffler has developed the OptiKit optimization tool and integrated it in Bearinx. With OptiKit, users can integrate development goals and restrictions of their choice in the optimization and thus quickly find a system adjustment that meets the optimum design and spectrum requirements.

The use of OptiKit can be demonstrated using the example of a friction-optimized tapered roller bearings on the pinion flange and pinion head of a rear axle transmission, Figure 8. In the end, friction was reduced again by more than 20 % compared to the X-Life-T29D designs which were already low friction. As criteria that are critical to functioning such as internal load distribution and contact pressure curves are included, the required service life is ensured.

In their initial condition, the two bearings analyzed have a friction loss of nearly 12 Wh in a reference cycle for the design. Here the optimization goal is to reduce bearing friction without having to touch other boundary conditions such as manufacturability, design space and service life requirements.



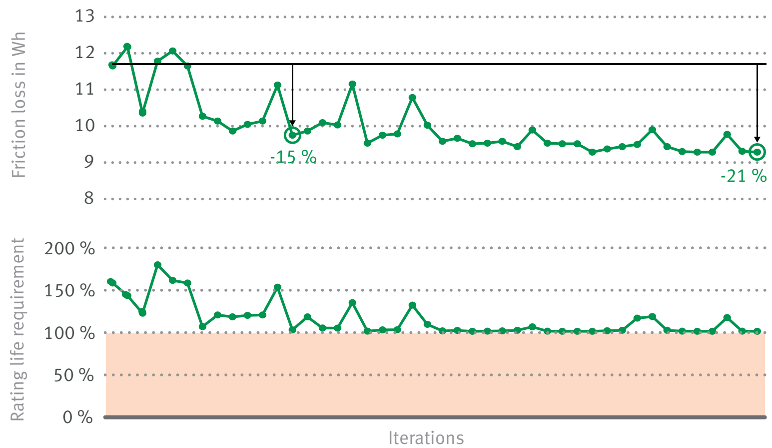
7 Comparison of the function of a classic optimization algorithm (left) and a real-life optimization based on development requirements (right)



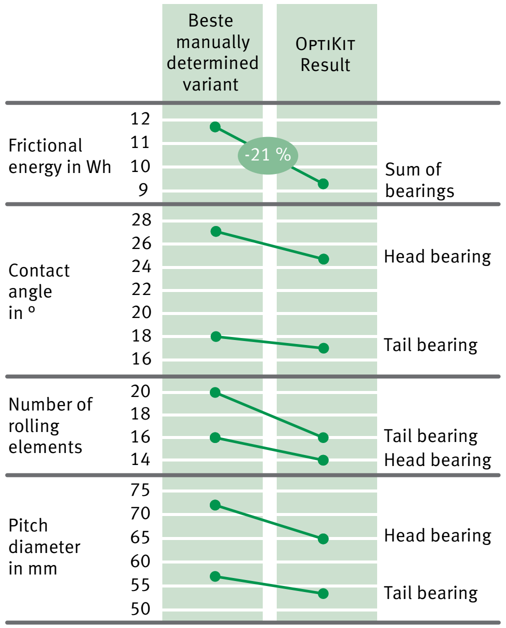
8 Model of the rear axle drive to be optimized in Bearinx

Step by step, the OptiKit algorithm now approaches other parameter combinations that allow the reduction of friction losses, which at the same time causes the service life of the bearing to decrease. It is only through an additional boundary condition that the service life requirement can be maintained during optimization. Figure 9 shows the development of the system friction and the service life evaluation over the course of the optimization run.

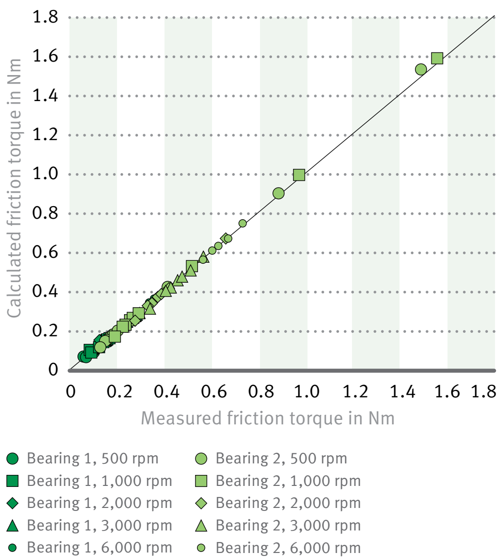
The optimization progression shows two areas that impressively demonstrate the new approach. The system is consistently tuned towards permissible design limits, which results in a 15% friction reduction compared to manual bearing design. Next, the system is optimized further by adjusting the bearing precisely to the application in question. At the end of the optimization, there is a bearing that was optimized with the specific characteristics of the adjacent system and the customer's development goals in mind. The parameters that play a role here are shown in Figure 10 for the pinion head bearing and the pinion flange bearing.



9 Curve of the determined friction energy and service life evaluation based on optimization Both values are calculated using different design cycles that represent the respective target requirements



10 **CAE-assisted optimization of tapered roller bearing parameters in the rear axle final drive analyzed**



11 **Comparison of calculated and measured friction torques for optimized on the rear axle final drive**

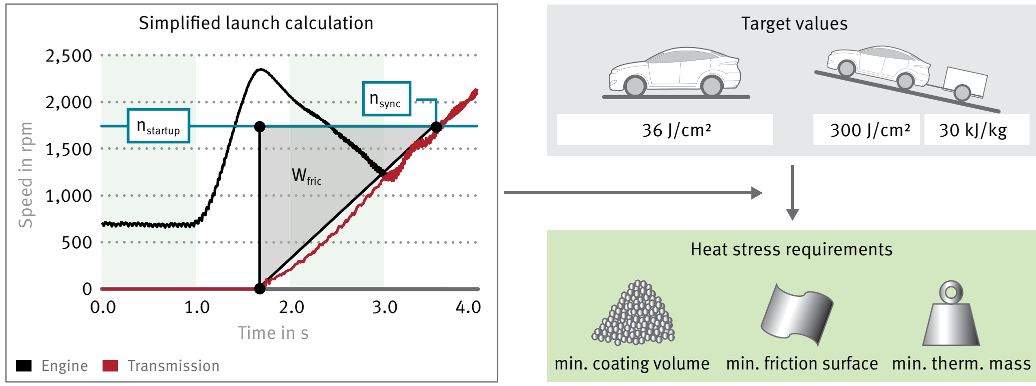
In total, the optimized bearings on the rear axle final drive show a 21 % reduction in friction energy that was achieved without having to adapt the design space or use special materials. In order to validate this calculated value, the optimized bearings were measured on the test stand, Figure 11. Different load points were approached for each speed. The final result shows that the optimized bearings on the test stand confirm the predicted friction behavior.

This results in a better and faster design that is made possible by combining the high modeling depth and predictive quality of Bearinx with the targeted parameter optimization through OptiKit. This potential can be utilized for all applications of Bearinx [4].

Thermal-mechanical clutch design

Previously, clutches were designed using recognized state-of-the-art methods, Figure 12. With regard to thermal dimensioning, the design was broken down to values for a specific work load. These values provide the minimum requirements with regard to thermal mass, lining volume and friction surface and have been very useful in the past [5]. FEM calculations and life time profiles did not play a role until later on in the optimization process.

In the development of modern clutch systems such as triple clutches [6], however, this classic approach is no longer sufficient because an enormous power density must be achieved in a very tight design space. Even for single clutches, the optimization potential must be analyzed in terms of function and costs.

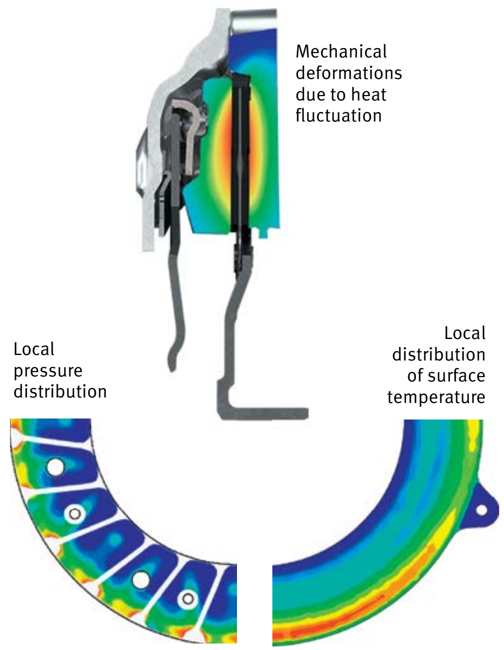


12 **Clutch design based on simplified calculation values**

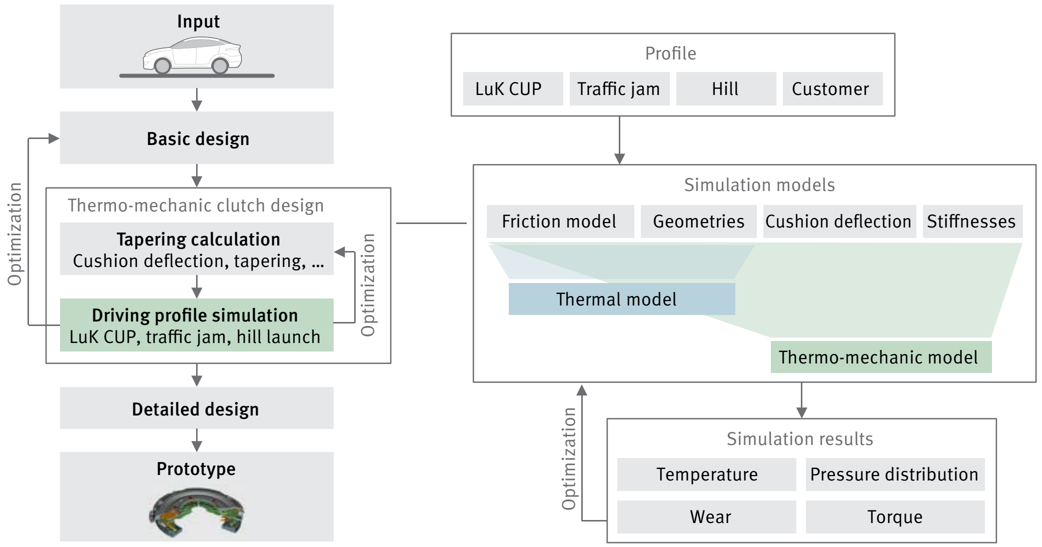
This interrelationship becomes evident when considering uphill starts – at first glance, this is a simple scenario, but when looking at it more closely, it can be seen that this involves a great deal of complexity. The friction power applied in the synchronization phase initially leads to a significant temperature increase on the friction surfaces. Resulting from the local, microscopic pressure distribution, the sliding speed varying across the radius and the prevailing surface temperature, the sum of all friction elements is the transmissible torque at a macroscopic level. The system is not stationary at any time and is influenced by the heat flow in the components generated by the power supply and the resulting mechanical deformations of the friction surfaces Figure 13.

The simulation of these effects requires enormous calculation work because the waviness occurring in a circumferential direction must be considered in addition to the radially occurring conicities of the friction surface. Another important factor is the interaction of the components such as the segmented design of the cushion deflection that provides a moving support and is able to compensate a certain amount of radial conicities and tangential waviness due to its axial flexibility.

Even with intelligent networking, calculation model distribution and sequential distribution of calculation processes, the required calculation time was only reduced to five weeks for the calculation of five consecutive starts. Based on this, a



13 **Interaction of pressure distribution with thermal and mechanical effects in the clutch**



14 **Schematic diagram of the optimized thermal clutch design**

variant calculation is not really an option. This makes this method primarily suitable as an analyzing tool and not for the normal design process that requires the calculation times to be reduced by a factor of around 1,000. At the same time, a flexible structure is necessary so that this method can be used for both single clutches and more complex systems such as double clutches and triple clutches.

Being aware of the hidden potential, an interdisciplinary team systematically analyzed the required parameter influences and their interactions and revised the calculation models to ensure that the required calculation time reduction was reached without corrupting the results. The new thermo-mechanical ATM simulation tool is based on an optimized finite element model and takes the following into consideration:

- Real-life conditions for uphill launches based on typical real-world accelerations
- Geometries and rigidities of individual components, in particular the tilting rigidities of the pressure plate and the flywheel with their radial conicities and tangential waviness as well as the pressure distribution of the cushion deflection
- Thermal and elastic material characteristics of metallic and non-metallic materials such as those of the lining
- Heat transfers in the clutch, in the clutch housing and into the environment
- A complex friction coefficient model that for the first time considers the history of thermal loads in addition to mapping the interaction of the temperature, pressure and sliding velocity
- Implicit temporal changes in thermal-mechanical load conditions.

A thermal-mechanical model is used for uphill starts and a purely thermal model for quasi-stationary starts on flat surfaces to calculate life time profiles, Figure 14. In cooperation with the customer, the implementation of customer profiles is possible, enabling the direct consideration of test cycles and a comparison of calculated and measured values.

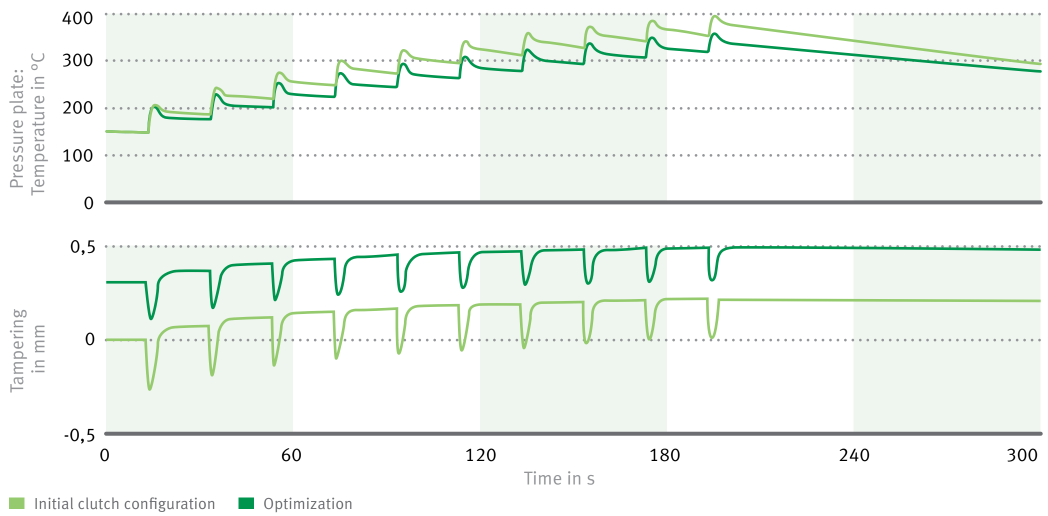
This makes it possible to discuss results with the customer as early as the design phase and define the optimization goals such as cost reduction, low lining wear and higher transmissible engine torque. The effects of different lining materials can also be predicted for various load scenarios.

Previously untapped potential can be utilized via the thermal-mechanical combination. Clutches can be optimized to achieve the target parameter of a preferably homogeneous pressure distribution of the friction surfaces and, as a result, not only show improved dynamic conicity behavior but also a maximum temperature that is about 40 °C lower as a result of surface pressure homogenization, Figure 15.

The reduced peak temperatures lead to an increase in torque capacity which can be used to reduced actuation forces and increase the efficiency of the system. At the same time, the improved conicity behavior and the lower temperatures in high-load areas result in reduced clutch wear. The achieved potential can now be used, for instance, to

- Integrate a centrifugal pendulum-type absorber by providing the required design space on the clutch disk, significantly improving the isolation effect in the powertrain
- Reduce the clutch size overall, thus using less weight and design space

Overall, this makes it possible to achieve higher power density as well as greater design reliability. The installation of additional protection mechanisms is recommended. This means that the current surface temperature of the friction surfaces



15 **Excerpt from uphill start simulation with the thermal-mechanical model**

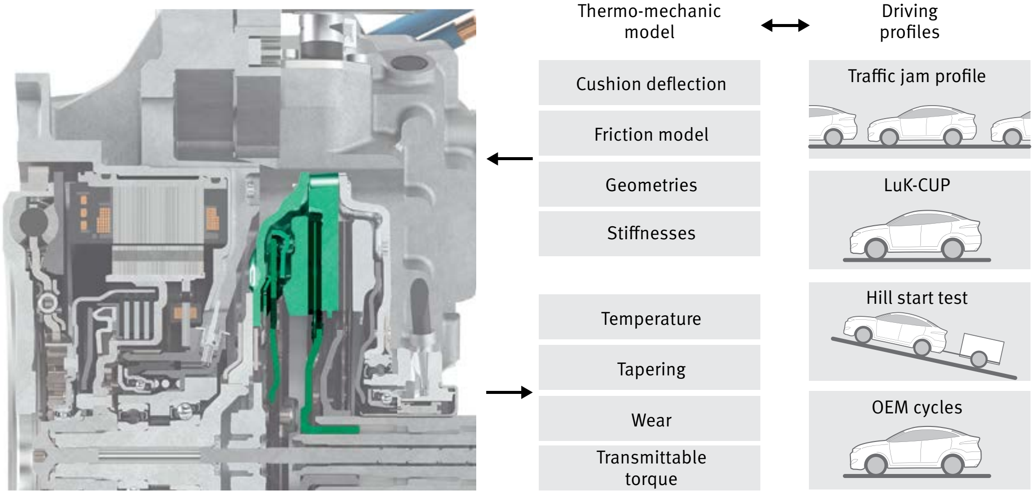
is monitored during operation by means of a small calculation program and the driver is provided with feedback as soon as critical values are reached. One example here is a thermal model already used in the engine control system that, given current developments, will also be available for conventional manual transmission applications. The design and software protection procedure described here can be easily applied to the development of triple clutch systems, Figure 16, [6], because active intervention in the clutch system is made possible while driving. Compared to manual transmissions, this results in even greater functional reliability for such systems.

With the successful introduction of the thermal-mechanical ATM simulation tool and its integration in the existing CLUSYS design program, the thermal optimization of the overall system has become the new standard. This means that the optimization of the mechanical force-travel design and the thermal-mechanical design are systematically merged in the early design phase.

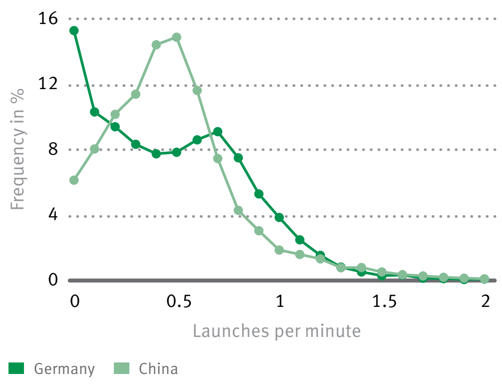
Virtual testing

Durability testing of vehicle components is tedious, time-consuming and expensive for customers. That is why durability tests frequently cannot be validated statistically. Since essential vehicle components often have not been finalized at the time of the durability test or have to be re-engineered in the event of failure, there is a remaining risk of unpleasant complaints occurring in the field even if the durability test was successful. Ideally, durability testing would be largely replaced by component tests and CAE calculations. To do this, loads under real driving conditions and their statistical variation must be known during the design of the components, such as those of the clutch system. In particular, driver variation and the associated influence of varying driving styles in different countries are cause for discussion again and again, Figure 17.

In order to ensure targeted design and testing of components under realistic operating conditions,



16 **Development and safeguarding of complex clutch systems by including thermal-mechanical effects**



17 **Distribution of start frequencies in the analysis of real driving data from Germany and China**

Schaeffler has carried out its own measurement programs in real operation for a long time now. These driving profiles such as the Schaeffler Driving Cycle and the LuK CUP (Customer Usage Profile) have been essential in the testing of product innovations such as new double clutch systems [7].

To reach the goals described above, this approach requires further development. This can be achieved by increasing and combining activities in all areas affected: Measured data, simulation models, statistical models and component tests.

Measured data on real operation are consolidated using a new database structure to permit the analysis of driver behavior and load spectra at a later time. This database includes special measurements such as the Schaeffler Driving Cycle and LuK-Cup as well as comprehensive long-term observations with data loggers for normal drivers, taxi and driving school applications.

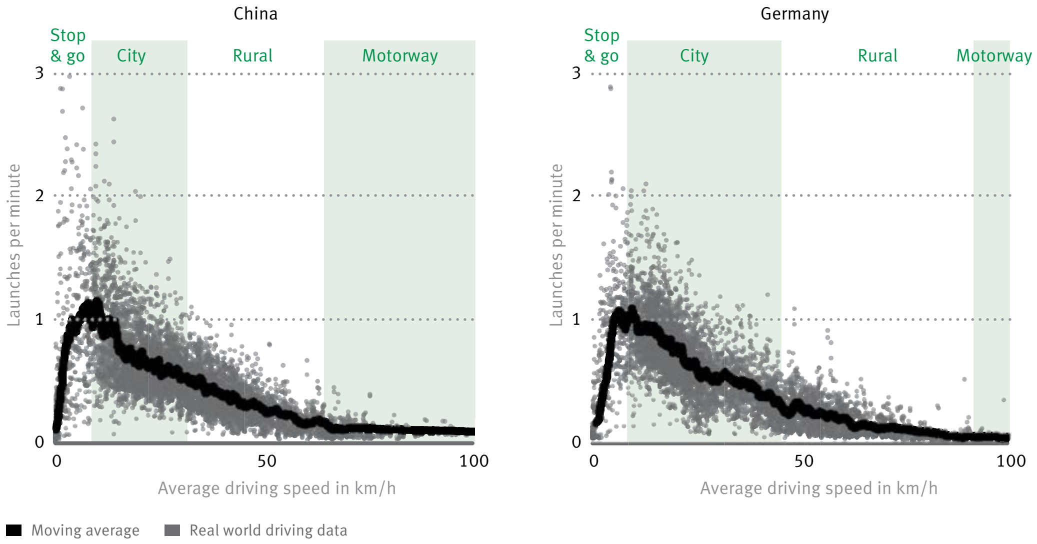
Simulation models allow a detailed data evaluation by re-simulating powertrain vibrations, friction power and temperatures that are not available as measurement channels at all times. In addition, the existing simulation chain has been expanded to include more tools that permit various simulation levels to be linked, even including traffic flow simulation.

Statistical models are used systematically to permit the conversion between test cycles, frequency distributions real-world driving and simulation and test-rig data. The available tool chain permits the inclusion of statistical data for various countries as well as the quantification of soft factors such as driving behavior.

Component tests in terms of wear or durability tests of individual components are linked to the existing tool chain. Unlike complete vehicle tests, these can be performed much faster and up to failure.

The consolidation of data and simulation methods are shown here using regional effects on design-related driving data as an example. This new approach allows large amounts of driving data to be analyzed systematically for relevant influencing factors, taking the regional dependency described above into consideration through suitable compensation factors. In this example, this was achieved by a driving data breakdown in which the start frequency of the average speed in the segment is compared for each driving segment. This results in comparable parameters for both regions shown and the difference observed previously is eliminated, Figure 18.

Traffic density plays a dominant role for analyses based on average speed, but it is not included as a measured variable in the available measured data. That is why a complementary analysis based on traffic flow simulations is required. For this purpose, a traffic flow model was set up that permits the simultaneous simulation of thousands of vehicles in a complex roadway system and thus the simulation of real-life driving situations including their interactions, Figure 19.



18 **Analysis of real driving data on the distribution of the start frequency of individual driving ranges based on average speed**

The traffic flow model maps the behavior observed in the measured data with adequate precision. This also permits the systematic variation of parameters such as traffic density, road type and driving style, Figure 20. Drivers only have very little influence if traffic density is high, but there is a large variation of event numbers due to external influences.

The combination of measured and simulated data allows the connection to traffic statistics of various countries. Available data is linked systemati-

19 **Traffic simulation model configured for city driving**

cally to data collected by Schaeffler, permitting a conversion to the relevant utilization areas, Figure 21. The conversion to individual vehicles is performed using Monte Carlo simulations.

This results in a conversion of the base data to variations and frequency distributions in various regions. This method is now being extended to include vehicle and component parameters so that database values on the recuperation potential of manual transmissions with P0 hybrids [8] or the resulting load spectra for belt drive components [9] can be derived. In the next step, the calculated field distributions are transferred to the system and component level, depending on the relevant damage mechanisms, where they serve to design and verify the service life.

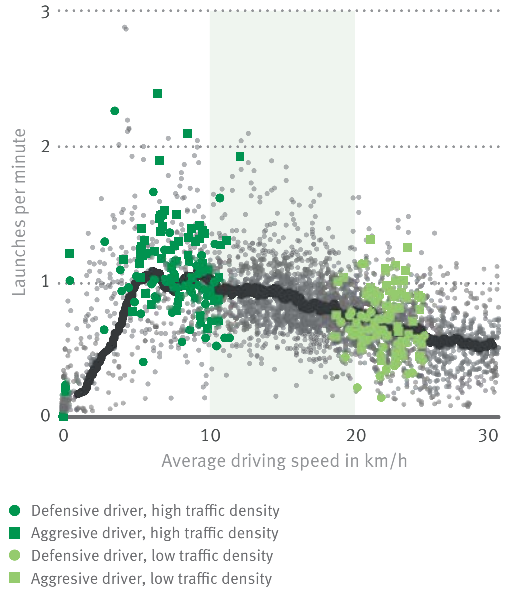
One example of this is the testing of the clutch actuation sub system (CASS), in which the synthesis of an application-specific cycle was able to improve the testing of rating life requirements compared to the previous general test cycle, Figure 22.

This helps reduce the testing requirements in the complete vehicle. The approach introduced here is used for many applications so that the new methods can be used step by step for both the development and launch of new products and the advancement and dimensioning of existing technologies.

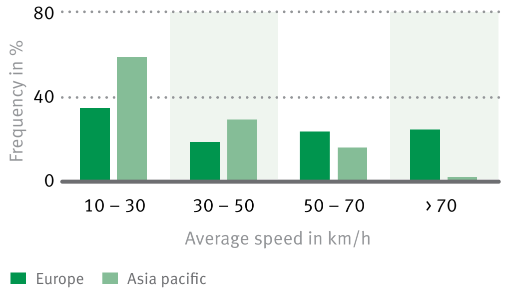
Summary and outlook

This paper has presented further developments of Schaeffler's CAE tools that allow accelerated design and development processes, improved predictive quality and higher functional and power density.

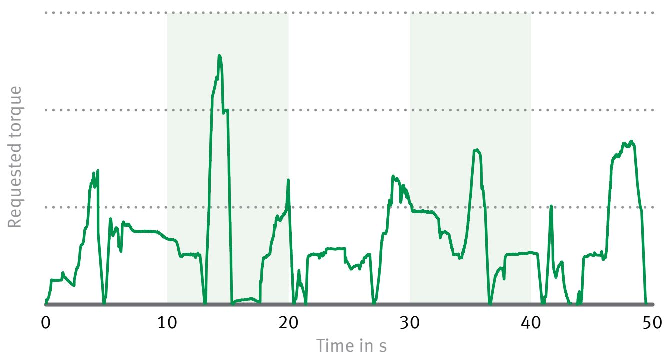
- The calculation of torsional vibration dampers in DYFASIM, expanded to include comprehensive and highly automated variation calculations for the optimum design of hybrid topologies with the aim of reducing development risks at an early stage
- The OPTIKITtool integrated in BEARINX, which allows the automated optimization of rolling bearings for the relevant transmissions and



20 **Traffic flow simulation results with dispersion based on various dependencies**



21 **Use of traffic statistics for data prostratification**



Application-specific test cycle (excerpt)

Improvement compared to previous test cycle

	Factor
Clutch actuations	1.2
Actuator travel	1.5
Hydraulic refill count	5 – 10

22 **Optimized synthetic test cycle for testing the clutch actuation system**

boundary conditions, permitting, for instance, a friction reduction of more than 20 %.

- The ATM model integrated in the CLUSYS clutch design, which permits the consideration of complex thermal and mechanical interactions in clutch design by reducing the calculation time by a factor of 1,000 compared to previous models
- The virtual testing method that allows accelerated approval tests and improved statistical safeguarding by linking driving data, CAE methods and component tests-

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Hybrid & E-Mobility

16

Schaeffler E-Mobility

With Creativity and System Competence in the Field of Endless Opportunities

Andreas Englisch
Thomas Pfund

Introduction

The field of e-mobility can look back on many years of expertise in the development and production of components for electrified drivetrains. The first concept vehicle was already built back in 2002. At the 9th Schaeffler Symposium in 2010, sophisticated solutions for mild and full hybrid drives were introduced [1]. Series production for hybrid components began in 2013. In the following years, the company focused on developing its expertise further, with the result that, at the 10th Symposium in 2014, the first complete solutions were presented, such as for electric axle and wheel-hub drives along with roadworthy prototypes [2; 3]. The technical competence gained in this manner has led to several series projects for hybrid modules and electric axle drives, with production launches between 2017 and 2019 – Figure 1. In the future, with China, Europe, and the US, all of the large markets for electrified propulsion will be served by our development and production. At the start of 2018, these activities were bundled in the new corporate unit of E-Mobility.

What Direction Is Powertrain Development Taking?

In the past few years, a large number of concepts have been developed in the automobile industry for hybrid and electric drivetrains. The driving force behind this is mainly CO_2 legislation and type authorization, which mandate compliance with limiting values for pollutant emissions. The gradual introduction of the assessment criteria relevant for this, including the new consumption cycles, started on September 1, 2017 (WLTC and RDE for new vehicle types). Beginning on September 1, 2019, it will be necessary for all new vehicles sold to demonstrate compliance with the pollutant limiting values on the road as well (Euro 6d TEMP, RDE).

1 **Series launches for electrical drives in 2017 to 2019**

The difficult task for the OEMs and the suppliers has to do with defining just the right fleet mix, while at the same time living up to customer demands for price and performance, albeit customer preferences vary greatly around the world. In addition, it is necessary to intelligently use the limited resources available for development.

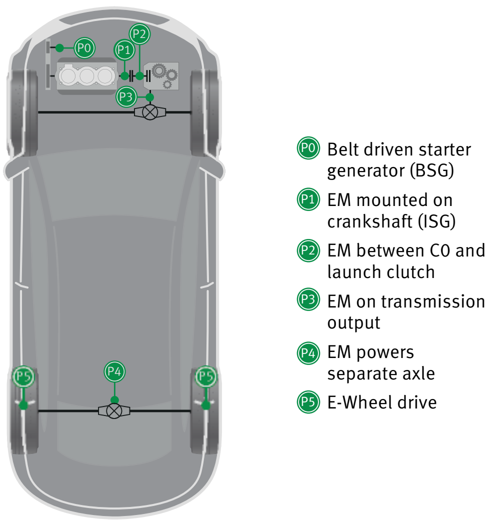
Based on current global and regional forecasts relating to technology development, the transformation of the energy chain, infrastructure, and the availability of resources, it must be assumed that there will continue to be a mix of different powertrain technologies tailor-made for certain usage profiles for the time being. The percentage of purely electric powertrains will be increasing, with this growth heavily dependent on the development of mobile energy storage systems and the associated energy chain and infrastructure. The solution space with relation to the drive unit is limited to electric axles, related near-wheel drives, and drives that are directly integrated in the wheel (wheel-hub drives). The area of digitalization and the development of autonomous driving represent another driving force in this area. New mobility concepts for urban areas are leading to new vehicle concepts where mobility is being offered as a service in specially tailored business models. Drives that enable optimum use of space and a high degree of maneuverability, such as the wheel-hub drive, are taking center stage.

Hybridization leads to the development of many more versions. These powertrains can be systematized on the basis of the architecture and topology and described with respect to the possible functions. The degree of hybridization has become established in everyday language (micro, mild, full, and plug-in hybrids, along with purely electric drives), and can be characterized based on the possible functions. Parallel, serial, and power-split architectures can be differentiated on the basis of the energy flow. Selecting which system is suitable ultimately depends on what functions are desired at the vehicle level (driving dynamics, driving performance).

Parallel architectures have also become established alongside the large number of power-split hybrid transmissions on the market (e.g. Toyota Prius). In addition, there are also structures that enable parallel or serial operation (e.g. Mitsubishi Outlander). Parallel structures are advantageous if
- the powertrain is primarily designed for recovering kinetic energy (mild hybrid).
- only one electric drive is to be used.
- the basic transmission is to be incorporated unchanged.
- a large percentage of the driving profile involves high velocities.
- a high level of functional redundancy is required (full function when the battery is low).

When electrical power is sufficient and redundancy limited, the transmission structure is not as demanding and will therefore be more cost-effective. Another advantage can be gained if it is possible to limit the operating range of the combustion engine. This can be achieved by the following operating modes:
- Stepless operation (eCVT, power split)
- Serial operation
- Limited driving range for the combustion engine (mechanical drive through of the combustion engine beginning at a limit velocity up to a top velocity)
- Purely electric operation in the lower velocity range

The best results can be obtained by combining the operating modes, which – in addition to special transmission structures – also requires dedicated designs of the combustion engine and electric drives. In these specially tailored powertrains, the operating strategy takes on a significant role in coordinating performance, consumption, drivability, and acoustic behavior.

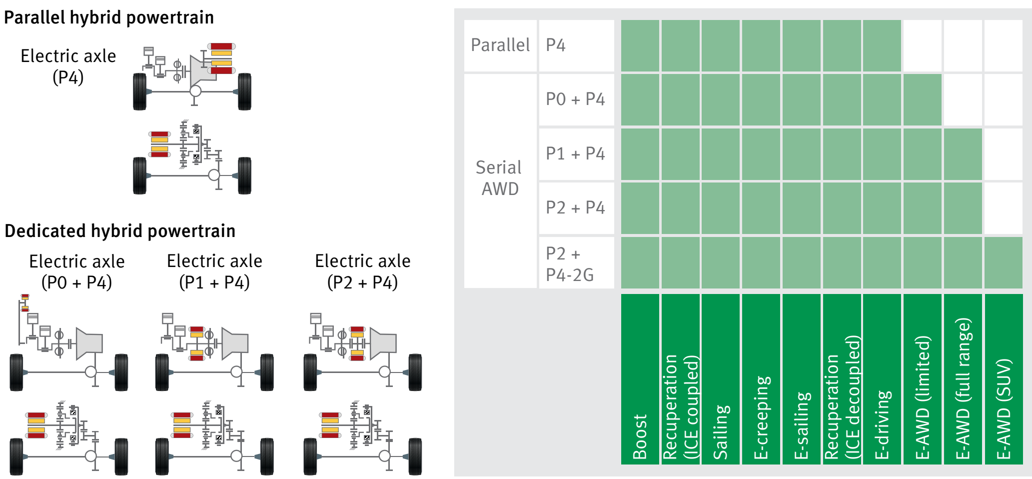


2 **Installation position of the electric drive in hybrid vehicles**

It is primarily with parallel structures that different topologies come to bear with regard to the position at which the output of the electric drive is coupled to the powertrain (P0 to P4) – Figure 2.

When other powertrain functions such as all-wheel drive are considered along with the pure vehicle driving function, then the question of architecture involves the function level as well. All hybrid functions, including electric driving, can be implemented using a P4 topology (electric axle). When combined with a combustion engine on the other axle, however, there is the possibility of all-wheel drive. If it is not acceptable for this all-wheel function to be lost when the battery is empty, then it will be necessary to combine the electric axle with a second electric drive, which in this case can work as a generator. Limited output when the battery is empty is conceivable in many applications, resulting in the possibilities shown in Figure 3.

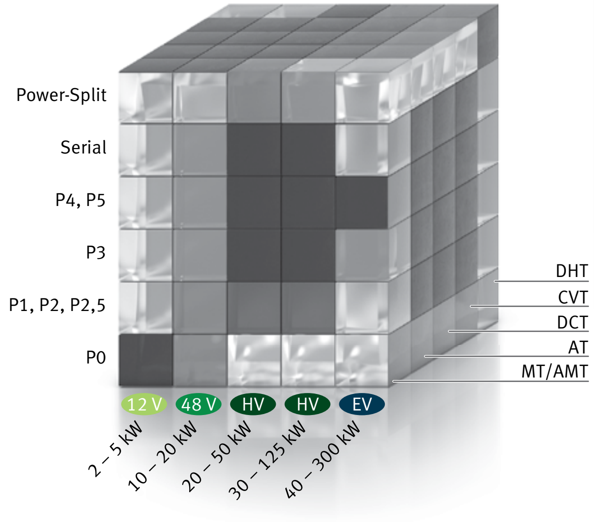
An electric axle has additional functional advantages. If a high level of drive-away torque and high final velocities are to be offered at the same time – such as in an SUV – then a shiftable two-gear axle can be used. Another option is the targeted distribution of the torque onto the wheels, thereby improving the driving dynamics considerably (“torque vectoring”).



3 **Functional scope of different P4 solutions**

	12 V	48 V	HV	HV	EV
	Micro	Mild	Full	Plug In	EV
Type	Hybrid	Hybrid	Hybrid	Hybrid	
	12 V	48 V	HV	HV	HV
P4					
P3					
P2,5					
P2					
p1					
P0					

16 possible Variants



4 **Combination of architecture, topology, and transmission options in a hybrid powertrain**

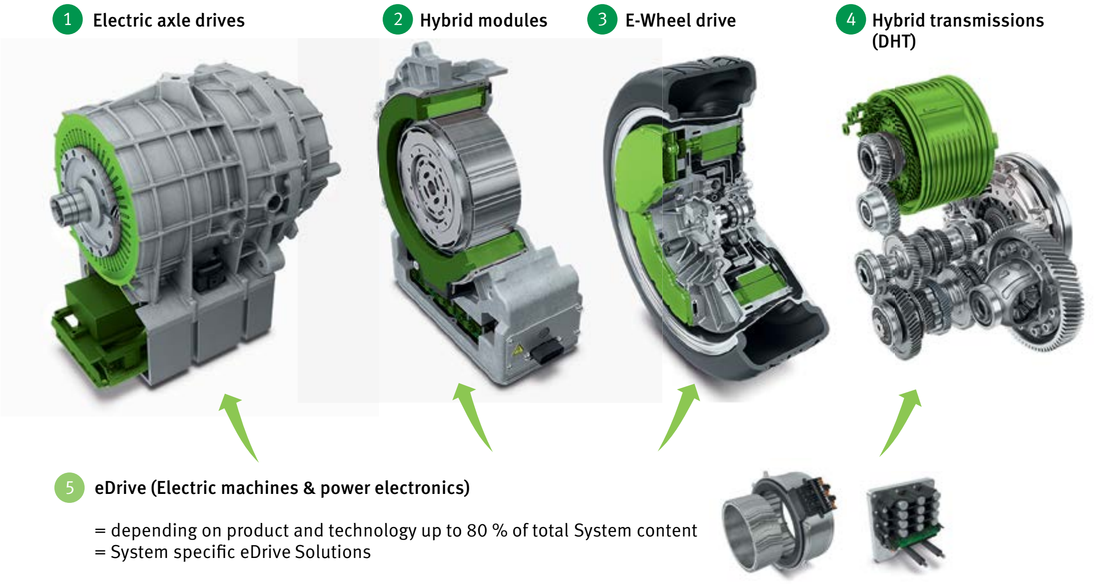
Combining the degree of hybridization and the topology already results in a total of 16 options. The complexity at the system level increases considerably if the transmission versions (serial and power split) provided in a drive module are also considered – Figure 4.

Challenges

In order to comply with the development goals with regard to function, cost, and time despite the high level of complexity and associated expense, automobile manufacturers have to rely on suppliers that have considerable system expertise going beyond their own scope of delivery. At Schaeffler, this system expertise involves all of the powertrain components mentioned (combustion engine, transmission, and electric transmission) and factors in the numerous interactions in the mechanical, electrical, information-technology-related, and thermally interconnected subsystems. The prerequisite for this is competence in incorporating and constantly optimizing simulation tools and test results from test stands or vehicle tests along the entire development chain.
A particular challenge for Schaeffler involves adding competence in electrical tractive drives to its already well-developed powertrain expertise and the know-how that it has built up over the course of decades in combustion engines and transmissions.

Developing Competence in Electric Drives

It is planned for the future product portfolio to offer electric axles, hybrid modules, wheel-hub drives, and even dedicated hybrid transmissions. The scope of services will include the entire system, i.e. even the electric machine, the power electronics, and the software. The resulting requirements for the electric machines and power electronics are therefore very different – Figure 5.



5 Platform approach for e-machines and power electronics in various applications

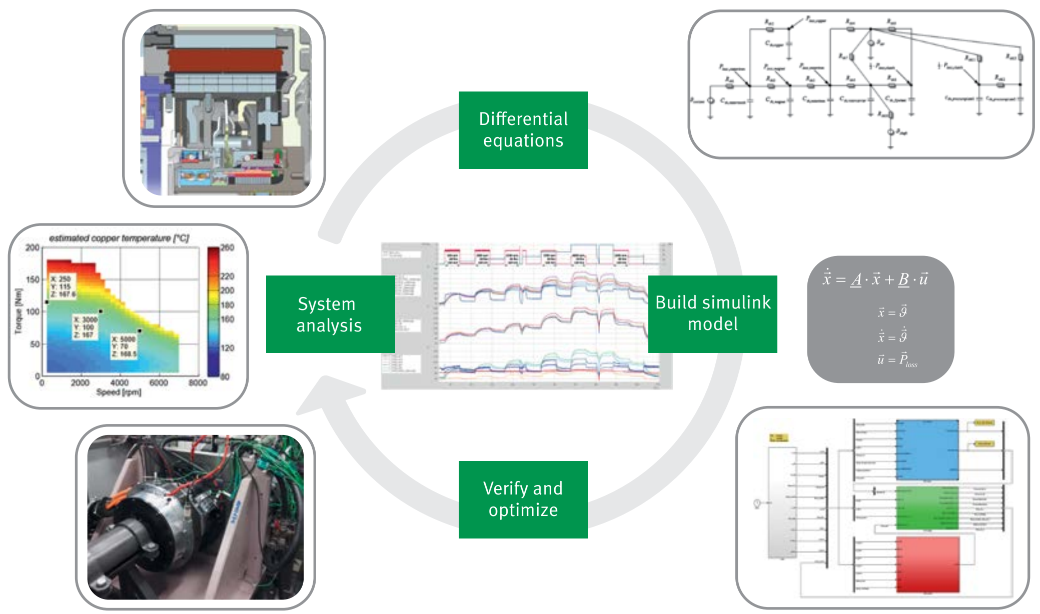
An ideal situation with regard to driving performance and consumption can only be achieved if the gear ratios, the electric motor, and the power electronics are all perfectly balanced. At the same time, the requirements for installation space, weight, and comfort need to be met. This is why Schaeffler has been consistently building up its development competence for around ten years in all relevant engineering disciplines.

In order to be able to make reliable decisions for the large number of individual influencing parameters, continuous model building and simulation is absolutely necessary in all key development steps. This becomes clear based on the supposedly simple question of how a PSM machine needs to be dimensioned in order to fulfill the requirements specification indicated in the requirements specification for a certain application. The design has proven to be a complex optimization problem, since in part opposing effects need to be balanced out, while boundary conditions such as minimal design voltage, maximum admissible current, available installation space, and the performance capacity of the cooling circuit are firmly defined through the application:

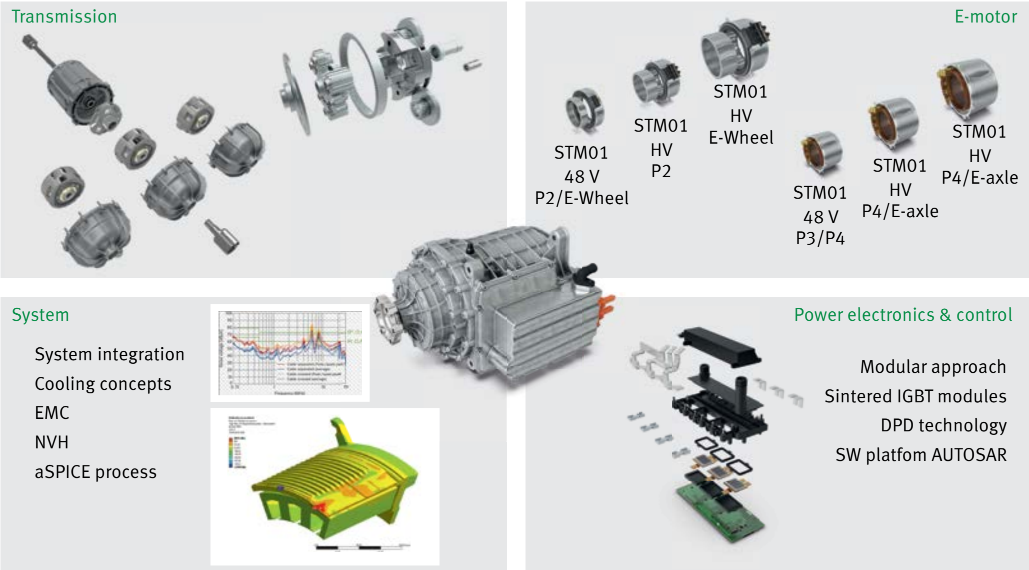
- The maximum torque needs to be achieved at the maximum admissible current. Since the installation space is limited, there is only a little leeway for the mechanical parameters such as the air gap diameter. Thus, the only remaining parameter is the surface force density, which is accompanied by a correspondingly high electromagnetic force (emf).
- This emf needs to be counteracted through field weakening in order to achieve the necessary maximum speed, whereby the output in the entire speed range is to remain at a high level. The optimum field weakening capacity often requires concessions to the maximum torque at the same level of power.
- In the case of a fault (an open gate at maximum speed or a phase short circuit), a high emf can also lead to critical conditions (overvoltage, overcurrent). This is only admissible if these conditions can be reliably intercepted by the power electronics (e.g. active short circuit and thermal stability with short-circuit current).

- The various losses are affected by the machine design as well (losses through ohmic resistance, hysteresis, current displacement effects, harmonics, etc.). The range of the good level of efficiency (the “sweet spot”) and its maximum value can be affected by this and harmonized with the load cycle within the physical limits.
- Limitations with this mainly electromagnetic optimization are due to the mechanical rigidity – including that of the rotor – and the heat balance. Special designs for the local reduction of mechanical tension in the pockets of embedded magnets or for winding cooling may cause the performance parameters to increase considerably.
- The characteristics of magnetic poles automatically lead to force effects that manifest as harmonic impulses in a spinning drive. At worst, they can result in an NVH problem. For this reason, the environment needs to be analyzed with regard to its transfer paths and natural modes. However, the impulses can also be affected by special design measures (a favorable pole/groove ratio, randomized groove opening width, rotor offset, etc.). The retroactive effect on the performance parameters needs to be checked here as well.
- The material quantity and quality not only affect the performance parameters, but unfortunately the costs as well.

The optimization loops needed for this combine calculations of the magnetic field simulation, the electrical design, the mechanical modeling, CFD calculations, and the heat flow simulation.



6 Continuous development of a thermal model for tractive motors



7 **Factors influencing the increase in power density in an electric axle drive**

Moreover, a large number of verified material characteristics are required in order to parameterize calculation models. The models devised and verified during the development process, such as with respect to thermal behavior, are transferred via model reduction to network models, which are ultimately employed for controlling the machine. The subsequent metrological validation is not only used for safeguarding, but also for continuously improving the model – Figure 6.

In order to be able to handle the wide variety of market requirements at an acceptable cost, Schaeffler has built up a development platform for electric drives [6]. The platform is just as suitable for electric axle drives as it is for hybrid modules, wheel-hub drives, and dedicated hybrid transmissions. However, the modular approach does not only include the electric components of the electric drive. Instead, modular concepts for clutches (for example, K0 or the triple clutch in the P2 hybrid modules), actuators, and mechanical transmissions have also been planned and developed. Figure 7 illustrates the application of these modular systems on an electric axle drive.

Considerable improvements are achieved in the four areas shown:

- The consistent optimization of the electric machine with regard to the power density and thermal stability.
- Integrated power electronics on the basis of sintered IGBTs and sophisticated, AUTOSAR-compatible control technology.
- A transmission concept that saves on installation space and makes it possible to simply adapt the gear ratio to the particular application concerned.
- A developmental approach in which key system properties such as electromagnetic compatibility, acoustics, and cooling are regarded in their entirety.

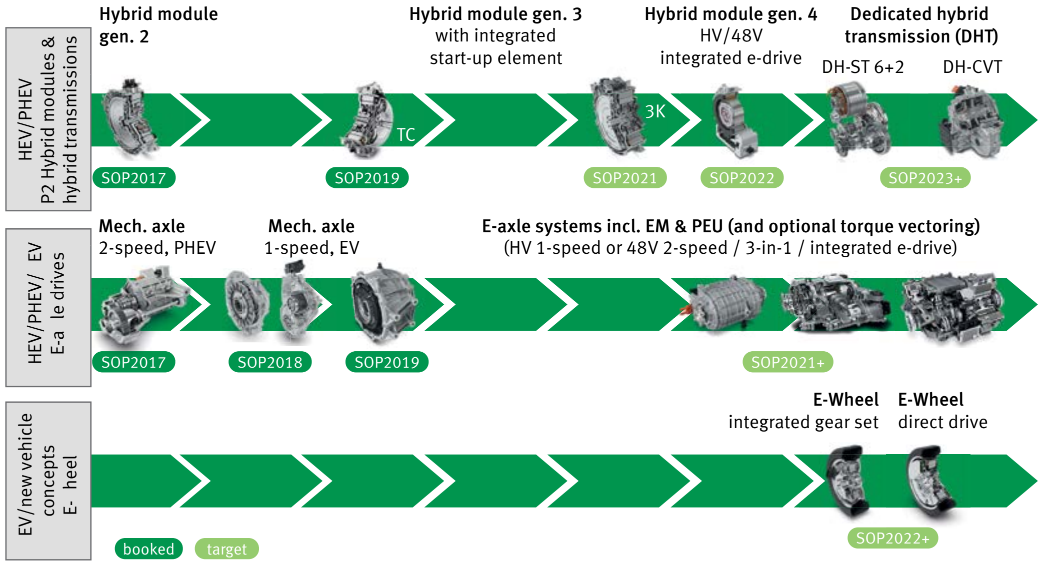
The consistent utilization of the latest technologies, optimum balancing of the subsystems, and further developed mechatronic integration in the current development stage of the electric axle have enabled the torque and power output to be doubled while cutting the weight by 15%.

The takeover of Compact Dynamics, completed in December 2017, has made it possible to broaden design expertise in the field of electric drives even further. Compact Dynamics has established itself as a successful development partner, such as for the drives used to equip the vehicles of the Audi Sport ABT Schaeffler Formula E team.

Not only the technical challenges need to be mastered, but also the commercial and scheduling challenges. It is only on the basis of flexible modular approaches that it is possible to individually tailor the solutions to various usage profiles while simultaneously shortening the development times. For this reason, Schaeffler has defined a technology module for the important subsystems. The underlying modularization allows for a high degree of solution flexibility despite standardization.

Road map

The second-generation hybrid modules in series production today will be supplemented in upcoming years by integrated launch elements (generation 3) and integrated power electronics. Dedicated hybrid transmissions represent an even greater degree of integration, with the electric drive taking on part of the transmission function. Transmissions for electric axle drives with one or two gears are already in production or are awaiting the start of series production soon. In the future, Schaeffler will also be developing and supplying axle drives along with the electric machine and power electronics – Figure 8.



8 **Product road map for electric drives.**

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17

P0 Mild Hybrid

With System Competence to Maximum Efficiency

Christoph Schröder
Dr. Andreas Stuffer

Introduction

The entire powertrain is currently in flux. The classic combination of a combustion engine and a transmission is increasingly being supplemented at various points by hybrid components and new electrical consumers. If a belt starter generator (BSG) is used in a classic auxiliary drive, this is referred to as a "P0 mild hybrid." While the focus was originally on 12-volt belt start-stop systems in the auxiliary drive, it has now shifted to P0 mild hybrid systems with a 48-volt or high-voltage belt starter generator, which can be used to fulfill stricter CO_2 regulations.

P0 Mild Hybrid as an Efficient Integration Strategy

The central argument in favor of P0 mild hybrids is their high functional scope combined with comparatively minimal integration time and effort, and low costs. This topology makes it possible to use more powerful electrical consumers, along with fast and comfortable combustion engine restart via the belt drive, and it is also capable of recuperating braking energy and feeding it back to the drive as needed, called "boosting."

According to the assessment of Schaeffler, these properties will lead to a considerable increase in the P0 mild hybrid segment in upcoming vehicle generations. For the year 2030, a global market of around 20 million units is anticipated for pure P0 drives on a 48-volt basis.

P0 hybridization is also used in combination with full or plug-in hybrid drives. This conceptual design places fewer demands on the system components involved compared to a pure P0 system. In this way, the P0 drive can be designed exclusively as a comfort start system.

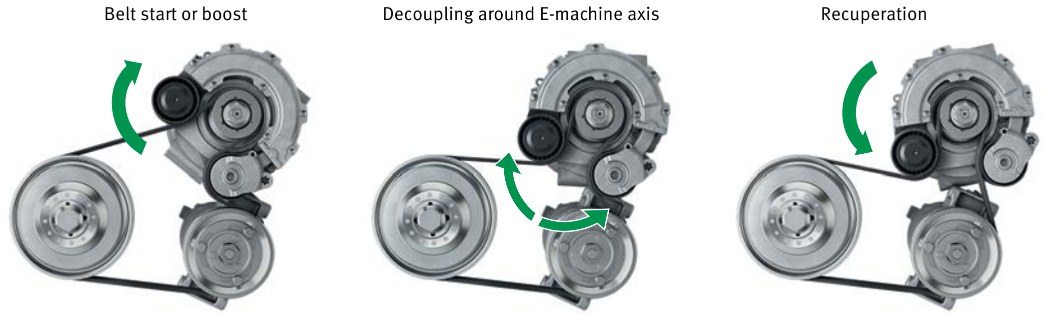
Thanks to the fact that Schaeffler offers key components for the belt drive and the powertrain, along with a wide variety of hybrid topologies, there is also extensive system expertise. Therefore, by working together with the automobile manufacturer, it is possible to design and implement an optimally efficient system for every application. By way of examples, this article shows how the subsystems interact, what repercussions this has on component selection and design, and what belt-driven components are ideally used in P0 mild hybrid systems. In the course of holistic system coverage, Schaeffler does not limit P0 activities to the belt drive alone, but integrates further studies and optimizations as well, such as a roller-bearing-mounted crankshaft bearing arrangement [1] for reducing friction loss or E-clutch systems [2] for manual shifting applications in order to completely incorporate the efficiency potential.

Effects on the Belt Drive System

When there is 48-volt or high-voltage hybridization in the belt drive, considerably higher outputs are transferred in comparison to conventional 12-volt applications [3]. For example, the generator's output goes up from approx. 3 kW to 12 kW. This output affects both directions of torque, and there is a frequent switching between "boosting" and "recuperating". This causes the overall strain on the system and the components to greatly increase, and the boundary conditions are very different compared to conventional solutions. Schaeffler responded to these challenges early on and has developed key components and development methods for integrating e-machines into the belt drive [4].

Key Components for the Belt Drive of the Future

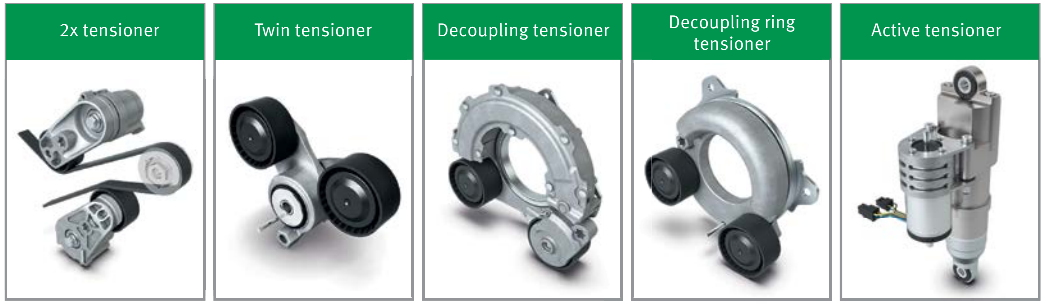
In order to fulfill the functions of a 48-volt drive – particularly "boosting," load point shifts, and the recuperation of braking energy that is as complete as possible – the torques applied to the belt drive by the e-machine increase considerably. Peak torques of up to 50 Nm are transferred to the starter generator both during generator operation in the idling speed range as well as when starting the combustion engine.



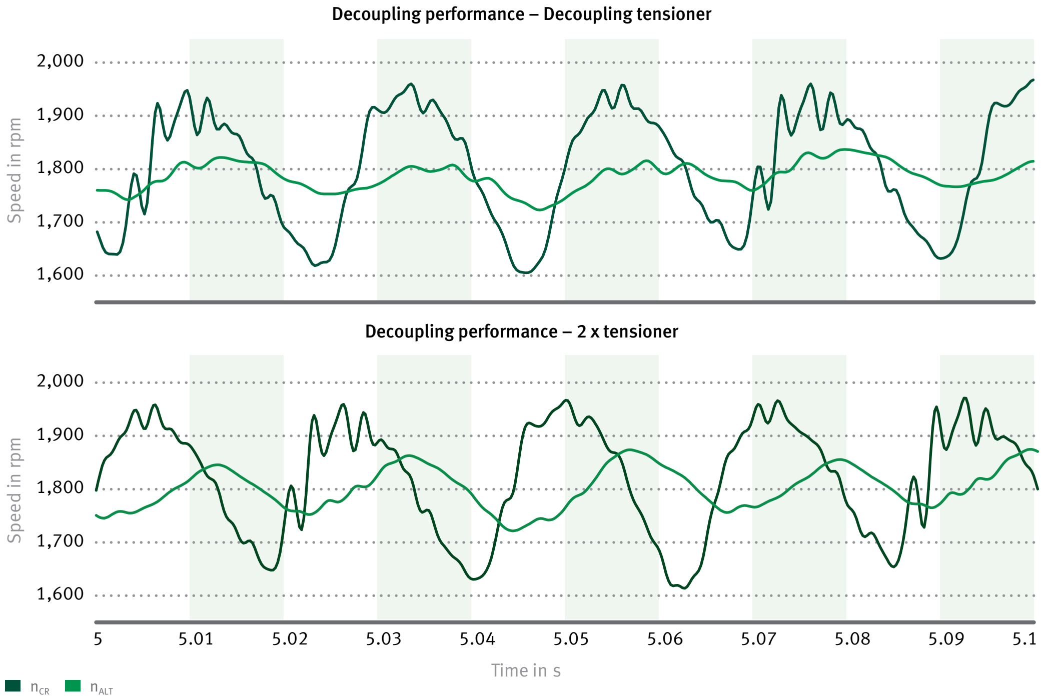
1 Operating modes of a decoupling tensioner

Depending on whether the starter generator is operated for generating power or as a motor, feeding torque into the belt drive, the tight and slack spans alternate, and the tensioner function is needed in different sections of the belt. This requires modified tensioning systems. The simplest way to implement this is with two separate tensioning elements, one positioned before and one after the pulley of the belt starter generator (BSG). Alternatively, both tensioning pulleys are combined to form what is called a "twin tensioner," in which only a tension spring connects both arms and only one attachment point is needed on the machine as well. Another design is the decoupling tensioner. When the torque changes direction, the belt tensioner changes its working position, making the optimum belt pre-tensioning force possible for the respective operating point – Figure 1.

Decoupling tensioners are attached coaxially to the BSG and bolted onto it to form a unit. The specific design depends on multiple factors, such as the arrangement of the auxiliary units and the installation space that is available in the vehicle. Schaeffler has developed various solutions and therefore offers a wide range for belt drives of the future – Figure 2.



2 Tensioning elements and decoupling tensioners for P0 belt drives



3 **Decoupling function of a P0 system, based on two individual tensioners (above), compared to a P0 system with a decoupling tensioner. Graphed here are speeds over time during generator operation; the BSG speed is converted to the crankshaft**

In addition, the irregularities introduced by the engine and their retroactive effect on the generator need to be examined. In a P0 application, a decoupling tensioner reduces the speed amplitude in the generator. Compared to two individual tensioners, this solution improves the overall dynamic behaviour of the auxiliary drive and leads to lower loads and smaller losses of power – Figure 3.

These irregularities from the decoupling tensioner can be readily managed below certain limiting values, which is the ideal configuration. However, heavily loaded combustion engines with a partly reduced number of cylinders already generate high torque levels slightly above idling speed. This results in much higher turning irregularities of the crankshaft. If the vibration angle exceeds a value of approx. ± 4° at 1,000 rpm and under a full load, then it is no longer admissible from a system perspective to introduce these irregularities from the crankshaft into the P0 mild hybrid belt drive. The specified value serves as an indicator and must be calculated individually depending on the design, the engine characteristic, and the auxiliary data. Such excessive irregularities already occur today in the overwhelming majority of modern three and four-cylinder engines and need to be isolated from the belt drive – particularly in a P0 mild hybrid system.

For this purpose, Schaeffler developed a crankshaft belt pulley decoupler [5] and launched its series production in 2013 for the first time, both for multiple conventional belt drive systems as well as for a P0 mild hybrid belt drive system. Like a dual-mass flywheel (DMF), the belt-driven de-

coupler – Figure 4 – is based on the principle of supercritical vibration isolation. The version with steel bow springs allows for a high power density, very good isolation levels, and a high degree of robustness, and it is designed to withstand more than 1 million start-stop events.

Compared to a non-decoupled system, the crankshaft belt pulley decoupler reduces the disruptive dynamics in the belt drive to a minimal residual amount as shown in Figure 5, which is based on the irregularities in the starter generator.

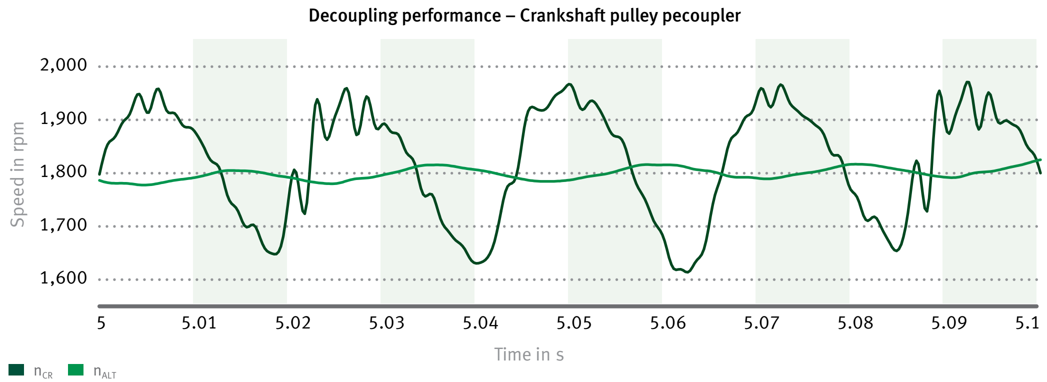
A crankshaft belt pulley decoupler can also be used before limiting values for the drive load are reached. In this way, the use of a decoupler can enable further potential for optimization, such as by reducing the width of the belt and lowering the pre-tensioning force, resulting in CO_2 savings [5].

4 **Belt pulley decoupler**

Design Criteria for Maximum Benefits

In order to satisfy future torque demands on the belt drive, it is possible to vary multiple parameters in the belt drive, such as the pulley diameter, drive ratios, pre-tensioning forces, belt width, wrap angle, and belt quality.

The selected drive ratio between the crankshaft and generator is decisive for the efficient use of the e-machine's map. The possible diameter for the crankshaft belt pulley is generally limited by the installation space. In a "map-optimized"



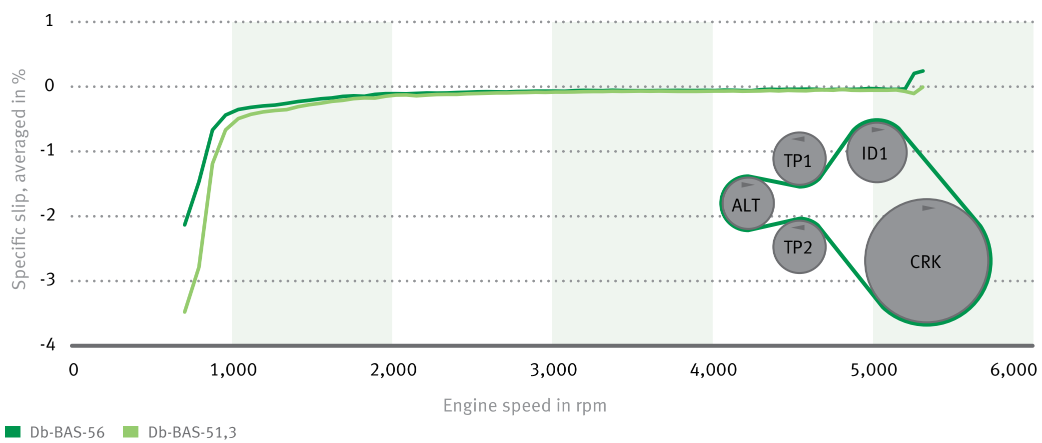
5 **Decrease in engine irregularities by using a crankshaft decoupler. Graphed here are speeds over time during generator operation; the BSG speed is converted to the crankshaft**

drive ratio, this can lead to the generator belt pulley having a small diameter. Particularly during "boost," this can result in excessive slippage in low speed ranges, thereby reducing the efficiency of the P0 application. Figure 6 compares the slippage during engine startup between two design versions with diameters of 51.3 and 56 mm. At low speeds, a smaller diameter of the generator belt pulley causes greater slippage between the belt and the pulley due to the reduced contact surface. Average slippage values of more than 2 % can lead to increased belt wear, disturbing noises, and a lower level of efficiency and are therefore not permissible.

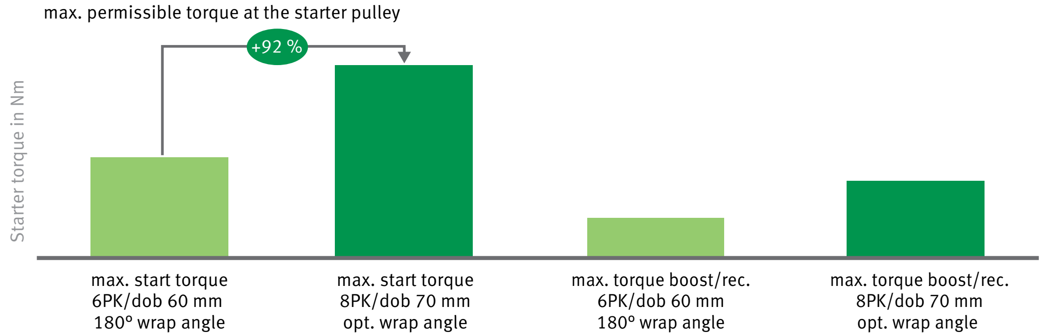
Keeping the relative slippage level down is a basic design goal. This is not only true of boost operation, as shown in the example, but also of all recuperation and start phases. In the current 48-volt hybrid systems with a P0 arrangement, the start torques are around 40 to 50 Nm and can be transferred using the current belts and tensioning elements, such as by means of a 6PK design.

Schaeffler anticipates that the power levels needing to be transferred will go up in the future. The peak torque during starting will temporarily reach values of up to 65 Nm, and the power levels during continuous and boost/recuperation operation will increase as well. The greater torques can also be used for starting the engine when cold, making it possible to do without a pinion starter. Calculations show that much greater torque can be transferred with a larger diameter, while increasing the number of ribs at the same time and optimizing the wrapping of the BSG pulley – Figure 7. Conversely, a given limiting speed for starting can already be implemented with six instead of eight ribs when the diameter is larger. Moreover, it is obvious that a reduced belt width also reduces the work needed for deforming the belt, meaning that lower power losses occur.

As a rule, it also makes sense to maximize the diameter of the belt pulley in the e-machine. In a given installation space, however, such an optimization strategy runs into limits due to the necessary drive ratio for the crankshaft output.



6 **Simulation of a P0 48V system during boost operation: Average belt slippage at the belt pulley on the generator side for two pulleys 51.3 mm (light-green characteristic curve) and 56.0 mm (green characteristic curve) in diameter.**



7 **Torque limits of the e-machine due to the maximum admissible belt forces, depending on the belt width, BSG diameter, and wrap angle.**

In the ContiTech-INA joint venture, Schaeffler also offers the system development of belt drives as a complete service. ContiTech is developing the start-stop belt further, with the goal of increasing its maximum load-carrying capacity and torque capacity.

The permissible loads of the belt in the BSG system need to be maintained for temporary and continuous operation at all times. Particularly in continuous operation, lower limiting values are involved, which are easier to maintain using decoupling elements from Schaeffler, since the dynamic load is reduced over all operating ranges.

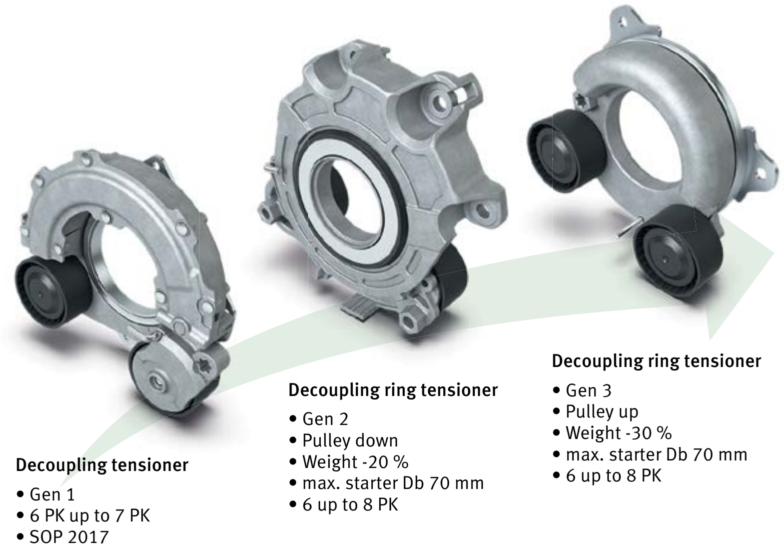
Further Developed Components for Future Requirements

Building on the belt drive / drive / vehicle system assessment, Schaeffler is consistently developing the components for P0 drives further, with an emphasis on the next generation of decoupling tensioners. In addition to general optimizations, such as reducing the required installation space and reducing weight, modular adaptation to various belts with up to eight ribs is at the forefront. Other than the version already produced in series for a P0 48 V belt drive based on a 6PK belt, a version with tensioner pulleys oriented towards the e-machine – middle of Figure 8 – is also being developed.

The version with tensioner pulleys facing in the direction of the e-machine leads to lower bending torques in the generator input shaft, thereby reducing the load on the generator shaft bearing. The belt is positioned. on the generator pulley prior to mounting the tensioner.

In addition, an active, electromechanically activated belt tensioner is currently in the advance development stage at Schaeffler – Figure 9. This kind of tensioner makes colder starts possible, as it permits infinite variability of the belt pre-tensioning force. Depending on the torque applied, the pre-tensioning force is regulated as needed at the generator. Basic tests are currently being conducted to check how the friction power advantage proves in practice at the various operating points when the pretension applied is ideal at all times.

Another point of emphasis is the constant further development of the crankshaft belt pulley decouplers. Today, the relevant range of a 5 to 8 PK belt width and a diameter of 140 to 200 mm for P0 mild



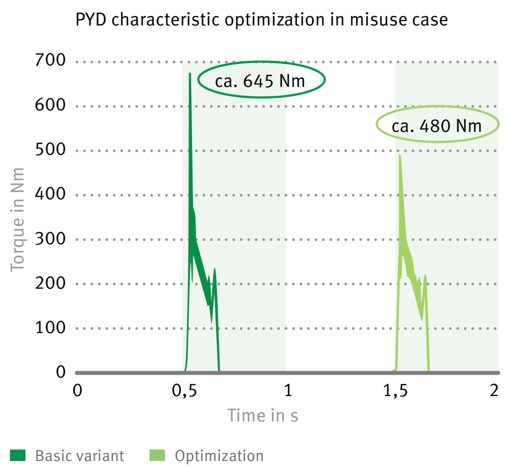
8 **Decoupling tensioners in comparison; left: series standard; middle: tensioner pulleys facing in the direction of the e-machine; right: tensioner pulleys facing away from the e-machine**

hybrids and standard applications is already covered. The main focus of more recent developments is on an increased power density and a greater transfer capacity of the decoupler so that even challenging P0 applications can be handled efficiently. Particularly with P0 applications, the interaction with the activation strategy of the starter generator is an important aspect, the effect of which is accounted for by the holistic system approach when defining the characteristic curve.

Thanks to the optimizations, it is possible for Schaeffler to offer belt pulley decouplers that provide the automobile manufacturer with additional degrees of freedom for its application strategy. The characteristics, ranging from overload protection during dynamic operation to the factoring in of "misuse" operating conditions, are heavily dependent on the system environment, such as powertrain rigidity levels and torque gradients when the clutch closes quickly. Since there is an extensive and equalized simulation environment, including the powertrain on the one hand and extensive expertise for designing and applying arc springs derived from applications of the dual-mass flywheel on the other, it is already possible to make high-quality predictions at an early stage with regard to how various combinations will behave later on. The upshot of these optimizations is that – even in cases of "misuse" – the admissible torque at the crankshaft / belt pulley interface is not exceeded – Figure 10. This is particularly important for central screw connections.

9 **Electromechanical belt tensioner**

With the "switchable decoupler," Schaeffler is offering a comfort extension in the portfolio of decoupled belt pulleys. This delivers all of the advantages of a standard decoupler for P0 mild hybrid applications and is also additionally equipped with actuators used to separate the torque flow between the crankshaft and belt drive as needed. This makes it possible to continue operating the air conditioner compressor located in the belt drive via the P0 starter generator when the combustion engine has stopped, such as during sailing or after stopping at a traffic light. The fact that temperature fluctuations in the vehicle interior are now prevented causes customer acceptance of start-stop operation to go up considerably. It is also possible to attain consumption advantages by stopping the combustion engine during real driving when air conditioning is needed. The combustion engine is comfortably restarted through the belt starter generator after the actuators close.



10 **Torque at the interface to the crankshaft**
Left: standard design; right: optimized characteristic curve of the "shifting error" load case

11 **Switchable belt pulley decoupler for air conditioning when the combustion engine is stopped**

The current optimizations of the switchable decoupler enable the possible applications to be extended to typical installation space situations, thereby reducing the need for space. Moreover, there is also the option of keeping the link to the crankshaft open even during combustion engine operation, allowing the belt drive to come to a standstill.

Schaeffler already has a broad portfolio of tensioners and belt pulley decouplers for standard and P0 mild hybrid belt drives. The further developments shown for the components and subsystems can be used to bring about additional advantages and thus more efficient utilization of the P0 strategy at the customer's. Nevertheless, the optimizations must not be seen as relating only to the belt drive as a system regarded in isolation.

System Behavior in the Drive

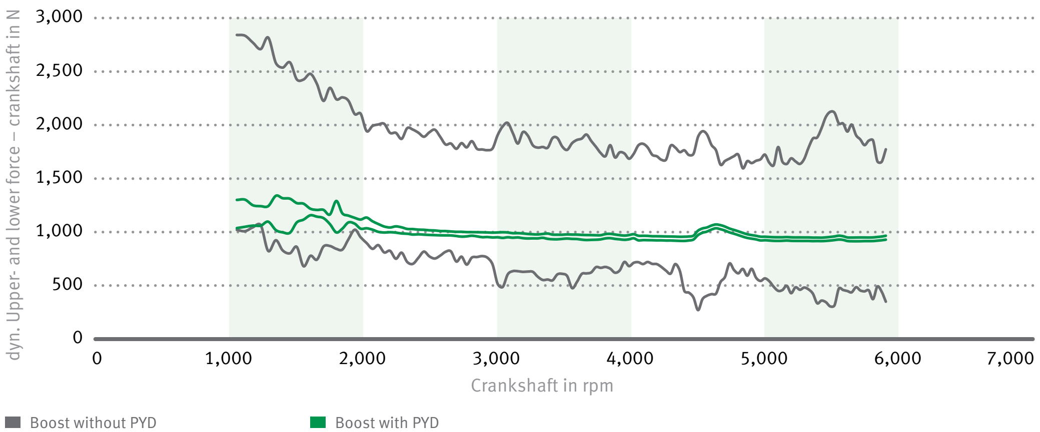
As a system provider, Schaeffler has good knowledge of the entire hybrid powertrain, from the belt drive to the wheel. The interaction of the individual systems with each other and with the vehicle is taken into consideration in order to efficiently coordinate the complete system. Key close-coupled interactions exist between the crankshaft bearing and the double clutch or dual-mass flywheel damper on the transmission side.

Crankshaft Bearing

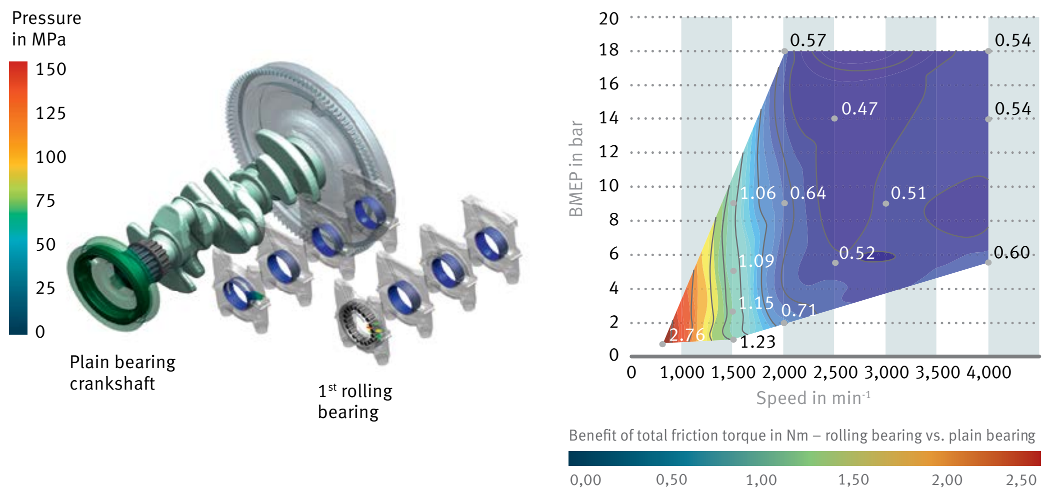
The retroactive effects of a P0 mild hybrid system on the crankshaft itself and – for instance – the first crankshaft bearing is taken into consideration for the various operating modes and optimized using different components, such as a crankshaft decoupler or even a rolling bearing arrangement.

The new operating modes, such as “boosting,” can lead to very high dynamic belt forces, which are also applied to the first main bearing. Figure 12 shows the very clear reduction in the dynamic force levels of the resulting hub load on the crankshaft when a crankshaft decoupler is used. At the same time, this ends up relieving the first main bearing and all other bearing positions with respect to the dynamic force peaks accordingly.

However, there is a higher pre-tensioning and operating force present in a P0 belt drive compared to a standard drive due to the torque transfer, which invariably leads to greater friction losses in the plain-bearing-mounted crankshaft. For this reason, Schaeffler is also taking the effect of the P0 systems on friction in the first crankshaft bearing into consideration. In tests on a three-cylinder engine with a displacement of 1.0 l, for example, it is evident that the increased friction power is caused by the higher loads acting on the front crankshaft bearing. It is possible to reduce these friction power losses again by using a rolling bearing instead of a plain bearing – Figure 13.



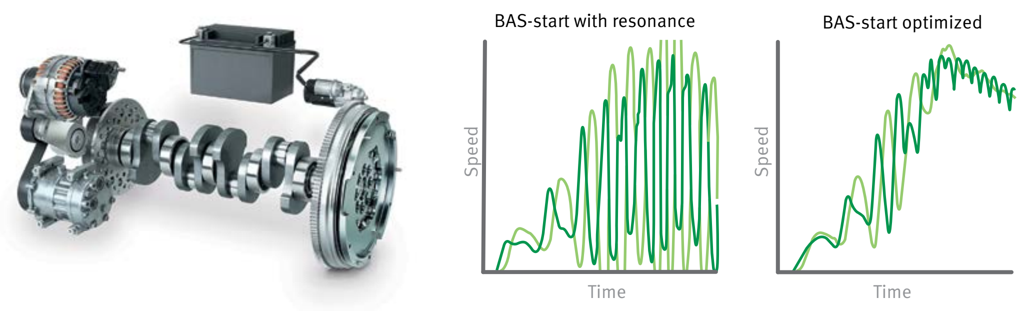
12 Reduction in the bearing forces on the crankshaft pulley by using a crankshaft decoupler



13 Friction power vs. effective medium pressure and speed in a three-cylinder engine with a displacement of 1.0 l and a P0 belt drive. Using color gradients, the diagram shows the advantage in the overall drive torque in the case of a rolling bearing arrangement in the first position of the crankshaft.

Interaction with the Dual-mass Flywheel

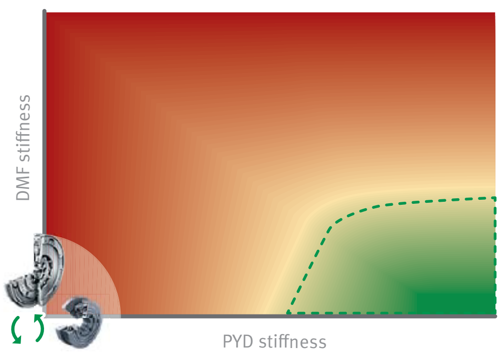
The integration of an e-machine in the auxiliary drive affects the behavior of the complete engine-transmission system. In the course of a development project, the challenge arose that the interaction between the belt pulley on the crankshaft side and the damper on the transmission side caused a resonance to occur in the powertrain when the belt starter alternator attempted to start the engine, which prevented the engine from starting. The characteristic was dependent on different parameters in the starter generator, the belt drive, and the damper on the transmission side. It was already possible in an early development phase to demonstrate the effect with the aid of suitable simulation tools – Figure 14. By coordinating the characteristic curves of the dual-mass flywheel and the belt pulley decoupler,



14 Vibration system of the belt drive – decoupler – crankshaft – dual-mass flywheel: rotational irregularities when starting the engine, both before (middle) and after optimization (right); green: crankshaft; bright green: belt pulley a comfortable engine start was able to be obtained, even though the output of the e-machine was below 2 kW and therefore in the lower limit for a four-cylinder diesel engine. The design of the Schaeffler belt pulley decoupler, which makes it possible to implement a wide variety of characteristic curves, was very helpful for this.

In mild hybrid systems, customers expect a particularly high level of comfort when the engine starts or restarts. It is vitally important to optimize the NVH behavior at an early stage. Schaeffler has suitable simulation tools for doing this, along with the expertise for harmonizing the components “on both sides of the crankshaft” – even before measurement data from real tests is available. For example, the Schaeffler-developed “DYFASIM” simulation program allows the interaction of vibration phenomena in the powertrain to be calculated quickly. Figure 15 shows the sample results of a test in which various dual-mass flywheel characteristic curves and various decoupler characteristic curves were tested for their interaction during engine start.

Viewed in isolation, all dual-mass flywheel and decoupler characteristic curves are in line with a good design. When considering the example of the belt start in the interaction, it is evident that



15 **Sample parameter test of dual-mass flywheel and decoupler characteristic curves**

only certain combinations of dual-mass flywheel and decoupler characteristic curves result in very good behavior when the belt starts, particularly when the battery voltage is reduced. The ability to narrow this down at an early development stage allows for a targeted design, fewer development loops, and an efficient use of development resources.

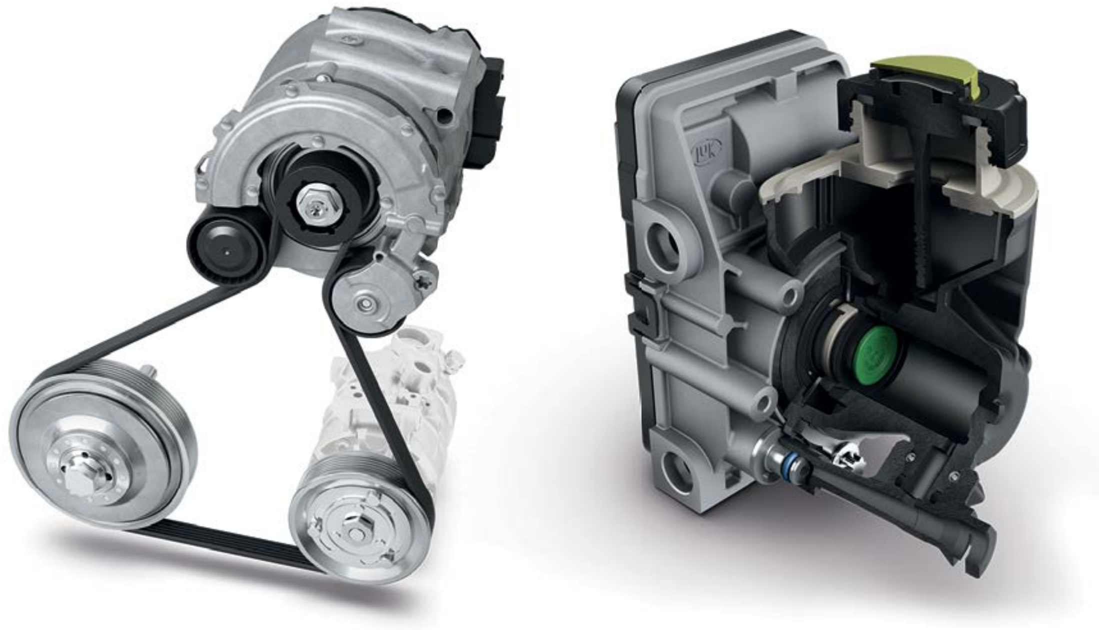
Vehicle System Integration

The CO_2 savings attainable in real road traffic play an important role. For this reason, Schaeffler has defined its own driving cycle for important development locations: one that reflects the requirements of the European RDE regulations.

Schaeffler uses this driving cycle for testing the integration of P0 hybrid systems in various drive topologies. Among other things, comparison drives are conducted with the following vehicles:

- P0: Schaeffler‘s “Gasoline Technology Car I – GTC I” technology demonstrator. It has a gasoline engine with a nominal output of 92 kW, a manual transmission, and an electrically actuated clutch, along with an e-machine with a peak output of 10 kW that is integrated in the belt drive – Figure 16.
- P0 + P4: A series passenger car with a 120-kW diesel engine, which has an electric rear axle with a peak output of 25 kW. The P0 machine used has a nominal output of 8.5 kW and is run at a voltage level of 200 V – like the P4 e-machine.
- P0 + E-clutch: A series passenger car with an 80-kW diesel engine, which is run with a 48 V P0 hybrid and an E-clutch. Here as well, an e-machine with a peak output of 10 kW is used.

The analysis of the measured driving data with respect to the load amount relevant for the belt drive shows that both the number of switching

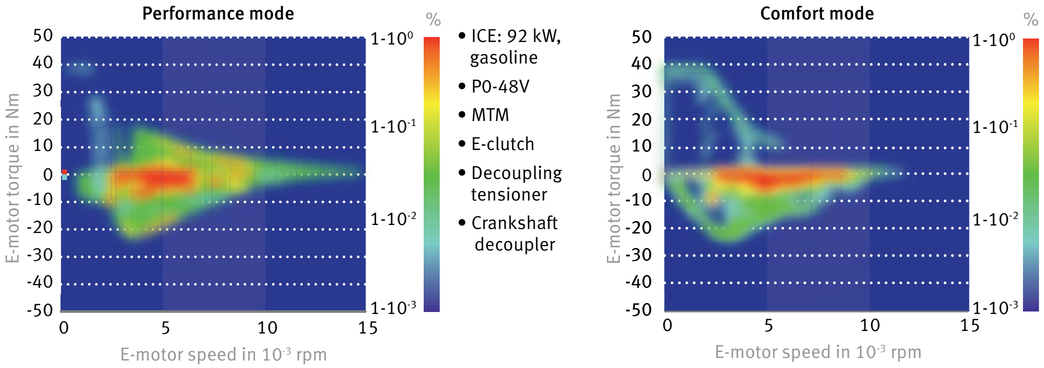


16 **Schaeffler components interacting in the GTC I technology car: P0 belt drive with decoupling tensioner and crankshaft decoupling on the left; E-clutch actuator on the right**

processes between the operating modes of “boost” and “recuperation” as well as the load level as such are clearly different. An initial dependency is inherent in the topology selected. For example, when a P0 is combined with a P4 e-axis, the e-axis takes on the vast majority of the boost and recuperation work. The P0 hybridization is responsible for supplying the vehicle power supply, charging during driving mode, and restarting the combustion engine. The focus of the P0 drive is on the start function and on optimizing the system in its interaction with the automatic transmission and the sailing function implemented. In contrast, a pure P0 system without P4 takes on the boost and recuperation work as well and therefore exhibits many more load changes between these two operating modes.

Both topologies have a direct effect on the load applied to the components in the belt drive and need to be taken into account for dimensioning. With a P0 and P4 drive combination, the components of the belt drive can be optimized due to the reduced loads.

There is a second dependency inherent in the operating modes of each vehicle. The following discussion of the effects of the existing “Sport” and “Comfort” modes takes the GTC I technology car with its pure P0 system as an example. Sport mode requires more power, and there are more frequent changes between the boost and recuperation torque of the e-machine. During RDE driving in this mode, there are a total of 620 boost and recuperation events over a route distance of 87.5 km, which corresponds to approx. 7.12 changes per km. Over an operating time of 240,000 km, this yields nearly 1.7 million load changes. In comfort mode, only 313 events occur in the same cycle, while the combustion engine is



17 Time-weighted occurrence of various loads in the P0 drive over the electric motor speed for two operating strategies: histogram analysis in sport mode on the left and in comfort mode on the right

restarted 160 times (sailing mode and engine start from a standstill), which corresponds to approx. 1.82 belt starts per km driven. In comparison, there were 0.74 starts in sport mode – Figure 17.

This analysis on the basis of a demonstrator vehicle is merely an example, yet it shows that the hybrid configuration – along with the newly added vehicle strategies themselves – lead to large differences in the loads applied to the components, and they need to be taken into consideration by Schaeffler in coordination with the customer.

The CO_2 emission of a vehicle equipped with a P0 hybrid drive is also heavily dependent on how large the percentage of the braking energy recuperated during real driving needs to be. Due to the system, this energy recuperation in only possible as long as the combustion engine is engaged in boost operation. In order to test the effect of driver behavior, further tests were conducted with a vehicle equipped with a P0 48-volt drive (peak output of 10 kW). The only thing that was varied was the time when the driver engaged the clutch pedal. With typical driving behavior, there is energy recuperation within the range of 17 kJ/km and 23 kJ/km. When the clutch is opened as late as possible, the entire energy recuperated in the RDE cycle averages 33 kJ/km – Figure 18.

This increase in mechanical recuperation output by a factor of 1.5 to 2 leads to estimated savings of approx. 0.2 l / 100 km.

			Mech. recuperation in kJ/km			
Date	**Driver**	**Optimised „delayed“ clutch opening**	**City**	**Country**	**Highway**	**Total**
12.09.2017	F4810	yes	37	39	23	**33**
12.09.2017	F4831	no	8	19	22	**17**
13.09.2017	F2924	no	28	38	17	**23**

18 Results of RDE test drives: influence of the clutch closure time on the recuperated energy in the same vehicle

From Schaeffler's perspective, the savings potential in a P0 hybrid with a manual transmission can therefore be very efficiently increased by an E-Clutch [6], since the conventional clutch will always be engaged at the ideal late moment by means of the overlapping activation of the E-Clutch actuator.

Maximum Efficiency

Reducing CO_2 emissions is one of the most important demands made by automobile manufacturers on future drives. Simulations by Schaeffler based on the WLTC show that the CO_2 advantage of the 48 V P0 hybrid drive is nearly double that of a basic configuration with a 12-volt starter generator capable of start-stop due to the more efficient use of boost and recuperation, and it has up to 7 % in CO_2 savings compared to a vehicle without start-stop.

If the various combination options are taken into consideration, such as the E-clutch or other hybrid technologies, a CO_2 advantage of up to 18 % is achieved for a C-segment vehicle with P0 + P4 (high-voltage), depending on the hybridization strategy. Here, the P0 mild hybrid makes important functions possible in the complete system, such as engine restart, load point shift, and range extender functionality, thereby contributing towards maximizing efficiency in the system.

In addition, it bears mentioning at this point that the underlying concept of a belt-driven starter generator and decoupling tensioner can also be utilized near the transmission in the axially parallel hybrid module. Schaeffler is developing corresponding modules and concepts based on 48 volts [7].

The increased vehicle electrical system, which is available very cost-efficiently by means of a P0 system with 48 volts, supports the use of other system components that enable additional CO_2 potential to be tapped into. Likewise, improvements in comfort can be implemented or the driving dynamics optimized. Even if many of the solutions already work with 12 volts, implementing them with 48 volts is simpler on the whole due to the reduced current levels. Examples of this are additional electrical charging, the active roll control system, level control, and the electric air conditioner compressor.

Summary

On the Way to Becoming the Standard

Regardless of the voltage level of the vehicle electrical system, it is apparent that the P0 mild hybrid is on the way to becoming a standard across the markets. The following are the most important reasons for this:

- The technology shows a clear customer benefit.
- It can be combined with various drive topologies and units.
- The complete system can be optimized in combination with other system components.
- The ratio of the work needed for integrating the P0 system to its benefits is very favorable.

This article has shown that the integration of powerful e-machines in the belt drive results in new requirements for all components of the belt drive and the complete system design. They can be mastered using the simulation methods and components developed by Schaeffler.

The use of P0 drives also makes sense in combination with other e-machines in the powertrain, whereby – as shown above – the design criteria change with the topology as well. From Schaeffler's perspective, therefore, P0 drives are more than just the introduction to electrification. They satisfy the prerequisite for establishing themselves as the standard in hybrid drives of the future. Due to the extensive system expertise at Schaeffler, maximum efficiency can be reached together with the customer in every application.

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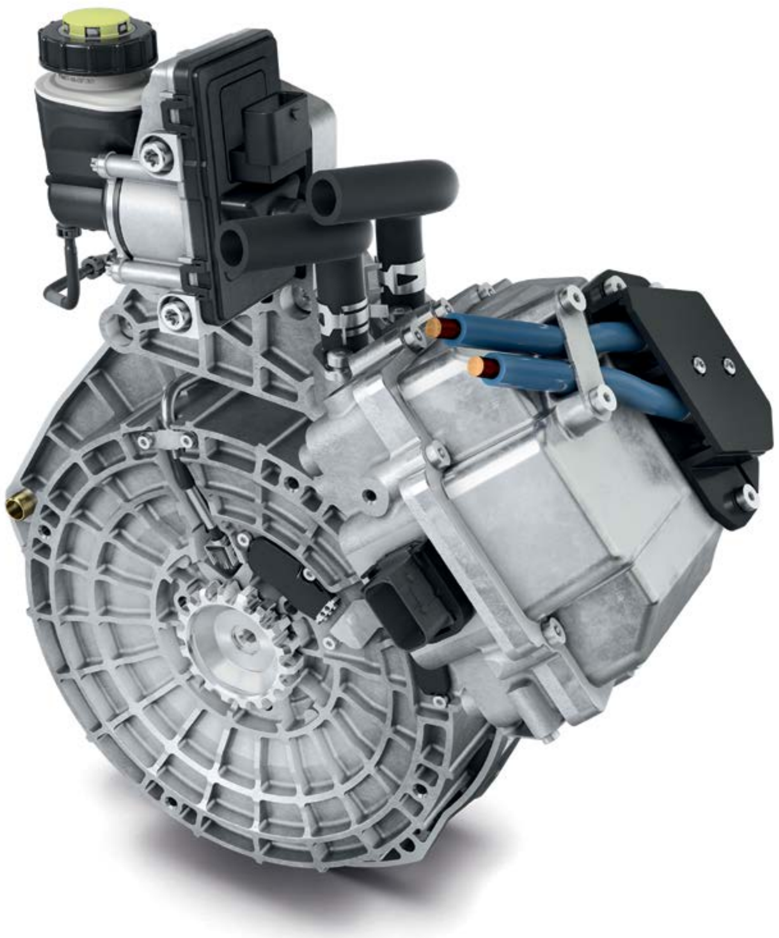


48 V Hybridization

A Smart Upgrade for the Powertrain

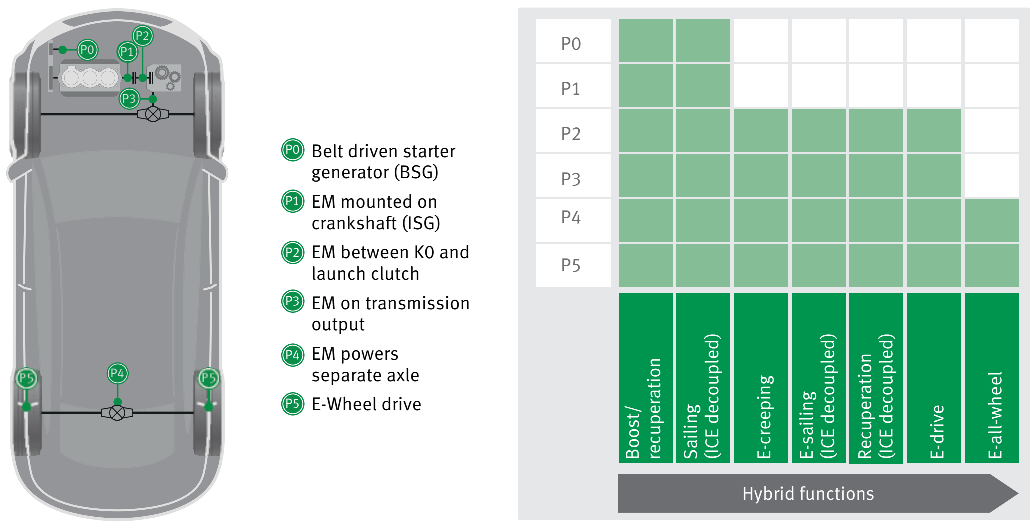
Thomas Eckenfels
Florian Kolb
Steffen Lehmann
Waldemar Neugebauer
Manuel Calero

Introduction

48 volt hybrid systems provide an opportunity to electrify conventional powertrains as parallel hybrids with peak outputs of up to 20 kW without having to fundamentally adapt the whole architecture. As the energy is almost completely recuperated in the delay phases, the use of this type of 48 volt hybrid system leads to significant reductions of up to 15 % in consumption and CO_2 in the WLTC (World-Wide Harmonized Light Duty Test Cycle) [1]. In actual road traffic, the savings achieved are highly dependent on the actual driving profile but the greatest potential for savings is to be found in city traffic. 48 volt hybridization also paves the way for meeting the ever-increasing demands on electric output, for example for increased comfort and driving dynamic functionality or autonomous driving [2]. The efficiency and performance of the combustion engine can also be substantially increased by inter-

a c -
ting with the 48 volt hybridization. This can be achieved by, for example, replenishing the torque at low rotational speeds, the so-called “boosts”, and by using electrically supported 48 volt-based charging systems combined with the corresponding dethrottling under partial load through adapted valve control times. From the perspective of the RDE (Real Driving Emissions) legislation, changing the load point in the acceleration phases and using electric catalysts can lead to a significant improvement in the emissions behavior.

When compared with high-voltage hybrid systems, 48 volt systems offer a particularly good cost/benefit ratio and also have the advantage that the protection measures in the direct current component of the on-board power supply associated with a high-voltage supply can be dispensed with. In addition to this, electric systems with short-term high output requirements,



1 **Various 48 volt architectures and the associated functionalities**

such as electric compressors, catalytic converter heating or active chassis systems, can be standardized and developed as a 48 volt system without the additional expense of high-voltage protection.

Integrating a 48 volt system into an existing engine and transmission module is financially feasible. The objective is to make the system as modular as possible and to insert it into existing production and assembly structures. As far as vehicle development is concerned, the question is, which 48 volt architecture is the correct one? The following article therefore compares the characteristics of hybrid systems for a range of 48 volt architectures. Particular focus is placed on CO_2 savings, functionality and driveability, particularly the restart comfort of the internal combustion engine.

48 volt architecture

Initial situation

Figure 1 shows various different ways of integrating the electric motor in the powertrain. The basic functions of boost and recuperation when the internal combustion engine is engaged as well as sailing when disengaged can be achieved with all architectures; an automatic launch clutch is necessary for disengaged operation here. The P2 to P5 arrangements have the following in common: They allow recuperation of braking energy when the internal combustion engine is disengaged (unlike the P0 and P1 arrangements which are linked to the crankshaft speed) and the performance capability of 48 volt systems makes purely electric driving possible. The P4 and P5 architectures also provide a 48V-based all-wheel drive functionality. This article does not address a P5 hybridization with a 48 volt wheel hub motor in any detail as this concept is only suitable as a stand-alone drive for lightweight vehicles as far as Schaeffler is concerned.

P0 and P1 hybridization

P0 drives based on a 48 volt system allow a significant amount of braking energy to be recuperated when compared to 12 volt systems as well as a dynamic and comfortable start for the internal combustion engine. A further benefit of this architecture lies in its extended comfort functionality such as the stationary air-conditioning by engaging a switchable pulley decoupler. Due to the comparatively low cost of integration, this technology will be used in significant quantities in the coming years, particularly in Europe, and will make a decisive contribution to meeting CO_2 fleet emissions targets in the near future. Schaeffler is optimizing its existing systems (pulley decouplers, belt tensioners) with the aim of increasing efficiency and comfort. A range of methods are used to validate these systems including the technology demonstrator vehicle, the “Gasoline Technology Car I”.

In addition to the classic 12 and 48 volt hybrid in P0 arrangement, various iterations of accessory drives (Front End Accessory Drives, FEAD) are also being developed for combination with other hybrid systems, see Figure 2. These iterations range from a beltless engine, through a pure accessory drive without an electric motor, to a fully-fledged belt drive with integrated 12 volt, 48 volt or high voltage electric motor. In the latter, the main functions are starting the internal combustion engine and generating electricity. Depending on the functionality, this results in a modified load profile which must be taken into consideration during the design phase [3].

The arrangement of the electric motor on the crankshaft as a P1 hybrid provides a ratio-free link to the internal combustion engine speed. Due space limitations, Schaeffler always uses a permanent magnet synchronous motor (PSM)

		Standard without alternator	beltless	BSG 12 V	BSG 48 V	BSG HV
Micro 12 V	P0			✓		
Mild 48 V	P0				✓	
	P1	✓	✓			
	P2	✓	✓	✓		
	P3			✓		
	P4				✓	
Full HV	Px	✓	✓	✓		✓
		FEAD				

2 **Possible combinations of hybrid architecture and accessory drives on the internal combustion engine side**

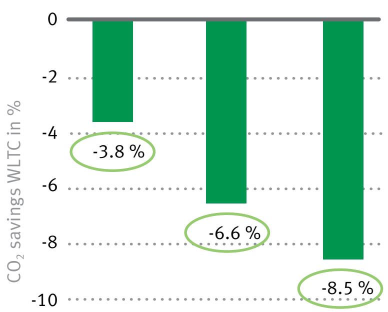
here which is characterized by both high efficiency and high torque capacity, particularly for cold engine starts. Unlike the asynchronous or claw pole machines used in P0 architectures, they bring a cost penalty with them. A P1 hybrid module is fundamentally similar to a coaxial P2 hybrid module. The only difference is that a launch device can be integrated in the rotor of the electric motor instead of the K0 disconnect clutch.

WLTC simulations completed by Schaeffler show that savings of 3.8 % are achieved in the P0 hybridization when using an asynchronous machine compared to a 12 volt micro-hybrid (smart alternator and start-stop function), see Figure 3. By replacing the machine with a powerful permanent magnet synchronous motor savings of 6.6 % can be reached. The P1 hybridization achieves savings of 8.5 % when using a PSM. The differences can be attributed to the efficient and powerful synchronous motors and the beltless design of the internal combustion engine in the P1 arrangement. In order to maintain comparability, no optimizations were carried out on the internal combustion engine in all the simulations. [4] shows however that further potential is available, particularly when combining P0 and P1 hybrids with a UniAir system and increasing the compression ratio.

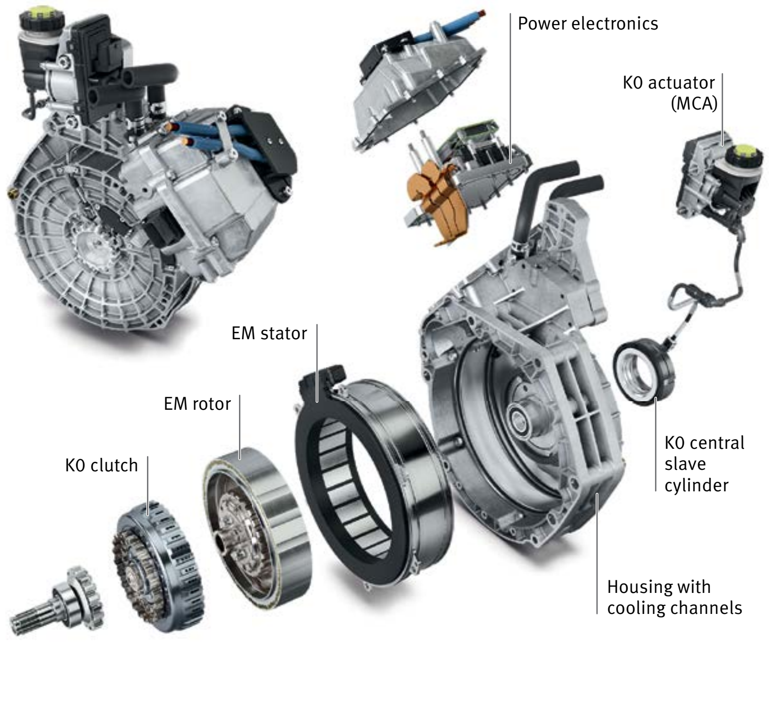
P2 hybridization

Schaeffler has been working successfully on high-voltage hybrid systems in a P2 arrangement for over ten years. An announcement was made in 2014 that this architecture is to move to a 48 volt system [5]. When compared to the high-voltage system, the 48 volt drive can be installed in both the coaxial and parallel axis designs as this arrangement makes use of electric motors which have been developed on a relatively modular basis and are already used in other architectures. The arrangement of the electric motor parallel to the drive axis is particularly relevant for front wheel drive, transverse power train designs as this does not extend the powertrain axially to any great extent. On the other hand, the coaxial design has a high degree of modularity to the high voltage P2 systems and also minimizes the friction losses as there are no force transmission elements when compared to the parallel axis arrangement.

Vehicle	C-Segment/FWD		
Drivetrain	Gasoline 3 cyl. 1.0 l/7-speed DCT dry		
Hybrid	Micro 12 V/smart alternator 2.5 kW		
	+ 48 V bordnet/1.4 kWh battery		
	P0	P0	P1
FEAD	48 V BSG	48 V BSG	beltless
Eta max (EM + PE)	84 %	94 %	94 %
EM @ 42 V/100 °C	ASM 8.5 kW (20 s)	PSM 16 kW (20 s)	PSM 15 kW (20 s)



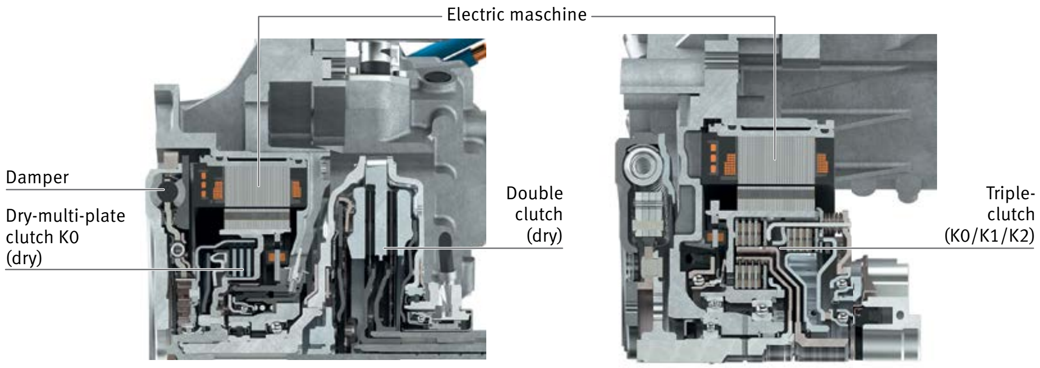
3 **Simulated CO_2 savings in a P0 and P1 hybridization compared with a basic vehicle in the WLTC**



4 **Design of the coaxial 48 volt P2 hybrid module with dry disconnect clutch K0**

The basic design of the Schaeffler coaxial hybrid module is identical to that of the high-voltage modules [6]. The disconnect clutch K0 can therefore be placed in the dry or oil chamber and either with or without a launch device. Figure 4 shows the construction of the 48 volt hybrid module with a dry multi-disk disconnect clutch K0. The electric motor and the clutch are located in the housing which also serves as a carrier for the power electronics and the modular clutch actuator “MCA” [7] for actuating the K0 disconnect clutch.

Figure 5 shows two typical installations for the 48 volt hybrid module in powertrains with double clutches. The hybrid module on the left-hand side is combined with a dry double clutch transmission. It supports the dry multi-disk disconnect clutch K0 and the electric motor rotor via a fixed housing wall. The housing wall also carries the slave cylinder (CSC) for actuating the clutch and the stationary component of the resolver. The link to a dry double clutch is provided by an intermediate shaft which has a series of splines on one end. All the force flow is retained internally due to the specific design with a fixed intermediate wall and double row bearings.



5 Cross-section through the coaxial 48 volt P2 hybrid module with damper on the engine side combined with either dry disconnect/double clutch option (left) or wet triple clutch (right)

The right-hand side of Figure 5 shows the same electric motor in a design with a wet triple clutch (K0, K1, K2). Due to the high degree of integration, this system offers even more significant potential to reduce the axial space further. This wet triple clutch in 48 volt hybrid modules is not discussed in any detail below: Further information can be found in [6] and [8].

The electrical system consists of a permanent magnet synchronous motor with a concentrated coil, a resolver and power electronics integrated into the housing [9]. The AC busbar is located inside the housing and cannot be accessed from outside. The system requires a low-temperature cooling circuit with a maximum inlet temperature of 85 °C and a volume flow of 6 l/min.

In the 48 volt design, the hybrid module with permanent magnet synchronous motor achieves a maximum output of 15 kW for 20 seconds and a continuous output of 10 kW when running off the motor. The dry disconnect clutch K0 has a torque capacity of 250 Nm. An additional 80 mm of axial space is required to integrate these elements: This is particularly easy to achieve when combined with a three-cylinder, front wheel drive, transverse application. The additional weight of the hybrid module of 31 kg must also be taken into consideration.

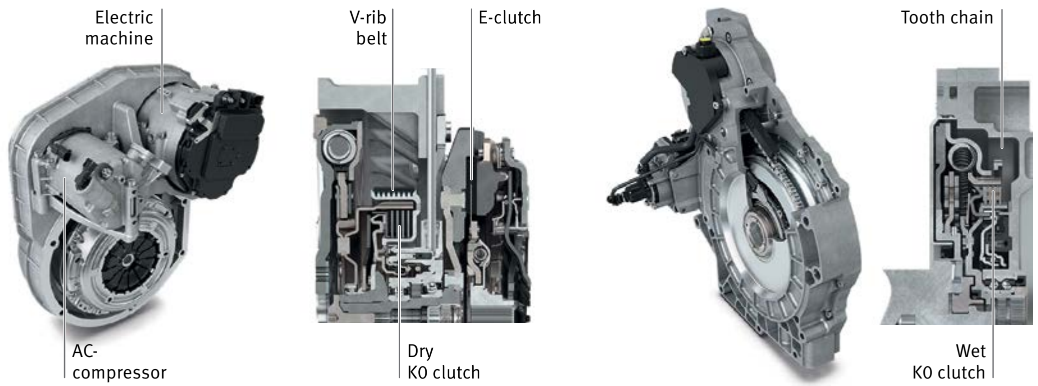
The coaxial installation means that there is no need for a belt drive on the internal combustion engine side as there is sufficient power for all start-up procedures can be comfortably accommodated by the P2 machine. This produces a further benefit in terms of CO_2 as there are no friction losses from the belt drive.

The parallel axis module uses the concept of moving the accessory drive, including the 48 volt starter generator, into the P2 position. The module consists of the subsystems of the electric motor, the K0 clutch system, the housing and there is an option for including an air-conditioning compressor and an integral K1 launch device. This type of hybrid module is in the process of vehicle validation in the technology demonstrator "Gasoline Technology Car II" in combination with a manual transmission.

The parallel axis module can be linked to the powertrain via a belt for dry systems or a chain for wet applications, see Figure 6. When used in combination with a belt drive, the same dry multi-disk clutch K0, as shown in Figure 5 for the coaxial system, can integrated in the pulley. The maximum torque capacity of a solution with a chain and wet clutch is up to 300 Nm. Even a parallel axis P2 drive needs additional axial space for the belt or chain drive but this can be restricted to 25 to 55 mm, depending on the design.

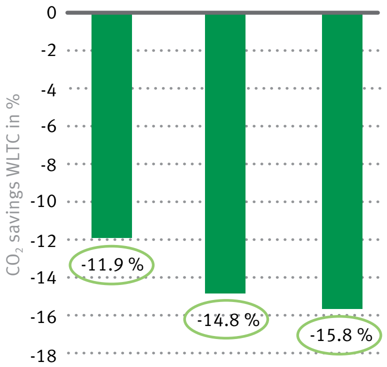
Arranging the drive component behind the damper significantly reduces the excitation from dynamic loads and provides maintenance-free operation for life; this is a necessary consideration at this position on the powertrain. The rotational speed of the internal combustion engine can be increased by a factor of 2.3 to 2.8 by the belt or chain drive which means that the high-speed electric motor can operate at maximum speeds of up to 18,000 rpm.

Simulations show that even 48 volt-based parallel P2 hybrids with a belt drive can achieve considerable CO_2 savings, see Figure 7. A saving of 11.9 % is achieved when using a relatively powerful permanent magnet synchronous motor and running the motor at 16 kW for 20 seconds. Schaeffler believes there is still potential in the parallel axis version to reduce the losses in the belt drive which are largely caused by the preload. The difference to the greater values of 15.8 % achieved with the coaxial concept is largely due to the belt losses. Furthermore, the difference can be attributed to the somewhat greater efficiency and the higher torque capacity of the coaxial electric drive system. The difference between a belt-free and a belt-driven internal combustion engine (without a generator/alternator) in a coaxial arrangement is approximately 1 %. The difference to a P1 hybrid with the same coaxial electric motor, see Figure 3, of 7.3 % is of particular note and this more than justifies the inclusion of a K0 system from a cost/benefit perspective. This significant difference can be explained by the greater potential for recuperation as well as the considerable electric driving at very low loading with a disconnected internal combustion engine, which is particularly



6 P2 hybrid module with belt (left) and chain (right) for parallel axis installation

Vehicle	C-Segment/FWD		
Drivetrain	Gasoline 3 cyl. 1.0 l/7-speed DCT dry		
Hybrid	Micro 12 V/smart alternator 2.5 kW		
	+ 48 V bordnet/1.4 kWh battery		
	P2 off-axis with belt drive (i = 2.3)	P2 coaxial	P2 coaxial
FEAD	beltless	Standard without alternator	beltless
Eta max (EM + PE)	93 %	94 %	94 %
EM @ 42 V/100 °C	PSM 16 kW (20 s)	PSM 15 kW (20 s)	PSM 15 kW (20 s)



7 **Simulated CO_2 savings by P2 hybrids when using a parallel axis and coaxial 48 volt hybrid module**

relevant in this vehicle segment.

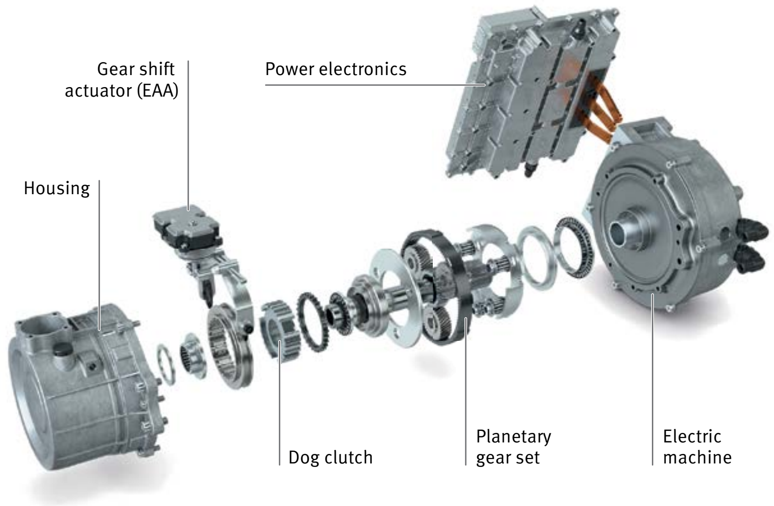
Concept for a P3 hybridization

The positioning of the electric motor on the transmission output (P3) is suitable for both in-line and transverse powertrains. Schaeffler is developing an additional module for vehicles with an in-line internal combustion engine and transmission. The P3 module, see Figure 8, has a coaxial design in order to make best use of the limited radial space in the tunnel beneath the passenger compartment, if required. The main transmission drive shaft is guided through the P3 module which not only saves space but also facilitates a modular manufacturing strategy. Schaeffler's modular strategy also allows the integration of a multi-disk clutch, if required, for an all-wheel drive variant.

The maximum performance of the electric, liquid-cooled electric motor is up to 20 kW in generator mode. The maximum output torque of the module in motor mode is 234 Nm with a peak torque on the electric motor of 60 Nm which is transmitted in a ratio of 3.9 via a single stage planetary gear. The downstream geometrically locked clutch is actuated via an electromechanical actuator and thus separates the electric motor from the powertrain if required. The maximum permissible speed of the electric motor is achieved at a vehicle speed of approximately 140 km/h. The dog clutch has to be opened at higher speeds. For a purely front wheel drive vehicle, the module weights 22 kg.

The hybrid module can also be used in an all-wheel drive vehicle with the addition of a multi-disk clutch. The clutch transmits up to 800 Nm and allows the cardan shaft to be connected and disconnected. When the clutch is open, the P3 module operates exclusively on the main transmission shaft and is therefore able to achieve the current hybrid functions. In all-wheel drive, the multi-disk clutch is closed and the P3 module acts on both vehicle axles.

If the P3 hybrid module with the electric motor described above, with 16 kW with the motor operating for 20 seconds, is integrated into a front-wheel drive vehicle and combined with a 12 V belt starter generator, the CO_2 emissions fall by 15.3 % compared to the basic vehicle, see Figure 10. Although the system only has a transmission and a neutral gear, it can still achieve high values. These can be explained by the targeted sizing on the WLTC and the avoidance of zero-load losses due to the neutral gear. However, in the single-speed solution, there are certain limitations with respect to the achievable CO_2 savings in real driving conditions and on a functional level compared to a two-speed solution due to covering such a large speed range. If the module is combined with a mechanical all-wheel drive system, which is possible due to the multi-disk clutch option, the additional all-wheel losses which occur compared to a purely front-wheel drive, can be more than compensated.

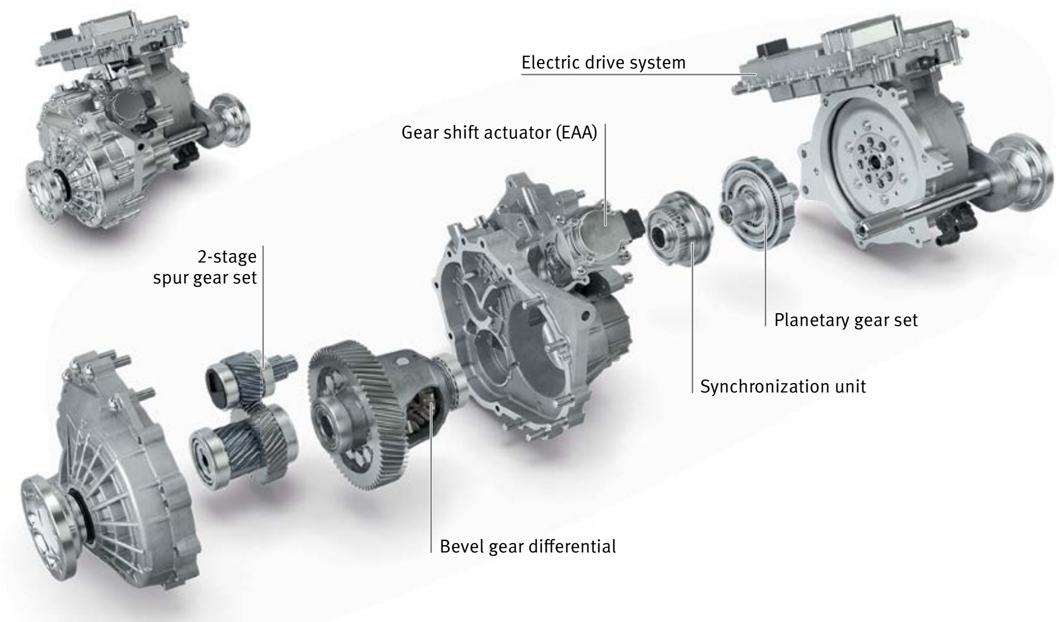


8 **Construction of the single-speed (1+N) hybrid module for a 48 volt P3 hybridization (without optional multi-disk clutch for AWD)**

P4 hybridization

The P4 architecture allows the mechanical all-wheel drive to be replaced by an electric drive system on the rear axle which supplements the internal combustion engine acting on the front axle. The significant benefit of this architecture is the representation of an all-wheel drive as well as the complete disconnection of the spaces for the motor/transmission link and the electric drive. Schaeffler is in the process of validating a corresponding architecture in the "High Performance 48 V" technology demonstrator.

A two-speed transmission which can be switched via a planetary gear set with a downstream gear stage and a bevel gear differential allows the electric motor to be adjusted to its optimum setting in a range of driving situations, see Figure 9. A classic selector sleeve and an electromechanical actuator "EAA" are used to switch between first, second and neutral gears [7]. In first gear (i = 15), the electric driving and recuperation functions are the main point of focus. With a 48 volt systems, axle torques of up to 1,200 Nm can be produced purely electrically at low speeds depending on the maximum torque of the electric motor. Furthermore, the axle is designed such that purely urban electric driving is also possible. Second gear (i = 5) is selected when the vehicle speed reaches approximately 70 km/h. Second gear is therefore aimed at interurban and freeway driving during which the focus is exclusively on the boost and recuperation functions. Which electric motor technology to use is generally an open decision and is highly dependent on the application. Maximum output, power density, space and functional safety are decisive factors here.

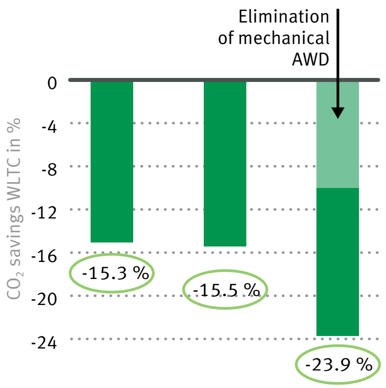


9 Construction of two-speed (2 + N) electric axle for a 48 volt P4 hybridization

The total weight of the electric axle is approximately 40 kg, depending on the electric drive system chosen [10; 11].

A 48 volt belt starter generator is used in the internal combustion engine component of the powertrain for the WLTC simulation of the P4 hybrid as this guarantees the availability of the all-wheel

Basic vehicle	C-Segment/FWD		C-Segment Conv. full AWD
Drivetrain	Gasoline 3 cyl. 1.0 l/7-speed DCT dry		see left
Hybrid	Micro 12 V/smart alternator 2.5 kW		see left
	+ 48 V bordnet/1.4 kWh battery		see left
	P3 1-speed	P4 2-speed	P4 2-speed
Fahrzeug	FWD	AWD "light"	AWD "light"
FEAD	12 V RSG	48 V RSG	48 V RSG
Ratio to wheel	i1 = 10 / N	i1 = 15 / i2 = 5 / N	i1 = 15 / i2 = 5 / N
EM @ 42 V/100 °C	PSM 16 kW (20 s)	PSM 15 kW (20 s)	PSM 15 kW (20 s)



10 Simulated CO_2 savings on a P3 hybrid using a single-speed hybrid module on the transmission output and a P4 hybrid (based on FWD and AWD) with a two-speed electric axle

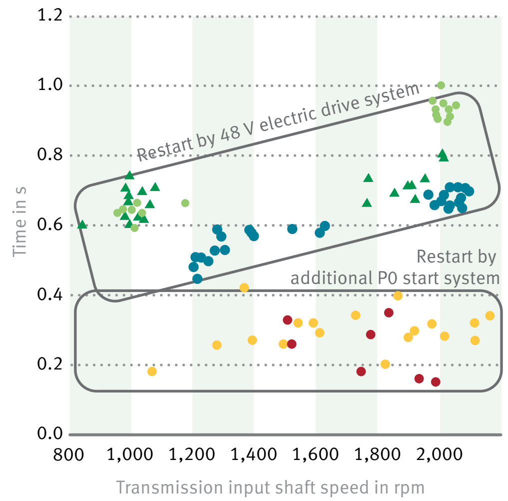
drive even when the battery has a low charge. This is based on a front-wheel drive variant and an all-wheel drive variant on a typical C segment vehicle. In this case, the increased consumption due to the use of a conventional all-wheel drive is approximately 11 %. When using an electric rear axle without a direct mechanical drive on a 48 volt basis, the emissions balance improves by 15.5 % compared to a front-wheel drive vehicle and by almost 24 % compared to an all-wheel drive variant, see Figure 10.

Functional optimization

The CO_2 savings to be achieved with 48 volt hybrid drives can largely be traced back to the good recuperation capability of this type of system. The electrical energy won is used in as many driving situations as possible either for boosting, purely electric driving or at the very least for maintaining the desired speed (active sailing). This means that the internal combustion engine will be disconnected and connected between 600,000 and 900,000 times over its lifetime and depending on the driving strategy adopted, particularly with P2, P3 and P4 hybrids. The comfort and dynamics of restarting are therefore particularly important for the acceptance of 48 volt hybrid systems. This criterion is defined by a time period which elapses from a specific action (depressing the accelerator pedal) until the acceleration felt on the rails of the driver's seat reaches a value of approximately 0.25 m/s2. This value corresponds to the acceleration immediately noticeable to the average driver.

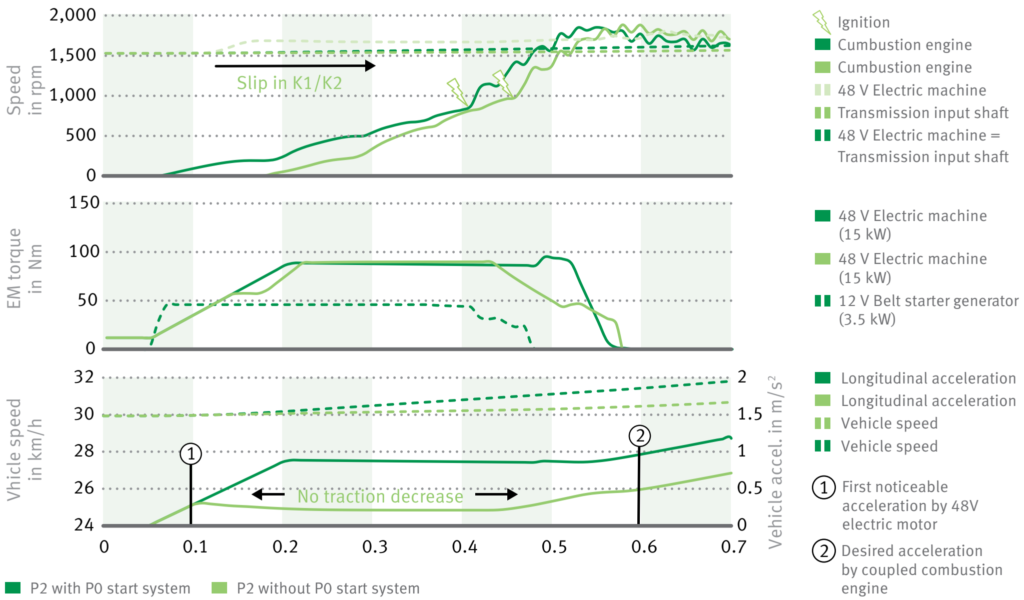
If a 48 volt drive without an additional starter system is used, the time to reaching this threshold value is purely dependent on when the internal combustion engine reaches the speed of the transmission input shaft and transmits the torque to the launch clutch. In the current prototypes, times of 0.5 and 0.8 seconds are achieved in P0 and P2 hybrids based on sailing with the motor disconnected and this is regarded as acceptable for volume production. The combination of a P0 starter generator as a pure additional starter system with a P2 and P4 hybrid leads to a significant shortening of the time period to 0.2 to 0.4 seconds, see Figure 11.

The difference in values shown in Figure 11 for the 48 volt drive with and without an additional starter system indicates that the design of the system has to be looked at in more detail if the architec-



48 V P0	Sailing with open K1	Demonstrator (MT) "Gasoline Technology Car 1"
48 V P2 + beltless ICE	Sailing with open K1	Demonstrator (MT) "Gasoline Technology Car 2"
	Sailing with open K0	
48 V P4 + 48 V P0	Sailing with open K1/K2	Demonstrator (DCT) "High Performance 48 V"
HV P2 + HV P0	Sailing with open K1/K2	Series (DCT) "Benchmark"

11 Measured times to achieving an acceleration from sailing of 0.25 m/s² on the rails of the driver's seat for various 48 volt architectures with and without additional starter system



12 Simulation of a restart from electric driving with a 48 V P2 hybrid with and without additional starter system

ture does not have a second electric motor. This includes, for example, the speed of the transmission input shaft when combined with automatic transmissions with a view to keeping possible restarts to a minimum in all driving situations. There is also an option of minimizing the compression torques when moving the internal combustion engine, for example by using a fully-variable drive [4].

When restarting from electric driving, as the example of a P2 hybrid shows in Figure 12, the driving strategy should be selected such that the electric motor always has torque in reserve for restarting electric motor. Furthermore, the disconnect clutch must also ensure there is a corresponding torque control accuracy for restarting. The challenge when starting from electric driving, compared to restarting from sailing, is to maintain the tractive force such that the driver is unaware of any negative gradients in the vehicle acceleration. The times for P0 hybrids which are now deemed acceptable for connecting the internal combustion engine must also be achieved. A P2 architecture with additional starter system allows greater initial acceleration as the electric motor does not have to provide any torque for moving the internal combustion engine. However, architectures with an additional starter system must also ensure that the driver's requirements can only be fulfilled in full once the internal combustion engine is connected, see Figure 12. In addition to this, it should be noted that if the traction motor and belt starter generator have the same voltage, the power which is taken off in parallel during restarting must also be taken into consideration when sizing the battery.

Summary and outlook

Comprehensive simulations have shown that sig-

nificant CO_2 savings can be achieved with a 48 volt hybrid drive. The savings are dependent on the architecture chosen and it is essential that they are evaluated from a cost-benefit perspective. Furthermore, the required functional scope, such as all-wheel drive, and the available space must also be taken into consideration when selecting the architecture. Looking ahead to the more stringent CO_2 limits which are to be expected in the next decade, P2, P3 and P4 architectures represent a further attractive type of hybridization from a cost-benefit perspective, and the fact that they are based on established P0 and P1 systems, and will take their place in the range of hybrid architectures.

In the past, Schaeffler has built a range of different concept vehicles to demonstrate the system behavior of 48 volt drives under real driving conditions. Schaeffler is currently working with partners on a new technology demonstrator which will combine a coaxial 48 volt P2 hybrid module with a three cylinder engine and a double clutch transmission in a front wheel drive transverse arrangement. The reduction in CO_2 emissions actually achieved with this drive concept will be published at a later date.

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P2 High-Voltage Drives

Efficient Hybridization for all Transmissions

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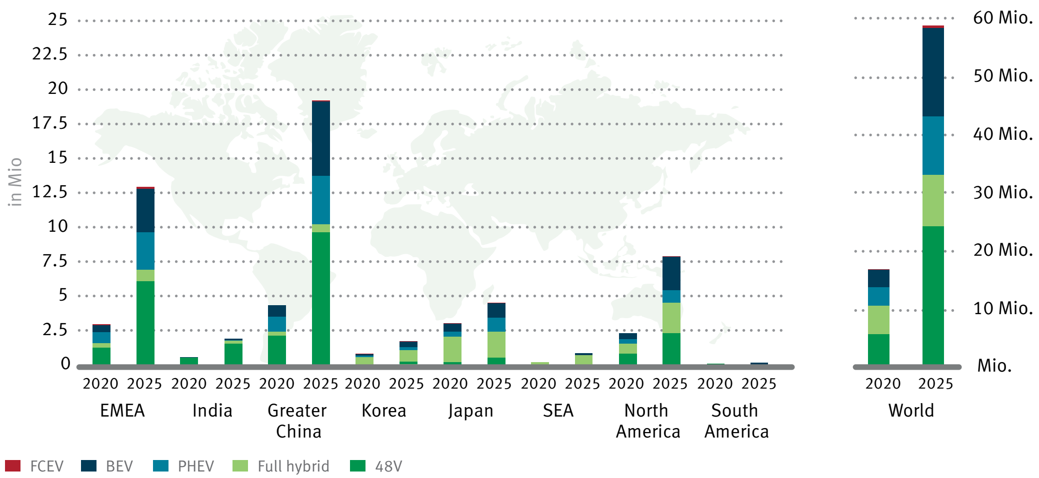
Willi Ruder
Martin Dilzer
Cedric Blaes

Introduction

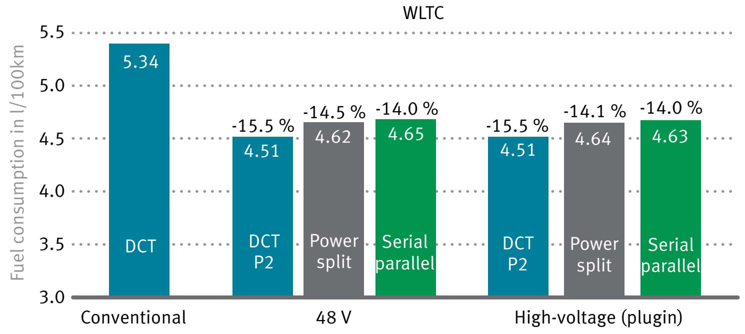
The answer to the question, which technologies will prevail in the electrification of the powertrain is largely dependent on future statutory requirements and developments in the costs of the relevant systems. The statutory regulations vary considerably from region to region throughout the world. Not only do fleet CO_2 targets have to be taken into consideration but also other factors such as establishing fixed quotas for vehicles with alternative drives. If this quota includes plug-in hybrid vehicles with a specific minimum range, as is the current situation in Chinese law, this can lead to high market share for these drives by 2025, see Figure 1. As the fleet targets are being tightened in China, it also is reasonable to expect increased volumes for the 48 volt technology. A similar trend can be found in Europe without the quota system due to stricter CO_2 targets. In Japan and the United States of America, on the other hand, the full hybrid without an external charging option is the dominant technology. 48 volt technology is unlikely to achieve significant distribution in these two markets for various reasons: In Japan, high-voltage technology has already been transposed to many vehicle platforms. In the USA, high vehicle weights limit the use of low voltage hybridization.

Considerable CO_2 savings are already possible with a 48 volt hybridization [1]. In the following comparison with a plug-in high-voltage hybridization in the WLTC (trickle charge), the installation position of the electric motor, transmission type and other vehicle parameters remain as they were. A battery capacity of 9.0 kWh is selected for the simulation so that an electric range of 50 km can be achieved. A consumption benefit of 15.5 % is achieved with the plug-in hybridization compared to the basic vehicle with double clutch transmission. Both the drive with power split and the serial parallel hybrid achieve very similar reductions of approximately 14 % and therefore not the potential of the P2, see Figure 2.

In vehicles designed as a full hybrid, a further reduction in the consumption is very limited due to the lower battery mass compared to the plug-in



1 **Volume scenarios for electrified drives in various global regions**



- P2 in combination with DCT in this configuration is the most efficient driveline with -15.5 %
- Due to higher battery energy density as well as operation in better efficiency areas, the penalty of the PHEV is considerably reduced
- The power split and the serial hybrid drivelines are nearly on the same level

Basis of the analysis	
Vehicle	C-segment
Weight	1,500 kg (konv.) 1,555 kg (Full hybrid) 1,616 kg (PlugIn)
Engine	1.5 l 4 Cyl. T
E-Motor	50 kW (Full hybrid) 90 kW (PlugIn)
Battery	3 kWh (Full hybrid) 9 kWh (PlugIn)
Rdyn	0.307 m
cw*A	0.59 m²

2 **Simulated fuel consumption for various high-voltage hybrid systems in the WLTC**

hybrid in trickle charge mode. The main reasons for the reduction are the greater output density of the battery cells and thus a reduction in the additional vehicle weight as well as the improved operating ranges with the larger 9 kWh battery in which the power flows for the relevant cycle take place.

The target saving from this perspective has very little to do with whether a continuously variable transmission or a double clutch transmission is used. However, the transmission's efficiency level is definitely extremely relevant for the efficiency of a P2 arrangement as can be seen when compared to other transmission concepts which operate with a power split or, for reasons of design, do not make all gears available for transmitting the torque provided by the electric motor.

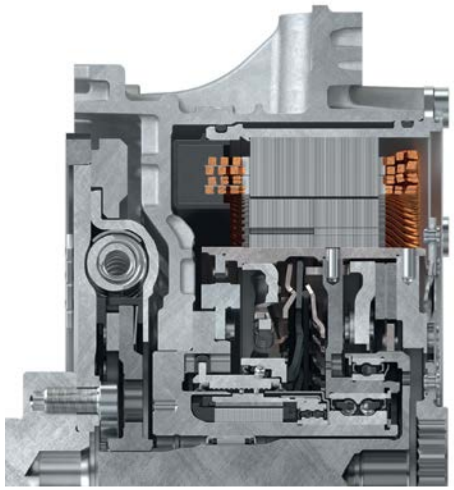
To illustrate this point, the consumptions of a power-split and a serial parallel transmission are shown as a comparison to the P2 hybrid. The consumption of a P2 drive based on electrification of a double clutch transmission in a four-cylinder, C segment vehicle is improved by optimizing the operating strategy of the individual drives.

Hybrid modules

Second generation hybrid module

Schaeffler has been supplying important components for a P2 arrangement volume hybrid module since 2010. The concept for a hybrid module was first presented at the Schaeffler Symposium, 2014, which takes account of the trend towards constantly rising output density [3]. This module was put into volume production at the end of 2017.

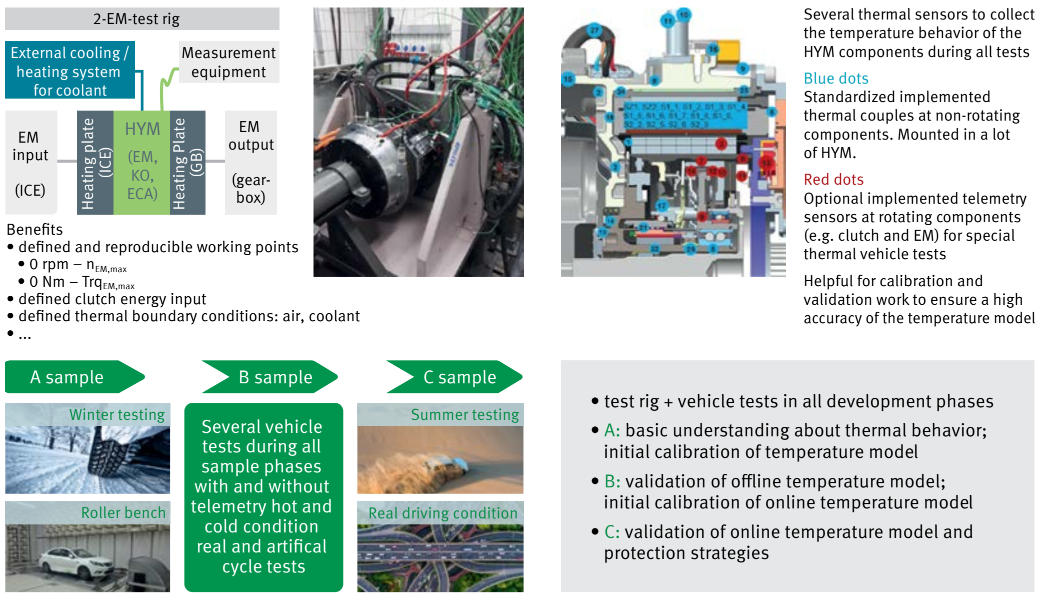
Hybrid variants are currently present in only small quantities in nearly all markets. The second generation hybrid module is therefore designed in such a way that it can be relatively inexpensively adapted to different internal combustion engines and transmissions. The core is a permanent magnet electric motor in various output levels, see Figure 3. By using a modular system, different outputs can be achieved by varying the active length and using



3 Cross-section through the second generation hybrid module in a dry clutch design

two external diameters of 270 and 300 mm. The electrical output is 82 kW and the maximum torque 300 Nm for the first volume production run.

The second generation hybrid module is characterized by a very compact design. Both the resolver for capturing the angular position of the rotor and the complete disconnect clutch K0, including the relevant actuation technology, are integrated into the rotor and are therefore accommodated within the structural length required by the electric motor. The clutch and actuator are designed to last for the life of the vehicle which, for plug-in hybrids, corresponds to approximately 650,000 actuations of the K0 clutch. A special clutch facing was developed in order to achieve the necessary wear reserve in this compact space. It is possible to connect a double clutch directly to the output side. The bearing support for the transmission input shaft can also be integrated if necessary. Due to the high target output density, the issue of thermal management is given particular attention from the very start of the volume development process. Schaeffler has therefore simultaneously developed a thermal model which simulates the source and transmission of the heat to the rotor,



4 Comparison of the thermal model during validation

stator and other components in the hybrid module as a function of the torque requirement, see Figure 4.

The model was initially produced in MATLAD-Simulink and then validated using measurements from test vehicles. The model is now of a sufficiently high quality that it can be placed in the vehicle's hybrid control device and used to control the drive.

The acoustic behavior of the hybrid module is a further priority in volume development. In order to ensure the attained level is maintained under volume production conditions, the hybrid module is subjected to 100 % testing after assembly. During testing, measurements are taken from the electric motor in both regeneration mode and under load in driving mode. The integrated actuation technology for the disconnect clutch is actuated during the test with a range of actuation travel distances.

Third generation hybrid module

Schaeffler intends to increase the degree of integration in the third generation of the hybrid module which is currently under development and thus to reduce the space requirement even further. In future, the module will also include the launch device. Schaeffler is currently developing a hybrid module with an integrated torque converter for an imminent volume application in the US market, see Figure 5. A significant part of the damping function in the converter is no longer required as there is a separate torsional vibration damper in front of the module. The converter can therefore have a considerably more compact design.

A second vibration damper, which can be partially integrated into the hybrid module, is required when this is used with diesel engines, and some-

5 Hybrid module with integral converter

times with highly loaded gasoline engines. The subsequent spatial gain with this type of design compared to using a P2 module with a classic converter is 30 to 50 mm.

A further increase in the output density, particularly with respect to continuous output, is possible due to improved heat dissipation. In plug-in hybrid vehicles, the thermal load registered on the stator is up to 2.8 kW. In the third generation, there is an option of connecting the hybrid module directly to the transmission oil circuit as an alternative to liquid cooling the stator.

The third generation of the Schaeffler hybrid module with integrated converter is currently in volume development and will be in volume production by the end of 2018.

The next stage in the evolution is to operate the rotor completely within the wet space. Combining this with a wet double clutch produces high integration potential, see Figure 6. Even at lower electric outputs, which are achieved within the concept by reducing the length of the motor, it is still possible to

6 **Cross-section through the complete oil-cooled hybrid module with integral triple clutch**

accommodate the disconnect clutch and the double clutch in the rotor. This produces a significant reduction in the axial space requirement.

The design shown in Figure 6 draws on a high pressure actuator concept in which each of the clutches is actuated by a CSC actuator (Concentric Slave Cylinder). Further details on these clutches and the actuation technology can be found in [4].

The oil feed is based on the principle of “from inside to outside”. The oil flows into the rotor space via the transmission input shaft and an additional hole in the housing of the hybrid module. Here, the heat from the clutches is initially dissipated and then the magnets on the rotor and the stator are cooled. The thermal characteristics of all the output levels currently offered by Schaeffler for P2 hybrid modules up to peak outputs of 125 kW can be controlled with this concept.

The bearing support concepts for the module variants shown in Figure 5 and Figure 6 are distinguished by the fact that a simple bearing support beneath the disconnect clutch is adequate for the converter module as a second bearing support on the converter collar is provided on the transmission input side. The triple clutch module is guided directly under the electric motor rotor by two mutually opposed ball bearings. A further bearing point at the transmission input is no longer required.

Further development of the electric drive for hybrid modules

Fourth generation electric drives

The concept of the hybrid module includes a new generation of electric drives which currently have an electric output of up to 85 kW and a maximum torque of 330 Nm. The output of the permanent magnet electric motor with an active length of approximately 60 mm will increase to up to 100 kW in the future which will allow a maximum torque of up to 350 Nm. The output density therefore increases compared to the current generation by 10 to 15 % depending on the design. The electromechanical characteristics of these drives are described in [5].

The mechanical integration of the electric motor into the relevant hybrid module and thus the vehicle powertrain is a decisive factor for comfort and operating life. This includes the connection for the power electronics; Schaeffler’s solution here is to assemble standardized output modules into the hybrid module housing, see Figure 7. The copper wiring from the terminal and stator is initially welded and then screwed to the connecting terminal on the power electronics. This provides an electrical contact which is watertight and sealed against dust.

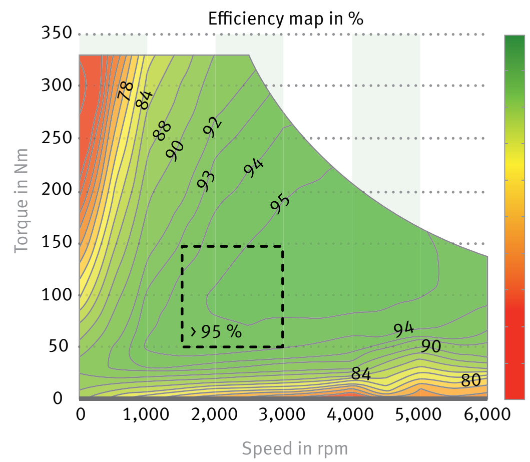
In a hybrid drive, the efficiency of the electric drive has a considerable influence on the consumption

7 **Hybrid module with integral power electronics**

savings achieved in trickle charge mode in the WLTC. It is also an important influencing factor when sizing the electric accumulator and this has a direct impact on the cost of the system. The new generation of Schaeffler electric motors achieves an efficiency of 96 % at its optimum operating point. It was therefore possible to noticeably increase the size of the data map in which the drive operates with an efficiency of more than 95 %, see Figure 8.

Axially parallel variants on the hybrid module

P2 hybrid modules with an electric motor arranged parallel to the axis reduce the axial space requirement and are therefore suitable for transverse front-wheel drive concepts. Schaeffler has developed a solution for 48 volt hybrid systems in which the torque is transmitted via a V-ribbed belt. Options for



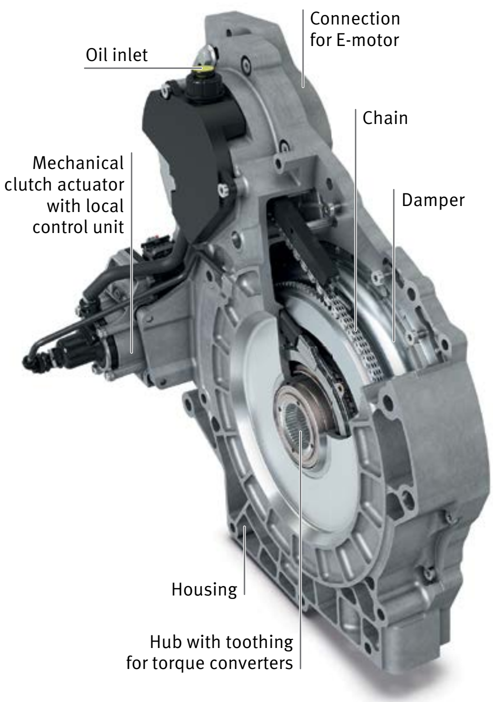
Boundary conditions	
Duration	≤ 20s
Cooling liquid	65 °C
Temperature magnets	120 °C
Temperature statorwinding	150 °C
EM phase-phase peak amplitude voltage of first harmonic	328 V_{peak}
EM max. phase current (≤ 20s)	400 Arms
EM max. phase current (cont.)	185 Arms
Max. power losses of e-motor for continuous operation	3.4 kW

Drag torque up to 0.6 Nm
from bearings and clutches excluded

8 **Efficiency level data map for a third generation electric motor**

transmitting the significantly greater torques which are produced by high-voltage modules are a chain or, alternately, a gear stage. Both of these can transmit the required outputs of up to 80 kW. The greater permissible circumferential speed of a chain compared to a belt also makes it possible to design the chain guidance around the clutch. The chain runs on the drive chain side on a diameter of more than 200 mm in an existing hybrid module. Figure 9 shows the essential subsystems of this module which has been designed for a torque of up to 400 Nm from the internal combustion engine side.

The module, which is supplied prefilled with oil, has been designed to make assembly on the vehicle manufacturer's vehicle or engine line as easy as possible. In this specific design, the damper,



9 Subsystems of a high-voltage hybrid module for axially parallel installation

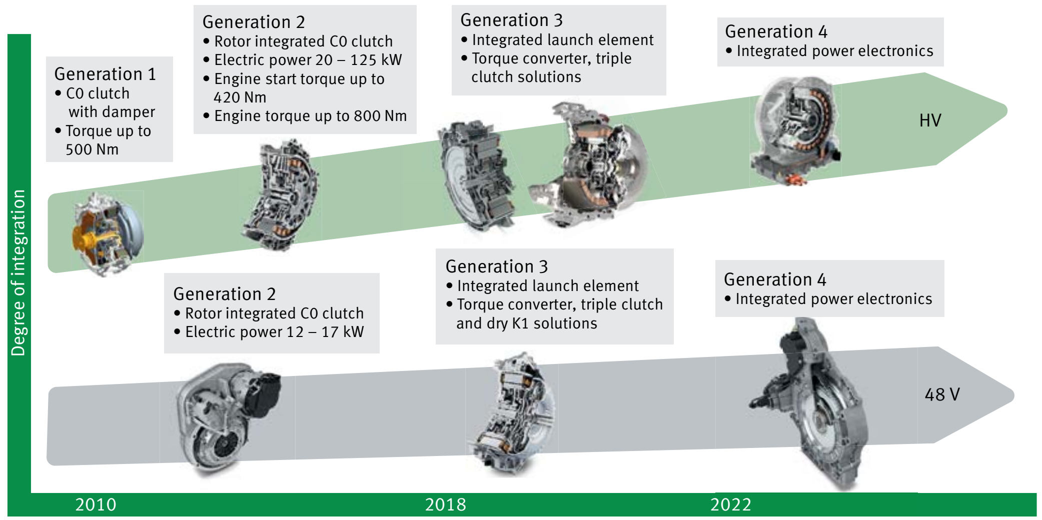
with integral centrifugal pendulum-type absorber, is pre-assembled in the module. For technical reasons related to oscillation, it makes sense to arrange the damper in front of the chain drive. The wet disconnect clutch between the damper and the chain drive is actuated via an integral electro-hydrostatic Modular Clutch Actuator (MCA) which is integrated directly into the module. The gearing on the transmission side of the module can be determined by the type of transmission to which it is to be combined.

The subsystems are arranged in such a way as to save as much space as possible. The disconnect clutch K0 is located directly behind the damper. The hydrostatic actuator for actuating the K0 is integrated into the envelope. The chain drive completely surrounds the clutch. The actuator on the housing for the main clutch K1 requires no extra axial space. The bearing support for the system is a double-row arrangement supported on the transmission side of the housing.

A second chain is provided for this module which drives the air conditioning compressor. This provides it with a direct mechanical drive from the electric motor and it does not have to be supplied as a completely electrified high-voltage component.

Summary and outlook

The P2 hybridization represents an attractive option for the future electrification of existing drive architectures by integrating a P2 module between the internal combustion engine and the transmission. With the second generation of the hybrid module, which is now in volume production, Schaeffler has succeeded in increasing the level of integration and therefore the output density even further. The third generation, which should be in volume production by the end of 2018, continues this development by integrating the launch device into the module. A particular feature is



10 Evolutionary phases for the Schaeffler modular system for P2 hybrid module

the further development of the individual subsystems as well as the optimum design of all the available space.

The electric motor offers a great deal of further potential due to the further increases in its output density and efficiency. The power electronics are integrated into the housing of the hybrid module as the next logical step, thus reducing costs further.

Moreover, Schaeffler is also introducing an attractive, solution for transverse front-wheel drive vehicles with reduced space requirements in the form of a chain-driven, axially parallel high-voltage module.

Schaeffler will continue to develop the modular system of P2 hybrid modules in the coming years, see Figure 10. Depending on their suitability, new basic technologies will be used for both high-voltage and 48 volt hybrid drives. The fourth generation of hybrid modules is already in the pre-development stage. Integrated power electronics will make even more compact designs possible in the next decade and simplify integration in the overall powertrain even further.

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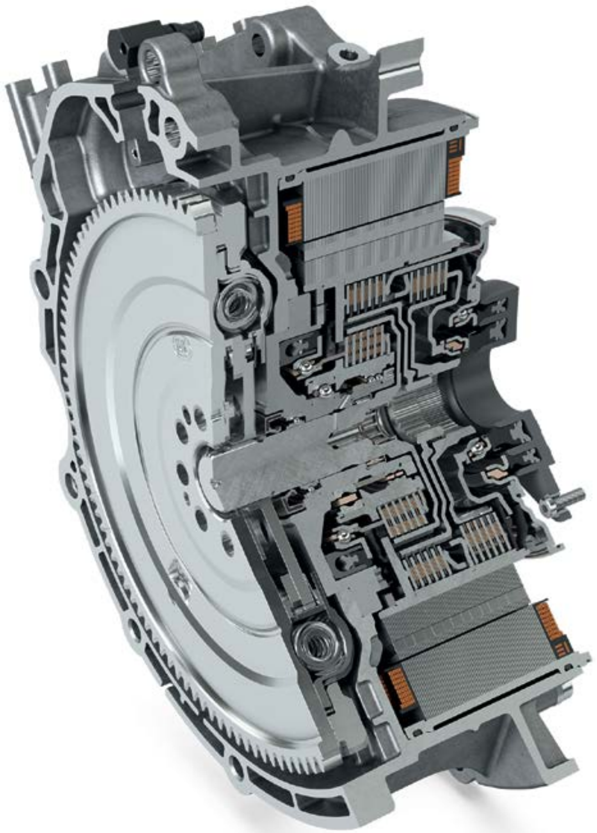


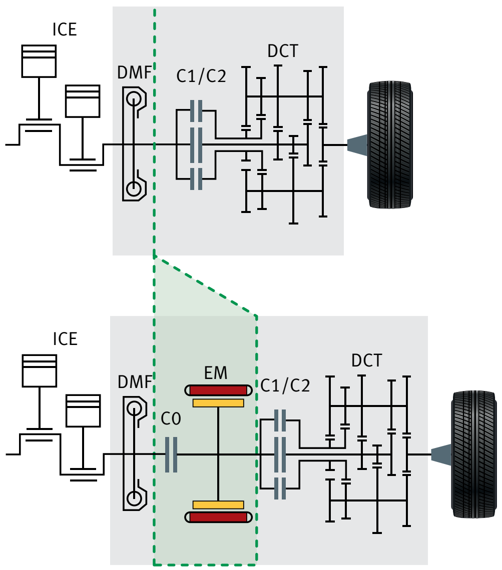
The Top 3 of P2

Space, Space, Space

Dipl. Ing. Joachim Hoffmann
Dipl. Ing. Karl-Ludwig Kimmig
Dr.-Ing. Andreas Baumgartner
Dr.-Ing. Knut Erdmann
Dipl. Ing. Wolfgang Haas
Dipl. Ing. Philippe Wagner

Introduction

By introducing double clutch transmissions, it has not only been possible to increase driving dynamics and comfort in the last ten years, but also to make great strides in the efficiency of modern vehicles. In order to further reduce climate-damaging CO_2 emissions and achieve future limiting values, however, it will be necessary to slash fuel consumption levels even more. A key concept for this is P2 hybridization combined with a double clutch transmission [1]. In P2 hybridization, a powerful electric motor is installed (coaxially or even axially parallel) between the combustion engine and the transmission – Figure 1. In addition, an automated C0 disconnect clutch is needed so that the electric motor can also be used independently of the combustion engine for pure electric driving.



ICE Internal combustion engine
DMF Dual mass flywheel
EM Electric motor
C0 Clutch for ICE
C1/C2 Clutch 1/clutch 2
DCT Double clutch transmission

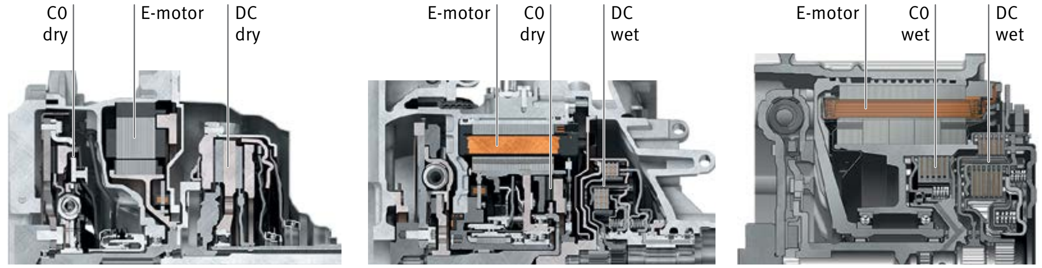
1 Schematic diagram of P2 hybridization in connection with a double clutch transmission

As explained in detail in [2], the particular advantage of combining a double clutch transmission with a P2 hybrid system rather than a P2.5 hybrid system with fixed coupling of the electric motor and a sub-transmission is that the gear ratio function of the double clutch transmission can be used in all operating modes (purely electric driving, hybrid mode, and battery charging). This results in a high level of overall energy efficiency. It is also possible to drive in any situation without interrupting the tractive force, making for extremely comfortable driving.

Particularly in smaller vehicles with a front-transverse powertrain, the big challenge posed by implementing P2 hybridization of double clutch transmissions is integrating it in the available installation space. This article therefore examines innovative further developments for dry and wet clutch systems in order to minimize the axial length of P2 hybrid drives. Since the overall length of the transmission is also heavily dependent on the synchronizations, Schaeffler has developed solutions for shortening the synchronization devices, which are presented below as well.

State of the Art

As an add-on concept, P2 hybridization only requires average interventions in the transmission structure, and it enables modular electrification within existing drive modules. The upshot of this is that it has already been possible to combine several hybrid powertrains with a double clutch transmission in series production in recent years. The systems shown in Figure 2 are different with respect to the arrangements and designs of the three clutches required for the system. For example, the system shown on the left shows an axial arrangement of the electric motor and disconnect clutch



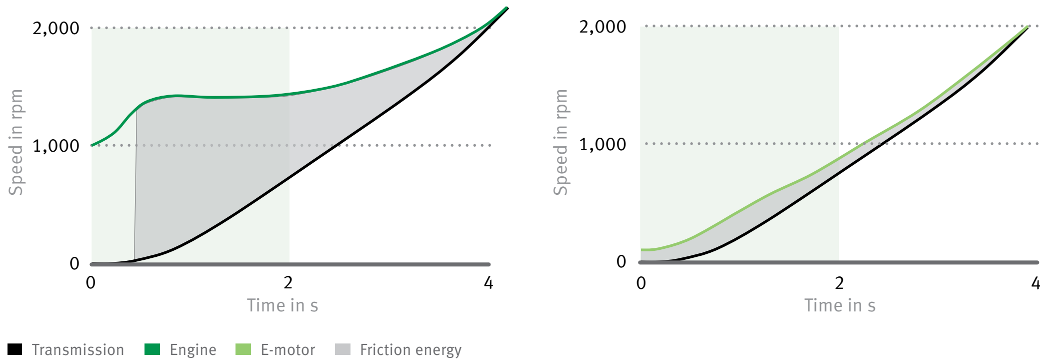
2 Implemented series solutions for P2 hybridization of double clutch transmissions

for a dry double clutch transmission. The representation in the middle shows the Schaeffler hybrid module with the dry C0 integrated in the rotor, which is used in combination with a wet double clutch. In contrast, the system on the right depicts three wet clutches that have already been very compactly coupled with the electric motor of the hybrid module.

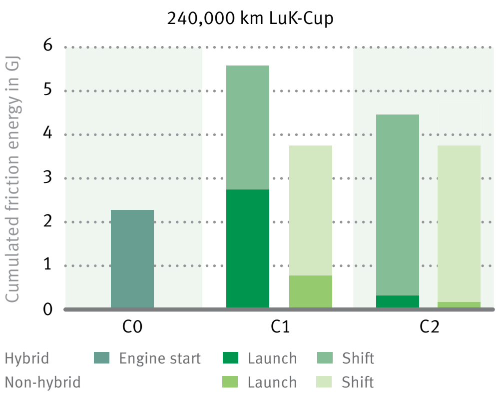
What all three systems have in common is that the axial length of the transmission is 80 to 120 mm longer than in a non-electrified transmission because of the hybrid module. The increase in length in a particular application is heavily dependent on the electrical output and thus on the size of the electric motor and the axial extension of the clutch system. The challenge for future solutions is to greatly reduce the additional need for installation space and, at the same time, satisfy the demands for operating life and torque capacity. Derived from this are fields of activity for optimizing the installation space in the clutch system and within the transmission.

System Assessment

The demands placed on the clutch system change through the electrification of the powertrain. If a high-voltage hybrid module is used in a P2 arrangement, then it is possible to use the



3 Clutch slip during vehicle launch with a combustion engine (left) and an electric motor (right)



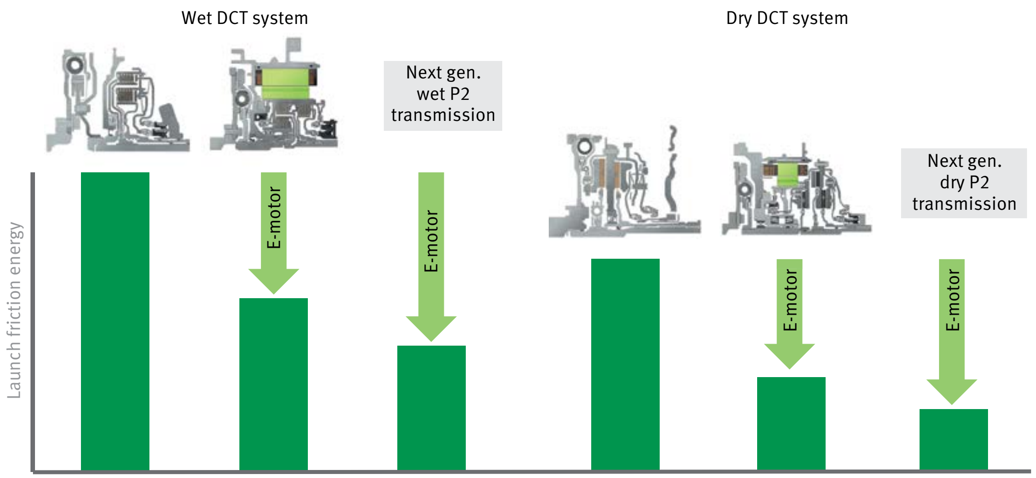
4 **Friction energy of the clutches for double clutch transmissions, with and without P2 hybridization**

electric motor to launch a vehicle even without significant clutch slippage. Depending on the battery capacity and battery management, the result is a considerably reduced clutch load. This is clear when the integral of the speed difference between the engine and transmission input shafts of a classic double clutch transmission is compared with that of a P2 hybrid transmission during the launch phase – Figure 3.

However, this raises new challenges. For example, the thermal power loss of the electric motor needs to be taken into account when designing the clutch. Moreover, additional friction energy is produced in the C0 disconnect clutch due to repeated engine starts while driving.

In order to quantify the changing requirements, Schaeffler conducted extensive measurements and simulations in the course of series development projects. The goal was to determine the cumulative friction energy registered for all three clutch types within a realistic driving cycle for each specific driving situation. Figure 4 shows the result of such measurements for a double clutch transmission with vs. without P2 hybridization.

Based on the measurements, it is evident that the current P2 double clutch transmissions in series production have already reduced the clutch load by 50% – particularly during launch. By developing the systems further, including with respect to optimum system dimensioning, and the output of the combustion engine, the electric motor, and the battery capacity, the possibility of further reducing the friction power and the friction energy in the triple clutch system can be assumed. Friction power and friction energy are important factors when designing a clutch and directly affect the size of it. In future P2 hybrid system generations, it will be possible to accomplish further reductions in the friction energy load with an even more reduced need for installation space through smaller clutches. As with today's non-hybridized double clutch transmissions, it is possible to employ wet and dry clutch solutions as a rule, whereby wet clutch solutions are generally used for larger and heavier vehicles, with dry solutions utilized for smaller and lighter vehicles – Figure 5.



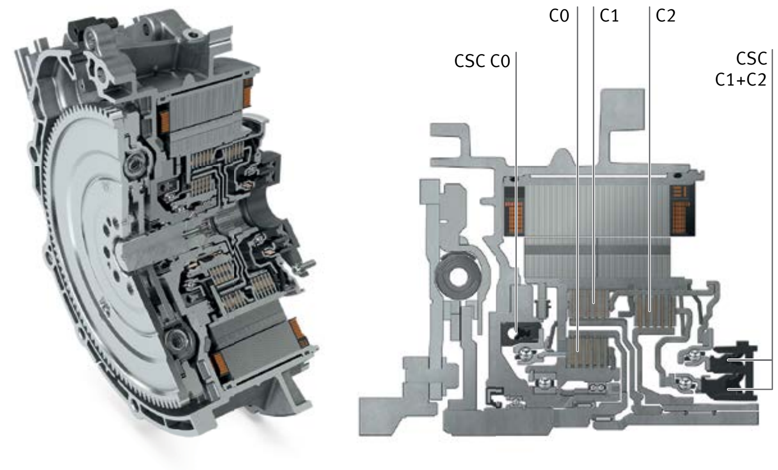
5 **Energy needed for vehicle launch with various stages of electrification of wet and dry double clutch transmissions**

Wet, Rotor-integrated Triple Clutch System

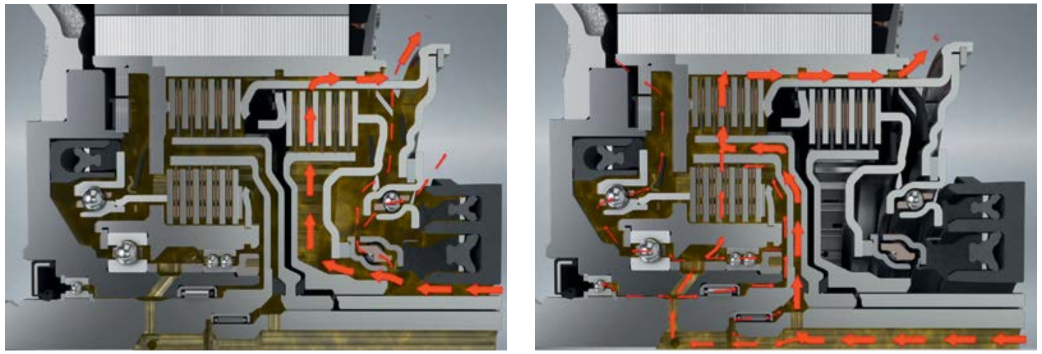
Schaeffler has developed a wet triple clutch system to series-production readiness according to the above-mentioned requirements with the C0, C1, and C2 clutches. It can be integrated almost completely in the rotor of the electric motor, thereby fitting in the latter's overall length [2]. The complete system consisting of the hybrid module, triple clutch, and transmission is very compact. For this reason, in comparison to non-rotor-integrated solutions, the axial transmission length can be reduced by 50 to 70 mm – Figure 6.

The basic design concept involves the radial and axial nesting of a wet triple clutch with the corresponding actuator system as much as possible to save space, thereby achieving the required torque and cooling capacity. The magnitude of the clutch torques that are to be transferred can be specifically set via the number of plates and the available actuating force. For example, the C0 disconnect clutch was situated radially inside the C1. In turn, the C2 is positioned axially to the C1. The bearing support of the triple clutch is via the main rotor bearing in the housing, which helps give the hybrid module a high level of overall efficiency and provides it with a very compact structure.

The three clutches are actuated via standing CSCs (concentric slave cylinders) with ring pistons, positioned on the transmission side for C1 and C2 and on the engine side for C0. The force is transferred to the spinning clutch by means of engagement bearings. Since this form of actuation is practically leakage-free, the clutch actuation



6 **P2 hybrid module with a rotor-integrated, wet triple clutch system**

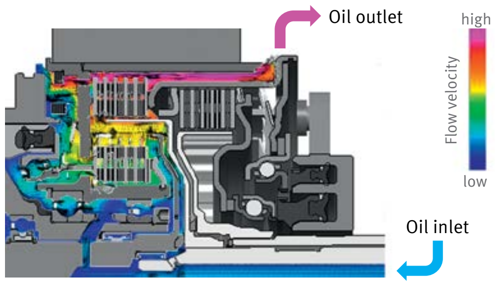


7 Oil flow inside of the rotor for the sub-clutches and the bearing positions

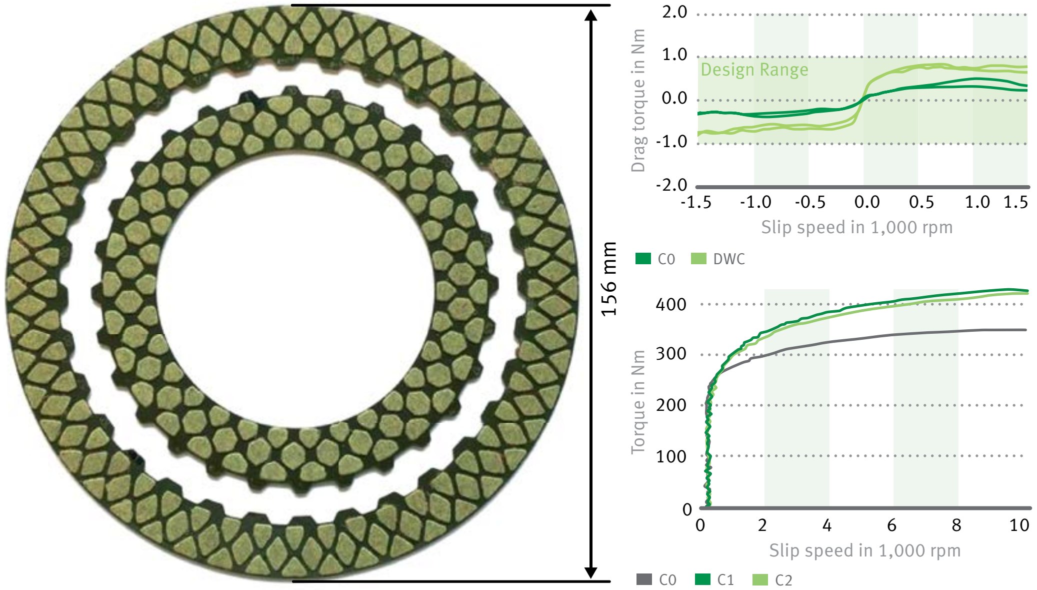
system works very efficiently. With the small radial dimensions of the plate pairs and optimized cooling oil flow, only minimal drag torques occur in the open clutches.

The clutches are completely integrated in the oil/cooling circuit of the hybrid module. For efficiency reasons, care has been taken to adapt the volume flow of the oil to the cooling performance actually needed in the clutches – Figure 7. A partial flow of the respective cooling oil path is used for cooling the C0 and supplying the bearings. The oil flowing out of the outer clutch housing also flows along the rotor and partly along the stator of the electric motor, thereby contributing to the extraction of heat from there as well.

While designing the thermal aspects, it was possible to ensure uniform heat dissipation from all plates of the multi-plate clutches through the design of the oil flow channels. Moreover, the oil is transported away very quickly from the friction contact so that no unnecessary drag losses occur. Using rotor cooling, we have succeeded in keeping heat-sensitive magnetic materials within an acceptable temperature range below 150 °C, even in high load situations, thereby increasing the power density of the electric motor. Extensive flow and temperature simulations with corresponding optimizations were conducted – Figure 8 – and the layout was validated using a testing system developed by Schaeffler. In addition to ensuring the torque and thermal capacity, the focus of the testing was also on optimizing the drag torques and losses in the complete system. On the basis of many realistic test stand trials, it was possible to be demonstrated that the compact wet triple clutch system reliably meets the requirements with regard to function and operating life.



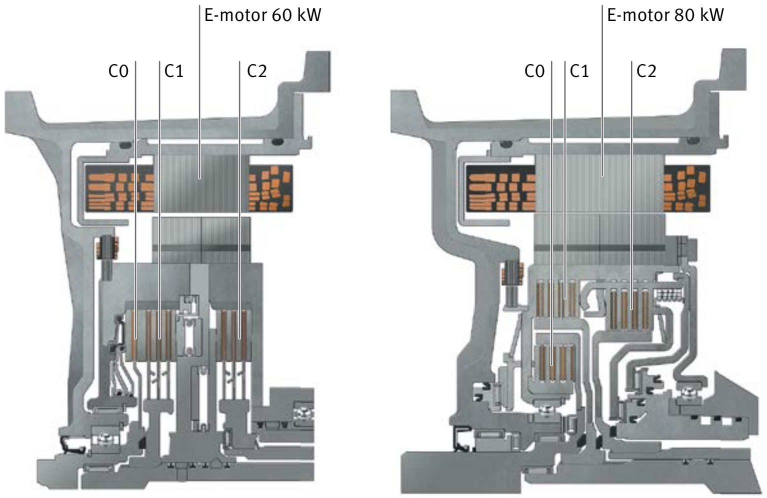
8 Flow simulation for a P2 hybrid module with an integrated wet triple clutch



9 Wet lining for double and also triple clutch systems

As with many clutch systems, a central role is attributed to the tribological system – consisting of a friction lining, counter friction faces, and cooling oil. The functional characteristics of torque capacity, thermal robustness, NVH behavior, and open clutch drag torque are heavily influenced by the friction lining. In order to really do justice to this fact, Schaeffler has also developed a suitable lining for wet double and triple clutch systems – in addition to linings for dry clutches – and has validated it in extensive test series – Figure 9. This has resulted in very good values for both the torque capacity as well as for drag torques in the sub-clutches. The drag torques in the C0 were able to be greatly reduced yet again through the geometric design, optimization measures in the friction plate, and minimization of the cooling oil volume flow.

Since even lower clutch loads are to be expected in future applications, it will be possible to raise the design values of the friction system and shrink the clutch packages even more. The two-layer Schaeffler wet friction facing, which is still in the developmental stage, will support this potential for optimization through its enhanced performance [3]. In this way, only the length of the electric motor will ultimately be decisive for the axial installation space required. Figure 10 depicts such concepts, including a version with hydraulic rotary transmissions on the right. Other innovative approaches for the next generation of wet triple clutches are installation space optimized needle roller bearings in the actuation system, a low hysteresis actuator piston, a rotor bearing arrangement that is very resistant to tilting, and low mass inertias on the output side. The installation space required by the hybrid module is reduced to a minimum when combined with the latest compact electric motor developed.



10 **Further development of the wet triple clutch**

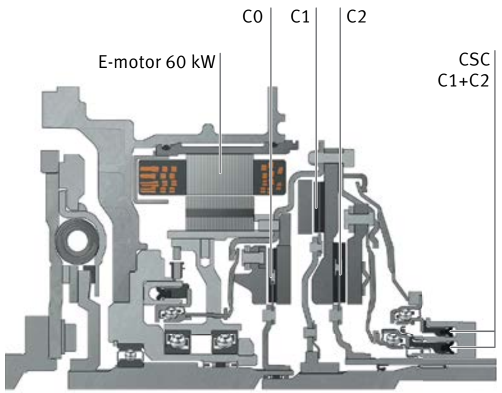
In this design, the electric motor is protected from heat and clutch abrasion from the dry triple clutch by a central sheet housing. In addition, the triple clutch unit can very easily be screwed onto the ready-mounted electric motor. This arrangement is particularly advantageous for medium electric engine outputs of around 60 kW, especially when combined with a short electric motor.

Dry Integrated Triple Clutch System

Primarily dry (i.e. air-cooled) double clutches are used for non-hybridized transmissions in the lower torque and performance classes. The question now arises as to how a dry triple clutch system optimized for the installation space can be designed in combination with a coaxial P2 hybrid system and whether a similarly high degree of integration is possible as with the rotor-integrated wet triple clutch.

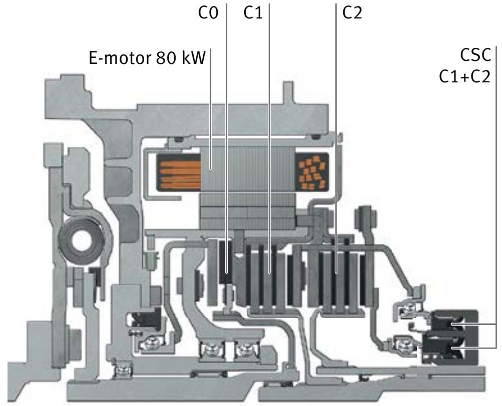
The first approach is to combine a rotor-integrated C0 disconnect clutch with a radially nested double clutch, whereby the double clutch is positioned on the rotor of the electric motor – Figure 11. Compared to a dry standard double clutch, this makes it possible to save 20 to 25 mm of axial installation space. The C0 disconnect clutch actuated on the engine side is integrated in the rotor of the hybrid module and therefore has nearly no effect on the installation space.

If higher electric motor outputs and thus electric motors with greater axial lengths are used, then it is usually possible for even dry triple clutches to be integrated into the rotor – Figure 12. The number of clutch friction faces varies depending on how high the required clutch torques are. This means that multi-disk clutches can be designed



11 **Radially nested dry triple clutch**

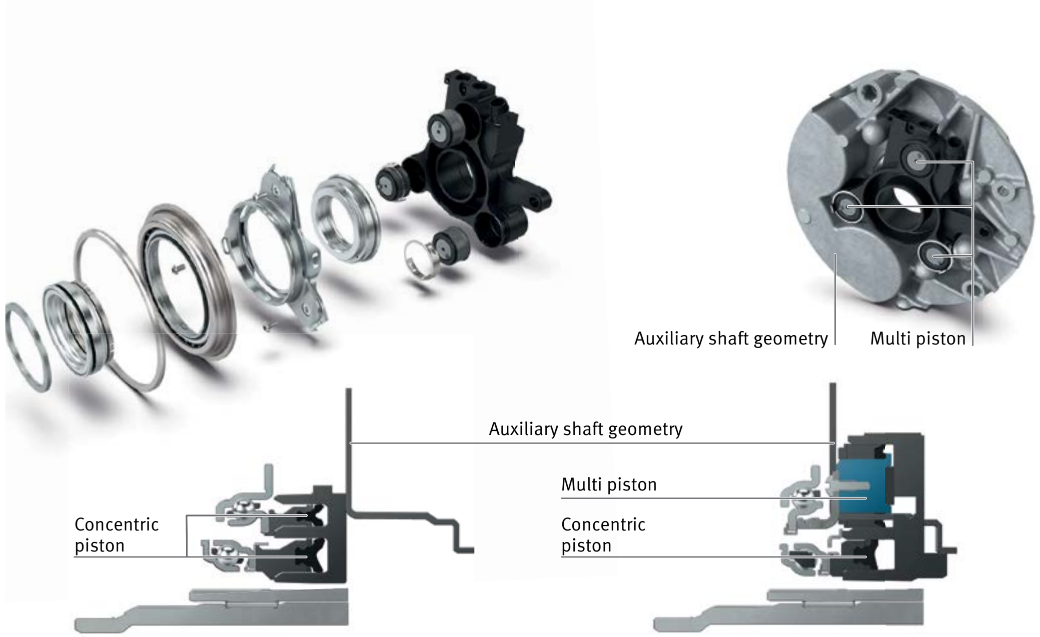
even for dry triple clutches, thereby allowing even more significant axial reductions in the installation space to be achieved.



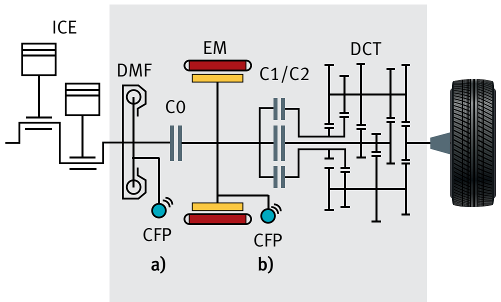
12 **Triple clutch integrated inside the rotor**

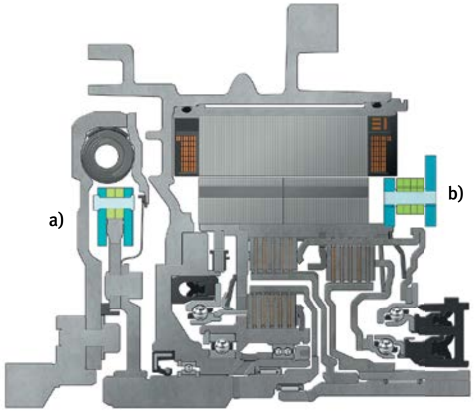
Further Development around the Clutch

In today's triple clutch systems that are ready for series production, two radially nested coaxial ring pistons are used for clutch actuation. The actuator system for the C1 and C2 clutches is ideally located between the main bearing of the transmission input shaft and the auxiliary shaft bearings. Due to geometric restrictions – mainly in a radial direction – caused by the position of the auxiliary shaft eyes, the ring pistons cannot be optimally integrated in many application cases. In order to compensate for the need for installation space despite this radial restriction, the outer ring cylinder can be replaced by three interconnected individual piston release mechanisms as an alternative. They can then be positioned between the auxiliary shaft eyes without having any effect on the installation space – Figure 13. An additional need for installation space of up to 30 mm can be avoided this way.



13 **Further development of the double actuator system for C1 and C2 (3-piston actuator system)**



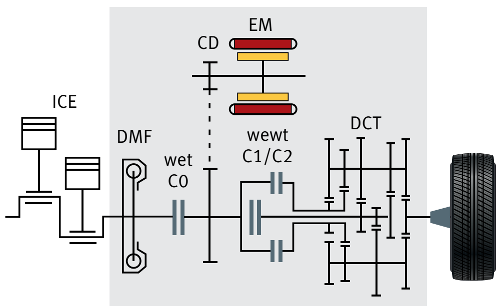


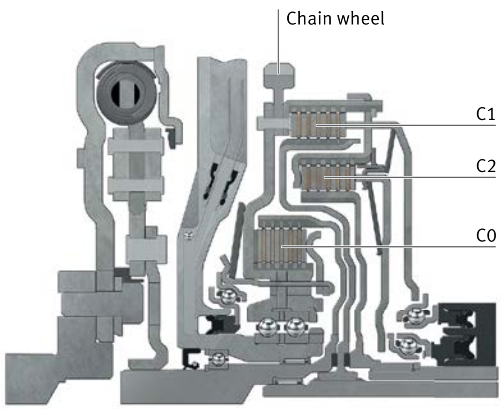
ICE Internal combustion engine
DMF Dual mass flywheel
EM Electric motor
CFP Centrifugal pendulum
C0 Clutch for ICE
C1/C2 Clutch 1/clutch 2
DCT Double clutch transmission

14 Arrangement variations for the centrifugal pendulum absorber in the P2 hybrid system

Due to the electric driving elements NVH performance is even more important in vehicles equipped with a high-voltage hybrid system. As shown in [2], the phase current variation of the e-machine can be used to optimize the vibration behavior of the entire powertrain. Depending on the vehicle class, the customer engine used, and the demand for comfort, it may make sense to integrate a centrifugal pendulum absorber (CPA) as an additional damping element. As an alternative to the classic arrangement at the dual-mass flywheel, position a in Figure 14, an installation position at the rotor/clutch flange on the transmission side, position b in Figure 14, is also possible. Since installation space needs to be provided at this point for the clutch actuator anyway, such a measure may represent another contribution towards reducing the overall axial length.

With regard to vibration-damping properties, both installation positions have specific advantages and disadvantages, depending on the driving situation. The arrangement on the transmission side is mainly advantageous when the





ICE Internal combustion engine
DMF Dual mass flywheel
EM Electric motor
CD Chain drive
C0 Clutch for ICE
C1/C2 Clutch 1/clutch 2
DCT Double clutch transmission

15 Axially parallel arrangement of a P2 system with a triple wet clutch

combustion engine is started via the electric motor of the hybrid module (drag start), while the arrangement on the engine side exhibits the biggest advantages when the combustion engine is engaged and running at a low engine speed.

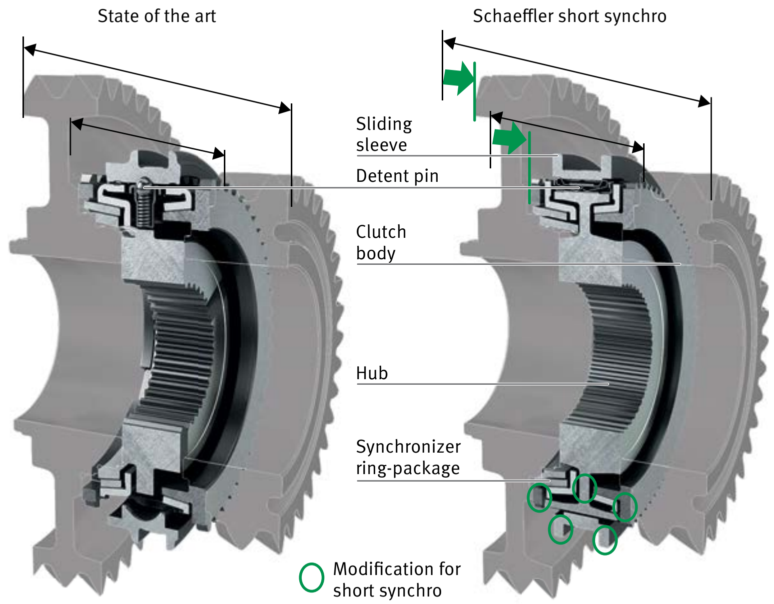
The P2 hybrid systems with a triple clutch that are described above are all arranged coaxially around the transmission input shaft. Depending on the size and capacity of the electric motor, an axially parallel arrangement of the electric motor may be advantageous – particularly for passenger cars with a front-transverse powertrain. Schaeffler has also additionally developed axially parallel P2 hybrid modules [2] that are well suited to being combined with a double clutch transmission. Figure 15 shows a possibility for further reducing the axial installation space needed for the hybridization through radial nesting of wet plate clutches.

The axially parallel arrangement poses special challenges for the mechanical design, particularly with regard to the bearing concept for the clutches and the rotor.

Optimizing the Installation Space within the Transmission

Synchronization Devices with a Minimal Overall Axial Length

If the transmission structure and number of gears in an electrified double clutch transmission remain the same, then the options for reducing the transmission length itself will be limited. The short synchronization devices already introduced by Schaeffler back in 2014 offer a solution for this [4].



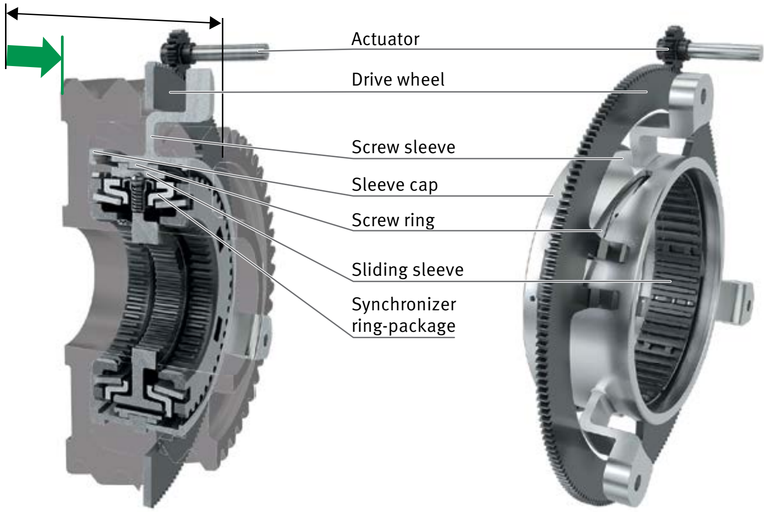
16 Optimizing installation space by using shorter synchronization units, taking the Schaeffler Short Synchro as an example

The concept, which has since been developed much further, makes it possible to reduce the need for axial installation space by up to 10 mm in each synchronization direction – Figure 16. This reduces the overall length by approx. 20 mm in a typical three-shaft transmission setup with two synchronization units arranged axially in a row. A key prerequisite for reducing the overall length is the production technology for producing gearing with a small module.

Gearing for synchronization devices is produced through machining – similar to larger gearwheels in transmissions. In contrast, Schaeffler is relying on sheet-metal-formed gearing, which makes it possible to increase the number of teeth by reducing the module from 2 down to 1 mm. Using a larger number of teeth makes it possible to achieve an identical torque capacity despite the fact that the teeth are smaller. This measure alone allows the overall axial length to be shortened by approx. 6 mm per synchronization device, since the smaller teeth lead to shorter axial point lengths with the given point angles.

Even though smaller gearing modules are used, relevant parameters are staying the same: width of the friction cones, load on the gears, the clearance for drag torque, and wear reserve. Depending on the application, the installation space gained can also be used for designing gearing for higher torques.

A further reduction of around 3 mm can be achieved with a new pressure piece design for the sliding sleeve. A radially flat pressure piece makes it possible to decrease the wall thickness of the sleeve carrier at the bar accordingly. This allows the cone surfaces of the synchronization rings that are decisive for the friction performance to remain the same. The power loss due to the friction is not greater compared to today's synchronization devices.



17 **Synchronization unit with a rotary actuation under the gearwheels (Schaeffler Ultra Short Synchro)**

The shorter axial length additionally reduces the shift travel by approx. 2 mm. This is equivalent to a shift travel reduction of up to 25%, allowing for a corresponding dynamic design. In vehicle tests, the shift quality was evaluated positively even for manual shifting. Another secondary effect is the weight savings obtained. If the axial installation space gained is used for shorter shafts and a correspondingly smaller housing, the weight savings add up to 3.5 kg.

Rotary Actuated Gearshifts

Another possibility for greatly reducing installation space involves innovative rotary actuated transmission shifting, in which the entire axial movement of the sliding sleeve takes place inside the gearwheel contour, meaning that there is no longer any need for external installation space. In transmissions with large gearwheels, such as when the gearwheel of first gear is situated directly next to the gearwheel of third gear, it is possible for such a system to be arranged completely under the gearing. This shrinks the idler gear distance significantly so that another 5 to 10 mm of axial installation space can be saved at each synchronization unit.

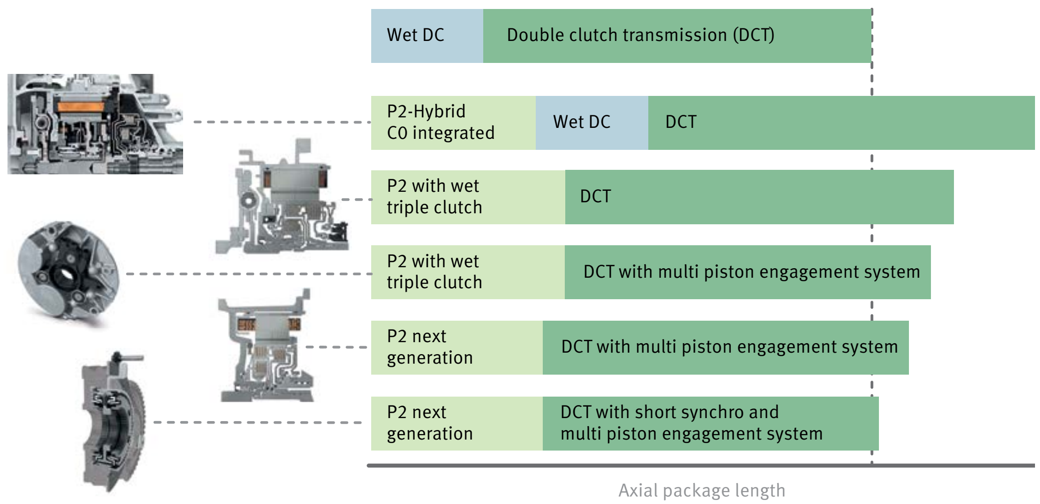
The module shown in Figure 17 shows such a shifting system. Actuation merely involves a swiveling rotation by an actuation wheel. The sliding sleeve on the inside is moved axially by a ring nut over a sliding surface. The ring nut has three slide cams and is turned from the outside via the actuation wheel. A threaded sleeve fixed to the housing absorbs the axial forces.

Another advantage is that the actuator for shifting – preferable in P2 double clutch transmissions – can be used in a direct acting actuating motor. In this way, shift rods and other actuation elements can be eliminated, thereby reducing the weight even further.

Currently, these ultra-compact synchronization units are at an advanced stage of development. However, initial tests with prototypes show considerable potential for this technology, which may be used in completely newly developed transmissions.

Compact Double Clutch Transmissions with P2 Hybridization

In order to maintain the given installation space with P2 hybridization of double clutch transmissions, Schaeffler – as shown in the statements above – is making a large number of product innovations and further development solutions available. Many of the product ideas in the P2 hybrid module and in the transmission part can be combined in such a way that the overall transmission length can remain nearly the same compared to existing double clutch transmissions, despite the powerful P2 hybridization – Figure 18.



18 **Measures for reducing the overall transmission length of P2 hybrid double clutch transmissions**

Summary and Outlook

Thanks to the combination of dynamics, efficiency, and comfort, double clutch transmissions are continuing to gain ground on the global transmission market. The systems designed so far are proving to be very suitable for electrification through the integration of P2 hybrid modules. However, the fact that the hybrid module is situated between the combustion engine and the transmission means that the powertrain is significantly longer. Schaeffler has developed a large number of product innovations for reducing the axial installation space, which will be able to be implemented in future series applications. An important basis for reducing the installation space has to do with the fact that P2 hybridization and a powerful battery reduce the maximum and average loads for the clutch system to less than 50 % compared to conventional double clutch transmissions since vehicle launches are largely carried out under electric power.

By taking these boundary conditions into account, it has been possible to develop and test highly integrated wet and dry triple clutch systems, in which the clutches are located almost completely within the installation space of the electric motor's rotor. To actuate the three clutches, compact, highly integrated actuating pistons needed to be developed, with a three-piston engagement device being used for the first time as well.

Other measures for reducing the installation space can be implemented within the transmission itself. For this purpose, Schaeffler has developed technologically new synchronization units that enable double clutch transmissions to be built that are 15 to 20 mm shorter. Since the various solutions for clutch, actuators, and transmission can be combined almost at will, a total installation space reduction of up to 100 mm can be achieved. In this way, P2-hybridized double clutch transmissions are able to preserve common vehicle installation space dimensions.

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21

Dedicated Hybrid Transmission

How the Transmission Becomes a Powertrain

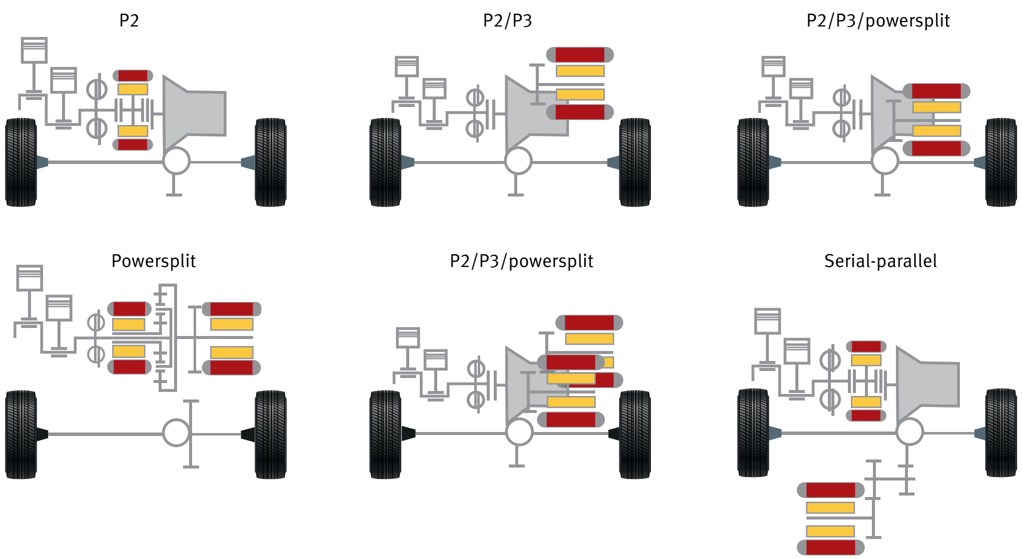
Andreas Kinigadner
Christian Lauinger
Martin Vornehm

Introduction

Plug-in hybrid vehicles combine local emissions-free driving with low consumption in hybrid operation and a high degree of driving pleasure. In addition to this, more stringent legislative requirements will see increases in the battery capacity and the electrical output as performance increases. This results in greater challenges with respect to spatial integration and the overall design of these types of drive trains. The greater costs of the electrical components make it necessary to take every opportunity to simplify the technical design.

The low overall quantities of hybrid vehicles have lead to the electric drive being primarily a P2 arrangement between the internal combustion engine and the transmission and the classic drive train components remaining largely the same. The expectation of increasing quantities means that optimization of the overall system is becoming a priority [1]. This also includes the option of simplifying the mechanical transmission, possibly by removing the reverse gear and integrating at least one of the electric motors into the transmission to take over this function completely. These transmission concepts are examples of Dedicated Hybrid Transmissions (DHT) [2].

Dedicated hybrid transmissions can be developed from existing transmission concepts, i.e. from double clutch transmissions, planetary automatic transmissions (AT), continuously variable transmissions (CVT) or automated manual transmissions. The electric motor becomes part of the transmission and it can then be connected to various different (drive) shafts, see Figure 1. In addition to the parallel (or serial) hybrid modes, it is possible to achieve one or more power split modes when combined with a planetary gear unit. Many drive structures can include more than one electric motor and thus provide a range of different ways of meeting the specific requirements.



1 **Function of dedicated hybrid transmissions with one or two electric motors**

As with concepts with only one electric motor, the basic question arises of whether the required comfort is achieved with continuously variable operation, for example by power split or serial functionality. As the high levels of efficiency in plug-in hybrids must be maintained when operating the internal combustion engine, this requires a corresponding gear ratio spread.

Two structures, each with only one electric motor, are selected for this investigation in order to reduce the complexity of the system.

The first concept is based on a CVT which is characterized by high comfort levels and good dynamics; the reverse planetary gear set and the associated shifting elements are dispensed with here. The second concept is based on an automated manual transmission and is intended above all for markets such as China and Europe where stepped transmissions reach significant volumes.

Dedicated hybrid transmissions with continuously variable ratio

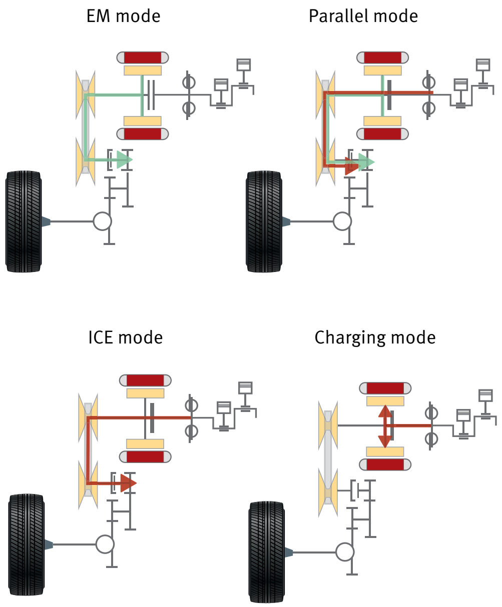
Schaeffler has been developing and manufacturing key components for continuously variable transmissions for almost 20 years, supplying more than 12 million chains and more than four million pulley sets over this period. Schaeffler has extended the modular concept for the CVT chain considerably so that the range now extends from the compact 05 chain to the 08 chain which can support torques greater than 500 Nm [3]. The 05 chain is being used for the first time in a CVT with 180 Nm torque capacity which Hyundai Motors is putting into volume production in the near future, see Figure 2. It is characterized by high power density and very good efficiency with low space requirements [4]. The small pitch of the chain ensures very good acoustic behavior. Due to many years of mass production development of CVT pulley sets and CVT hydraulics Schaeffler also has detailed knowledge in the design of continuously variable transmissions systems.

2 **Hyundai Motors CVT transmission with the Schaeffler 05 chain**

DH CVT in P2 arrangement

The basic characteristics of continuously variable force transmission are particularly well suited to the power delivery of an electric motor whose operating points can also be freely selected over a wide range of speeds. In addition to this, the CVT also allows the internal combustion engine to operate efficiently.

Continuously variable transmissions which incorporate an additional electric motor on the engine side are already in volume production. However, these transmissions use a planetary gear set for driving in reverse. In the future, it will be possible to achieve reverse driving purely via the electric motor due to increasing torques in the electric



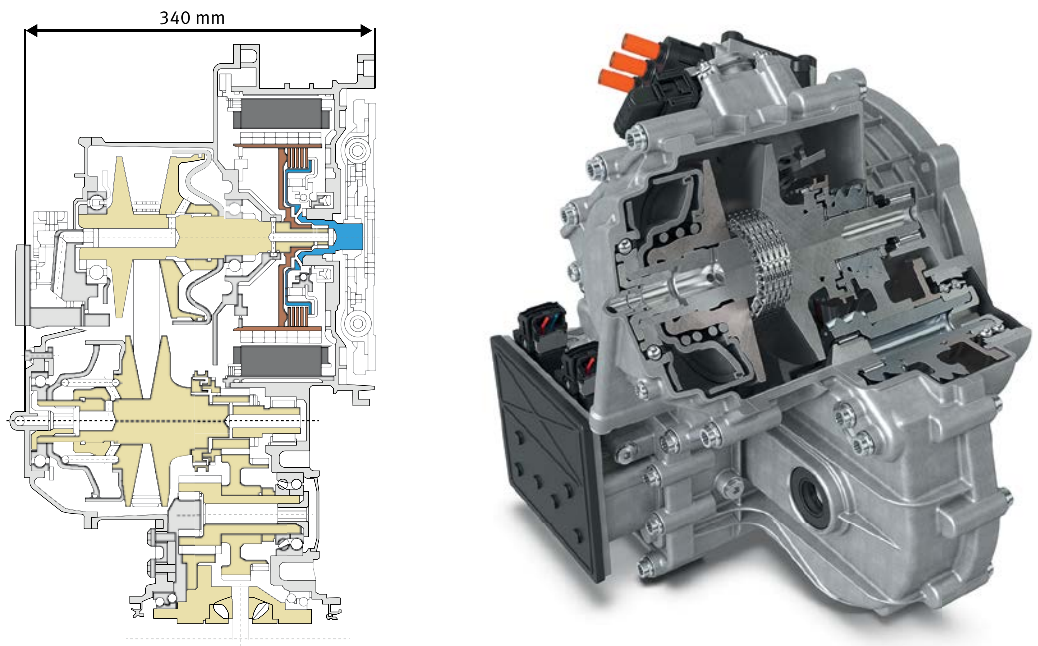
3 Operating modes of the DH CVT with electric reverse gear

drive and greater battery capacity. This dispenses with the planetary gear set, the actuator required to shift the gear set and a clutch. The characteristics of the electric motor allow the torque converter to be removed, too. All other operations correspond to those of a classic P2 hybridization, see Figure 3.

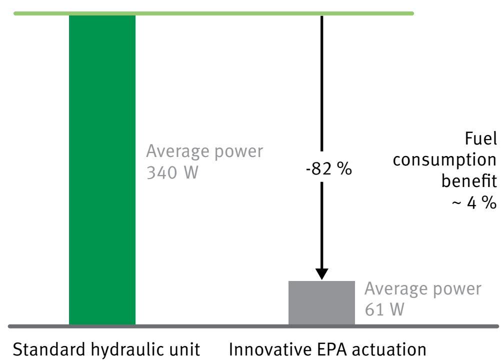
In the DH CVT shown here, the electric motor has a peak power of 80 kW and a maximum torque of 330 Nm and is completely integrated into the transmission housing, see Figure 4. This achieves a compact axial section length of approximately 340 mm. Compared to other hybrid transmissions in this performance class and with a P2 architecture, this axial length represents the lower limit. The influence of the ratio spread on consumption was investigated separately. Up to a ratio spread of 7, there is a proven and noticeable consumption benefit without a reduction in acceleration capability. A ratio spread of 7 was therefore chosen for the concept. The variator can be separated from the wheel side using a dog clutch after the second pulley set. This means that the battery can still be charged, even when the vehicle is stationary. This is an emergency function to be able to drive backwards after only a short charging period when the battery is completely discharged.

Hydraulics which include a mechanical high pressure pump are typically used in conventional CVTs. Based on the WLTC consumption cycle, approximately one third of the energy losses occur in the pump. The reason for this is the design of the pump size for extreme driving conditions such as fast adjustment of the variator following a kick-down downshift. Furthermore, as the degree of hybridization increases, it is also becoming necessary to have an electric auxiliary pump.

Electrification of the transmission allows the actuator technology for clamping and shifting to be designed to be significantly more energy-efficient as well. The new Schaeffler actuator technology concept is a logical further development of existing approaches [5]. It proposes separating the “clamping” and “shifting” functions and installing two electric pump actuators (EPA) which can be controlled as required. BLDC motors are provided for the actuators. A further electric oil pump provides cooling and lubrication for the transmission components. The average power demanded of the pumps in the WLTC is thus reduced from 340 W with standard hydraulics to 61 W for the EPA actuator technology. This represents a consumption benefit of approximately 4 % in the WLTC, see Figure 5. The components in the CVT drive train model are characterized by measurements. Data maps for the internal combustion engine, the wet-running multi-disk clutch and the electric motor with power electronics and lithium ion battery are thus included in the simulation. Measurement data for various operating conditions are also used for the variator unit, the hydraulics and the bearings. The EPA actuators and their electric motors are described by existing data maps.



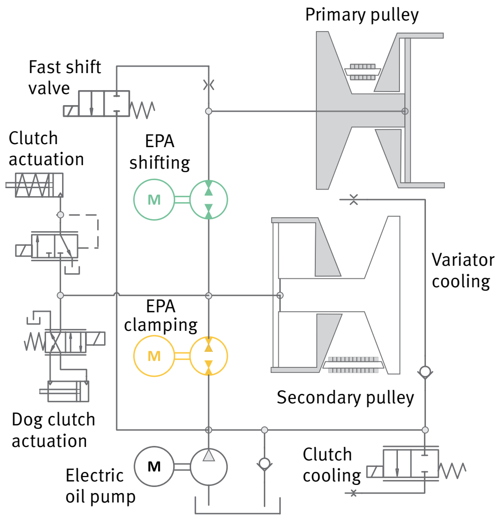
4 Cross-section and construction of the DH CVT



5 Comparison of the of the output requirements of a standard hydraulic unit with EPA actuation technology

When using the innovative EPA concept in a hybrid transmission, the specific driving situations must be identified which are relevant for the design of the pump actuators and their power electronics. These include full-load acceleration and the associated quick ratio changes or situations with constant maximum torque load at the variator.

A fast shift valve is used in order to relieve the shifting pump of high volume flow from the primary pulley set during fast variator slew events, see Figure 6. This is either controlled electronically or regulated via the volume flow through the shifting pump. The fast shift valve allows the same electric motor to be used for both actuators. Preloading the clamping actuator by using the cooling oil pressure improves the efficiency and the dynamics.



6 **DH CVT hydraulic unit with innovative EPA actuator technology**

Figure 7 left shows an example for the time dependence of the variator ratio and the variator input torque during a fast variator adjustment resulting from a quick build-up of torque after a tip-in event.

Figure 7 right shows the operating points of the two EPA electric motors for the same situation. Due to the selected design, the maximum motor power for each actuator is approximately 200 W in this situation. The continuous line marks the maximum continuous mechanical output for the electric motors, reaching a maximum of approximately 450 W at the corner point.

7 **Tip-in maneuver.**
Left: Time dependency of variator ratio and torque.
Right: Output uptake for both EPA actuators (dotted line) and maximum continuous mechanical output for the electric motors The previous section shows that the simulation tools developed by Schaeffler allow an optimum design of the pump size, the motors and the actuator electronics to be found for the specific customer requirements.
The compact EPA enables such highly efficient CVT-actuation concepts.

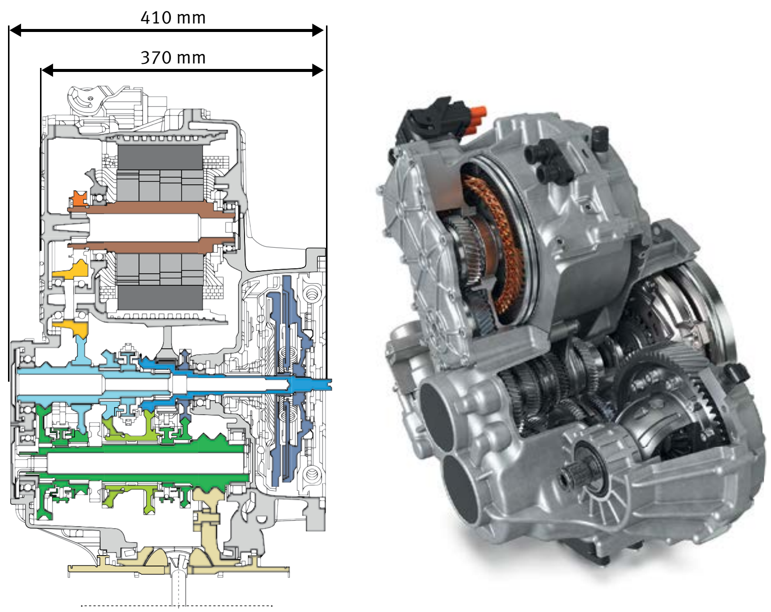
Dedicated hybrid drives based on automated manual transmissions: Dedicated Hybrid Shift Transmission (DH-ST)

Schaeffler has been supplying decisive electromechanical subsystems since 1997 which convert conventional manual transmissions to automated manual transmissions and DCTs. Clutch actuation was developed for different technologies [6], as well as gear-shifting actuation.

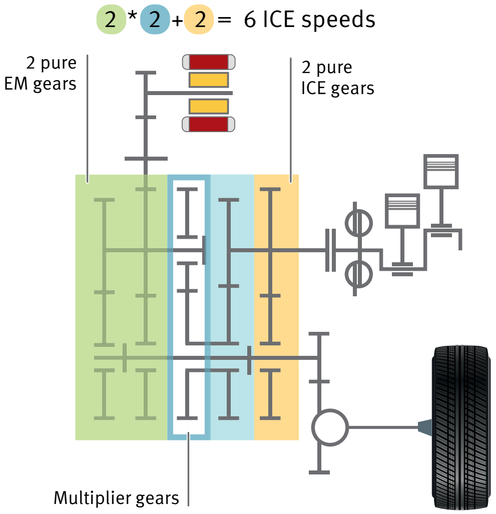
Based on this volume production experience a new transmission concept is now in pre-development. This exploits the benefits of an automated manual transmission in an electrified drive. Complexity and costs are reduced compared to a P2 double clutch transmission. The disconnect clutch (K0) between the P2 motor and the crankshaft, and one of the two double clutches as well as one drive shaft with bearings and gears are no longer present. This also reduces the actuation effort.

In functional terms, the power shift is achieved via the interaction of the internal combustion engine and the electric motor. The electric motor output is therefore either at or greater than that of the internal combustion engine. The illustrated design combines a six-speed transmission with a powerful electric motor (peak output 147 kW). The transmission works with only five gear planes on two shafts. The ratio spread is 6 in order to achieve a gradeability of over 25 % and vehicle speeds of over 200 km/h. Figure 8. This results in a compact axial envelope of 410 mm which is compatible with the envelope of a typical 350 Nm front-wheel transverse powertrain.

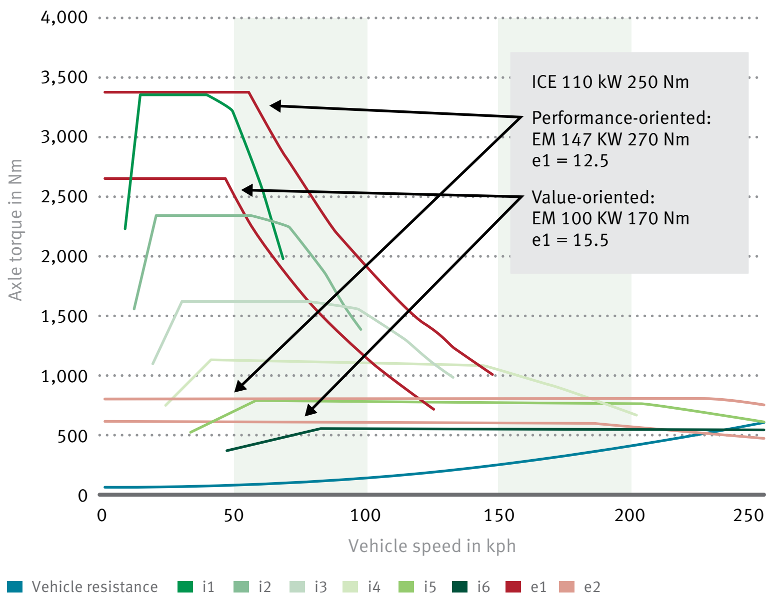
The transmissions can be split into two partial transmissions. The electric motor which is running parallel to the internal combustion engine operates in one of the partial transmissions. This is integrated into the transmission structure in such a way that it has two ratios available via two pairs of gears, see Figure 9. Two gear stages are available in the other partial transmission when the system is running the internal combustion engine only. The internal combustion engine can also make use of the partial transmission for the electric path either directly or via a ratio by deploying two further wheel pairs. These form a



8 **Cross-section and design of the DH ST 6+2 with six combustion engine and two electric gears**



9 **Transmission schematic for a hybrid transmission with six combustion engine and two electric gears**



10 **Axle torque as a function of the driving speed for two variants of the electric motor (147 and 100 kW) as well as two different ratio stages in the electric gears**

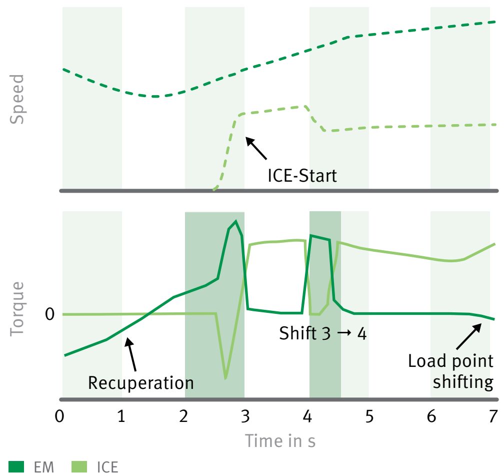
type of multiplier transmission between the two partial transmissions. This means that four further gears are available. The internal combustion engine can therefore operate in a total of six gears while the electric motor is driven in two speeds. As one wheel plane is being used twice, only 5 wheel planes are required for the six gears.

This combination of a single clutch and three shift elements allows further drive conditions and power flows to be achieved, for example a purely electric reverse gear. Gear selection is via a single actuator which can also operate the parking lock.

A significant benefit of this design is that the tractive force can be augmented by the electric motor during combustion engine shifts. This structure provides purely electric driving similar to existing purely battery-operated electric applications. After switching modes from electric to hybrid driving and the associated internal combustion engine start-up, the electric motor can be shifted to second gear. This allows comfortable shifting in the upper gears due to the augmentation of the tractive force. At high driving speeds, the six gears used with the internal combustion engine and the second electric gear provide favorable consumption and noise reduction.

When designing the transmission, it was assumed that the target maximum speed is greater than 200 km/h and that electric driving up to 140 km/h should also be possible. If the acceleration target and, if necessary, the requirement for shift comfort with a high accelerator pedal position are reduced, smaller electric motors can be used. A value-oriented variant with a peak electric output of, for example, 100 kW (maximum torque 170 Nm) is feasible in addition to the performance variant of the transmission, see Figure 10.

There is a basic assumption that plug-in hybrid vehicles will cover significant distances in electric mode, given sufficient battery capacity. Beyond a certain driving speed or a required driving torque, the internal combustion engine can be started dependent on the battery charge. This is achieved by pre-selecting a suitable gear and closing the friction clutch. A pinion starter is therefore not required, see Figure 11. As soon as the internal combustion engine achieves driving torque, the electric motor can be shifted up. Moreover, the shift point from first to second electric gear can be adjusted depending on the load condition.



11 **Internal combustion engine gear selection and start-up process**

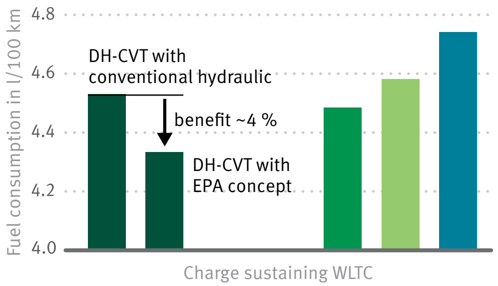
Further arrangements of the DH ST solution are feasible. A coaxial electric motor could be used or the number of gears for the internal combustion engine reduced to three or even to the point where the clutch is no longer required as far as the vehicle launch is achieved exclusively via the electric motor.

Potential for driving performance and consumption

In order to quantify the driving performance and reductions in consumption to be achieved with dedicated hybrid transmissions, Schaeffler completed a comprehensive series of simulations. The consumption simulations were carried out based on the WLTC. They are based on a D segment plug-in hybrid vehicle with a mass of 1,670 kg with a 1.4 l turbo gasoline engine (maximum torque 250 Nm). The storage capacity of the lithium ion battery for determining the electric range is 8.7 kWh. The approaches outlined above for the DH CVT and the DH ST are compared in Figure 12 with a hybridized 6-speed double clutch transmission and a power split drive train.

Data maps are also used here as there was a very high emphasis on the accuracy of the model in these simulations. So, for example, a hybridized double clutch transmission with wet-running multi-disk clutches from a volume application with actuator technology optimized for efficiency [7] is measured on a test rig and the data map is subsequently scaled. The shifting strategies for various wet-running double clutch transmissions are also already known from comprehensive vehicle tests. The model is also compared to these strategies.

If the charge condition is to be sustained, then the two Schaeffler concepts are superior to both the hybrid drive with the six-speed double clutch



Transmission	R/C	max. system power	EM power (Peak)
DHCVT	7	140 kW	80 kW
DH-ST 6+2	6	220 kW	147 kW
P2-6DCT (wet)	6	180 kW	80 kW
Powersplit	–	184 kW	50 kW (MG1)/100 kW (MG2)

12 Comparison of the fuel efficiency of various hybrid drives in charge sustaining mode

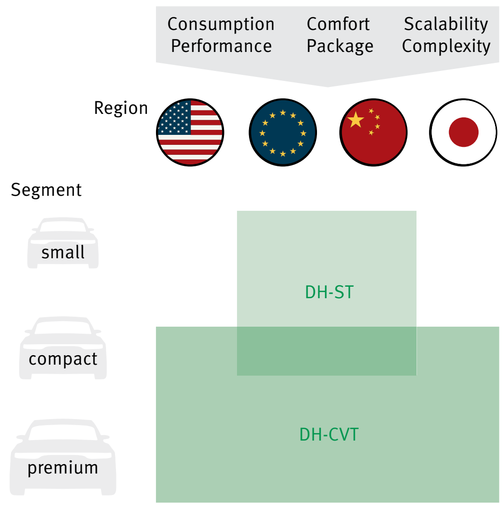
transmission and the power-split transmission: With approximately 4.3 l/100 km, the CVT compares more favorably than the 6+2 transmission with approximately 4.5 l/100 km. The main reason for this is that the internal combustion engine can operate at a greater ratio spread in more favorable operating fields. There are therefore no losses from power conversions such as occur in power-split concepts.

This very positive assessment of the CVT concept may come as a surprise. A recently published study [8] for a B segment vehicle platform showed that modern CVT and double clutch transmissions have comparable efficiencies. This has also been confirmed through in-house studies. Due to the transition from conventional hydraulics to the EPA 'power-on-demand', the components with the greatest energy consumption are no longer included in the CVT, see Figure 5. The improvement in consumption in hybrid mode and in the electric range of the DH CVT is based on the associated increase in efficiency. Optimization of the hydraulic unit as a necessary step for the future competitiveness of the CVT is also considered in [9]. In purely electric driving, the dedicated hybrid transmissions achieve the greatest ranges over the course of the WLTC. On the other hand, the value for the DH CVT of 55.4 km is somewhat better than that for the DH ST 6+2 of 54.9 km which shows the benefit of the EPA actuator technology for the DH CVT. Furthermore, the electric motor within the DH CVT can use a transmission ratio spread of 7 due to the P2 arrangement whereas the DH ST 6+2 drives very efficiently using mechanical gears.

A comparison of the vehicle performance values for a purely electric full-load acceleration from 0 to 100 km/h shows that the DH ST achieves the best value of 6.8 seconds by a significant margin. However, this is not that surprising as the electric motor used in this concept is by far the most powerful (147 kW). The dedicated hybrid CVT and the hybridized double clutch transmission are practically identical in terms of model accuracy.

For a full-load acceleration using the electric motor and the internal combustion engine, the DH ST 6+2 achieves the best result of 5.9 seconds due to the high system output of 220 kW. The DH CVT achieves 6.4 seconds which is a similar result to a current hybridized double clutch transmission.

Figure 13 shows further criteria for comparing the dedicated hybrid transmission with current volume-produced vehicles. The concept based on CVT is a competitive solution for the mid to upper vehicle segments and SUVs in more than just the Asian markets. It combines high levels of comfort with low axial space requirement and can be relatively easily scaled to various system outputs. The dedicated hybrid transmission DH ST 6+2 represents a straightforward, equally convincing solution for achieving dynamic and



13 Criteria with allocation by market and segment for the DH CVT and DH ST

efficient plug-in hybrid transmissions for small and medium vehicle segments as well as compact SUVs.

Summary and outlook

Dedicated hybrid drives offer considerable potential to further increase the efficiency and driving dynamics of plug-in hybrid vehicles. Focusing on the use in hybrid vehicles allows system characteristics to be improved whilst simultaneously reducing the overall complexity of the drive train.

The hybrid transmissions presented here based on a CVT represent the logical further development of the current P2 arrangement. The mechanical reverse gear, which is usually achieved with a planetary gear set, is dispensed with completely. A new concept in actuator technology based on separate pump actuators provides further significant reductions in the hydraulic losses and therefore high levels of efficiency.

Schaeffler's dedicated hybrid transmission based on an automated manual transmission takes the concept of a transmission a step further by offering a very compact design with six ratio steps for the internal combustion engine and two gears for the electric motor. The mechanical power path allows a very high degree of overall efficiency in the drive train.

Both the concepts discussed here, the DH CVT and the DH ST 6+2, are by no means the final say in the field of hybrid transmission technology. They open up an attractive field of collaborative work in the development of drive trains.

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22

The Schaeffler eDrive Platform

Modular and Highly Integrated

Thomas Pfund

Introduction

The development of electric drives goes way beyond the electric motor. Optimal efficiency, range, and system costs can only be achieved through the interaction of the motor, power electronics, sensors, mechanical integration, and control strategy, which is why a system-oriented approach is needed.

Schaeffler already proved back in 2011 that it possesses system expertise with the "Active e-drive" concept vehicle. The combustion engine in a Škoda Octavia was replaced with two electric axle drives, each with 105 kW of nominal power. Even then, these motors were units that had been developed within the company. While the gear ratio of the motor and axle speed was fixed initially, a third generation rear axle drive with a two-speed transmission and nominal power reduced to 65 kW was employed in the same concept vehicle beginning in 2014. The drive now included new power electronics developed within the company. Both generations of the concept vehicle had torque vectoring in each wheel independently. While the power electronics for the tractive drive was still a separate system at this time, it was already installed on the axle for torque vectoring.

Meanwhile, it is now foreseeable that electrified vehicle quantities will reach large proportions during the current decade and that the degree of mechatronic integration will increase considerably. At the same time, very different topologies are being implemented for the powertrain, which can be classified based on its installation position [1]. Four classes of aggregates are needed for the design implementation of these powertrains:

- Hybrid modules for integration in the powertrain of the combustion engine
- Dedicated hybrid transmissions for specifically realizing hybrid and electric driving modes
- Electric axle drives for dedicated hybrid powertrains and purely electric vehicles
- Wheel-hub drives for new mobility concepts

Their complexity is increased by the fact that electric drives can be operated at various voltage levels, ranging from 48 V for initial hybridization to 400 V in purely battery-driven vehicles. The first manufacturers have already launched projects with an 800-volt on-board electric system in order to achieve short battery charging times at acceptable current levels. Accordingly, the power spectrum of electric drives is very broad – from 20 kW to more than 400 kW.

What all future aggregate concepts have in common is that they will integrate the mechanical and electrical/electronic systems to a large degree – Figure 1. However, the requirements for electric motors are very different. For example, the axial installation space is limited both in hybrid modules as well as in wheel-hub motors. Moreover, the speed is directly coupled to the powertrain, meaning that power scaling mainly needs to happen via the torque of the e-machine. While electric axle drives have greater axial installation space available, the diameter is generally limited due to the installation position. Coupling to the axle is always via at least one transmission stage in order to guarantee the required drive-away torque. This makes it possible to increase the capacity of the electric motor over higher speeds, which in turn makes smaller designs and material savings possible. The same applies for most dedicated hybrid transmissions. In contrast, power electronics is largely independent of the powertrain type; it is mainly defined by the voltage level, the amount of current needed for maximum output, and the installation space specified for the powertrain.

The large number of requirements listed necessitates a modular approach for various installation

1 **Powertrain concepts with an integrated electric drive**

spaces and power classes. This is even more true since – depending on the design – the drive's electric and electronic components make up to 80 percent of the total added value and are therefore key elements in the total cost.

Modular Approach

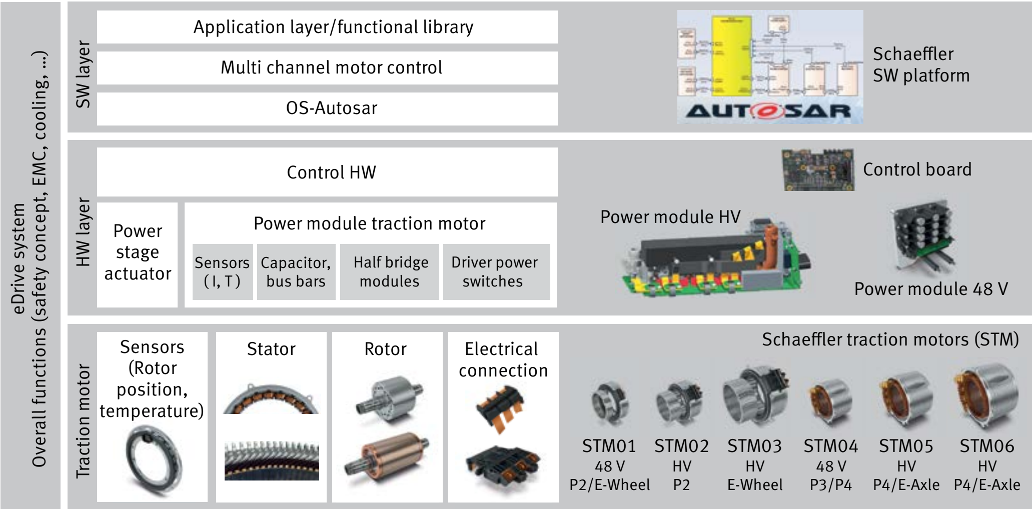
Developing a Technology Platform

In order to support a large number of vehicles and drive concepts on the one hand and yet minimize the development expenses on the other, Schaeffler has developed a modular technology approach for electric drives. Its three levels – Figure 2 – not only include electrical components, but also the hardware and software needed for controlling them.

The electric motors form the base of the platform. Due to the different requirements described at the beginning, multiple series need to be defined, each with a scalable output. According to the current estimate, the complete spectrum of future applications can be covered completely with six series. Depending on the necessary power density and other requirements, both permanently-excited motors as well as asynchronous machines will be used for this.

The middle level, to begin with, includes the power electronics with all key components, such as the power switches, capacitor, bus bars, driver stages, and sensors. Another part of the platform concept is a carrier frame that serves to channel the coolant and as a heat sink for all components of the power electronics. Also pertaining to the second level is the hardware for the drive control, the specifications of which not only depend on the electric motor and power electronics, but also on the functions carried out at the vehicle level, thereby requiring it to be implemented specific to the application. A key example of this is the communication network, which may be executed as a FlexRay, CAN, or CAN FD network.

For this reason, it cannot be separated from the third level where the software platform is located, which has likewise been developed according to a function-oriented approach. The software includes a functional library based on AUTOSAR,



2 **Modular technology platform for electric drives**

which also defines the requirements for the associated hardware.

Platform Components

Electric Motors

When regarding highly-integrated powertrains such as hybrid modules, electric axles, or wheel-hub drives, it is difficult to identify the electric motor as an independently functioning unit. The complete motor function often cannot be tested until the powertrain has been assembled, since multifunctional modules are also determining factors for the functioning of other subsystems. Examples of this are bearing arrangements that also provide support for an integrated drive, or rotor carriers that are also plate carriers for a clutch. The upshot of this is the high requirements for the test concept during production. It is necessary to utilize the measurement of parameters such as winding resistance, inductance, or magnetic field distribution of the components to assess the quality of the magnetic circuit assembly (stator and rotor) in order to immediately sort out faulty parts and not fail to identify them until during the end-of-line test.

In addition to the main function, that of representing defined speed-torque behavior, other demands need to be handled by this magnetic circuit as well:

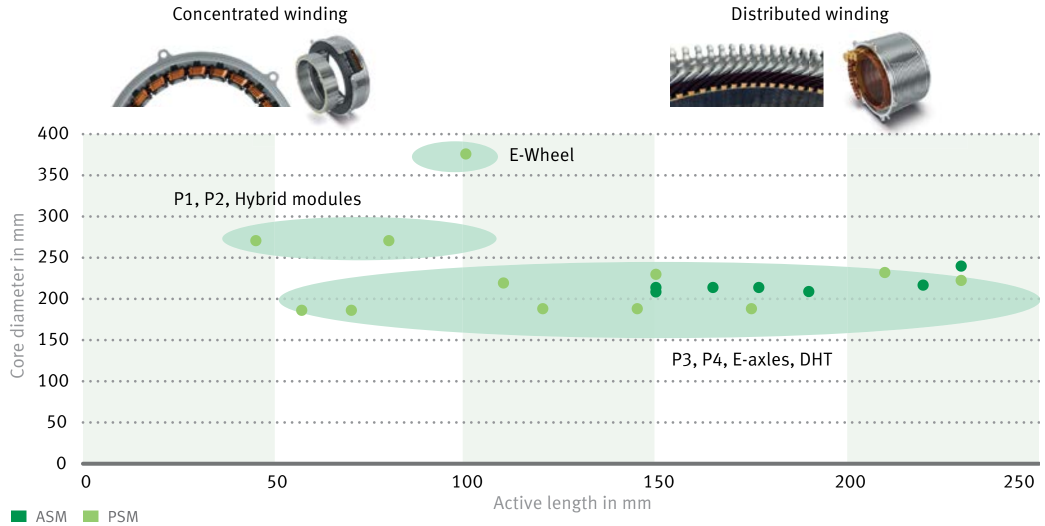
- Ideal cooling-down capacity and a high copper filling ratio to ensure long-lasting continuous performance
- Minimal use of materials in order to optimize costs
- Minimized harmonically occurring radial and tangential force effects in the stator for NVH optimization
- Minimized harmonics, which for their part likewise generate force effects, but also induce eddy-current-driving voltages and ultimately contribute towards losses and stator/rotor heating
- Minimal cogging and ripple torques

And the list goes on and on, as any detailed analysis of the individual phenomena will show. Thus there is an optimization problem, the solution for which must be oriented towards the requirements of the application and the expected load cycle.

Distributed winding has proven to be advantageous for a high torque density along with low harmonics and good heat flow from the current-carrying winding into the stator's laminated core. Since the coils extend over slots at different angles, the coil ends are large compared to concentrated winding. Round wire distributed winding, known from industrial electrical engineering, has proven to be poorly suited for use in automobiles. More and more solutions are shown that use what is referred to as hairpin or I pin technology, which involves inserting copper bars and welding them to the end faces of the laminated core. The requirements for this production technology are high, since there is a high number of weld points for each stator. Compared to coiled wires, the bars have a much larger cross-section, leading to eddy current and skin effect related losses during operation that greatly increase the more frequent changes in polarity become.

While the fact of larger coil ends compared to concentrated winding is considerably alleviated through hairpin technology, it nevertheless continues to persist. For this reason, there are still applications for electric machines where the advantages of distributed winding no longer outweigh the disadvantages, such as applications with extremely short axial installation spaces, e.g. 48 V hybrid modules, in which the active motor length is far less than 50 mm – Figure 3. At the same time, this is a good example of the strong influence of the available production technology. The smaller the coil ends can be made, the further the application limit of distributed winding will shift in the direction of shorter axial lengths.

Schaeffler has investigated whether there are alternatives to hairpin winding that utilize the advantages, yet minimize the disadvantages. One good alternative is wave winding, in which the



3 **Application areas of concentrated and distributed winding, depending on the active length of the electric motor**

	Concentrated	Distributed	
			
	Single tooth	Hairpin winding	Wave winding
Copper fill factor	+	++	+
Utilization of active material	0	+	+
Dynamic stator resistance	+	–	+
Number of slots	–	+	++
Rotor losses	––	+	++
Stator cooling	–	+	++
Size of winding heads	++	0	0
Number of welding points	+	––	++
Production flexibility	++	–	0

4 **Application areas of concentrated and distributed winding, depending on the active length of the electric motor**

(distributed) winding is produced in a kind of braiding process and then joined in the stator slots . By allowing certain concessions with regard to the copper filling ratio, it is possible to work with smaller cross-sections. The potential number of slots is thereby increased and the effect of the eddy current losses reduced. Figure 4 shows a qualitative comparison of the three winding types.

The extent to which the advantages and disadvantages come into force depends on the specific application case. In order to quantify the differences, the comparison between the hairpin and wave winding was depicted using a specific example, namely their use in an electric axle of a purely electrical powertrain in the C segment. In coordination with the downstream gear stage, this resulted in the following specification values:

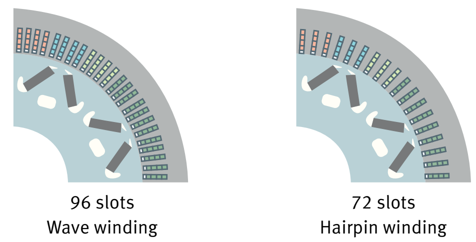
- P_{max} = 147 kW
- M_{max} = 265 Nm
- n_{max} = 18,000/min.

The table in Figure 5 shows the performance data of the e-machine with wave winding at selected, application-specific load points, with a outside stator diameter of 220 mm, 96 slots , and an active length of 110 mm.

Operating point	OP2	OP3	OP4	OP5	Partial load	Max. cont. power
Speed in rpm	5,300	18,168	4,542	18,168	3,750	10,000
Torque in Nm	265.1	64.5	146.5	36.5	3	95.5
Power in kW	147	123	70	69	1.18	100
Efficiency in %	94.9	94.9	96	95.3	91.9	97

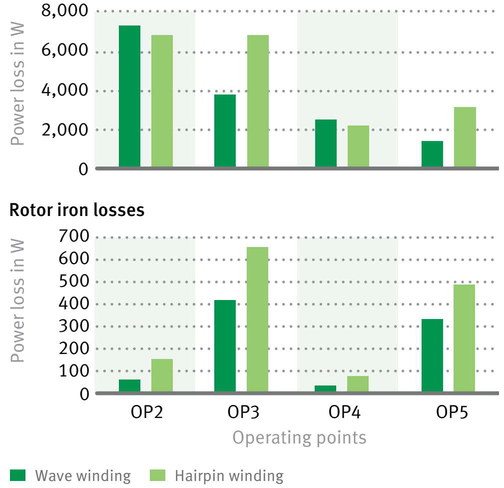
5 **Calculated level of efficiency in an electric motor with wave winding for various operating points**

Variants



Operating points

Operating point	OP2	OP3	OP4	OP5
Speed in rpm	5,300	18,168	4,542	18,168
Torque in Nm	265.1	64.5	146.5	36.5
Power in kW	147	123	70	69
Efficiency in %	94.9	94.9	96	95.3



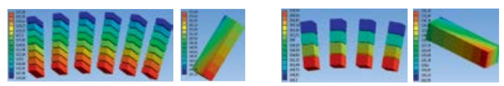
6 **Comparison of the stator and rotor losses for the electric motor in an electric axle application with wave and hairpin winding**

A second motor with the same installation space requirements was produced using hairpin technology, whereby 72 slots were able to be implemented. Subsequently, the stator and rotor losses at the same operating points were compared with each other – Figure 6. This shows that the stator losses in hairpin winding are somewhat less in the lower speed range than in the motor with the wave winding. In contrast, hairpin winding scores much worse in the upper speed range due to the high frequency losses. Moreover, the rotor losses are less at all operating points compared when wave winding was used, an effect that is mainly due to the lower harmonics.

In addition, the greater number of slots for wave winding results in a larger overall surface, which is useful for heat dissipation. This becomes evident in the temperatures that occur in the rotor and stator when the losses occurring in the actual design are considered. Figure 7 shows a comparison at a low load and high load operating point.

Finally, the efficiency was compared in the cycle (WLTP). For wave winding, there was an average level of efficiency of 94% for the motor, while the

Thermal behavior



Identical losses assumed		Winding temperatur	Magnet temperatur
OP4	72 Nuten	164 °C	160 °C
	96 Nuten	151 °C	151 °C
OP5	72 Nuten	137 °C	157 °C
	96 Nuten	133 °C	153 °C

Losses according topology		Winding temperatur	Magnet temperatur
OP4	72 Nuten	155 °C	175 °C
	96 Nuten	151 °C	151 °C
OP5	72 Nuten	210 °C	232 °C
	96 Nuten	133 °C	153 °C

7 **Comparison of the stator and rotor temperatures at two operating points in an electric motor with wave and hairpin winding**

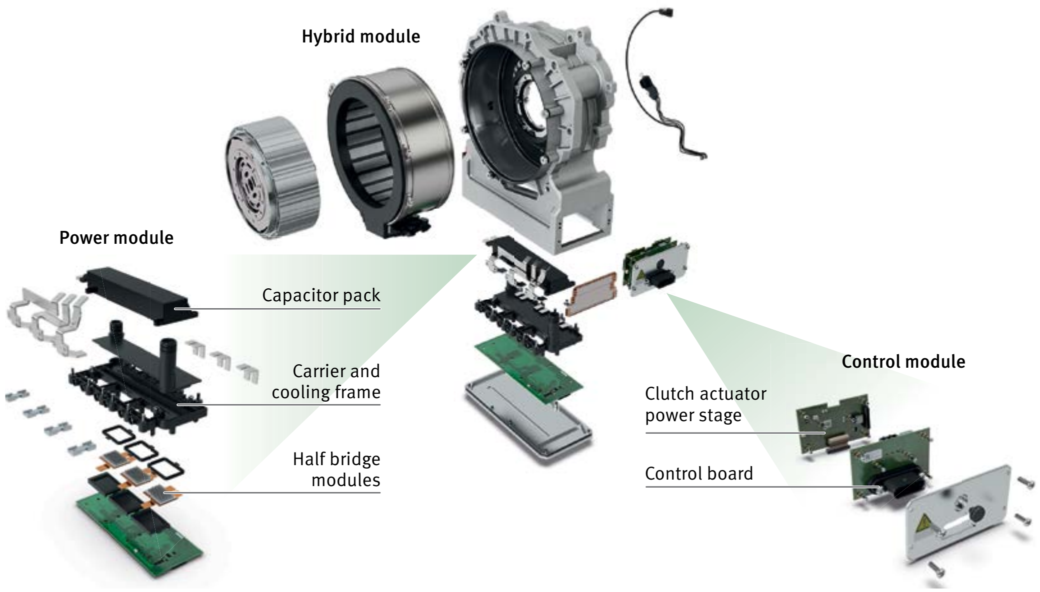
value for the motor in hairpin design added up to an average of 89%. One general advantage of the distributed winding is that the stator can be used for permanently-excited synchronous motors, asynchronous motors, or even separately excited synchronous motors, thereby making it suitable as the basis for a modular system.

Power Electronics

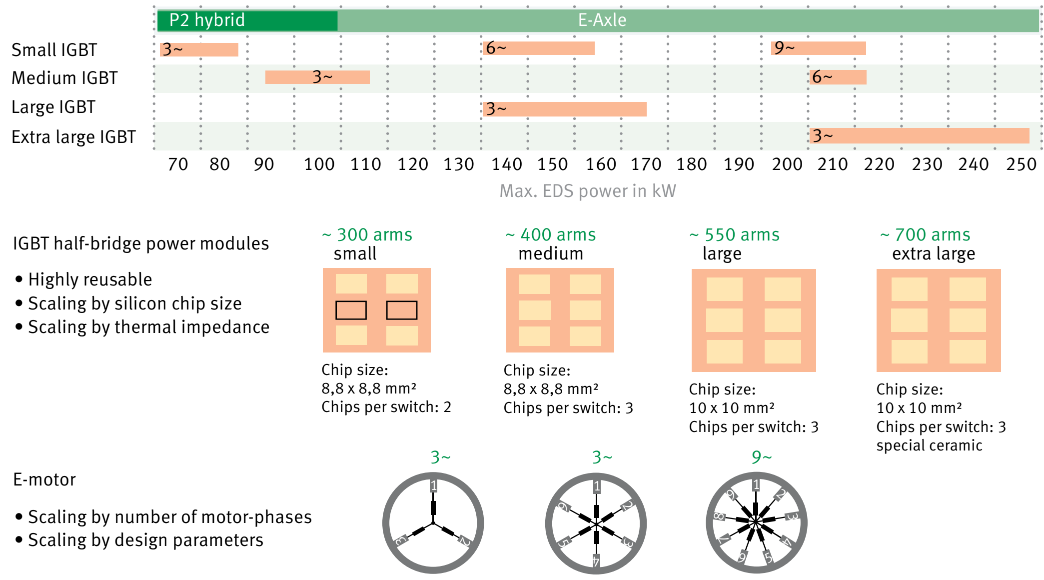
Due to the use of power electronics in automobiles, its integration in powertrain components such as electric axles, hybrid modules, and dedicated hybrid transmissions, and its high quantities, it is also subject to specific requirements that must be carefully borne in mind:

- High level of robustness, since the variance of the user profiles increases at high quantities and there are greater environment-related requirements in highly-integrated powertrains (vibrations, ambient temperature)
- High power density in narrow installation spaces, resulting in special requirements for leading away power losses
- Flexibility of the design, as the installation spaces vary in the different applications
- Optimal current formation to minimize harmonic losses
- High torque accuracy under all operating conditions
- Operational reliability in a wide range of operating voltages in order to respond to specific battery configurations
- Functional safety

As with electric motors, a detailed analysis would be able to add to this list indefinitely. In addition, power electronics need to be adapted to the application-specific power requirements. To scale power electronics in various applications in light of the specified requirements, Schaeffler has developed a modular concept, which is explained as follows on the basis of a



8 **Modular concept for power electronics, taking a hybrid module as an example**



9 **Power scaling through the size and number of IGTB semiconductors and the number of phases in the electric motor**

sample implementation in an 85-kW class hybrid module – Figure 8.

To begin with, there must be strict separation between the control module and the power module. The control module consists of the control board, the actual intelligence of the power electronics, and an optional actuator power amplifier. As a rule, the control board can be used to depict two engine control channels. In the sample case, the second channel is provided for controlling the K0 actuator in the P2 hybrid module. As an alternative, it can also be used for a gearshift actuator in a two-speed axle, a parking lock, or a second tractive drive in a power split transmission.

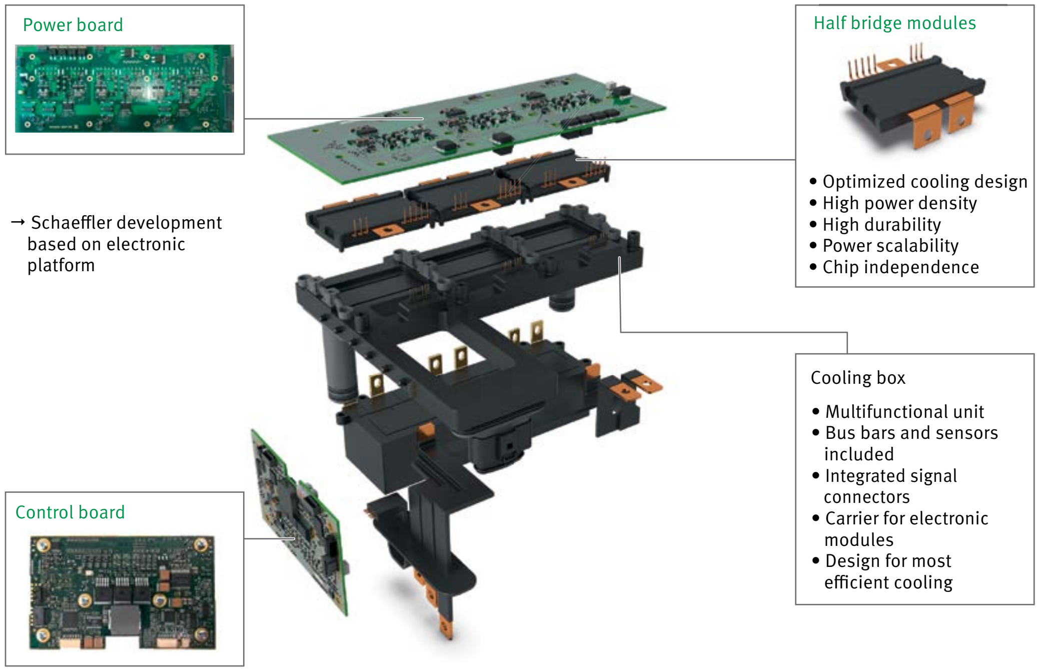
In the power module, it is first necessary to consider the scaling of the power semiconductors, which make up a large portion of the value and entail a lot of the effort needed for the qualification. Schaeffler has decided to use half-bridge modules as the smallest unit, since they also enable the construction of multiphase drive systems (with more than three phases). In order to satisfy the range of requirements, two mechanical sizes and one basic population quantity (chip size and chip number) were defined. The interfaces to the outside for contacting and cooling are always the same. Moreover, for the sake of robustness, special importance was placed on avoiding soldering connections and aluminum bond wires in the mounting and connecting techniques used at critical points in the module. In the current design, the IGBTs (Insulated-Gate Bipolar Transistors) and parallel diodes are made using silicon semiconductors. However, the use of wide-bandgap semiconductors is being considered as an alternative, as it is conceivable in the same module at activation frequencies of up to 20 kHz. Frequencies even higher than this require adaptations to be made to the basic design of the power electronics. Figure 9 illustrates the basic scaling approach.

The consistent use of sinter technology increases the cycle stability of the IGBT modules by a factor of 10 in comparison with conventional aluminum bond wire technology. Other basic components involved in carrying current are the capacitor and the bus bars, which also lead to a loss of power due to the flow of current.

The flexibility of the installation space is controlled by the central carrier frame (Figure 8 “Carrier and cooling frame”). This injection-molded part controls the flow of coolant, connects the current-carrying components to the heat sink, and arranges the basic components with respect to each other. CFD simulation is used to optimize this component in order to ensure symmetrical warming of the half bridges. Moreover, the electrical connections are optimized with regard to parasitic inductances and capacities. In the specific example, a power density of more than 30 kW/l was able to be represented.

Thanks to the flexibility of the installation space and the modular approach, it is easy to react to deviating requirements. Figure 10 shows an example of a design for use in an electric axle with a maximum output of 150 kW. An EMC filter can be integrated on the DC side as an option and can be designed big enough to enable shielding for the DC supply line to be done without. This represents a good option for optimizing the system costs at the vehicle level.

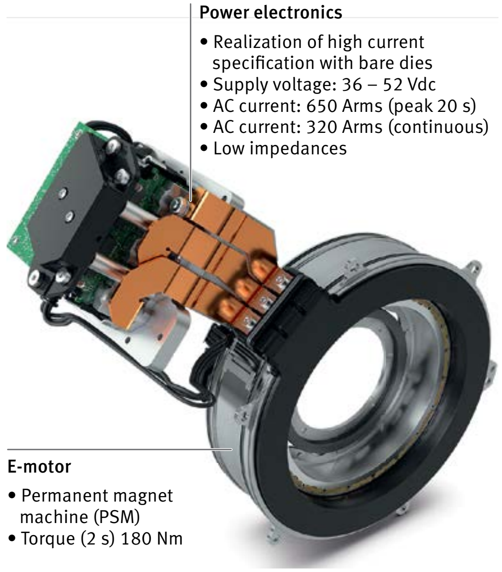
It was able to be demonstrated that the basic concept selected for the power electronics has the necessary flexibility for meeting the different requirements of P2 and P4 high-voltage drives with regard to the installation space and electric output.



10 Power electronics setup for an electric axle drive

In a further evolution step, it was checked whether the selected approach can also be used for integration in a coaxial P2 hybrid module with a voltage level of 48 V. Used as power switches in this case are MOSFETs (metal-oxide semiconductor field-effect transistors). The components are attached directly to a ceramic substrate (called “bare dies”). The substrate is connected with the coolant-carrying carrier across its entire surface. This makes for very good heat dissipation and thus very high power density, which is important for the relatively high current levels that can occur in powerful 48-volt drives. In the specific application with 15 kW of nominal power (20-second value), the maximum current that occurs is 650 Arms. The capacitors are attached directly via the MOSFETs in order to keep impedance low.

Since it was only possible to have an active length of 45 mm due to the available installation space, the electric motor was also designed with concentrated single-tooth winding. Despite the differences compared to a high-voltage application, key technology platform components were able to be used in this case as well, thereby allowing for a compact hybrid module structure – Figure 11.



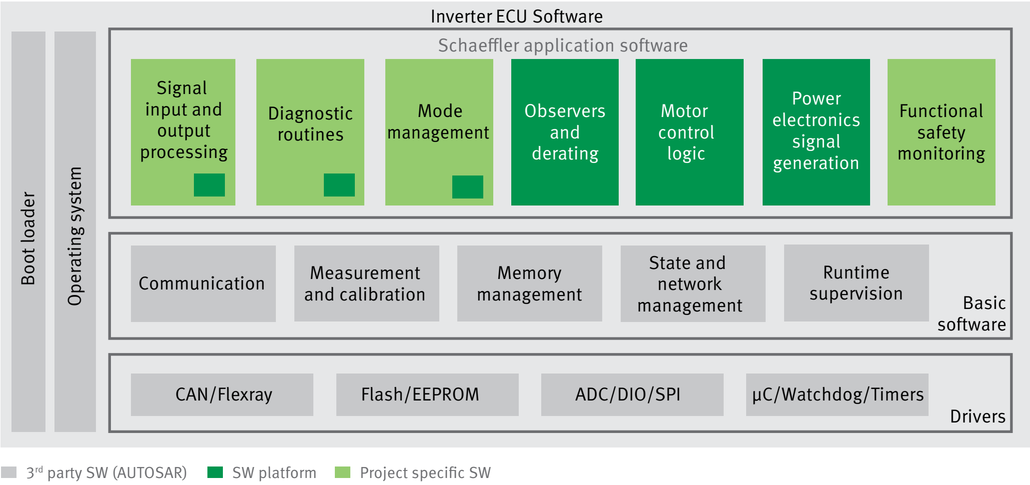
11 Integration of the power electronics in a P2 hybrid module at the 48-volt level

Software

The approach pursued by Schaeffler involves the development of an extensive function-oriented software library, which also includes the specifications for the required hardware. The following are key elements of the library:

- Analysis of sensor signals, such as for determining the rotor position, the phase currents, or temperatures at defined points
- Functions for motor control as a function of the motor type used (PSM, ASM)
- Functions for current control, such as a field-oriented control system that factors in all relevant influencing variables (e.g. field weakening)
- Superordinate controllers for functional integration in the powertrain, which can also be integrated as customer modules upon special request
- Monitoring functions, such as for controlling power derating for thermal reasons and for providing functional safety

In addition to software modules, the library also contains any necessary hardware circuits, along with their definitions and preferred components. Special rules define the implementation in the final layout to guarantee optimum heat dissipation or electromagnetic compatibility. This layout is prepared specific to the application in order to react to special customer re-



12 **Software architecture**

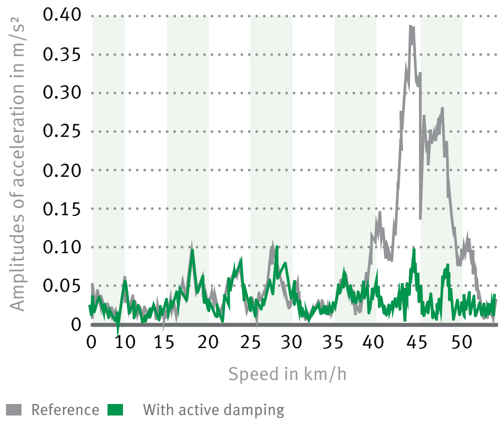
quirements. The various communication network options (CAN, FlexRay) have already been mentioned as examples.

With regard to the architecture, the software strictly follows the AUTOSAR paradigms in order to guarantee a high level of reutilization – Figure 12.

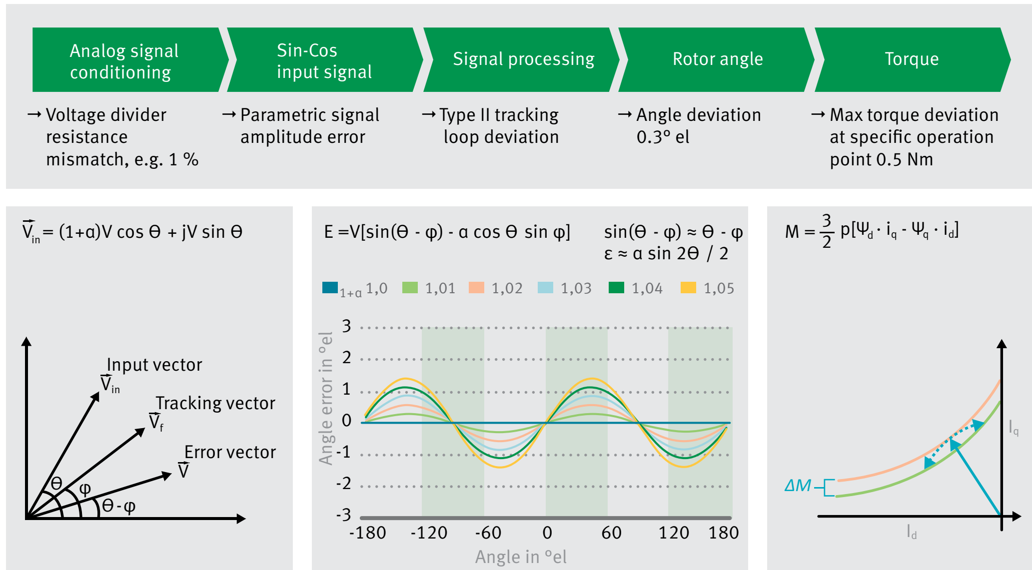
System Development

The system development for a specific drive unit based on the technology platform always depends on the boundary parameters required by the overall powertrain and – in part – the vehicle concept as well. This can be illustrated by a wheel-hub drive design in which disturbing acoustic phenomena occurred during operation. A systematic analysis of all of the components, the software, and the transfer paths that was conducted together with research partners KIT, FAST, and ETI revealed the cause: Magnetic field fluctuations generated longitudinal and transverse forces in the electric motor's stator, which were transferred to the bodywork via the chassis.

The initial countermeasure that presented itself was a change in the electric motor's design in order to prevent excitations in critical frequen-



13 **Vibration reduction in a wheel-hub drive through targeted torque variation in the e-machine**



14 **Propagation of torque generation inaccuracies in a typical e-machine**

cy ranges or orders. When this method reached its limitations, a reduced physical model was able to be used to adjust the engine control system. In the remaining critical frequency ranges, the electric motor worked as a damper by ma-king a slight, targeted change in the motor torque. Figure 13 illustrates the reduction in vibration amplitudes achieved in relation to the vehicle velocity.

Another example of the high relevance of a system-oriented development approach is torque accuracy, which is part of the specifications for every e-machine. However, each physical component in the torque generation chain is subject to certain tolerances that can cause the output torque to fluctuate. If the voltage in an analog signal generator for the rotor position varies by only 1 %, this can cause the angle during activation to be incorrect, resulting in a torque deviation of 0.5 Nm. Since the complete chain has been modeled – Figure 14 – it is possible to determine the effect of each individual component on the torque accuracy. It is necessary to estimate the anticipated deviation in relation to the operating point, which will make it possible to initiate any targeted countermeasures that are necessary.

Summary and Outlook

Due to a growing variety of electrified powertrains in ever greater quantities, new solutions are needed for the electric drive. Schaeffler has found an answer to the balancing act between variety and standardization with its scalable technology platform for electric drives.

This platform includes both the electric motor as well as the power electronics and the hardware/software for drive control. On the basis of different applications (P2 hybrid modules with high-voltage and low-voltage technology, P4 axle

	E-Axle 2011	E-Axle 2017
Power electronics (traction)	separately	integrated
Peak torque	2,000 Nm (10 s)	4,000 Nm (60 s)
Peak power	60 kW (10 s)	145 kW (60 s)
Overall lenght (flange to flange)	525 mm	515 mm
Weight	90 kg	80 kg

15 **Comparison of electric axle drives from 2011 and 2017**

drives), it has been shown that a targeted approach is able to cover a very broad application spectrum.

The technologies used in the platform correspond to the very highest demands for efficiency, power density, and scalability. It has been developed in a modular fashion so that hardware and software from third-party providers can be seamlessly integrated.

Future generations of electric drives will be characterized by increasing integration of electrical, electronic, and mechanical elements. If the opportunities offered by the Schaeffler technology platform are consistently put to use, it will be possible to obtain considerable improvements at the system level. For example, the current development stage of the axle drive mentioned above at the beginning, with a 15-kg reduction in the overall weight, has doubled the maximum torque, which is now available for 60 seconds instead of only 10 – Figure 15 – while the nominal power has more than doubled, going from 60 to 145 kW.

Literature

[1] Englisch, A.; Pfund, Th.: Schaeffler E-Mobility – With Creativity and System Competence in the Field of Endless Opportunities. 11. Schaeffler Kolloquium, Baden-Baden, 2018

[2] Reitz, D.: One Idea, Many Applications – Further Development of the Schaeffler Hybrid Module. 10. Schaeffler Kolloquium, Baden-Baden, 2014

23

The Innovative Schaeffler Modular E-Axle

Thorsten Biermann

Introduction

In 2011, Schaeffler presented a future-oriented powertrain in a purely electrically driven vehicle as part of the Active eDrive electrically driven concept vehicle. The demonstrator was based on a Škoda Octavia Scout and had two axle drives which provided not only electric driving but also an intelligent electromechanical lateral distribution of torque within both the front and rear axle. A particular USP of this axles drive at the time was the coaxial arrangement of the motor to the transmission and a differential with a planetary design. As this point in time, the permanent magnet synchronous motor was unable to operate at higher speeds which meant the maximum velocity was limited to approximately 150 km/h despite having two powered axles. However, the dynamic potential of electrically driven vehicles with additional torque vectoring functionality could now be experienced for the first time.

Both the electrical and the mechanical components of the electric axle drive have been developed further since then. This has produced a family of electrically powered axles which offers an optimum solution for a wide range of customer applications and platforms. The drive concept is suitable for use not only in purely electrically driven vehicles but also in hybrid applications. Shiftable, two-speed P4 solutions with high ratios and an additional disconnect functionality are available in order to cover the performance spectrum of both full and plug-in hybrid vehicles. The mechanical elements or the transmission are designed such that a comparatively high torque can be produced even when combined with an engine from the lower performance segment (48 volt). This means that the electric axle is not only the preferred choice, when attempting a significant reduction of CO_2 emissions, but it also allows additional all-wheel drive and thus provides optimum traction, even under difficult driving conditions.

In purely electrically driven vehicles, the concept offers both single-speed solutions as well as two-speed axles with powershift capability which provide an optimum power density with respect to the required start-up torque and the final speed due to an intelligent choice of ratio and the technical design of the drive motor. Due to the choice of either a coaxial or parallel axis arrangement, it is also possible to operate an extremely wide range of platforms from out-and-out sports vehicles, with high demands on the power density, to SUVs with high demands regarding the angular displacement of the side shafts.

It is also possible to equip each drive axle with additional functional modules irrespective of the powertrain arrangement or the output class. The possibility of adding subsystems, such as a shifting or even a torque vectoring module, allows the electric drive unit to be tailored to customer requirements and thus to generate maximum functionality if required. The multiple application of mechanical and electrical subsystems in different powertrains helps to reduce both development times and the associated costs considerably.

Basic idea – lightweight differential

A significant feature of the Schaeffler electric axle is the integration of a differential with a planetary design as an alternative to the bevel gear differential [1]. In terms of its functionality, there are no differences in comparison to the classic bevel gear differential. It provides uniform distribution of torque to the wheels and compensates differences in rotational speed during corning.

In terms of installation space requirements, the Schaeffler lightweight differential offers significant benefits compared to the classic bevel gear differential which is characterized by a compact radial design due to the lack of a drive wheel but requires significantly more space in the axial direction than the lightweight differential with the same torque capacity.

1 Lightweight differential with planetary design with O-arrangement bearing support for parallel axis drive axles

The reduction in axial space is a particularly good reason for using the new differential gearing in the coaxial electric drive systems as it leaves significantly more axial space for the drive motor than would otherwise be possible with a classic bevel gear differential.

In addition to this, the lightweight differential, as the name implies, is approximately 30 % lighter than the bevel gear differential which can lead to a weight advantage of up to 3 kg in higher torque powertrains. This weight advantage is the main reason for its use in the Schaeffler parallel axis electrically driven axles.

Concept for coaxial drives

The transmission in the coaxial powertrain is designed such that one or more planetary gear units are connected to a differential. This means there is no final drive in the differential and the differential itself is driven directly via the housing. An output shaft for the differential is connected to the wheel through the rotor on the drive motor which means that the rotor shaft has to have a hollow design. Even in a powertrain with a coaxial arrangement, the main point of focus is to design the drive system to be as short as possible in the axial direction in order to be able to give the side shafts as much space as possible. If the side shafts are too short, this may lead to the angle of deflection exceeding the value which can be technically achieved on the CV joint. For a given space, the objective is therefore to design the transmission to be as axially compact as possible in order to provide more space for the drive motor and thus to achieve a greater output torque.

Even in drive axles with a coaxial arrangement, the narrow design of the spur gear differential therefore offers significant benefits when compared to the bevel gear differential with its spherical differential cage. The differential gearing of the lightweight differential described above is also particularly well suited to being connected to a further planetary gear stage. This design of the spur gear differential thus allows simultaneous use of the differential housing as planet gear carrier for the transmission ratio gearing. The gearing of the differential is thus located in a housing with a planetary gear unit. The torque at the differential cage in this design is increased by the factor $(i_0 - 1)$ as a function of the stationary gear ratio of the epicyclic gearing i_0. Here, I_0 equals – $Z_{Ringgear} / Z_{Sun.}$ The “Z” here stands for the number of gear teeth. The stationary gear ratio has a negative sign as the number of gear teeth in the ring gear by definition has a negative sign. The total

2 **Stepped planetary gear set with integral compensation gearing on the differential**

ratio of the transmission is therefore a positive value irrespective of the type of differential gearing.

In an actual application, however, using just the spur gear compensation gearing combined with a two-speed planetary gear unit did not achieve the required results. The motor was still too long to fit in the available space. Using a so-called stepped planetary gear set offered a solution.

In a stepped planetary gear unit, the planet is provided with an increment and has two levels of gearing, see Figure 2. The use of a stepped planetary gear set allows ratios of more than 10 which would otherwise only be possible with two linked planetary gear units. A second glance however reveals a further benefit which is not feasible with an unstepped planetary gear set.

While the gearing for the differential is arranged axially adjacent to the planetary gear stage on classic planetary gear units, using the stepped planetary gear set allows the spur gear on the differential to be shifted into the space occupied by the planetary gear set. This produces a highly integrated and extremely short transmission concept, see Figure 3.

It is possible to integrate the differential because the stepped planetary gear set has some special design features which are not included in standard planetary gear sets. These include the fact that the sun gear only partially protrudes into the planetary gear set space and meshes with only the large planet gear. As the differential housing and the planet carrier are combined in a single

3 **Transmission with highly integrated stepped planetary gear set**

Technical data	
Weight transmission in kg	16
Power in kW max./continuous	190/100
Output torque in Nm max./continuous	3,960/2,250
Input speed in rpm	18,200
Dimensions transmission in mm	Ø 300 x 150
Ratio	9

Customer benefits

- 1-speed system
- Compact axial space due to combination of stepped planetary gearset with spur gear differential
- Different optional ratios
- Low weight
- Series development

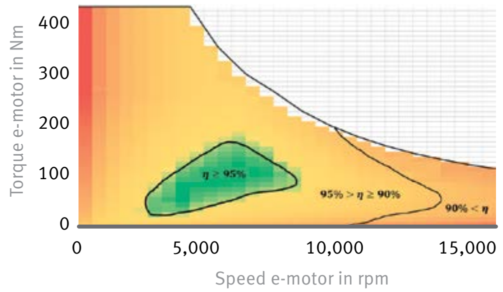
component, it is possible to position the sun gear of the differential radially within the small planet gears. In a further step, the differential compensation gears are embedded partly between the small planet gears of the stepped planetary gear set which allows the stepped planetary gear set to overlap approximately half of the differential without contacting the gears. The planetary gear unit and the gears of the differential are effectively nested within each other.

In actual applications, it has been possible to produce a transmission which has an axial extension of no more than 150 mm and which has a differential cage with an available nominal output torque of approximately 4,000 Nm. Due to this compact design, there is an option of integrating longer, powerful motors into the drive axle without infringing the installation space specifications in the vehicle.

A permanent magnet synchronous motor is currently in development which produces a torque of up to 440 Nm at a maximum power of 190 kW. The nominal torque is still 250 Nm at a nominal power of 100 kW. This motor will soon be available as a powerful electric drive in combination with the very compact transmission. This drive also has a comparatively high efficiency due to its design as can be seen from Figure 4.

The drive is extremely efficient: This is reflected in efficiency levels greater than 95 % in the relevant WLTC operating ranges between 2,500 and 7,500 rpm, which represents a velocity of approximately 30 to 90 km/h. The efficiency only drops to less than 95 % at high torques and speeds. However, these operating conditions occur very infrequently in the WLTC. The PSM can operate at speeds of up to 18,000 rpm which allows the vehicle to run at a maximum velocity of over 200 km/h.

Figure 5 shows a schematic overview of the electric axle drives for the coaxial installation. At this



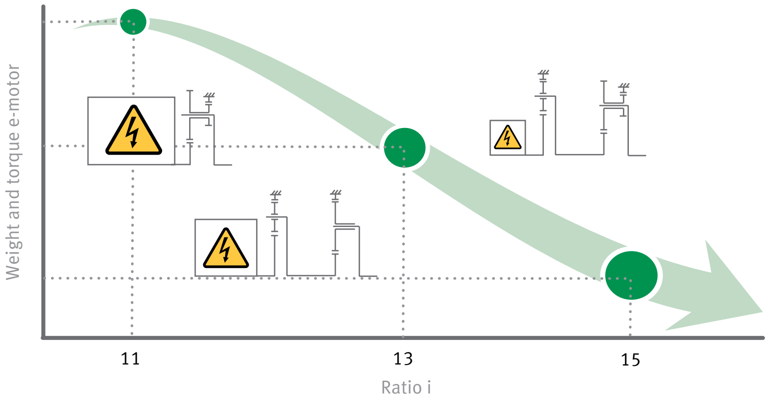
4 **Efficiency of the drive axle with 440 Nm PSM and power electronics**

point, it only shows the planetary gear set in half-section without the differential which is connected to the planet carrier. The stepped gear set is shown on the far left in combination with a powerful drive motor such as the PSM described above with a short-term output of 190 kW and a maximum torque of 440 Nm.

Stepped gear sets are currently in development which allow transmission ratios of up to 12 and which have been optimized in terms of the gear contacts to provide high profile and step overlaps.

A high end speed is not always necessary or appropriate with respect to the target markets for electric drive systems as the legislators in many regions have maximum speeds of considerably less than 200 km/h.

A possible criterion in the decision-making process is the dynamic driving behavior in the lower and mid speed range. At lower maximum speeds, it makes sense to raise the transmission ratio further. This is achieved by coupling with an additional planetary gear set. The transmission ratio is thus raised in a defined manner based on the speed range of the drive motor and the motor torque simultaneously reduced. These measures reduce the mass and space requirement of the drive motor which, amongst other things, has a positive effect on the production costs.



5 **Schematic representation of the coaxial drive concept**

In the trade-off between the required continuous power, the maximum speed, the desired acceleration in the low speed range and the voltage, adjusting the transmission ratio in this way provides a means of designing a system with good overall cost efficiency. The relationship between the total transmission ratio and the dimensioning of the electric motor is shown in Figure 5 in a considerably simplified schematic. In practice, there should be no detrimental effects on the acoustics, cooling, and the level of efficiency of the whole system.

Figure 6 shows the actual design implementation of the concept in which permanent magnet synchronous motors, with continuous outputs of up to 80 kW and maximum motor torques of 250 Nm, are combined with a range of gear sets. By varying the gear stages, the drive systems are matched to various different end speeds and driving torques as well as wheel diameters and vehicle weights.

The solutions described above are suitable for vehicles with high end speeds as well as those with a high drive torque. In the drive axles shown, the interfaces between the electric motor and the transmission and the first planetary gear set have an identical design. Furthermore, all the gear sets have the same differential gearing. The only differences to be found are in the running gearing in the second planetary gear stage. The differences between the variants are therefore an overall ratio of 15 for the sun and 19 for the large planet while the ring gears and the differential housing have an identical design. Both variants have a stepped wheel set which overlaps the differential gearing.

The drive axle with an overall ratio of 11 does not have the stepped gear set but retains the gearing of the small planets and the corresponding ring gear which is also used in both the other variants. In the low ratio variant, the differential gearing is arranged in the housing axially next to the differential compensation gearing and is not nested with this component. The transmission layout is the same as the middle schematic in Figure 5. The gear stage has a high overall ratio of 19 and is a single-speed system suitable for urban traffic only due to the low output speed; however, it does provide a starting point for multi-speed systems. This is discussed in more detail elsewhere.

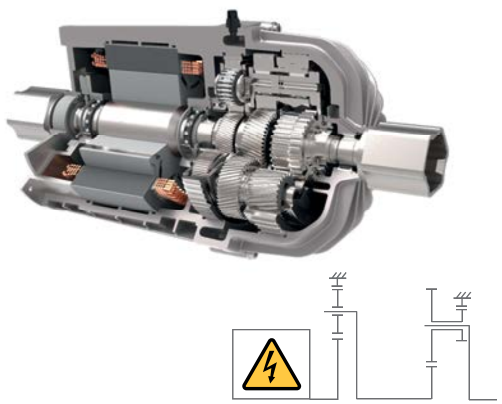
Figure 7 provides a detailed representation of the output data for the drive axle with an overall transmission ratio of 15. The electric axle has a permanent magnet synchronous motor which has a maximum torque of 250 Nm at a continuous output of approximately 80 kW. The maximum vehicle speed with a transmission ratio of 15 is approximately 150 km/h. This results in a maximum available torque at the output shaft of 3,750 Nm whilst retaining a continuous torque of 2,250 Nm. This provides dynamic driving for a C-segment vehicle both within and outside the urban environment. In spite of these ambitious performance data, the electric drive system weighs only 75 kg, including the integrated power electronics.

6 **Schematic representation of the coaxial drive concept**

In heavier vehicle platforms, for example a purely electrically driven SUV, it is possible to install two drive axles. In this case, it makes sense to design the drive axle with a lower transmission ratio of, for example, 11. By increasing the continuous power to 160 kW it is easily possible to achieve end speeds in the region of 200 km/h despite the greater road resistances. The lower output torque of 2,750 Nm is more than compensated by the simultaneous powering of the front and rear axles.

Shiftable systems in coaxial drives

Single-speed electric axles should always be designed such that both the starting torque and the vehicle end velocity are safely achieved within the fixed ratio. A single-speed axle with a ratio of 19 has the disadvantage that the maximum speed is limited to approximately 120 km/h due to limitations on the speed of the PSM to approximately 18,200 rpm. However, if the solution is designed with a shifting system, the drive unit can be an interesting solution for more powerful plug-in hybrids.



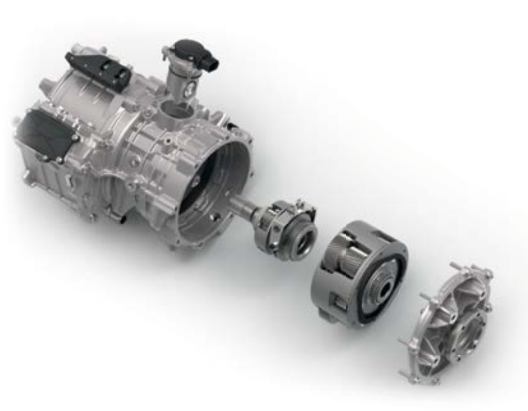
Technical data

System weight in kg	75
Power in kW max./continuous	150/80
Output torque in Nm max./continuous	3,750/2,250
Input speed in rpm	18,200
Dimensions in mm	Ø 285 x 425
Ratio	15

Customer benefits

- Modular system
- Compact design with high power density
- Different optional ratios
- Customer acquisition

7 **Electric axle drive for C segment with reduced maximum speed**



Technical data	
System weight in kg	69
Power in kW max./continuous	75/45
Output torque in Nm max./continuous	2,850/1,615
Input speed in rpm	18,200
Dimensions in mm	Ø 285 x 450
Ratio	19/6.4

Customer benefits

- P4 with full e-drive option for midsized SUV
- WLTC driving cycle without combustion engine possible
- Electric AWD option w/o mechanical cardan shaft, less mech. losses
- 1st gear for high output torque and acceleration
- Boost and recuperation at high speed due to 2nd gear
- Optional with 48 V technology

8 **Two-speed electric axle drive for powerful PHEVs**

By integrating the shifting system, the drive will gain a neutral gear position and a second gear with a ratio of 6.4. The neutral gear position allows the drive to be disconnected and the second gear allows the vehicle to be driven at velocities of more than 120 km/h. In second gear, the first planet gear stage is bypassed and the 6.4 ratio is made available using only the stepped gear set. The drop in tractive torque during the shifting process is an impediment to using this system in purely electrically drive vehicles.

In hybrid powertrains, the short-term loss of tractive force can be compensated by adopting a suitable operating strategy for the conventional powertrain. This allows conventional shifting with an electric actuator to be used. A drive motor with a peak torque of 150 Nm is thus sufficient to provide extra torque of up to 2,850 Nm via the second axle in first gear. The auxiliary drive therefore allows the acceleration process and traction performance to be optimized by an electrical boost. In urban traffic, the axle can also be used as a standalone electric drive for speeds up to 120 km/h if, for example, driving with a conventional internal combustion engine is not allowed by law. The range of the vehicle depends on the capacity of the battery. A fully-fledged electric all-wheel drive is also available for difficult driving conditions. However, the drive's primary aim is to reduce CO_2 emissions compared to a conventional powertrain. A less powerful 48 volt drive is also available for this purpose in addition to the high-voltage drive described above. Further details on possible reductions in CO_2 by using corresponding P4 solutions are provided as part of the contribution on the 48 volt systems [4].

Concept for parallel axis drives

Portability

The Schaeffler transmission concept is not restricted to just coaxial drives. The basic concept of the planetary design can also be transferred to parallel axis drives [3] in which the differential is arranged parallel to the rotor shaft, see Figure 9.

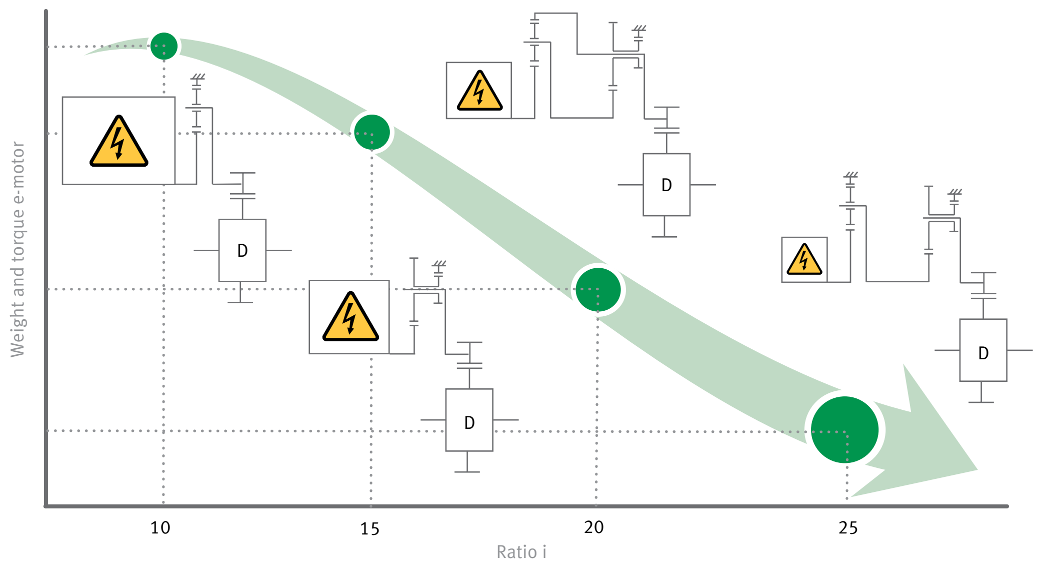
Particular attention is given in the design of the parallel axis concept to redeploying the gear set components and subsystems from the coaxial concept. Some of the components which have been developed for the coaxial application can also be used in the parallel axis drives with only marginal changes to the design. An advantage of redeployment is the existing depth of testing. Using planet gear sets which have already been validated or which are already proven in volume use considerably reduces the cost of developing new drives. This practice reduces development times and costs and simultaneously increases product safety and quality.

Shiftable systems in parallel axis drives

As already explained in the section "Shiftable systems in coaxial drives", there are considerable advantages to be had in hybrid vehicles by using a shiftable two-speed drive. This means the hybrid drive can be designed in such a way that a high output torque is achieved at a relatively low electrical capacity. A further way to ensure a comfortable drive for hybrid vehicles, as an alternative to the operating strategy described above of topping up the tractive force via the internal combustion engine, is to reduce the effect of the break in tractive force by modifying the design of the powertrain.

This is achieved by displacing the gearshift to a speed range in which the output of the electric drive is already significantly less. Compared to the capacity to recuperate braking energy, the tractive force made available at this point is less significant.

Figure 10 shows a two-speed solution for the parallel axis arrangement. The combination of planet and spur gear drives produces a very compact design. The wheel distance of the rotor shaft from



9 **Schematic representation of the parallel axis drive concept**

the differential output shaft is only 127.5 mm. The transmission has a ratio of 15 in first gear and 5 in second.

The design of the shifting module, including the actuator, is therefore identical to the solution already described for the coaxial powertrain. At the current stage of development, the system described above is fitted with a high-voltage synchronous motor which provides a maximum power of 100 kW and a continuous power of approximately 60 kW.

The first gear covers the speed range up to 120 km/h while the second gear is intended exclusively for longer distance travel at higher speeds. Here it mainly serves to recuperate electrical energy in the braking phases and to provide an electric sailing function (active sailing). The system is mechanically decoupled in neutral and this improves the efficiency during passive sailing as well as when driving using only the internal combustion engine.

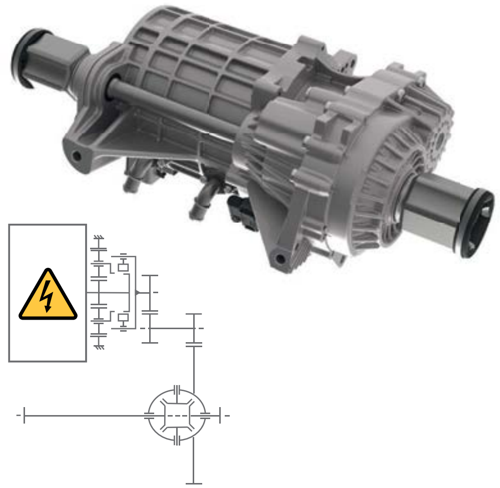
Due to the high overall ratio in first gear, the drive has a maximum output torque of approximately 3,000 Nm at low speeds. A vehicle equipped with this unit therefore achieves very good values for driving dynamics, even when driving purely electrically in urban environments. The active components in the motor can also be used in the two-speed coaxial electric axle. At the same time, the parallel axis drive can also be fitted with the less powerful drive motor from the coaxial variant described above, see Figure 8.

Torque vectoring

Torque vectoring allows the torque produced by the drive to be dynamically distributed to all wheels of the vehicle. To understand how torque vectoring works, it is first necessary to look at the function of the mechanical differential in detail: Leaving aside the lock values and the internal friction, the differential distributes the input torque in equal measure to both the wheels on an axle. The speed of the wheels changes with the radius of the corner. The wheel on the inside of the corner rotates more slowly than that on the outside which has to cover a greater distance. If the lock value of the differential is now considered, this produces the following scenario: A greater lock value usually produces poorer vehicle steering behavior. This is due to the fact that the wheel on the outside of the corner is slightly deaccelerated with respect to the differential housing by the internal friction in the differential and the wheel on the inside of the corner is correspondingly accelerated. The produces slightly greater torque on the wheel on the inside of the corner.

This effect is also used by the torque vectoring system which accelerates or brakes at least one of the two wheels. This option is achieved in the Schaeffler axle drives by using an electromechanically-driven, three-stage planetary gear drive, the so-called superimposing transmission. By using this transmission and an additional traction motor, this produces a specific “dynamic bracing moment” between the output shafts. The wheel, which is then accelerated by the normal adjustment range of the mechanical differential, transmits a greater torque and produces the correspondingly greater tractive force. This tractive force thus ensures there is a corresponding yawing moment around the vehicle’s vertical axle. The Schaeffler torque vectoring module can be combined with all drive axles. Figure 11 shows this in combination with the coaxial concept.

The superimposing transmission of the torque vectoring module basically consists of two identical planetary gear units. Both of these identical planetary gear stages share a common sector hub; the output is via two identical ring gears



Technical data	
System weight in kg	74
Power in kW max./continuous	100/60
Output torque in Nm max./continuous	3,000/1,650
Input speed in rpm	18,200
Offset/System length in mm	Ø 127.5 x 485
Ratio	15/5

Customer benefits

- P4 with full e-drive option
- WLTC driving cycle without combustion engine possible
- Electric AWD option w/o mechanical cardan shaft, less mech. losses
- 1^{st} gear for high torque and max. acceleration
- Boost and recuperation at high speed due to 2^{nd} gear
- Optional with 48 V technology

10 **Two-speed electric axle drive in parallel axis arrangement**

which are connected via a further planetary gear stage with an output shaft and the differential housing. The sun of one of the planetary gear units is connected to the housing while the other is connected via a spur gear stage to the servomotor which is arranged in parallel.

When the vehicle is traveling in a straight line, neither the differential nor the superimposing transmission or the rotor of the servomotor rotates which minimizes any losses. No differential torque is produced unless the servomotor is activated. The wheel torques are identical and the servomotor is not subjected to any moments. Due to the inertia of the superimposing transmission and the servomotor, the actively controllable differential behaves like a conventional differential with an increased lock value when deactivated.
If the servomotor turns the sun of the superimposing transmission to which it is connected, this produces a relative rotation between the ring gear and the differential output shaft. This in turn produces the differences in speed at the wheels. The superimposing transmission allows the servomotor to have a ratio of around 40 to the wheel. If a torque of approximately 30 Nm is produced at the servomotor by this gear ratio, this produces a differential moment of 1,200 Nm between the wheels. This order of magnitude is required to ensure the system functions adequately across almost the whole speed range. In this type of system, significantly less electrical system power is required for lateral torque distribution than with drive concepts which have one electric drive per wheel. A maximum output of 6 to 7 kW is therefore sufficient for the servomotor to be used in the vehicle.

In addition to this, a system architecture which features a combination of a traction drive unit and an additional torque vectoring module has significantly better acoustic characteristics than systems which have two main drive motors with the relevant gear stages. One reason for this is that two identical systems have an acoustic overlay within a single axle suspension point which must be moderated by expensive secondary measures in the vehicle.

11 Coaxial drive concept with additional torque module

The electromechanical torque vectoring system can be combined with both the coaxial and parallel axis drive concepts described above. The “Active eDrive” is thus born from the purely electric axle drive and, based on current vehicle tests and investigations, this represents a mile stone in terms of functionality and dynamics. The system combines the electric axle drive and an intelligent lateral distribution of torque which can be used to support the steering, to aid traction, or to stabilize the vehicle when it is at its limits.

Summary

Drive technology for vehicles is currently in a state of enormous technological development. The transition from conventional powertrains to hybrid and purely electrically driven solutions represents a major undertaking for everyone involved due to the short development times available. Both suppliers and OEMs have an urgent interest in generating overarching standards and solutions which should lead to synergies across the individual platforms.

Schaeffler is therefore proposing an axle concept which takes into consideration the demands on drive units for both hybrid and purely electric vehicles. The resulting modularity allows cost reductions to the extent that, on the one hand, they become manageable and, on the other, they lead to attractive development times and costs. It is therefore not enough for the technical solutions merely to reflect the current state of technology but instead they should offer greater power density and functionality.

Schaeffler is therefore taking the step from a pure electric axle concept to a flexible electric axle configurator which uses adaptable systems to meet the various demands of function, maximum speed and driving dynamics, and thus also produces an optimum result with respect to costs, mass and envelope. A largely modular approach allows individual components and modules to be used irrespective of the installation position and functionality of the drive. In the future, the electric axle configurator will allow extremely compact and highly integrated systems with a high power density and performance to be compiled in short development cycles and in accordance with customer specifications.

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Intelligent Thermal Management for Hybrid Powertrains

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Michael Weiß

Introduction

In order to maximize the efficiency of future powertrains, it is necessary to take more than just the thermodynamic, mechanical, and electrical energy paths into consideration. Indeed, it is vital to optimize the heat balance of the drive system and of the individual components as well. Targeted control of the heat flows is essential for this. Schaeffler has been involved in series production since 2011 when it introduced the first thermal management module for use in a gasoline engine powertrain [1]. Since then, both the module as well as its components have been consistently developed further, with the expertise for system development and validation growing considerably at the same time.

Components for Active Thermal Management

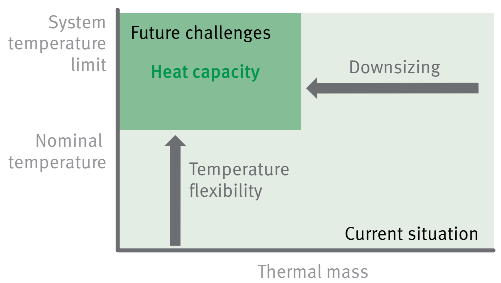
New Challenges

The requirements for thermal management will continue to go up in the future. On the one hand, this is attributable to the trend of developing especially efficient combustion engines. The engine cooling circuits of "downsized" engines with a high specific output are characterized by high heat flow inputs, which result from the lower heat capacity of the base engine and the integration of the components for exhaust turbocharging, among other things – Figure 1. On the other hand, the widespread use of start-stop systems and increasing hybridization are leading to discontinuous heat inputs after cold start.

These requirements make it necessary to respond much more quickly to changes in the operating mode. It must be possible to activate heat sinks much faster, i.e. in less than a second. Classic thermostat control is too sluggish for this. Instead, the engine and vehicle operating modes are taken as the basis for calculating the energy quantities that are initially present and proactively setting the temperature accordingly. It is already possible to implement a corresponding control strategy in the first generation of the thermal management module. This results in CO_2 savings of approx. 3.5 percent in the NEDC. In addition to directly reducing the CO_2 emissions, the thermal management module also enables the implementation of functions such as "active engine heating" or "active transmission heating" for which additional credits are issued according to the US CAFE standards. For each of these functions, 1.5 g of CO_2/m are credited to a passenger car certified in the US.

Second Generation of the Thermal Management Module.

The current second generation of the Schaeffler thermal management module continues to be based on rotary slide valves, which control the flow of coolant depending on the driving situation [2] – Figure 2. Thanks to the new actuator concept, considerable improvements with regard to functionality and installation space requirements have been achieved compared to the first generation.



1 **High heat inputs and low heat capacity will characterize the engine cooling circuit of the future.**

2 **Cross-section of the second generation of the thermal management module**

Two independent rotary slide valves are utilized as actuators in the second generation, one of them for volume flow regulation of the engine cooling circuit and the other for separating the sub-circuits in the cylinder head and cylinder block. This makes it possible to implement a "split cooling" principle, whereby as much heat as possible is initially stored in the cylinder block following a cold start in order to reduce the friction, while the temperatures in the cylinder head are able to be adjusted to the admissible thermal loads of the fuel mixture systems. Another advantage that results from the use of two actuators is the greater flexibility with regard to the arrangement in the engine peripherals.

The actuator for the split cooling regulation is designed as a smart valve, which only requires the target setting for the angular position from the engine control system. Based on the temperatures that occur and additional information, the integrated control system calculates the correct position and constantly adjusts it. The actuator used is a smart single valve, which will be described in greater detail in the following section. It meets the requirements of on-board diagnostic capability.

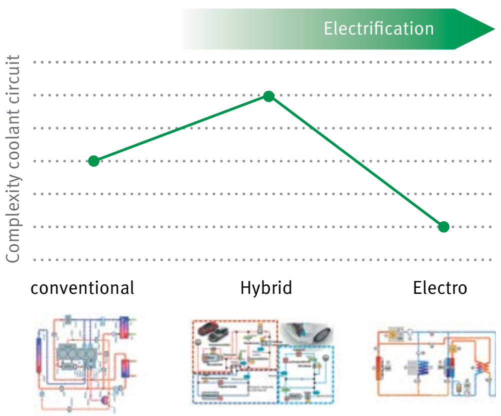
The two-actuator concept makes a different design and considerably greater flexibility of the rotary slide actuation possible. The first generation still used a continuous shaft supported by bearings on both ends, which resulted in higher flow resistance. This was replaced by two shaft studs (Figures 3 and 4), which act directly on the two rotary slides from the actuator side, meaning that they are not inside the flow of coolant, thereby reducing the loss of pressure.

3 **Main valve**

4 **Auxiliary valve**

Decentralized Thermal Management in the Electrified Powertrain

Electrification causes the complexity of thermal management to greatly increase, since additional components and cooling circuits need to be integrated, as shown in Figure 5 based on the schematic diagram of typical cooling systems. For example, there are already hybrid vehicles on the market that have more than three cooling circuits operating at different temperatures for the combustion engine, the electric drive, and the battery. Nevertheless, these circuits are interconnected via heat exchangers. The individual components are no longer assigned fixed roles as heat sinks and sources, but can readily alternate these roles. The large number of valves and thus switching options that occur in a complex hybrid powertrain make it clear that complete integration into a single thermal management module will probably not be possible. Both the installation space needed for the lines and module as well as the different coolant temperatures in the individual circuits argue against this.



5 **Complexity of the different drive cooling systems**

6 **Smart single valve in near-production design**

Smart single valves (SSV) coordinated via a central control system represent a solution for decentralized thermal management. Moreover, they also make it possible to modularize powertrain variations that automobile manufacturers use to react to different legal requirements for the target markets. The first generation of the SSV has been used in a new generation of three and four-cylinder gasoline engines at BMW since 2017.

The smart single valve is characterized by a very compact design – Figure 6. The core feature of the SSV is a 12-volt DC motor that actuates a rotary slide valve via a worm or spur gear. The motor works in a broad temperature range of up to 140 °C. It is controlled by a LIN interface to the engine control unit. An inductive rotary angle sensor is used for position detection, since this measurement method – unlike a Hall sensor – is characterized by a high degree of insensitivity to electromagnetic disturbance variables. The process of electrification results in increased current flows and a greater number of cables. Since these cables emit discontinuously, Hall sensors are subject to an external influence that can hardly be mastered. It is already clear today that the EMC requirements for drive components will continue to rise.

The structure of the SSV itself is modular: While the electrical part contained in the cover is always identical, regardless of the intended use, the lower part, particularly the coolant connections, can be adapted for the coolant flows required for the particular application – Figure 7.

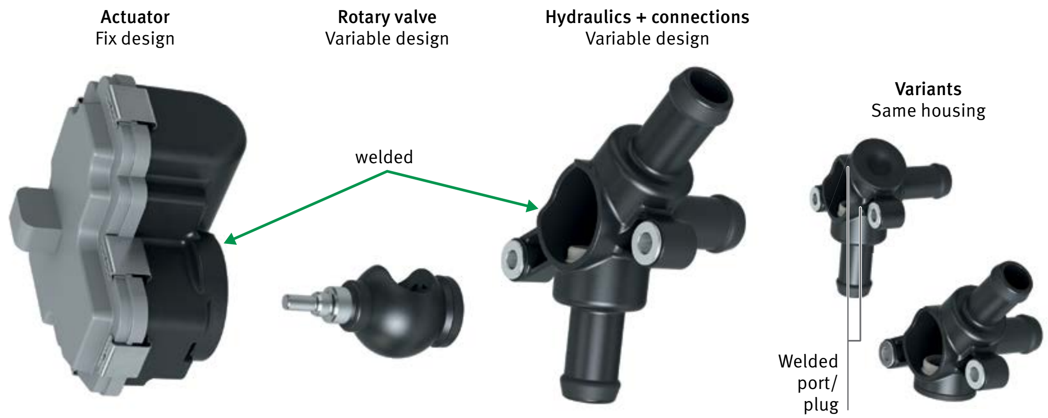
Freedom from internal leaks is a key requirement for the hydraulic part of the SSV during its operating life. This must be ensured at operating points where no coolant should be flowing. For this reason, Schaeffler has collaborated with a supplier to develop a sealing concept adapted to the mode of operation that makes leakage rates of less than 20 ml/min. possible, in the range of 0.5 to 3 bar.

The shaft for moving the rotary slide valve is also designed to be very robust. Used for this is a rigid, continuous shaft that has a direct actuator connection via its high level of rigidity. The ball is molded, which makes clearance possible between the drive and ball, supporting the control accuracy accordingly.

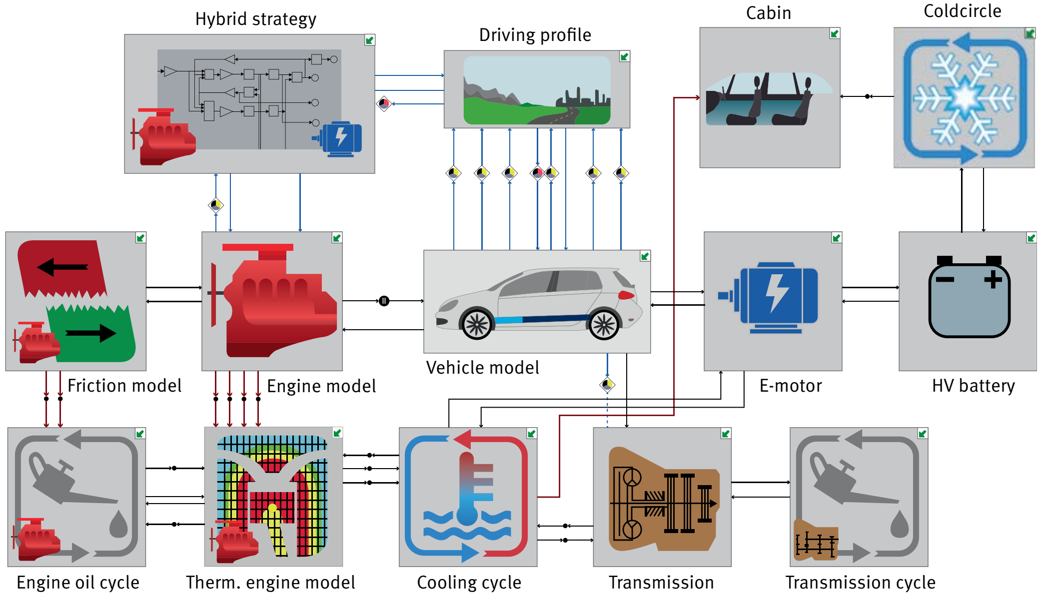
System Development and Validation

There is an intensive interaction between active thermal management for the powertrain and the vehicle and its operating conditions. For this reason, Schaeffler has used specially designed demonstration vehicles in the last ten years to build up its system expertise. An overall model for vehicle thermal management is being developed and validated on the basis of numerous measurement runs – Figure 8.

The individual blocks of the simulation model are transferred to more detailed models of the individual components. They are linked via freely definable heat flows that arise from the required

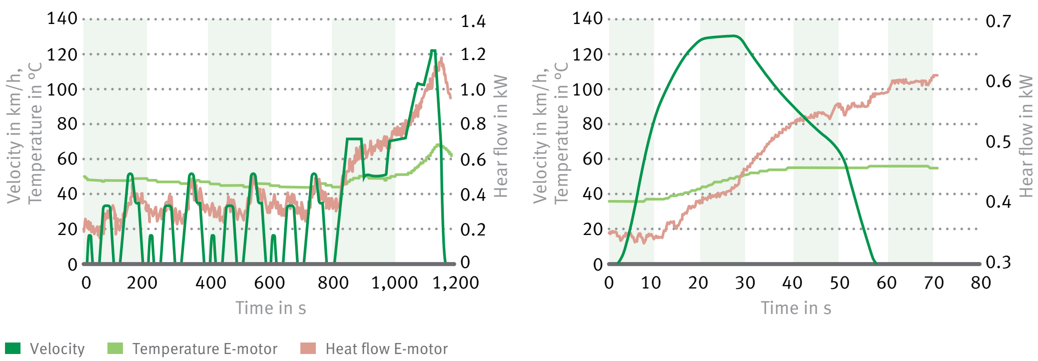


7 **Modular design in intelligent valves for coolant control**



8 **Simulation model for a plug-in hybrid vehicle**

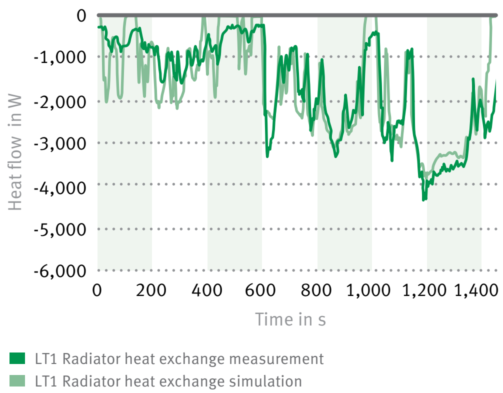
cooling output of the powertrain components or the required heat output, with the latter not only referring to the vehicle interior, but also to the desired minimum operating temperature of the individual powertrain components. This is particularly relevant for hybrid vehicles, since, on the one hand, the heat inputs provided by the combustion engine are reduced through the electric drive components and, on the other, the electrical energy stored in the battery is to be used as much as possible for propulsion alone and not for heating processes. In the practical design of a thermal



9 **Model calculations for the heat entry through the electric motor in the NEDC (left) and during full-load acceleration**

management system, the upshot of this is that most powertrain components can be used both as a heat source as well as a heat sink.

Linking a thermal management control system to the driving and hybrid strategy is also essential because the cooling output in electrical operation is a decisive factor with regard to performance. For example, the temporary peak performance of electric motors is generally devised with a view to the admissible thermal load. To this end, it is necessary to be able to simulate the heat inputs occurring in cycles and in certain driving situations and the heat dissipation capacity with as much accuracy as possible at an early development phase already. Figure 9 shows an example of the heat inputs of an electric motor during the NEDC and during full-load acceleration. It can be seen that, despite moderate acceleration levels, the average heat input at 600 W is greater in a specifically designed tractive motor during the driving cycle than during a single full-load acceleration (500 W). The background for this is a very rapid derating of the electric motor in combination with a relatively high thermal mass.



10 **Measured and simulated heat exchange in the low-temperature heat exchanger of a plug-in hybrid vehicle**

To validate the simulation model, extensive measurements were taken on a standard plug-in hybrid vehicle and compared with the simulation results. The measurements included the following parameters:

- Volume flows in all cooling circuits
- Coolant temperatures
- Temperatures of the engine and transmission oil
- Component temperatures in the electric motor and battery
- Charge status, voltage, and currents in the battery
- Heat inputs of the relevant components
- Heat outputs of the heat exchangers

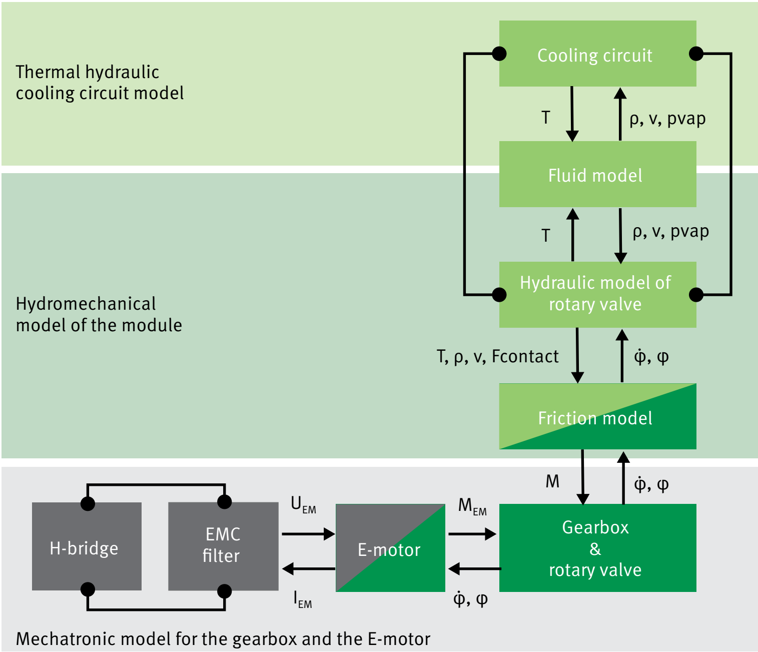
Based on the sample comparison of measurement and simulation results for the heat exchanger in the low-temperature circuit (battery cooling), Figure 10 shows that the current state of development has already achieved good model quality.

The system assessment answers the question regarding at what points it makes sense to use controllable actuators for optimizing the efficiency of the overall system. This gain in efficiency is to be demonstrated and quantified soon on the basis of a standard plug-in hybrid vehicle modified by Schaeffler.

Models of the Cooling Circuit and the Thermal Management Module

Beneath the complete system level, a detailed simulation of the material and heat flows and the electrical signal flow is necessary for designing actively switchable and dynamically controllable cooling circuits. To this end, hydraulic operating modes, electric loads, and mechanical parameters are coupled in a physical model that can already be used for virtual tests as a “digital twin” even before the first prototypes are built – Figure 11.

The basis of the physical model is a mechatronic model for the actuator transmission and electric motor that factors in both the electric loads that occur as well as the torques and heat flows down to the level of individual rotary valve positions. It interacts with a hydromechanical model of the thermal management module or the individual actuators, which takes into account the friction caused by the valve movement. This submodel is in turn linked to the thermohydraulic model of the entire cooling circuit.



11 Structure of a physical model for designing thermal management

The purpose of this kind of a "digital twin" of the actual components of a thermal management system is not only for achieving greater accuracy during the concept phase, but it can also shorten the development time considerably, particularly for design variants, since the effect of individual parameter variations on the cooling circuit can be calculated very quickly.

Test Equipment

In order to validate the above-mentioned simulation models and test modules and components for thermal management during the development phase, Schaeffler has been setting up its own test centers for thermal management in Germany, China, and the US over the past few years. Other test equipment is located in Korea and Japan. Various test stands have been built with the assistance of the in-house Special Machinery Department. The most important test equipment includes the following:

- A test bench called "Typhoon," which is used to test complete modules under extreme conditions – such as greatly fluctuating coolant temperatures of between -40 and +125 °C. Valve movement, temperature gradients, and volume flow of the pump are recorded on this test bench. Accordingly, the test bench is suitable for durability tests, such as a test with a duration of more than 1,200 hours, corresponding to around 6,000,000 valve movements in a thermal management module.
- Equipment for electric function tests, mainly used for validating actuators under dynamic operating conditions. In addition to the motor function, the signal quality of the position sensor can also be checked.
- A special test bench for testing sensors with respect to accuracy, temperature behavior, and hysteresis.

Along with long-term durability and robust temperature behavior, especially for plastic components, electromagnetic compatibility is also an important goal of validation. Schaeffler is making every effort to develop future product generations that will satisfy EMC protection class 5.

Summary and Outlook

Schaeffler was entering virgin territory with the first series launch of the thermal management module in 2011. By integrating the control system in smart actuators, the second generation of the thermal management system is forging a path to become a mechatronic module.

In Schaeffler's opinion, future electrified powertrains will lead to central thermal management, combined in a single unit, being gradually replaced by decentralized actuators. The smart single valves (SSV) already represent a suitable technology to this end.

The development of thermal management for the drives of the future will be closely coupled to the overall heat balance of the vehicle. For example, the precise knowledge of the heat inputs and flows in hybrid vehicles can be utilized for predictive regulation, such as for preventing the vehicle interior or exhaust gas cleaning system from cooling off. Schaeffler is currently working on the development tools needed for complex system designs. At lower levels, they already have a high degree of maturity, making it possible for the heat balance to be designed for maximum efficiency.

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Chassis

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Mobile in the City of Tomorrow

The Fusion of Drive and Chassis

Dr. Christian Harkort
Dr. Dirk Kesselgruber
Dr. Manfred Kraus
Dr. Jan Moseberg
Benjamin Wuebbolt-Gorbatenko

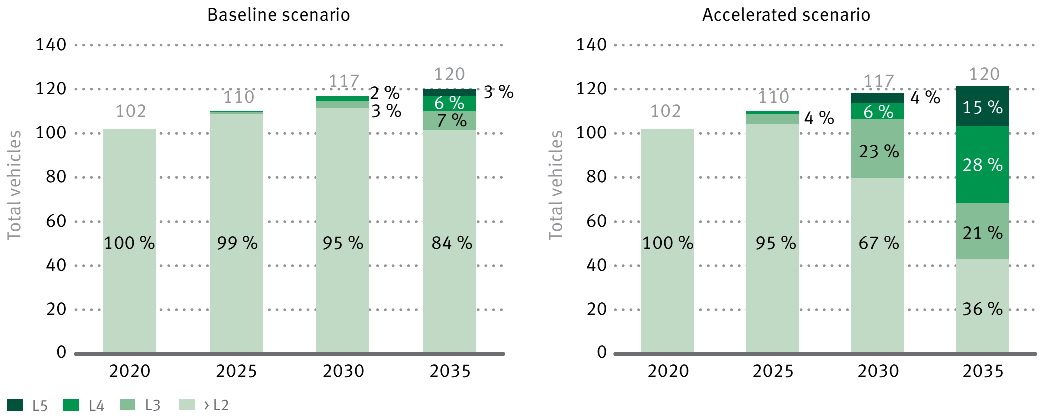
Introduction

Today already, nearly half of the world's population lives in cities. And further increases in the population agglomeration are to be expected in the future as well. For instance, the UN assumes that two thirds of all people will live in a city by the year 2050 [1]. Particularly in Asia, the number of megacities with more than ten million inhabitants has been growing rapidly for years. The result of this is a specific need for mobility, making special vehicle concepts necessary. One solution approach is a highly automated, i.e. driverless, passenger transportation vehicle that Schaeffler is currently developing as a feasibility study. As part of a networked mobility concept, the aim of the electrically driven Schaeffler Mover is to one day bring up to four passengers to the desired destination comfortably, quietly, safely, and without any emissions locally. Due to an extensive data connection with the surroundings, the passengers will be able to simply request the vehicle via a smartphone app, specify their destination, and book the trip.

One of the most important prerequisites for introducing driverless transport systems is the development of driver assistance technologies that allow for automated driving. The autonomy level for this is typically classified in six levels, ranging from level 0 (no assistance, with the driver directly driving the vehicle) to level 5 (complete autonomy, with no more driver intervention necessary) [2]. The Schaeffler study corresponds to level 5. According to studies [3], such systems will be widespread in road traffic in less than 20 years' time. As Figure 1 shows – depending on the underlying scenario – between 8 and 15% of all vehicles produced will already be capable of highly automated driving in 2035. Therefore, economies of scale with dropping development and system costs are to be anticipated in upcoming years, a fact that will support the market penetration of driverless transport solutions even further.

Vehicle Concept

The technical implementation of the autonomous transport vehicle requires a completely new development, which fits with the requirements of highly automated driving and takes advantage of the resulting technical freedoms as much as



1 **Market scenarios for the introduction of automated driving**

possible. For example, operating elements such as the steering wheel and cockpit can be eliminated in favor of providing the passengers with more space. In particular, maneuverability, safety, and driving dynamics stability are important development parameters for a highly automated vehicle for urban use. On the basis of these requirements, a vehicle concept in which all four wheels can be driven and steered independently of each other already took shape during the first potential assessment. This kind of control system provides a lot of agility with a high level of driving comfort, enabling cornering with nearly no noticeable lateral forces to the passengers.

2 **The wheel module from Schaeffler**

Schaeffler has developed an innovative wheel module for this task, condensing the drive into a compact unit via a wheel-hub motor and wheel suspension, along with the vehicle suspension system and the actuator of the electromechanical steering: the Schaeffler Intelligent Corner Module – see Figure 2. On the basis of a cost-saving modular approach, identical units are installed in front and in back. With regard to control, the wheel modules open up the option of an integrated driving dynamic approach that combines the function of the ESP driving dynamics control system, force distribution between the driven wheels (torque vectoring), and all-wheel steering, thereby offering high safety reserves. In the case of a technical defect, steering and traction still remain intact if one or two wheel modules are defective, because the wheel-selective drive and the wheel-selective steering provide a large degree of redundancy.

Complete Vehicle

The complete vehicle concept is based on a stable and light frame structure made of aluminum that holds the wheel modules. The traction battery is housed in the underbody of the vehicle. This arrangement offers the advantage of a high degree of crash safety with good utilization of useable space. In addition, a lower vehicle center of gravity is achieved as well. The vehicle platform concept – see Figure 3 – opens up maximum flexibility with regard to the vehicle setup. In the simplest stage of expansion, the system scope includes the chassis element (consisting of the frame, the four wheel modules, including the

3 Platform concept of the Schaeffler Mover

wheel hub motor, suspension, and steering systems), the software for the driving dynamics control system, the vehicle electrical & electronic systems, and the traction battery. Additional modules supplement this substructure with outer covering, the interior, a lighting system, and the sensor package necessary for highly automated driving, including control software for a drivable vehicle that already offers all driving functions of the transport system. Attachments and installations, such as a cabin with windows, an air conditioning system, operating display and connectivity functions complete the operational vehicle. As an alternative, other special installations are conceivable as well, such as for goods transport.

Since the vehicle is exclusively used for short distances in the urban area, a relatively small and therefore light and economical traction battery is fully sufficient for fulfilling the transport tasks according to Schaeffler's calculations. This is all the more true, since it will be possible in the future to use standstill and wait times – such as at red traffic lights – for intermediate charging via inductive charging systems. This will have a positive effect on the weight balance: The drivable vehicle with its chassis, four wheel modules, interior, and 150-kg high-voltage traction battery – but without the cabin – weighs only 1,150 kg.

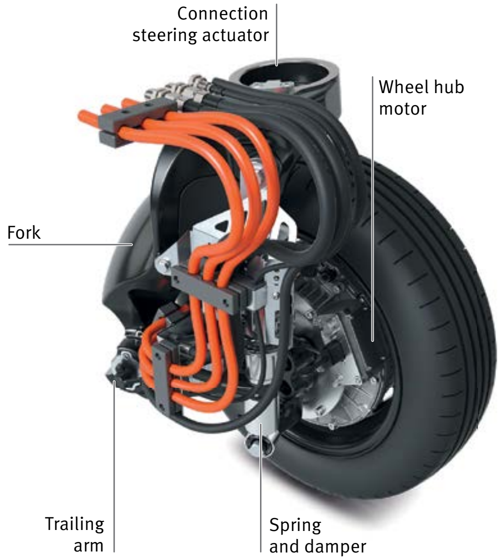
Its turning circle of less than 5 m with four-wheel steering makes the vehicle very maneuverable in city traffic, with the outer wheels describing a steering angle of 21.8° and the inner wheels a steering angle of 45°. Even in front steering mode, the wheel modules, with a turning radius of less than 10 m, still offer great advantages compared to conventional axle designs thanks to the large steering angle. Moreover, the wheels can be swiveled out up to 90° when the vehicle is at a standstill, even making driving perpendicular to the vehicle's longitudinal axis possible and allowing for maneuvering, parking, or turning around in very compact spaces.

Wheel Module Setup

The steering system of the wheel module is designed as an electromechanical steer-by-wire system. An actuator integrated in the wheel module coaxially to the steering axis – see Figure 4 – turns the complete unit during a steering operation, thereby executing the direction specified by the control unit. As described above, the module's steering angle is limited to 45° max. in normal driving mode, while it can be adjusted to 90° during maneuvering. Here, the 48-V motor of the actuator can theoretically generate up to 1,000 Nm of torque. In practice, much less steering torque is required, meaning that the system offers considerable power reserves. The actuator used for the steering function is from the Schaeffler's existing large series modular system and has proven to be extremely robust in other applications.

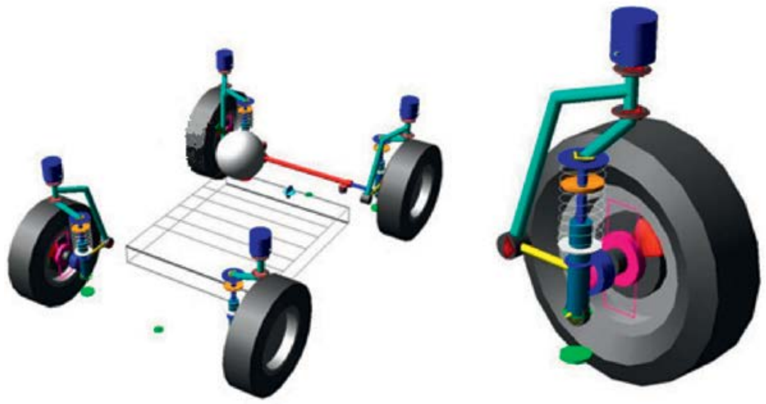
The individual parts of the chassis are shown in Figure 5. It consists of a trailing arm, which is suspended on the frame via a forked carrier. On the wheel side, the trailing arm is connected with a strut, which is supported from above via a strut bearing on the fork carrier. As an option, the wheel module can be equipped with electromechanical ride height adjustment, which raises and lowers the vehicle, such as for helping passengers to enter and exit. During the development of the axle kinematics, it was particularly the bearing supports of the trailing arm and the steering axle that were in need of extensive simulations. This is because the position of the trailing arm is determined by the entire axle geometry, including the wheel's camber angle. The position of the vertical steering axle determines caster and king-pin offset. Moreover, the bearing support of the vertical steering axle must support high torque levels during braking processes, for example.

4 Actuator for the steer-by-wire system



5 Chassis components

In order to work out the optimum wheel module configuration and chassis kinematics, Schaeffler was already carrying out multiple-body simulations during an early concept development phase – see Figure 6. With this help, it was possible to analyze and optimize the dynamics of the movable parts and the distribution of the loads and forces. The simulation for this included risk of tipping, directional stability at higher speeds of up



6 **Vehicle setup in the multiple-body simulation**

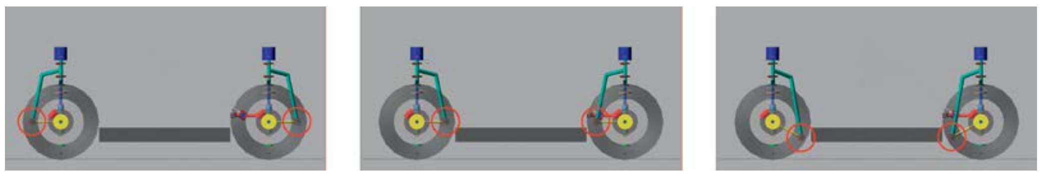
to 80 km/h, the selection of the correct tire dimension, and general calculations for driving stability. In addition to this, it was already possible to conduct initial comfort investigations during the early development phase.

Numerous design solutions were considered and compared in the simulation series. As an example, three versions are shown in Figure 7, on which basis the vehicle behavior with braking at 0.4 g was compared. With version A, the rotation point of trailing arms of the wheel suspensions is located outwards, i.e. in the direction of the back and front of the vehicle. Version B represents the inverse arrangements, with rotating points of the trailing arms pointing towards the vehicle interior. With respect to the trailing arm arrangement, version C corresponds to version B; however, the rotating point of the trailing arms are farther down.

In other words, the fork of the wheel carrier is pulled more in the direction of the vehicle floor. While version A offers the advantage of requiring little installation space, the simulations showed a clear tendency towards pitching motions during braking and acceleration processes. In contrast to this, an unpleasant torque around the vehicle’s transverse axis was apparent in version B, which was reminiscent of the sagging effect on ski lifts. As shown by the results of the simulations, version C offers the greatest advantages in the form of little to no existing pitching torques. When developing the wheel module, these versions were pursued further and implemented in the designs for the feasibility study.

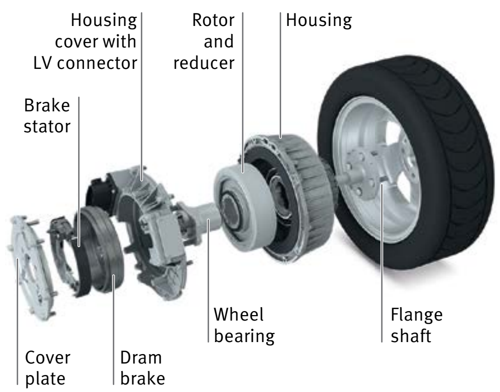
Wheel-Hub Motor

The traction motor of the wheel module – Image 8 – has been designed as a brushless, permanently-excited synchronous machine. In comparison with comparable asynchronous motors, this type offers the advantage of greater power density. This is of particular significance when used in the wheel hub with a limited amount of installation space. In the current design for the feasibility study, each of the four electric motors – with 300 V of operating voltage – supplies a continuous output of 13 kW and a temporary peak output of 25 kW. The nominal torque of 250 Nm per motor can be increased for a brief period of 60 s to 500 Nm max. The top revolution speed of the electric motor is 1400/min. The drive torque of the motor is transferred to the wheel hub via a three-stage planetary gear with a gear ratio of 3.35. At 90 %, the system efficiency of the wheel drive is at a very high level. The liquid cooling system ensures that the traction motor is thermally robust. To this end, a large number of channels through which the coolant flows have been embedded in the motor housing. A central heat exchanger leads away the heat energy from the wheel-hub motors.

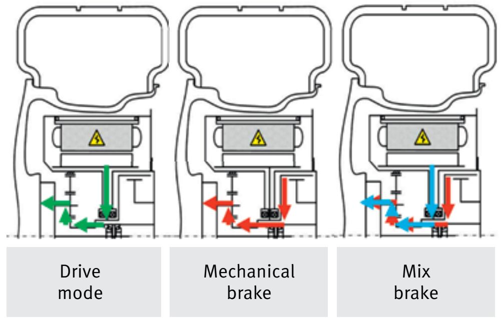


7 **Simulation of different trailing arm configurations**

When the vehicle is braked, the traction motors work as a generator and feed electrical energy back to the vehicle battery – Figure 9. The system is supplemented with mechanically operating drum brakes on all four wheels. Thanks to an adapted operating strategy, Schaeffler has optimized the interaction of regenerative and mechanical brakes to implement a recuperation rate that is as high as possible without sacrificing comfort for the passengers.



8 **The wheel-hub drive components**

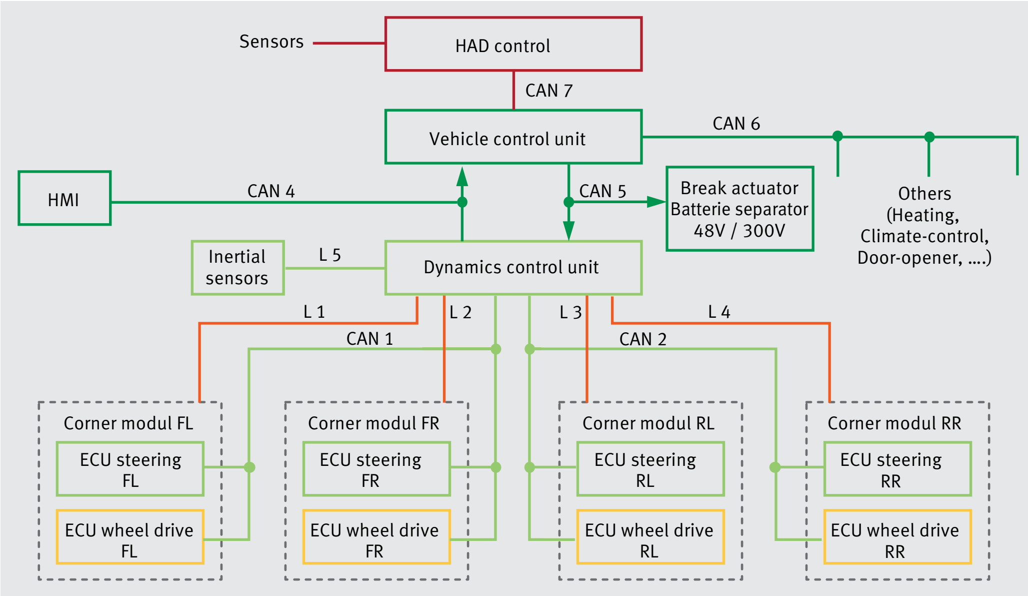


9 **Interaction of regenerative and mechanical brake**

Based on the complete vehicle model, the attainable driving performances according to the simulation were determined for various unladen vehicle weights. Accordingly, a very light vehicle weighing only 600 kg will already reach a velocity of 50 km/h in less than 3.3 s. The maximum velocity is around 140 km/h. It is possible to drive away on a hill with a gradient of up to 31 %. Even a heavier vehicle with an unladen weight of 1,000 kg requires less than 5 sec. to accelerate to 50 km/h. Moreover, the top speed of 135 km/h and the ability to climb a gradient of 21.5 % are still at a high level.

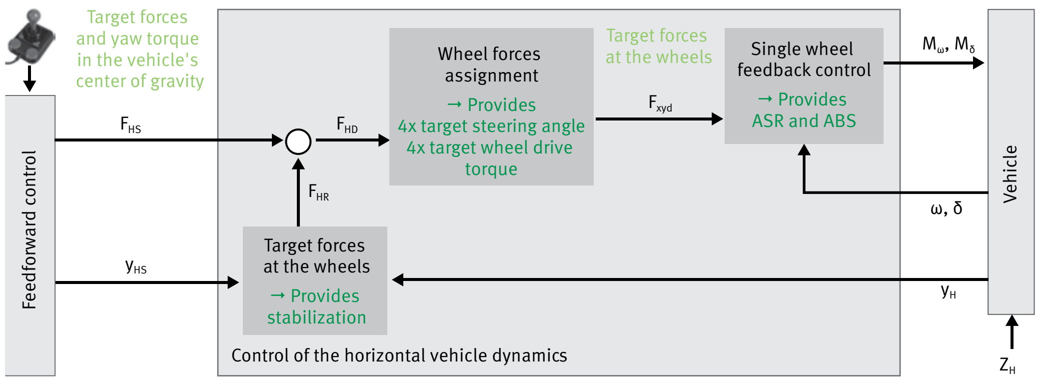
Control Algorithm for the Steer-by-Wire System and the Drive

The complex control algorithm for the steer-by-wire system and the drive constitutes a central component of the software development. On the one hand, the large functional scope resulting from the individual wheel steering needs to be reflected in the control software; on the other hand, it is necessary to implement numerous safety circuits and redundancies in the software. The development of the steer-by-wire control software was implemented completely within Schaeffler. Figure 10 shows the signal flows that are relevant for activating and controlling the wheel modules.



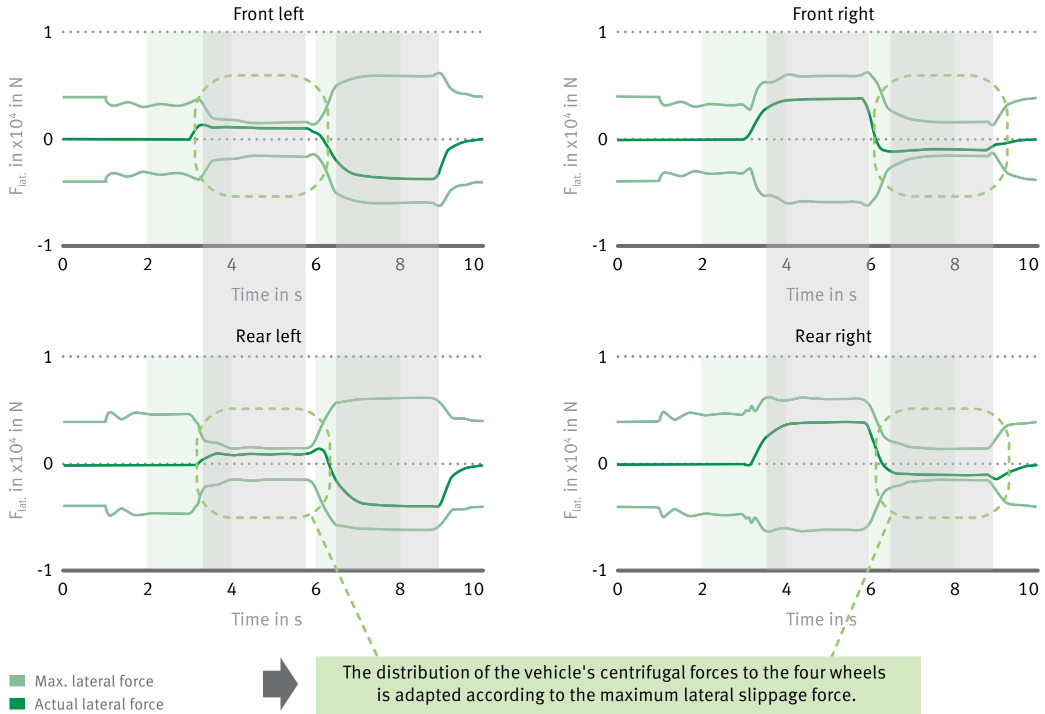
10 Signal flows for controlling the wheel modules

The control unit for the highly automated driving functions – marked in red in Figure 10 above – is directly connected with the controller for the complete vehicle (blue) via a CAN bus. Among other things, this controls the braking and recuperation function, along with comfort elements such as the air conditioner. During the development and test phase, a driver will be on board to monitor the driving function and intervene using a central joystick if necessary. Later on, in regular operation, this module – integrated via CAN bus 4 in Figure 10 – will no longer be necessary. The vehicle control system has reciprocal data contact with the driving dynamics controller. The traction and steering specifications of the driving dynamics controller are implemented in the wheel modules in corresponding torques and steering angles. The steer-by-wire system monitors the function of the individual wheel modules via separate signal lines and activates corresponding safety functions whenever there are any defects.



11 Signal flow in the electronic module and the individual components of the controller

Figure 11 gives a detailed view of the algorithm structure for the driving dynamics controller. The block diagram shows the signal flow in the electronic module and the individual components of the controller. In pre-control, the trajectories when driving around curves and the total forces related to the vehicle's center of gravity that need to be applied for the individual driving maneuver are calculated. The driving dynamics controller uses the target and actual values of the vehicle's longitudinal and transverse guidance system to separately calculate the required torque of the wheel-hub motor and the steering angle of the steer-by-wire actuator for each wheel module. To do this, it puts together the individual traction torques of each wheel in such a way that the driving dynamics control functions of the ESP and the torque vectoring are reflected. For the steering angle of the four wheel modules, it takes in account in the vehicle's longitudinal acceleration, the curve progression, and – as an option – the inclination of the vehicle, among other things. One of the target parameters is for constantly adapting the lateral wheel forces to the dynamically changing slippage limits in order to guarantee maximum driving stability and therefore safe-

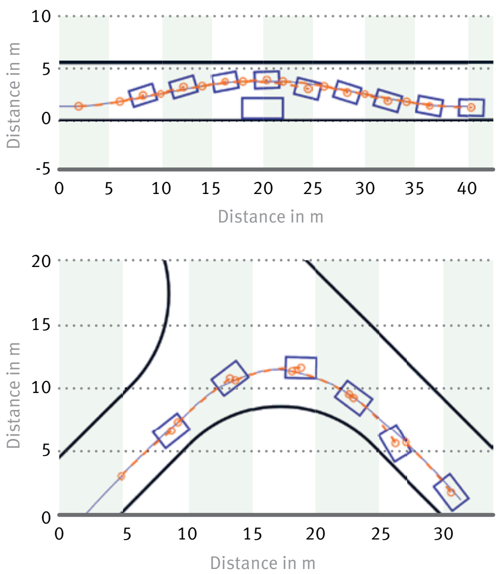


12 The driving dynamics controller controls the force distribution of the four wheel drives in such a way that the actual force is always below the friction contact limit

ty – see Figure 12. Maneuvering and turning the vehicle in place by using a 90° steering angle in the wheel module is also controlled and monitored by the driving dynamics controller

Safety Concept

Highly automated vehicles with steer-by-wire steering need to meet especially high requirements for the functional safety of the individual components and the complete system. It has been possible to achieve maximum safety though a multilevel failure strategy, which – depending on the severity of the fault – involves various measures that build upon each other. If the vehicle electronics discovers a fault in a wheel module that impairs the steering, the drive, or the data communication, then the defective module will be shut off. Due to the axle kinematics of the wheel caster, the module will automatically assume a straight-ahead position. The other three wheel modules can take over the traction and steering tasks of the defective wheel module without any problem, with the result that the function of the complete vehicle will not be impaired at all. The next safety level involves faults in the driving dynamics controller or the failure of the 48-V or 300-V vehicle electrical system. In this case, emergency braking is initiated via the hydraulically actuated drum brakes. If the failure involves the steering system, then it will be deactivated and will remain more or less in the same position, with the vehicle coming to a standstill while following the progression of the curve. The third escalation stage is activated when there is total system failure. The hydraulically activated emergency braking is then triggered mechanically, with the steering remaining more or less in its position here as well. A simulation in Figure 13 shows the effectiveness of the safety strategy. The scenario is based on the worst fault that can be assumed (worst case): mechanical emergency braking, triggered by failure of the vehicle electronics or all vehicle electrical systems. Even then, the vehicle still has maneuverability, and it is possible to avoid obstacles as well as go around curves.



12 **Vehicle behavior during emergency braking due to total system failure**

Summary and Outlook

The urbanization trend that is already visible today will continue to increase in the future. Through the greater population density of living spaces, cities are developing into megacities. This raises completely new questions, especially in the area of mobility. With its electric vehicle experience, Schaeffler has developed a highly automated, electrically driven vehicle for local passenger transport and has implemented it in a feasibility study as a roadworthy prototype. The vehicle is designed to transport four passengers comfortably, quietly, safely, and locally without emissions. On the one hand, the technical key points of the project include the modular vehicle design, allowing for a large number of passenger and goods transport tasks to be implemented with a single chassis structure. On the other, Schaeffler has developed the innovative Intelligent Corner Module, which combines a wheel-hub motor with an electromechanical steering system in a compact unit. This approach, consisting of all-wheel drive and steering, gives the vehicle high driving stability while making it extremely maneuverable in city traffic. The very efficient and powerful traction motors help the vehicle to achieve a driving performance that is more than sufficient, along with a large battery range. A key aspect of highly automated vehicles of the future is the control algorithm for the steer-by-wire system. Schaeffler has completely designed and implemented the software all by itself, including the extensive safety concept. Schaeffler will be successively optimizing the feasibility study of the electric vehicle experience in the near term, and the current, fully roadworthy chassis of the prototype will be equipped with a cabin and air conditioning to form a fully-fledged passenger transporter in the course of 2018.

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