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ATZ WORLDWIDE

04 April 2010 | Volume 112

COST POTENTIALS in the Assembly of Modularized Electric Vehicles

DUAL CLUTCH TRANSMISSION with Dry Clutch and Electro-mechanical Actuation

ACTIVE ACCELERATOR PEDAL as Interface to Driver

INFLUENCING VEHICLE DYNAMICS by Means of Controlled Air Spring Dampers

ELECTRICITY FROM EXHAUSTS Development of Thermoelectric Generators for Use in Vehicles

THERMAL MANAGEMENT

GOOD CLIMATE WITH COLD AND HEAT

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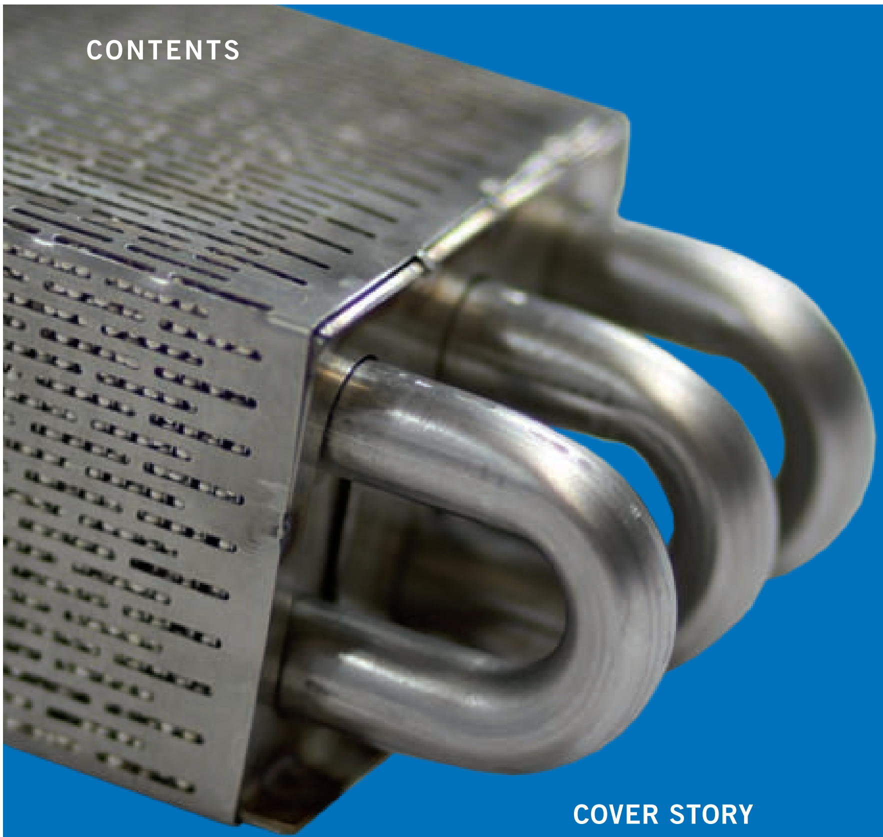

4, 8, 12 I Since the combustion engine of an automobile must be cooled, the waste heat for many decades was quasi in vain. In winter this energy was used for the heating of the interior. If it was too hot in summer, the interior had to be cooled with air conditioning systems. But with energy-efficient diesel engines, sharper emission laws in the cold starting phase and because of the trend to electric cars this traditional procedure comes gradually into staggering. The Forschungsgesellschaft Kraftfahrwesen Aachen (FKA), Eberspächer and Behr light up the topic "Thermal Management" from different perspectives.

COVER STORY

THERMAL MANAGEMENT
GOOD CLIMATE WITH COLD AND HEAT

COVER FIGURE Behr

FACT AND FANCY

Dear Reader,

A few days ago at the Geneva Motor Show, I stood awestruck in front of the Porsche 918 Spyder. Anyone who can remember the racing cars from Zuffenhausen for the "Targa Florio", the legendary Sicilian endurance race, would have fallen in love on the spot. Only later did I notice that this racing car is not only equipped with a V8 engine (368 kW/500 PS) as befitting its status but, as a plug-in hybrid, can also be driven purely electrically for 25 km. In accordance with EU rules, Porsche claims a CO_2 emission of just 70 g/km, which is equivalent to a fuel consumption of around 2.5 litres per 100 km.

Customers will never achieve this fuel economy. Neither will they manage a fuel consumption only 20 % above the standard figure, as is the case with today's vehicles with an internal combustion engine. If the car were to go into series production, the actual consumption would be far higher than that indicated in the brochure, probably by a factor of four. After all, it would be a shame if such a nice car could only be driven for a few kilometres. Porsche is not alone; other car makers presented similar concepts – and in doing so, I believe, they are doing a disservice to the entire industry. Raising customer expectations to such an extent inevitably leads to a loss of credibility.

There was a positive example from the same group: in Geneva, Audi showed an electrified version of the compact A1. The 12 kWh battery is sufficient for a range of at least 50 km (a conservative figure). After that, a small range extender enables the car to carry on driving. At the rear axle, a very compact rotary engine developed by AVL powers a generator. In view of the distances actually covered by vehicles in the A0 segment, the internal combustion engine is likely to be used only intermittently, with the result that many customers will be able to achieve – or even undercut – the claimed CO_2 emission of 45 g/km.

Apart from that: on no account should the CO_2 emissions for the electrically driven section of the EU cycle be set at zero, but must be calculated according to the electricity generation mix in Europe. More intelligent customers – and all fuel consumption concepts are aimed at such groups – are under no illusion that electricity simply comes from a wall socket.

Customers are cleverer than some might think!

JOHANNES WINTERHAGEN, Editor-in-Chief
Frankfurt/Main, 4 March 2010

HEAT TO COLD
ADSORPTION CHILLERS FOR AUTOMOTIVE CLIMATISATION

With thermally driven adsorption chillers, waste heat can be recovered in order to provide cold in the vehicle. Thus, the additional energy consumption for climatisation purposes can be reduced and the overall vehicle efficiency is raised directly. Furthermore, due to their energy storage ability of adsorption chillers, this technology can be used for park cooling. The Forschungsgesellschaft Kraftfahrwesen mbH Aachen (FKA) develops compact adsorption chillers for mobile applications as they can be taken into account for hybrid vehicles and electric cars.

THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN ATZ.
ATZ PEER REVIEW
RECEIVED 2009-08-26
REVIEWED 2009-09-24
ACCEPTED 2009-11-03
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

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INTRODUCTION

Air conditioning compressors are one of the main ancillary consumers in today's cars and thus have an important impact on fuel consumption and overall efficiency. Also, more and more cars are currently being equipped with air-conditioning (AC) systems for comfort and security reasons. Therefore, overall energy consumption for cooling purposes in automotive applications will rise in the near future. The cold demand for future car concepts, such as hybrid electric (HEV) or battery electric vehicles (BEV), will be even larger than for actual cars, as the electric storage unit must be climatised. Lithium-ion batteries, for example, should not exceed a critical temperature of about 40 °C, for lifetime and consequently for cost reasons. Even if not under load, the battery has to be climatised as the decomposition reaction is mainly temperature dependent. Thermally driven sorption chillers are a promising technology for active cooling in automotive applications.

With thermally driven adsorption chillers, as they are developed by FKA, waste heat can be recovered to provide cold for cabin and battery climatisation. Consequently, energy consumption can be lowered and overall efficiency improved. Furthermore, due to the energy storage ability of adsorption chillers, they are well suited for cooling the battery in park mode, for example.

WORKING PRINCIPLE OF ADSORPTION CHILLERS

Similarly to vapour-compression refrigerators, adsorption chillers provide cold in an evaporator and part of the heat rejection to the ambient takes place in a condenser. However, for thermally driven chillers the vapour compression is achieved by a so-called thermal compressor instead of a mechanical one. For adsorption chillers, this thermal compressor consists of at least one adsorber heat exchanger. Adsorption is the ability of porous solids to bind a fluid (refrigerant vapour for the automotive cooling application) onto the surface under heat release (exothermal reaction). The uptake of refrigerant vapour will increase with rising vapour pressures and with decreasing adsorber temperatures. ❶ shows the schematic of a two-bed adsorption chiller – with two adsorber heat exchangers mounted in parallel.

The first adsorber heat exchanger (bed 1) of the two-bed adsorption chiller, shown in ①, is filled with the adsorbent (like zeolite) and is assumed to be dry (no vapour uptake) and thus, can bind refrigerant vapour (for example water steam). Bed 2 has the maximal uptake and therefore has to be dried. This setup has been chosen exemplarily and is arbitrary. Alternatively, both adsorbers could be completely dry. For the chosen start configuration, all valves (V1 to V4) are closed. This means that in the evaporator as well as in the condenser, a two-phase equilibrium (liquid / gaseous) is found. The pressures correspond to the saturation pressures of the refrigerant at the given equilibrium temperatures. For the evaporator, this is the cold supply temperature (for example 5 °C). As part of the heat rejection occurs in the condenser, the required temperature has to be above ambient condition.

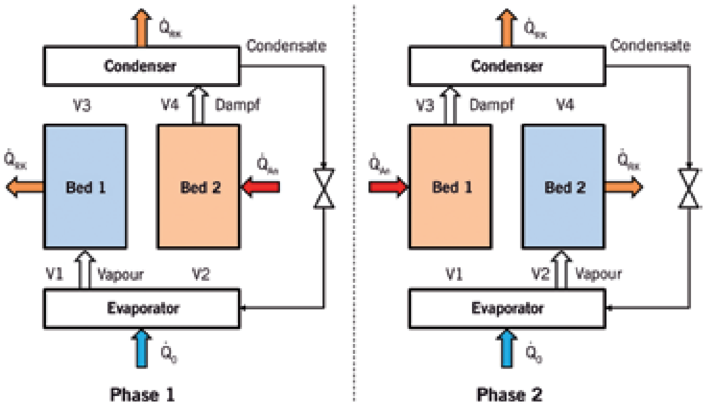
By opening valve V1 refrigerant vapour from the evaporator gets to the adsorbent and is adsorbed. This will result in a temporary pressure drop in the system "bed 1/evaporator". Thus, liquid refrigerant will evaporate to compensate this pressure drop and because of to the heat of evaporation cold is supplied.

The heat of adsorption (exothermal reaction) has to be rejected in order to avoid a temperature rise of the adsorber, which would result in curbing or even halting the adsorption process. Besides the heat rejection in the condenser, the adsorption heat is the second part of the heat rejection at middle temperature. Thus, the adsorption end temperature has to be adapted to ambient conditions.

Simultaneously to the uptake of bed 1, bed 2 is dried (desorbed). The heat of the drive is added. The released refrigerant vapour will result in a pressure raise in the gaseous phase. The pressure is increasing till the vapour condensate at the coldest place in the system "bed 2/condenser" (valve V4 open). It has to be taken into account that the condenser temperature is low (boundary conditions included) in order not to brake or stop the desorption.

With bed 1 fully loaded and bed 2 completely dry, the process can be reversed by opening valves V2 as well as V3 and closing V1 as well as V2. However the end temperatures of the ad- and desorption process do not correspond to the required start temperatures. Thus bed 1 has to be heated up and bed 2 to be cooled down.

In the following, the challenges for the development and the integration of adsorption chillers are presented more in detail. Therefore, high temperature heat source, cold supply, heat rejection, cycle times but also packaging and integration were investigated.



❶ Schematic of a two-bed adsorption chiller

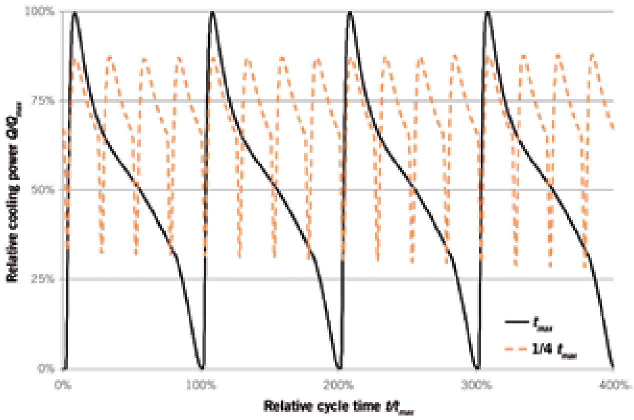
HIGH TEMPERATURE HEAT SOURCE

The time characteristics and the temperature level of the high temperature heat source (for the desorption) are essential for the choice of the adsorbent/refrigerant material pairing. In general, higher temperatures are favourable but there is a reasonable limit depending on the material pairing. For combustion engines, mainly two heat sources are available.

Exhaust gases are usually available above 200 °C, but their availability is subject to highly dynamical fluctuations, depending on the operation point. As thermally driven chillers have a high inertia compared to mechanical ones, this heat source is not favourable. Furthermore, the pressure losses over an exhaust gas heat exchanger would have a direct impact on the operation of the combustion engine. Moreover, the relatively weak heat transfer coefficient of exhaust gases would lead to large heat exchangers. For these reasons, exhaust gas is not recommendable as a driving heat source.

Engine cooling water is available at lower temperatures around 90 °C. Nevertheless, this temperature level is sufficient to reach good changes in uptake with modern zeolites and with silica gel (both in combination with water as refrigerant). Furthermore, engine cooling water is constantly available and will lead to compact heat exchangers, due to water's favourable heat transfer coefficient. For combustion engines, about a third of the fuel is found as waste heat in the cooling water. Thus even for small engines (for example 40 kW driving power), sufficient heat is available for driving an adsorption chiller. For assumed coefficients of performance (COP, ratio of cold to driving heat) of adsorption chillers of about 20 to 60 % (see also section "Cycle Times"), a minimum of about 8 kW of cold could be made available through using the cooling water as heat source.

For BEVs, most available heat sources do not come into consideration because their temperature level is too low (beneath 40 °C). At these low driving temperatures, few adsorbents are available, but these are very strongly sensitive to the heat rejection temperatures (see also section "Heat Rejection"), thus they are not possible for BEVs. Hence the only available heat source for adsorption chillers in BEVs is the waste heat of the electric motor, which is usually available above 100 °C. For an assumed driving power of 40 kW for a small car and a maximum efficiency of the electric motor of 95 %, about 2.1 kW of high temperature heat would be constantly available. With the above given COPs 0.45 to maximum 1.26 kW of cold could thus be provided. This would be sufficient to cover the base load for cabin climatisation or to climatise the battery.



❷ Influence of the relative cycle time on the relative cooling power

COLD SUPPLY

Air conditioning systems in cars do not only cool down but also dehumidify the cabin. This dehumidification is reached by cooling below the dew point in the evaporator heat exchanger. Thus, the evaporator temperature of actual vapour-compression refrigerators is about 0 °C. However, thermally driven chillers are highly temperature-sensitive and their efficiency will strongly depend on the evaporator temperature. For a driving temperature of 90 °C and a heat rejection temperature of 35 °C, the COP of a thermally driven chiller could, for example, be doubled by raising the evaporator temperature from 0 to 15 °C. For the climatisation of the electric battery, this evaporator temperature is sufficient.

However, for cabin climatisation purposes, this cold supply temperature would require an additional dehumidification unit like for example an open sorption wheel. In comparison to the buildings sector, open sorption systems for automotive applications are at a very early development stage, realistic cold supply temperatures for adsorption chillers at present are around 5 to 10 °C. With an evaporator temperature of 10 °C and a temperature difference of 5 K in the evaporator, a surplus of about 5.5 g water-vapour per kg dry air are let into the cabin, compared to 0 °C saturation. However, the absolute humidity of 10.8 g/kg still lies below the sweating limit.

HEAT REJECTION

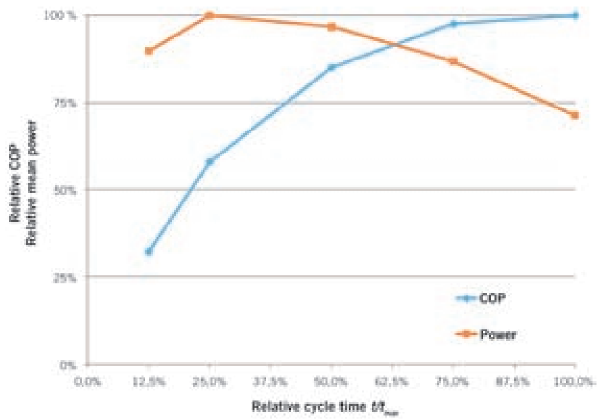
For the integration of adsorption chillers into vehicles, the temperature levels of the cold supply and the driving heat source are generally determined by the automotive concept and are mostly constant. The ambient temperature however is subject to fluctuations and has a big impact on the COP of the adsorption unit. Moreover, the change in uptake and thus the available cold supply is directly influenced by the ambient temperature. A raise in ambient temperature (that means a raise in the heat rejection temperature) implies a higher condenser pressure and thus the adsorbent cannot be dried that strongly during the desorption (see also section "Working Principle of Adsorption Chillers").

Furthermore, the adsorbent will not take up as much refrigerant vapour during the adsorption phase as the adsorption end temperature will rise. For adsorbents which are designed for low temperature desorption, a raise of the heat rejection temperature from 35 to 45 °C for example will reduce the change in uptake so radically that nearly no cold can be supplied.

CYCLE TIMES

As adsorption chillers are non-stationary units, the cycle time for the adsorption and the desorption phases is an important design and control value. The influence of the relative cycle time (t/t_{max}) on the relative cooling power (compared to the maximum cooling power at t_{max}) is exemplarily shown in ❷ for a fourth of the cycle time.

Due to the shorter cycle times the adsorbent is neither completely dried during the



❸ Influence of the relative cycle time on the relative COP and the relative mean cold supply

desorption phase, nor completely loaded during the adsorption phase. As the ad- and desorption kinetics are the quickest shortly after switching, as the driving potentials are maximal, the influence of shorter cycle times on the maximum cold supply is small. As shown by the simulation results in ②, maximum cold supply is reduced by about 12 %. At the end of every process phase, performance tends to zero, as nearly no change in uptake happens (see solid curve in ②). Thus by reducing the cycle time, the average cold supply can be enhanced. The influence of the cycle time on mean cold power is shown in ❸. Below a relative cycle time of 25 %, mean cold supply drops, as during the switching phase, the bed to be desorbed has to be heated up and the dry bed has to be cooled down. The cool down heat of the dry bed is a disprofit for the unit. This critical value for the cycle time depends on the technical realisation of the adsorption chiller (thermal masses).

For short cycle times, the potential of the adsorbent is not used to full capacity. Thus, the COP will decrease with decreasing cycle times. This influence is shown in ③. The maximal COP that can be reached in a thermally driven chiller is a function of the three temperature levels discussed above. For a two-bed adsorption chiller, the COP cannot exceed 100 % and will be around 60 to 70 % for real mobile applications.

The influence on the cycle time on the COP and the mean cooling power is used for partial load control. By extending the cycle time, the chiller is throttled in power. Contrary to vapour compression units, the COP will rise due to this procedure.

PACKAGING AND INTEGRATION

Due to the low exergetic driving source, the cooling power density of adsorption chillers will be lower than for vapour compression units. Within the European research project "Thermally Operated Mobile Air Conditioning Systems" (Topmacs) [1] three adsorption chillers for mobile applications were developed. The chosen material pairings are ammonia/activated carbon, zeolite/water and silica gel/water [2]. For a cold water temperature of 10 °C the volumetric cooling powers of those units with water as a refrigerant are about 20 W/l (related to the volume of the thermal compressor).

The FKA, in cooperation with the Chair of Technical Thermodynamics of RWTH Aachen University, has improved the volumetric cooling power within the framework of an industrially funded project. The ammonia/activated carbon unit has a considerably higher volumetric cooling power (about 70 W/l for the same boundary conditions). However, the COP of this unit is quite low with a value of 15 %. At first sight, the COP of a thermally driven chiller for automotive application is not essential, as sufficient high temperature heat is available. However, the amount of heat to be rejected to ambience is directly influenced by the COP (inversely proportional). As the heat rejection is crucial, it is clear that the COP is very important for the integration of an adsorption chiller.

The volumetric cooling powers of the Topmacs project show for state-of-the-art, adsorption chillers for automotive applications can only be used to cover the base load due to missing available space for integration. However, covering the base load by means of waste heat recovery with adsorption units, about 60 % of the additional energy consumption for cooling purposes can be avoided [3]. With an assumed average consumption for cabin climatisation of 0.5l/100 km the possible fuel savings of 0.3l/100 km are thus determined. Calculations were made for a cabin temperature of 22 °C, the New European Driving Cycle (NEDC), and yearly ambient conditions of Frankfurt/Main, Germany [4].

CONCLUSION AND OUTLOOK

As the FKA analysed in a research project, thermally driven adsorption chillers have a big potential for improving the overall efficiency of vehicles – regardless of the powertrain concept. For today's vehicles with combustion engines, a saving potential of about 60 % of the additional energy consumption for cooling purposes is expected. Furthermore, this technology features further comfort applications, as for example park cooling, due to the energy storage ability of the adsorbers.

However, the relatively poor volumetric cooling power of today's adsorption chillers avoids covering of the whole cold load. In particular, peak loads have to be covered by conventional mechanical vapour compression units. In future developments, this peak load unit will be displaced by consequent reduction of the cold demand and by technological improvements of adsorption chillers.

Adsorption chillers have a non-stationary operational behaviour. To achieve the potential described here, a smart control of the unit is necessary. As thermally driven units have slower dynamic characteristics than vapour compression units, and because of the small temperature differences at heat and material transfer in the beds, the control concepts are crucial for maximal energy savings. This is why next to the technological development of adsorption chillers and heat rejection heat exchangers, emphasis should be put on the development of innovative control concepts.

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Eberspächer heating systems distribute the heat via well designed air vents (left) in the interior of the concept electric vehicle Rinspeed „iChange“ (source: Rinspeed)

HEATING CONCEPTS FOR VEHICLES WITH ALTERNATIVE POWERTRAINS

Electrification of the powertrain demands a decoupling of the heating from the engine’s waste heat. Electrical and fuel-operated heating systems are already based on independence from the engine, and – ideally combined – can guarantee rapid and efficient heating of the passenger compartment. Eberspächer presents the state-of-the-art for both heating systems and concepts for passenger cars with conventional, hybrid and electric drive.

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ELECTRIC MOBILITY IN THE FOCUS OF GERMAN FEDERAL GOVERNMENT

The bandwidth of electrified powertrain technologies for reduction of CO_2 emissions is considerable, ❶. This is the case both for currently in-development projects and already available series-production. It ranges from combustion engines with electrified ancillary units over hybridized drives and on to pure electric vehicles. The significance behind the megatrend of electric mobility is clearly illustrated by the agendas approved by the German Federal Government:

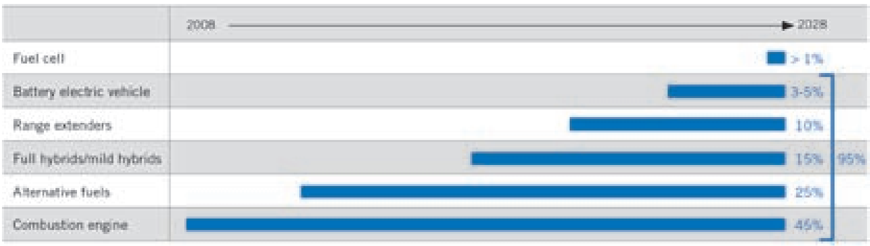
: Integrated energy and climate program (2007) [1]
: National innovation program for hydrogen and fuel-cell technology (2006) [2]
: German Federal Government national development plan for electric mobility (2009) [3].

Electric mobility will only have a future if, despite every endeavor for boosting the efficiency of the drives, customer demands for a comfortable climate in the vehicle can also be met.

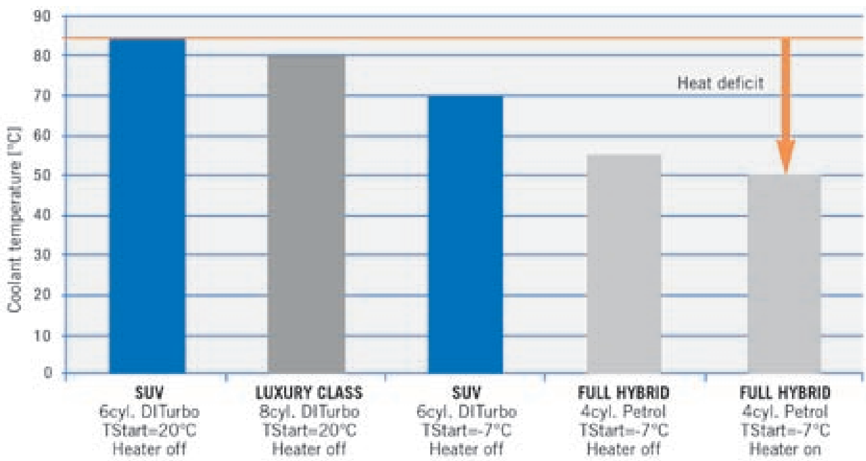
HEAT DEFICIT OF VARIOUS DRIVE CONCEPTS

A comparison of coolant temperatures in vehicles with efficient drives clearly demonstrates the dilemma associated with heating the passenger compartment using coolant heat. It is shown in ❷ for five drive types. Without passenger-compartment heating, none of the vehicles achieved the specified temperature for the coolant of 85 °C at the end of the new European driving cycle (NEDC). The reduced consumption through the engine's start/stop system is limited in winter, when the coolant fails to reach the minimum temperature. At -7 °C, a full hybrid vehicle has such a great waste-heat deficit throughout the entire NEDC urban driving cycle that the heating of the passenger-compartment has to be decoupled from engine-related waste heat, if the customary level of comfort is to be maintained in the vehicle.

Battery electric vehicles (BEV) no longer have a sufficient source of waste heat. The search for alternative heat sources requires answers from the sup-



❶ Market trends for electric drives and technology road map – what will the mobility of the future look like?



❷ Heat deficit of conventional drive with six-cylinder engine versus a full hybrid drive – coolant temperature at the end of the urban NEDC

plier to the questions of the BEV customer and the vehicle manufacturer. Will a BEV customer accept:
- limited thermal comfort?
- a reduction in operating scope in excess of 40 % with electric heating?
- the necessity for visiting a filling station with a battery-preserving fuel-powered heater?

The vehicle manufacturer could pose the following questions:
- Can the heating system assume the thermal management of the battery?
- What expense is associated with the fuel tank system exclusively required for the heating system of a fuel-powered heater?
- Which fuel would best suit the environmentally-friendly image of an electric vehicle?

HEATING CONCEPTS BASED ON ENGINE-INDEPENDENT SYSTEMS

Engine-independent heating systems require an energy source which is independent from the engine's waste heat. The heating output deficit that has to be compensated for by an engine-independent system is as follows:
- cars with consumption-optimized combustion engines 400 to 2000 W
- full hybrid vehicles around 3500 W
- BEVs approximately 6000 W.

With the exception of the pure BEV all the above vehicles are equipped with two energy sources: battery and fuel. Electric PTC heaters [4] (PTC: positive temperature coefficient) and fuel-powered heaters use these energy sources. With an efficiency level of more than 80 % both types of heater can be designed as auxiliary heaters to back up the classical vehicle heating system, or as fully independent heater systems. Both systems are available as blower heaters and as independent coolant heaters that heat air or coolant, either electrically or through a burner.

HEATING CONCEPTS FOR CONSUMPTION-OPTIMIZED VEHICLES

Practically all modern diesel cars are nowadays equipped with PTC auxiliary heating. Decisive for the success of this heating measure is the high quality and reliability, combined with the outstanding intrinsic safety against overheating. Compact dimensions and a high level of flexibility in connection technology enable integration into the central heating, ventilation and air-conditioning system (HVAC) downstream of the water heat exchanger. PTC auxiliary heaters support passenger-compartment heating when the coolant temperature in the water heat exchanger is too low. Basically, two systems have currently established themselves [5]:
- mechanical heater, which can be switched on or off in stages using external relays
- electronic heater with a continuously variable, integrated electronic power control.

FULL HYBRID VEHICLES

The increased efficiency of the hybridized drive means that in cold start conditions the use of engine waste heat, backed up by electric auxiliary heater, is unavoidable. In order to be able to drive electrically in the winter, full hybrid or electric vehicles with range extender (E-REVs) require a solution to compensate for the heater output deficit – without using valuable electrical energy. One possible solution is the engine-independent, fuel-powered coolant heater, which utilizes the given infrastructure in the water circuit and the climate system to heat the passenger compartment more comfortably, ❸.

From the moment the engine is started in cold conditions, a fuel-operated coolant heater can either support the engine as an auxiliary heater or run the heating function on its own in electrical driving mode. Also possible is heating during an combustion engine start/stop mode. A quick-start combustion process enables the full heater output of 5000 W within 25 s.

The devices comply with the increased life-time requirements through state-of-the-art alternative fuel-burning combustion procedures and electronic-commutated blowers and water-pump motors. With a view to the ecological perception, it remains to be seen whether customers will accept fuel-powered auxiliary heating during purely electric service.

BEVS

The heating of BEVs represents new conceptional and technological territory. Apart from the solution to the problem of the operating range and battery costs the success of these vehicles shall be determined by the climate comfort as customers will only accept minor reductions to the customary level of comfort compared

❸ Fuel-powered heaters for heating the passenger-compartment air (left) and coolant (right)

❹ High-voltage PTC heaters for heating the passenger-compartment air (left) and coolant (right)

VEHICLE MODEL	HEATING-CONCEPT RECOMMENDATION	SELLING POINTS
Consumption-optimized Vehicle	PTC air heater 400 – 2000 W	: 12 V vehicle electric system : Installation in given heater climate module (HVAC) : Weight
Mild Hybrid (4-5 seater)	PTC air heaters 2000 W	: Option 12 V, 24 V, 48 V : Installation in given HVAC : Weight
Full Hybrid (4-5 seater)	Fuel-powered heater, coolant PTC coolant heater 3000 W	: Fuel periphery given : CO_2 balance : HVAC without modification : Heat balance/engine support : Platform option
Electric Vehicle / Range Extender (4-5 seater)	Fuel-powered heater, coolant PTC coolant heater 5000 – 6000 W	: Operating range : Water circuit given : No high voltage in passenger compartment : Platform standardization : Engine pre-heating
Electric Vehicle Compact Size (4-5 seater)	Fuel-powered heater, air/coolant PTC coolant heater 5000 – 6000 W PTC air heaters 5000 – 6000 W	: Operating range : With given coolant circuit : No high voltage in passenger compartment : Platform standardization : Battery temperature control : Parking heater : Short trip heating : Weight and packaging
City Electric Vehicle (2 seater)	PTC air heaters 3000 W	: Short trip heating : Weight and packaging
Fuel Cell Vehicle (4-5 seater)	PTC coolant heater up to 10,000 W	: Utilization of given coolant circuits : Load dump and recuperation function

❺ Heater recommendation depending on the vehicle model

with existing vehicles. Currently, three concepts are competing against each other: electrical high-voltage (HV) air heater, electrical HV coolant heater, ❹, and the heat pump, which is not viewed here.

Electrical HV air heaters can replace the conventional coolant heat exchangers in the passenger compartment and convert electrical energy straight into heat. The required outputs of 5000 to 6000 W can be achieved in the given installation space without involving any critical overheating in the heater element in the event of any malfunctions or faults. The complete heating comfort is available after approximately 20 to 30 s. To comply with the requirements for a constant temperature in the passenger compartment, a linear (continuously variable) control with as yet to be developed control settings is required in the climate control system. This enables outputs to be regulated in combination with low thermal masses.

HV PTC coolant heaters represent an ideal solution for a BEV's complicated thermal management. They use the given heater climate module including the climate control of a conventional vehicle, and permit the energy-efficient utilization of waste heat from traction motor, power electronics, voltage transformer and battery – including their temperature control at low temperatures through a water circuit.

However, this obvious technology for heating by means of an electrical heating system reduces the operating range of the car by more than 40 %. Vehicle electric systems of up to 500 V direct-current voltage (DC) require special insulation techniques and layouts to comply with international safety standards and to guarantee reliability in automobiles.

The most-efficient means of handling current as a resource is offered by fuel-powered air heaters with a heater output of 900 to 4000 W at a maximum current consumption of 40 W, ③. They can be arranged in the passenger compartment or on the periphery. The distribution of the heated air can be effected through a variety of systems ranging from simple air vents up to complicated distribution systems with multiple air vents. BEVs and E-REVs without a coolant circuit can therefore be comfortably heated. Due to their original application geared towards commercial-vehicle heating, current air heaters already fulfill the life-time requirements for BEVs. It remains to be seen whether customers and legislative body shall accept fuel-powered auxiliary heating in BEVs, because they will not fulfill zero emission criteria.

OVERVIEW AND OUTLOOK

As energy and thermal management in the vehicle grows increasingly electrified, it is becoming the main challenge for development departments in the automotive sector. Consumption-optimized engines and new additional components in the vehicle and therefore new heat sources and heat sinks require changes to be made in the area of heating technology. Therefore, Eberspächer developed modern auxiliary PTC heaters and fuel-powered heaters but also electrical high-voltage heaters for hybrid and electric vehicles. ❺ contains a summary of recommendations by Eberspächer for heating current and future drive concepts.

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INTEGRATION OF A LITHIUM-ION BATTERY INTO HYBRID AND ELECTRIC VEHICLES

Electrical energy storage units are the key technology to electrified powertrains. They determine driving range, weight, package and costs substantially. Lithium-ion batteries are particularly suitable for meeting these challenges, but at the same time they also place high demands on the temperature management. Behr presents the integration in the entire vehicle and the different system concepts for battery cooling and illustrates the specific benefits and drawbacks in each case, but also how the refrigerant and coolant circuits could be optimised.

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LI-ION ACCUMULATORS NEED A SPECIAL THERMAL MANAGEMENT

Due to their exceptional properties, lithium-ion batteries will in future be used as an electrical energy source in many hybrid and electric vehicles. Owing to their high energy density these batteries are superior to their currently used Ni-MH counterparts; but they also require a special cooling and heating concept. While the optimum temperature range depends on the cell chemistry employed by the different manufacturers, as a general rule temperatures of over 60 °C during storage and 40 °C during operation need to be avoided. There is an appreciable reduction in the potential charging and discharging rates at temperatures of below -5 °C.

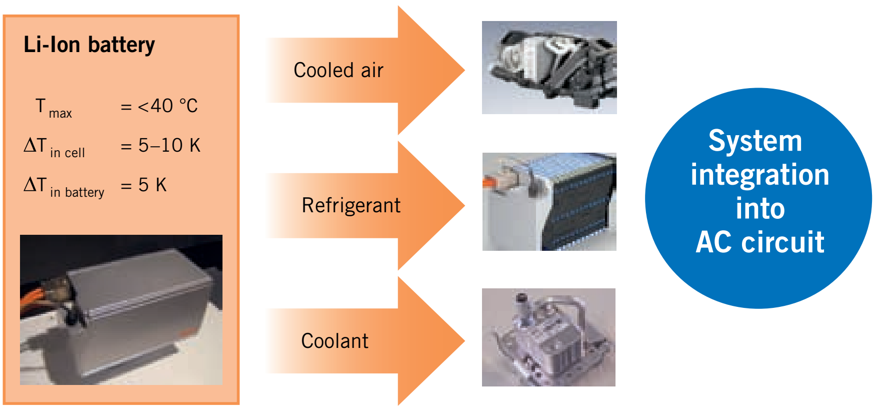
Operation outside this temperature range shortens the battery life due to irreversible damage occurring within the Li-ion cells and should therefore be avoided. In hybrid vehicles, this would result in restricted electrical support for traction, while in electric vehicles it can even lead to a breakdown if this temperature interval is exceeded and the battery shut down. The thermal management of the battery thus ensures the safe operation of the vehicle and reduces the loss of value of these expensive batteries, particularly in the case of large batteries in plug-in hybrid vehicles and electric cars.

At 95 %, the efficiency rate of modern Li-ion cells is extremely high. However, the degree of waste heat that results from the internal electrical resistance of the cells during acceleration and braking is a factor that cannot be ignored. The various cell cooling designs have already been presented and discussed in ATZ [1]. This article examines the integration of the battery's thermal management system into the vehicle.

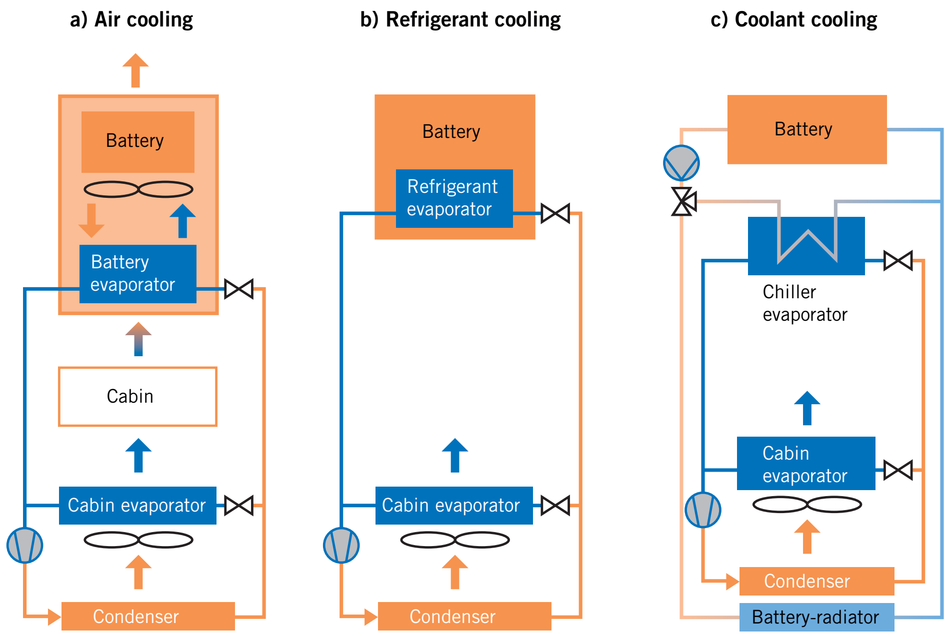
Particularly in summer, ambient air is not suitable for cooling the Li-ion battery; the difference between ambient air temperature and the maximum permissible temperature of the battery is too small to dissipate battery heat employing reasonable means. Accordingly, the air conditioning circuit, ❶, is the only available heat dissipation method in the vehicle that can be actively influenced. The heat from the battery can be transferred from the refrigerant circuit of the A/C system using conditioned air, a dedicated coolant circuit, or refrigerant itself. The method used is dependent upon the application profile of the vehicle type, for which it is intended. Each cooling method has its own particular advantages and disadvantages. All three are currently used in production vehicles, and are presented below.

COOLING METHOD WITH AIR

With air cooling, the cooling air is fed through large ducts to the battery. Once the air has warmed up in the course of passing through the battery and along the cells, it is often discharged directly into the outside air. Basically, there are several drawbacks with this very simple cooling method. They include the large air ducting system to and from the battery, the



❶ The refrigerant circuit is the only actively influenceable heat sink available in the vehicle – the heat transfer from the battery to the refrigerant circuit can be effected using air, directly via the refrigerant or coolant



❷ Schematic diagram of the different battery cooling concepts using air, refrigerant and coolant

weight of the blower, and, in some cases, the annoying noises that the latter produces in the passenger compartment, as well as certain safety aspects relating to the use of cabin air when there is a direct connection between the cabin and the battery. To prevent internal fouling of the battery, which, in combination with moisture, can cause creepage currents or compromise heat transfer, the air has to be filtered.

If cabin air is used for cooling, the battery cannot be cooled independently of the cabin. At certain operating points this leads to a conflict of objectives between cabin comfort and battery cooling. This drawback can be avoided by installing a separate, compact air conditioning unit for the battery, similar to a rear air conditioning system in luxury class vehicles, that is switched in parallel with the cabin air conditioning unit. This method is presented in ❷ (a). The refrigerant evaporator chills the battery via an air flow or a ventilator. This method increases weight and package size and further reduces system energy density.

However, the benefit of this solution is that battery cooling can be operated in recirculated air-only mode, which obviates the need for filtration. Air cooling of the battery is used principally in vehicles with sufficient installation space, for example in sport utility vehicles (SUVs).

COOLING METHOD WITH REFRIGERANT

Refrigerant cooling is the most compact method of battery cooling. A compact evaporator configured as a battery cooling plate is installed inside the battery and is in heat-conducting contact with the Li-ion cells, ② (b). The heat required to enable evaporation of the refrigerant is drawn off from the battery cells, which are cooled very effectively as a result. The cooling ducts must be configured and arranged in such a way as to ensure that evaporating refrigerant is available everywhere and at all times in order to achieve the required temperature homogenity.

Only two additional refrigerant pipes are required to connect the battery cooling plate to the refrigerant circuit: firstly the pressure pipe to the battery, and secondly the suction pipe from the battery back to the compressor. The battery evaporator is switched in parallel with the main evaporator.

The system energy density of the Li-ion battery is reduced only very slightly by the cooling process because very little equipment is required outside the battery. The cabin and the battery each have different cooling requirements, so that specific coordination of the overall circuit is needed. However, the use of a variable-speed electric air conditioning compressor makes this coordination process easier compared to conventional belt-driven compressors.

The battery cooling process always requires the use of a refrigerant compressor. But the additional power consumption compared with the cabin air conditioning system is low.

Refrigerant-cooled battery systems are generally used in vehicles that require a compact battery cooling system, and where the additional compressor power draw does not detrimentally effect overall efficiency. Since the refrigerant circuit can only operate at temperatures down to around -5 °C, battery cooling at lower temperatures is restricted. However, in practical applications this can usually be tolerated.

COOLING METHOD WITH COOLANT

Cooling the battery with coolant is the most flexible method. At the same time, it is also extremely energy-efficient if an additional battery radiator is used, ② (c). In such systems, the battery is also equipped with a cooling plate, in this case, though, with coolant (water/glysantine in a so-called dedicated “secondary loop”) flowing across it. A carefully designed ducting system ensures the greatest possible temperature homogenity across the plate and compensates for the heating of the coolant.

❸ The Behr chiller: an extremely compact heat exchanger, in which the refrigerant evaporates, causing the coolant to be cooled

The temperatures in this secondary loop will depend on the operating strategy and on the properties of the battery cells, but are typically between 15 and 30 °C. For closed-circuit cooling purposes, Behr has developed a special, very compact heat exchanger, known as the chiller, which connects the refrigerant circuit with the secondary loop, ❸. The refrigerant evaporates in this chiller and the heat required for the process is withdrawn from the coolant in the secondary loop. The high power density of the Behr chiller enables it to be packaged in a small space.

The energy efficiency of the battery cooling system plays a key role in electric cars, and also in plug-in vehicles with high electric drive ratios. In the case of a secondary loop, an additional battery cooler in the cooling module is suitable for ensuring the range is not reduced by unnecessary operation of the electric compressor. A changeover valve then allows drivers to use the chiller in summer, and the battery cooler when external temperatures fall.

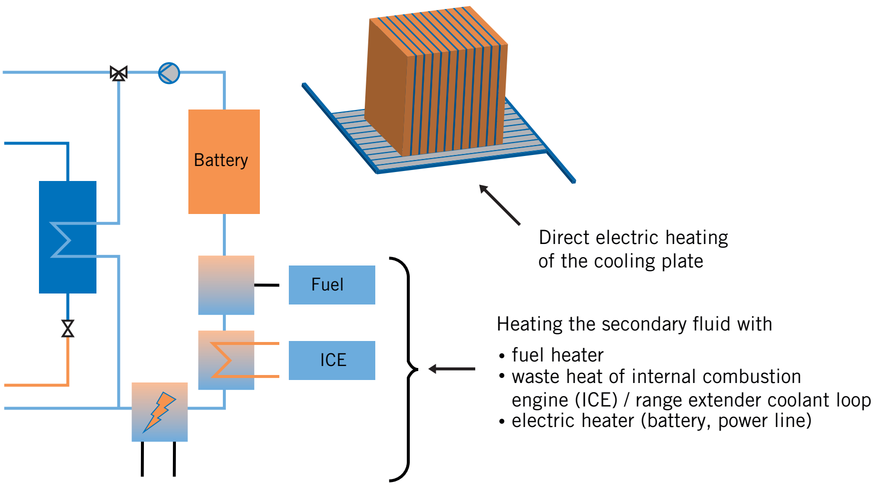
The advantages of the coolant cooling go along with the disadvantage that this system takes up a relatively large space outside the battery. This is because, in addition to the chiller, a pump, lines and, in the case of bivalent operation a further cooler and a valve are required.

BATTERY HEATING

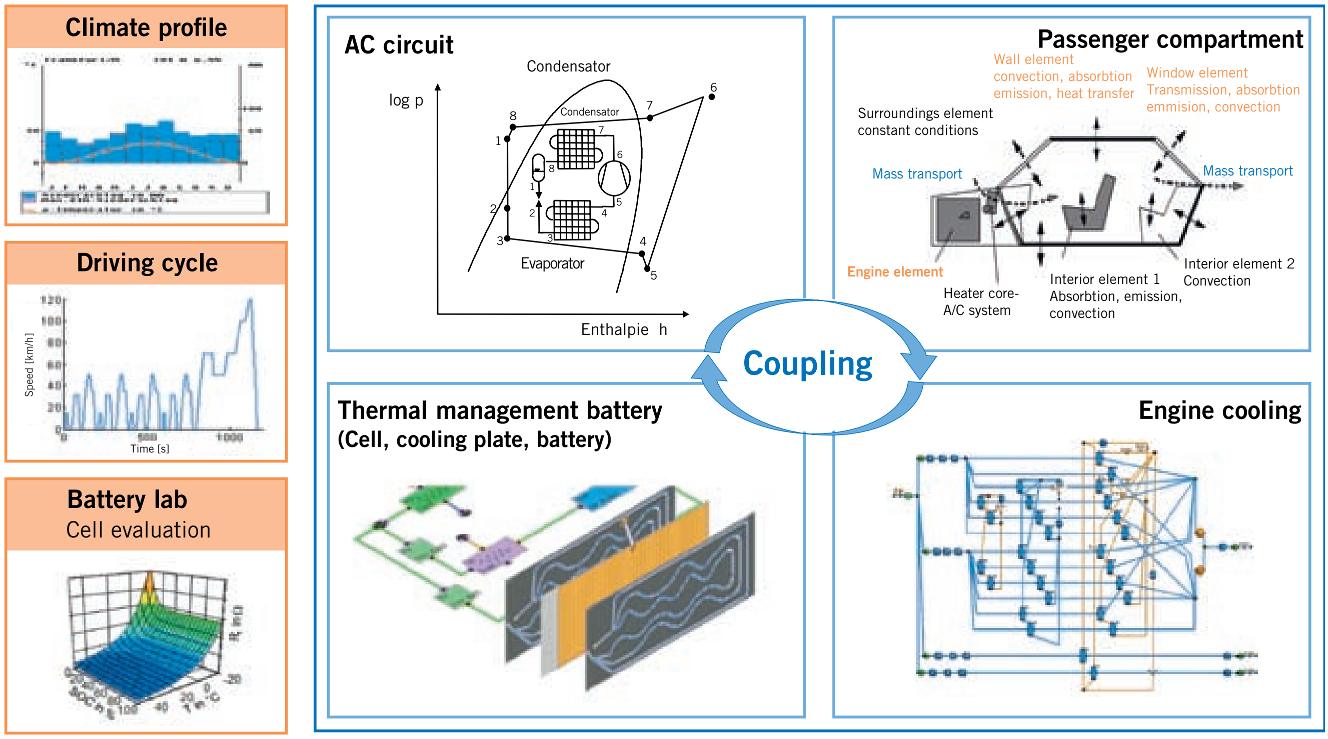
Battery thermal management also involves heating, as well as cooling. This is due to the fact that at low temperatures, a Li-ion battery will severely degrade in performance and, during charging, may even sustain damage. Here again, no standards have been established as yet, and various methods are used depending on specific requirements, ❹.

The coolant can be heated directly if a secondary loop is used; in this case, the heat flow flux is reversed. Available heat sources include electric heaters, or fuel heaters in hybrid vehicles. Heat from the engine cooling circuit can also be used via an additional heat exchanger. The drawback, here, is the thermal inertia resulting from the secondary loop, and the long way from the heat source to the cell interior. Losses in this context must be minimized with good insulation.

It is more practical to generate the heat closer to the cells. This can be achieved by direct electrical heating of the battery cooling plate, enabling thermal losses to be reduced. This heating method can also be used with refrigerant cooling.



❹ The battery heating can be effected through an integrated direct electric heater for the battery plate or by heating the secondary fluid using various heat sources



❺ The computer program Behr Integrated System Simulation (BISS) permits the system simulation of the thermal and energy management as a function of, for example, climate profile, driving cycle and cell characteristics

SYSTEM DESIGN VIA NUMERICAL SIMULATION

The options described here for the thermal integration of a battery into the overall vehicle call for a numerical simulation for virtual analysis and design, ❺. Accordingly, Behr has enhanced its group-wide simulation program BISS [2] to include battery simulation modules. The new modules encompass electric and thermal simulation of the battery cell as an additional heat source, as well as the design of the battery cooling plate. Together, they comprise the battery component that can be incorporated into the stationary and non-stationary system simulation. As well as climate profiles, additional data and time series, such as cell data and driving cycles, are required as input values. The thermal data for the battery (for example waste heat, temperature gradients inside the battery) and electrical data (for example charge rates, charged status) are calculated from this input data. Using the system simulation allows the impact on a secondary loop to be simulated, and also the impact via the chiller on the refrigerant circuit and cabin cooling. Furthermore, the energy simulation allows conclusions to be drawn on the energy expended for cooling and the resulting reduction in range.

The simulations allow to make an initial design for battery cooling in the overall system; however, further system measurements are required to verify these simulations. For this purpose, Behr has enhanced an HVAC test bench to include a battery test bench. This not only allows the air conditioning system to be designed for different climate profiles, but also facilitates the simultaneous integration of battery cooling and testing with standardized or customer-specific driving cycles.

SUMMARY

With the Li-ion battery, a further component has been introduced into the vehicle that has special requirements in terms of thermal management. Owing to the wide variety of cell and battery types used, and the specific operating strategy of the different automotive manufacturers, quite different cooling methods are used.

Behr offers optimally matched battery cooling designs for all vehicle models and operating strategies and has the experience acquired from production orders for all three designs presented before: battery cooling with air, refrigerant, and coolant. Therefore, numerical simulations of the thermal and electrical energy flows as part of the BISS simulation tool are available, and also a combined system test bench for cabin air conditioning and battery cooling.

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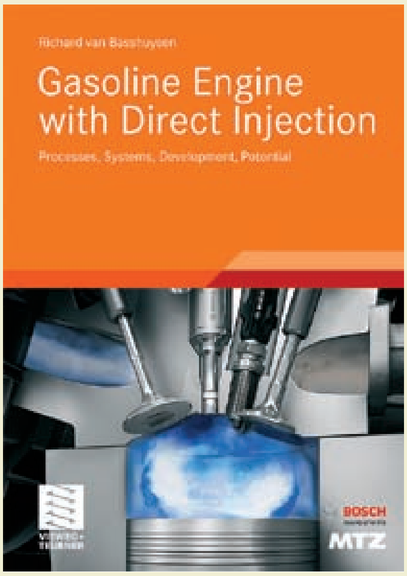
Gasoline Engines are the answer to the challenges of future



Richard van Basshuysen

Gasoline Engine with Direct Injection

Processes, Systems, Development, Potential

2009. xviii, 437 pp. With 399 Fig. Hardc. EUR 49,00
ISBN 978-3-8348-0670-3

Direct injection spark-ignition engines are becoming increasingly important, and their potential is still to be fully exploited. Increased power and torque coupled with further reductions in fuel consumption and emissions will be the clear trend for future developments. From today‘s perspective, the key technologies driving this development will be new fuel injection and combustion processes. The book presents the latest developments, illustrates and evaluates engine concepts such as downsizing and describes the requirements that have to be met by materials and operating fluids. The outlook at the end of the book discusses whether future spark-ignition engines will achieve the same level as diesel engines.

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Dr.-Ing. E. h. Richard van Basshuysen was Head of Development for premium class vehicles and for engine and transmission development at Audi. Today, he is editor of the magazines ATZ and MTZ. The editor was supported by a distinguished team of authors consisting of 22 experts and scientists from industry and universities.

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COST POTENTIALS IN THE ASSEMBLY OF MODULARIZED ELECTRIC VEHICLES

Whether and how quickly electric vehicles become established as a serious alternative to conventional vehicles depends above all on their overall production costs. According to RWTH Aachen University, a radical rethink in the design of electric vehicles is required in order to compensate for the cost disadvantages of the electric powertrain. In the “StreetScooter” project initiated by the university, the complete vehicle has been designed for optimum producibility right from the start.

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NEW REQUIREMENTS REGARDING INDIVIDUAL MOBILITY

Urban populations are growing significantly worldwide. A study conducted by the UN reveals that 40 % of the world's population will live in urban areas by the year 2015 [1]. For that reason, conventional vehicles are not included in the plans for the mobility of the future. A new generation of compact cars that require less space, are emission-free and are optimized for the traffic in urban areas is needed.

Due to increasing living costs, the demand for low-cost cars is rising significantly. Pushed by public subsidies to scrap old cars, most customers bought new ones at a low price, causing the average selling price for cars in Germany to drop last year for the first time in many years. Therefore, this trend can be observed even in countries with an established automotive market.

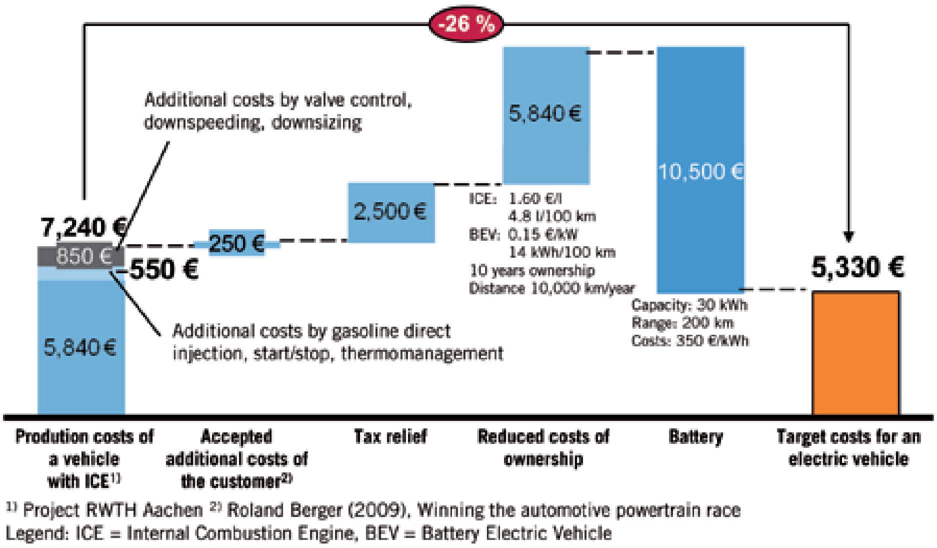
The third trend is the growing attention to ecological issues. Today, every fifth customer in Germany is putting off buying a new car and is waiting for an electric one [2]. Furthermore, there is a shift in the mind-set from the cultivation of heavy and powerful cars as a status symbol to ecological mobility. This is encouraged by government subsidies and laws.

As a result, there is significant potential for electric cars fulfilling urban traffic requirements. Electric mobility is no longer a niche market. 100,000 to 400,000 electric vehicles are forecasted for Germany in the year 2020 [3]. In the medium term, a number of units that requires industrial mass production appears to be realistic.

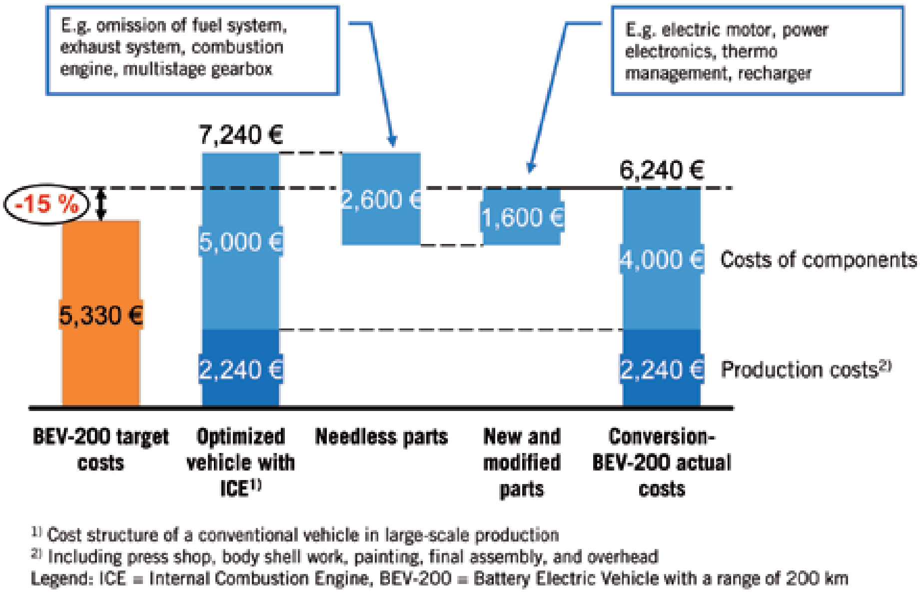
COST TARGETS AND TARGET COSTS

Electric vehicles are in competition with conventional vehicles right from the start. Despite a few niche markets, the selling price is the critical purchase factor for the customer. Based on the production costs of a conventional vehicle with an internal combustion engine, ❶ shows the derivation of the target costs for a compact electric vehicle with a range of 200 km in the year 2015. Even taking into account the fact that customers are willing to accept a slightly higher selling price of 250 euros and that governments have announced a fiscal subsidy of 2,500 euros to compensate for the high battery costs, the production costs of an electric vehicle are still 25 % above those of a conventional vehicle. Therefore, the production process itself must make a major contribution towards reducing the overall costs.

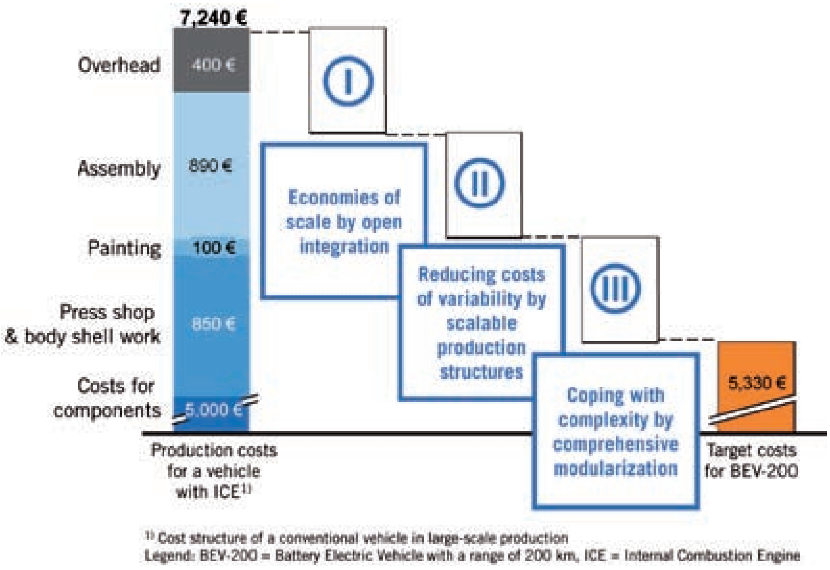
Basically, there are two design principles for the development of an electric car [4]. First, electric vehicles can be produced using a "conversion design" approach. In such designs, an established vehicle with an internal combustion



❶ Calculation of the target costs of an electric vehicle



❷ The required target costs cannot be achieved in a conversion design



❸ Potentials for cost reduction in a purpose design

engine is converted to an electric powertrain. The vehicle can then be produced in existing production facilities. The Mini E is an example of such a vehicle. The car was originally devised as a vehicle with an internal combustion engine and is now produced in small series as an electric vehicle.

Second, the "purpose design" approach makes it possible to consider the specific requirements of production at an early design stage. For instance, the package of the electric powertrain can be optimized with regard to the corresponding production processes. This offers the chance to redesign the value-added process.

Analysing the target costs of a vehicle developed in the "conversion design" approach demonstrates that the additional costs of the battery cannot be compensated for by the cost-savings of the components of the internal combustion engine that can be omitted (fuel system, exhaust system, internal combustion engine and multistage gearbox), ❷. Therefore, electric vehicles in a conversion design will only have a market potential as a temporary solution.

Competitive electric vehicles can only be produced in a "purpose design" approach. A completely new concept of the car allows design engineers to rethink the traditional car manufacturing process. Established production processes with the classic value chain of press shop, bodyshell work, painting and final assembly can be redesigned. Electric mobility is a chance for automotive manufacturers, especially small and innovative ones, to create potentials in production by achieving new economies of scale, reducing capital tied up in equipment and material and increasing productivity. Examples of three starting points from which the cost advantages of production-oriented "purpose design" can be exploited are illustrated in the following, ❸.

ECONOMIES OF SCALE THROUGH OPEN INTERFACES

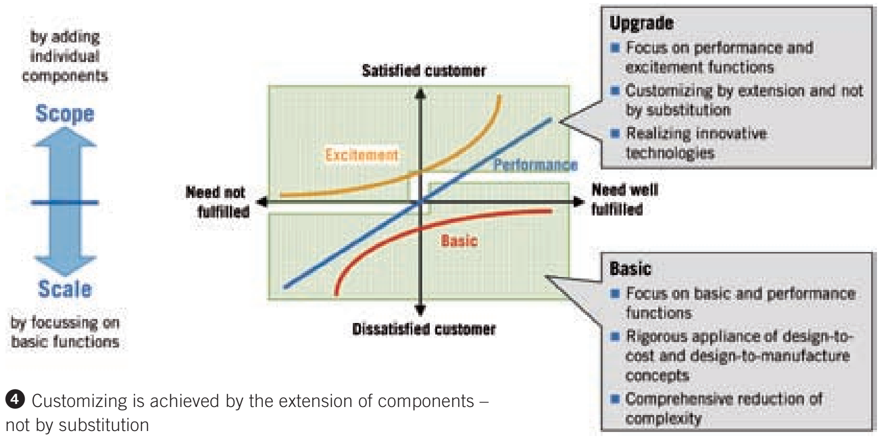
Currently, all automotive manufacturers develop the components of the electric powertrain in cooperation with suppliers. For example, Daimler and the German conglomerate Evonik founded a joint venture called Li-Tec for the development of lithium ion batteries. The essential economies of scale cannot be achieved by designing proprietary systems. This has been analyzed by Bosch for the development of the battery. A manufacturer producing 50,000 units has additional costs of 450 euros per battery compared to a manufacturer that produces 500,000 [5]. Therefore, the market for core components must be opened up and manufacturers must be able to integrate their products through standardized interfaces. Open integration is essential, particularly at the beginning of the market introduction of electric vehicles. Moreover, the focus has to be on components that are not relevant for the customer.

One example of open integration in industrial history is the IBM PC. The PC is characterized by a simple system architecture of a motherboard and extendable plug-in cards that can be produced by third-party suppliers. This system architecture has evolved into an unofficial industry standard that is still present today. The standardization process was supported by the fact that IBM made it usable without a license. As a result, the market for computer components has grown and the costs of the components have been reduced. The effect of transferring this idea to the automotive industry would be the cross-manufacturer utilization of components. One example could be the motor of an electric vehicle. The same electric motor could be integrated into different vehicles of different manufacturers. The customizing itself could then be done by software. Another option for the motor is to be used not only by different manufacturers but also in different industrial applications. Economies of scale can be created by increasing the number of units on the component level. The results of industrial projects at manufacturers of electric motors and electric systems have proven that approximately 25 % of the costs can be saved if the number of units is doubled.

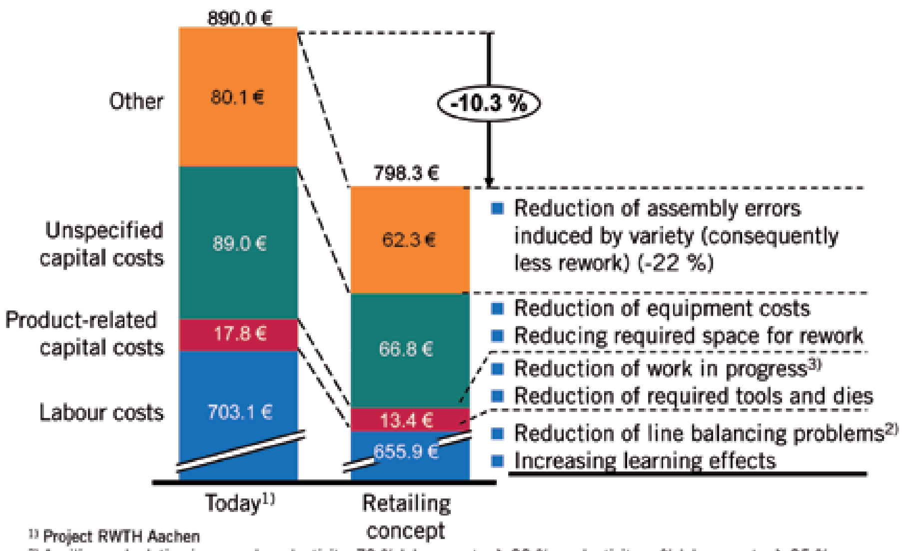
REDUCING THE COSTS OF VARIABILITY

One of the main challenges in automotive production is to cope with the large product variety. For the VW Fox compact car, the customer is able to configure up to 7.53×10^{12} different variants. The resulting situation in final assembly is extremely severe. In the operational process, a larger number of variants will result in assembly errors and quality problems, balancing problems, and tied-up capital. In order to maximize the capacity utilization of the major investments, automotive manufacturers are compelled to produce numbers of units based on forecasts. Thus, production plans must continuously be adjusted, since the forecast demand rarely meets the actual one. As a result, inventories are created and these must be pushed onto the market at discounts. Furthermore, promised delivery rates cannot be met.

By classifying product features into excitement, performance and basic, as in the Kano model, the vicious circle can be broken and the point of order can be shifted to the end of the value chain, ❹ [6]. Basic performance functions must be integrated into a basic version of the car that also includes all the safety and basic comfort features that are relevant. Additional performance and excitement features are optional extras that must be conceived as extensions and not as substitutions. Scaling the range of the car could be an additional option in the electric vehicle. The basic car includes only a minimum range and the customer pays only for the essential requirement. Subsequently, additional batteries can be mounted into an existing space with the corresponding thermal management. Shifting the point of order to the end of the value chain has a positive effect on the overall costs, which can be reduced by around 10 %, ❺.



❹ Customizing is achieved by the extension of components – not by substitution

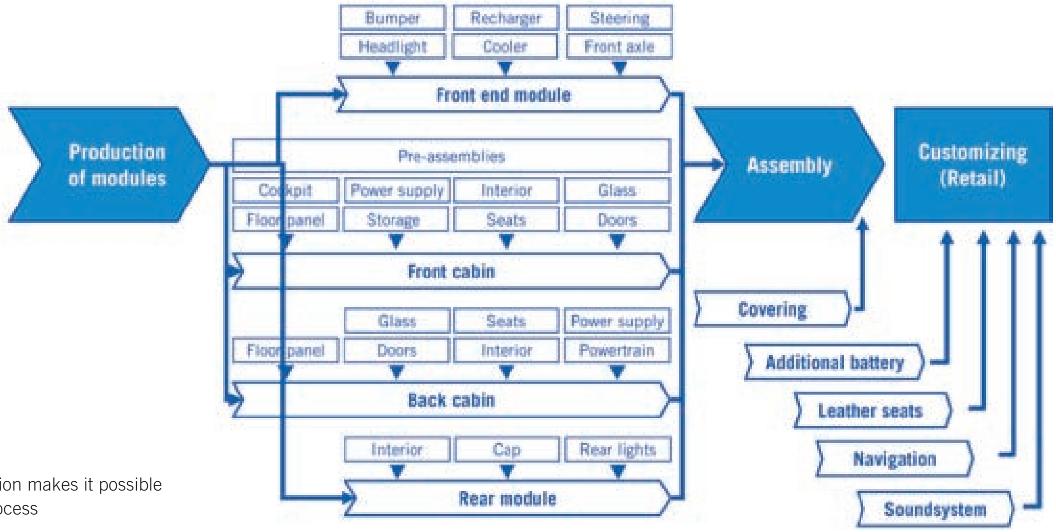


1) Project RWTH Aachen
2) Auxiliary calculation increased productivity: 79 % labour costs → 90 % productivity, y % labour costs → 95 % productivity → y = 74.84 % labour costs → relative difference: -5.3 %,
3) Auxiliary calculation work in progress: ca. 25 % of product-related capital costs is work in progress and 2/3 of the product functions can be shifted in the retailing concept → relative difference: ~-0.5 %

❺ Reducing variability in the process has a positive effect on the cost structure

REDUCING COMPLEXITY BY MODULARIZATION

In contrast to other industries, the production rate of the value chain in the automotive industry is defined by the production plant and not by the customer. The objective is to stabilize the complex value network. Often, the unstable process in the value chain is the painting process, which has a process stability of only 80 to 85 %. The subsequent final assembly is decoupled by a buffer storage. Thus, primarily in the final assembly, a stable production plan exists and suppliers receive a specification of the required module no longer than eight hours before installation. Sourcing from low-cost countries over a large distance is impossible. If the point of a stable production plan is shifted further to the beginning of the value chain anyway, the size of the buffer storage between the final assembly and the painting process will increase.



❻ Comprehensive modularization makes it possible to redesign the value-added process

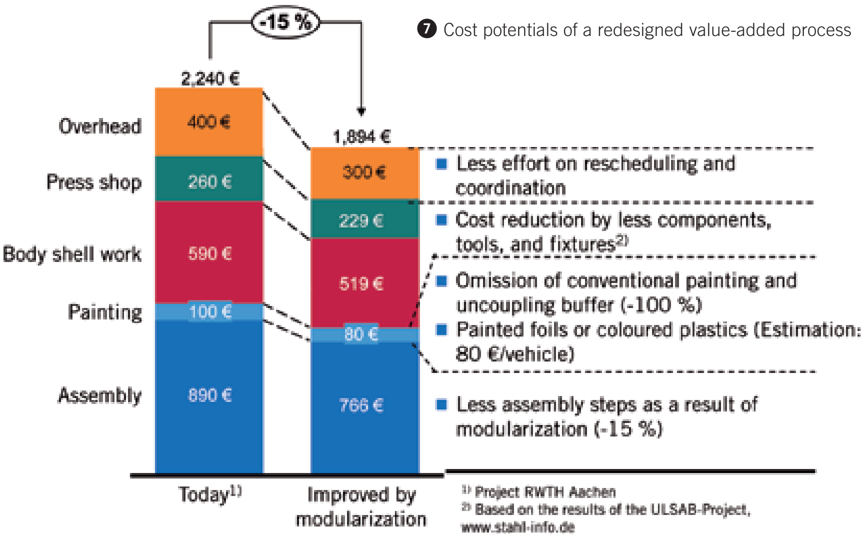
By modularizing the structure of the product, the conflict described above can be solved. This modularization has to be more comprehensive than existing concepts for the interior. The basic idea is to decouple the load-bearing structure from the painted body shell. This eliminates the problem of process stability in the painting shop. One example of a comprehensive modularization concept is demonstrated in the results of the EU research project ILIPT. The load-bearing structure is divided into four modules, which are scalable in length and width. Modules for the powertrain and bodyshell are added to the load-bearing structure. Modularization makes it possible to redesign the value-added process, ❻. The painting of the vehicle can be shifted to the end of the process and can be substituted by painted foil or coloured plastics. The different modules can be assembled by CMT welding (Cold Metal Transfer). The results are lower production costs and scalable product structures, ❼.

CONCLUSION

Electric mobility is no longer a niche market. There is an evident market potential for electric cars, although they have to face competition from conventional vehicles right from the start. Customers are not willing to pay more for electric mobility. Therefore, the selling price is crucial for market success. Due to the high additional costs caused by the battery, there is enormous cost pressure in production. Considerable cost-cutting effects can only be achieved by a "purpose design" concept. For this, the following principles are essential:

: economies of scale by open integration
: reducing variability costs by differentiating between basic, performance and excitement functions
: less complexity by a comprehensive modularization of the product structure.

In this way, considerable cost potentials can be opened up. The design principles demonstrated are extensively applied in the StreetScooter project of RWTH Aachen University [7]. A range of companies from medium-sized mechanical engineering companies to large automotive suppliers are developing a low-cost electric vehicle for urban traffic in a cooperative network. A cost-effective design is ensured by the simultaneous development of the product and the production processes.



❼ Cost potentials of a redesigned value-added process

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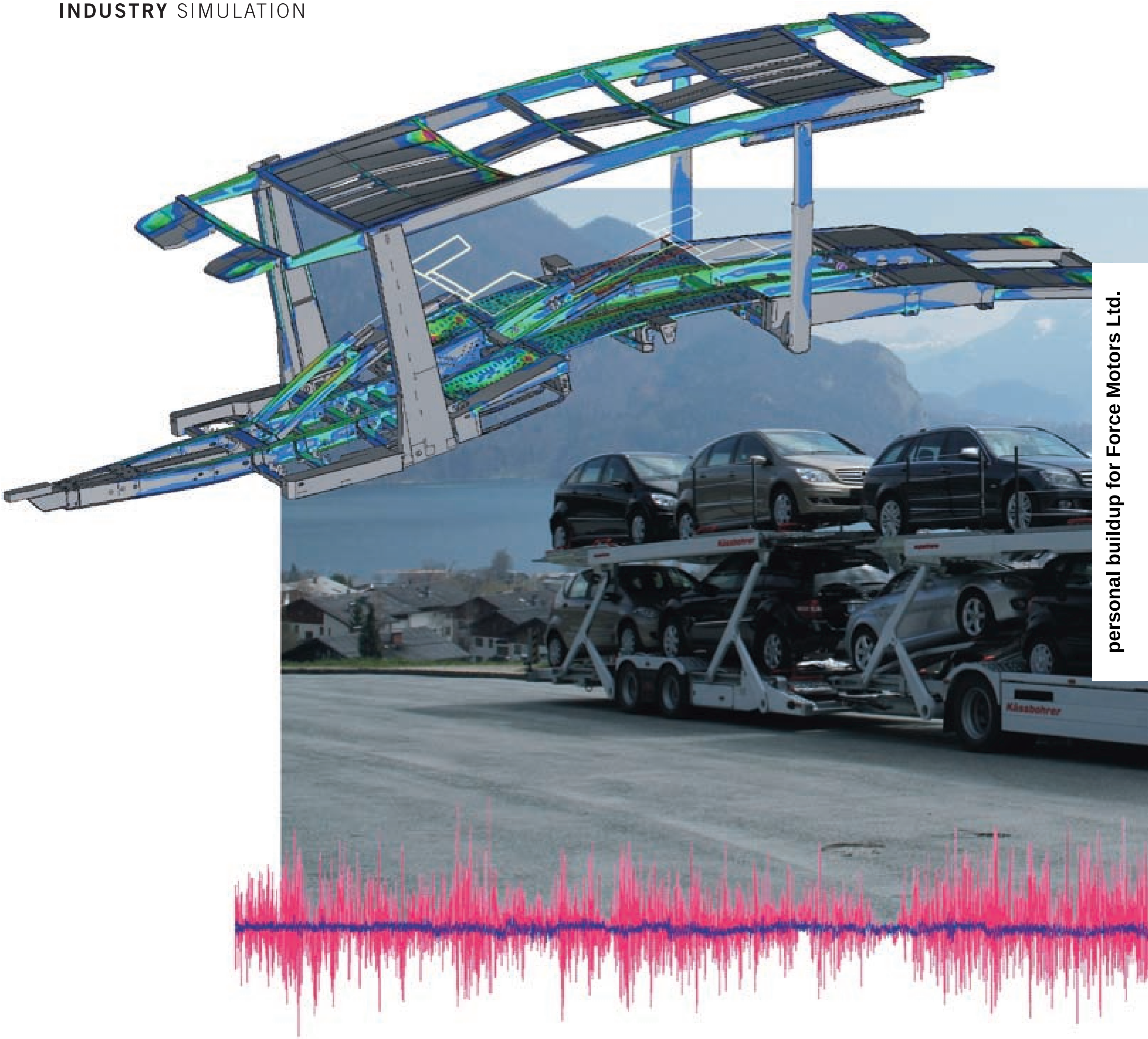

INFLUENCE OF DIFFERENT ROAD CONDITIONS ON THE LIFETIME OF A CAR CARRIER

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In the last few years new chances have emerged to manufacturers of vehicle transporters due to the growing globalization and the opening up of new markets in eastern Europe. But these chances came along with new problem statements as well. For example the bad road conditions in eastern Europe cause increased load to the structures hence reductions in product lifetime compared to western European roads. Kässbohrer and dTech Steyr have developed a method to forecast the influences on the product lifetime.

OPTIMIZATION OF THE PRODUCT LIFETIME

Kässbohrer Transport Technik is investing great effort in further development of their vehicles due to the usage of new methods and technologies (like FE-Simulation and measurement technique (test trials, fatigue testing)) to go on increasing the product durability. The complexity in designing a car carrier is due to legal restraints in different countries regarding certain dimensions (restrictions in height, length and width) plus the gross vehicle weight rating and on the other hand the customers demand to carry as many vehicles as possible on a single ride (loading factor) while maintaining maximum handling comfort for the driver.

These – to some extent divergent – requirements are directing the development engineers to designs which are reflected in the special dynamics of the system and the significant interaction of the components.

Given the fact of the above-mentioned issues, this publication covers the method development of a vehicle transporter specific lifetime calculation matched to the manufacturing process at Kässbohrer Transport Technik. This method development did result from a close cooperation of Kässbohrer Transport Technik with dTech Steyr – Dynamics and Technology Services GmbH.

TEST RUNS

A proper lifetime simulation is subject to verified stress calculations for the different load directions. Up to now stress- and deformation optimisation of components have been carried out using estimated extreme loads. But these extreme load conditions are of limited relevance for lifetime calculations. For a lifetime simulation it is necessary to know realistic load conditions in all spatial directions plus the frequency of their occurrence, summed up in a load spectrum. To get these load information under different operating conditions, Kässbohrer Transport Technik initiated a recorded test run thru the East of Europe. The individual route sections have been divided into four load classes based on different road conditions:

I) selected sections of roads western highways
II) eastern good condition country roads
III) eastern average condition country road
IV) eastern bad condition country road.

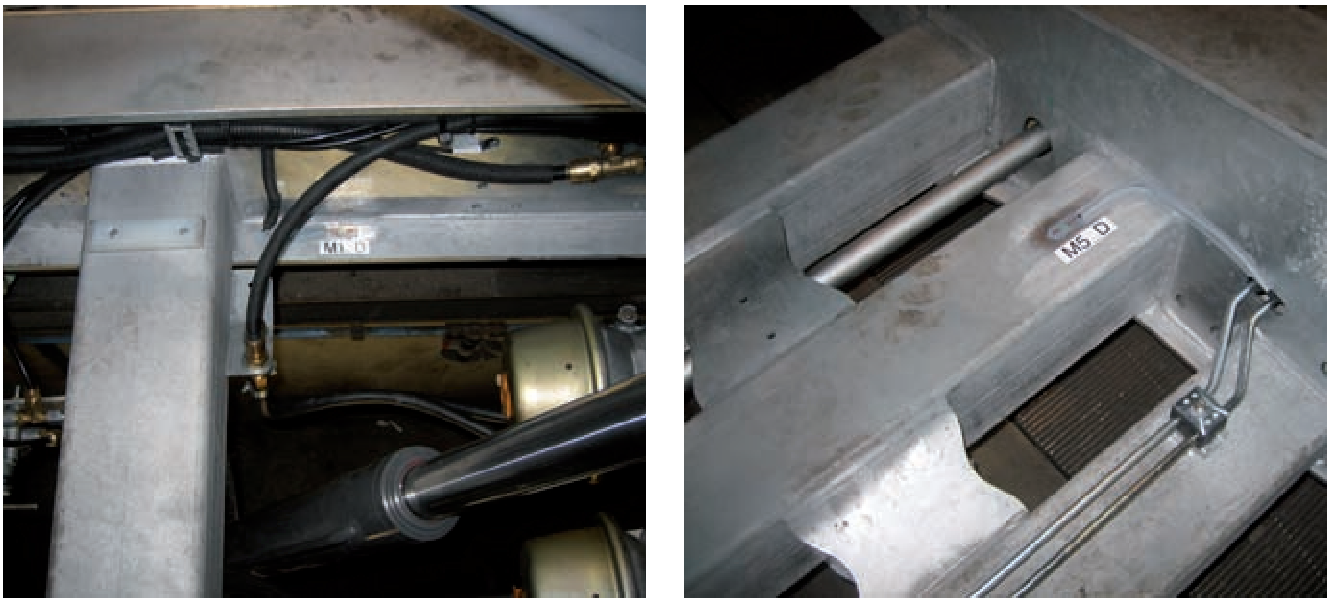
The truck as well as the trailer have been equipped with 45 measuring pickups in total. Accelerometers and resistance strain gauges have been applied. The accelerometers have been used to gauge accelerations at peripheral structures, while the strain gauges transmitted the central elastic deformations of the main beams.

To calculate deterioration, individual load spectrums are necessary. The load spectrum should expose the loads occurring per driven kilometre within different operating areas. This means the information about intensity and number of load amplitudes are stored in the load spectrums. Certain pick up points out of 45 have been chosen to get as representative as possible quasi-static loads for the frame`s main components in all three spatial directions. ❶ shows the measurement points M1_D and M5_D which are storing the stresses and accelerations on the longitudinal beam in the area of the trailer´s axles.

ANALYSING THE MEASUREMENT DATA

Mean stress as well as amplitude stress are the main factors for a simulation of duration. From the FE-model the mean stress in the chassis rail was determined in the area of the DMS sensor adhered in the real vehicle.

The amount and the magnitudes of the load cycles are much less on western terrain than on eastern roads. To exemplify, ❷ shows a comparison between measuring records of a good western highway (blue) and a bad condition country road in the east (red). Here you can see the big difference in the amount of the stresses, which highly affects the lifetime.



❶ Illustration of the measurement points M1_D and M5_D at the longitudinal beam of the trailer

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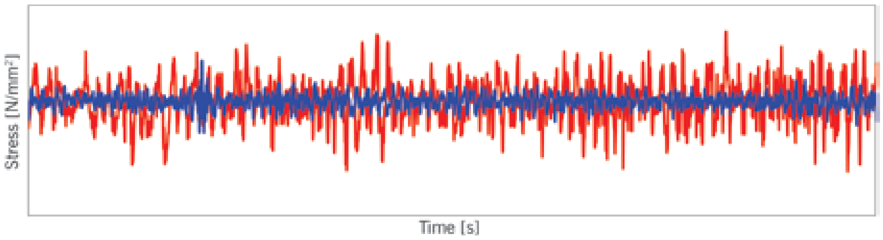


❷ Cutout of a comparison between a measurment record of a good conditioned (blue) to a bad conditioned road; strain gauges measurement point at the chassis frame

The analyzing table in ❸ shows the allocation of the load change on the four fields divided in loads from three directions in space. It´s cognizable that there is a big difference in their compositions. On western highways more than two-thirds of the load cycles are caused by soft vertical concussions. In contrast, on very bad country roads the load cycle frequency is shifting to transversal loads (65 %) and to longitudinal loads (11 %).

ADJUSTMENT FIELD OBSERVATIONS/ SIMULATION TO THE DETERMINATION OF LOCAL LIFETIMES

To get the local deflection stresses, FE simulation on the complete structure of the trailer is used. The „range count" classified conditions are simulated with the FE model, ❹, and are evaluated with the special fatigue software developed by dTech Steyr (dTechWeldEndurance). Thus it is possible to simulate each critical point on the chassis and evaluate the local endurance.

A realistic material model is absolutely essential for a significant durability simulation. The data basis of the Kässbohrerspecific Woehler curve was defined by guidelines and norms respectively by provided data of fatigue tests [1, 2, 3, 4].

To see the influence of typical strain conditions to the fatigue, the load spectra are separated. The principle approach is that laboratory S-N curves are adjusted so that good correlation between simulation and practical observation is achieved for critical areas of the trailer structures. For each step of adjustment all partial damages are compared quantitatively in a matrix of the loading conditions against the crack critical areas.

During the adjustment work the collective portion with the high number of cycles and low stresses configured the fatigue limit. The collective portions with the higher stresses are used to determine the slope of the S-N curves. To find the best correlation with detected cracks, evaluation of trends and their consequences on the partial and total damage is performed. ❺ shows this Woehler curve.

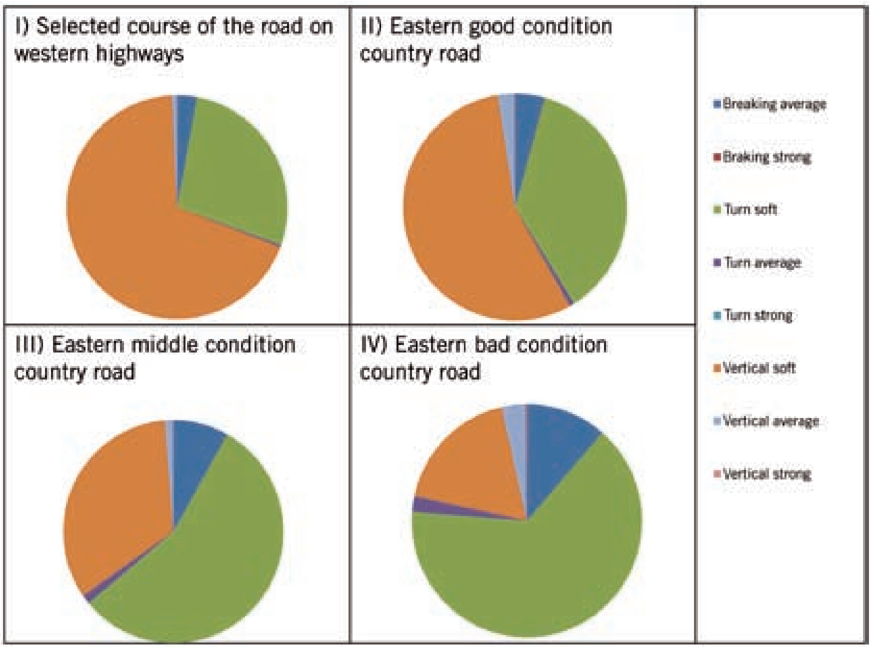
ANALYSIS OF THE INFLUENCES OF DIFFERENT ROAD CONDITIONS

The load conditions from the longitudinal dynamics are low enough for all track categories that the fatigue simulation results show zero damage. The amplitude stresses are below the level of the endurance limit.

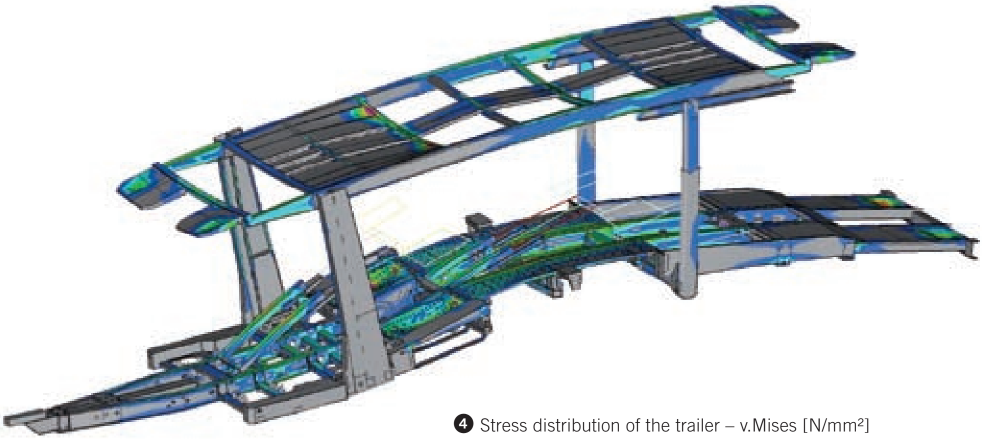
Loading conditions I)
selected sections of roads western highways:
For the cruising range the most damaging mechanisms are from frequent occurrence of light to medium vertical hits. The lateral accelerations resulting from turns for instance are relatively rare in these loading conditions. So the damage from cornering is relatively marginal.

Loading conditions II)
eastern good condition country roads:
The amount of the stressed load cycles rises by the factor of three in comparison to western roads. Primarily the higher strain conditions are increasing. Damage from the lateral acceleration is rising remarkable stronger compared to the vertical hits. The matter of this behavior can be that the bad road conditions make the driver turn around the chuck-holes or that vehicle roll movements are higher because the roads are uneven.

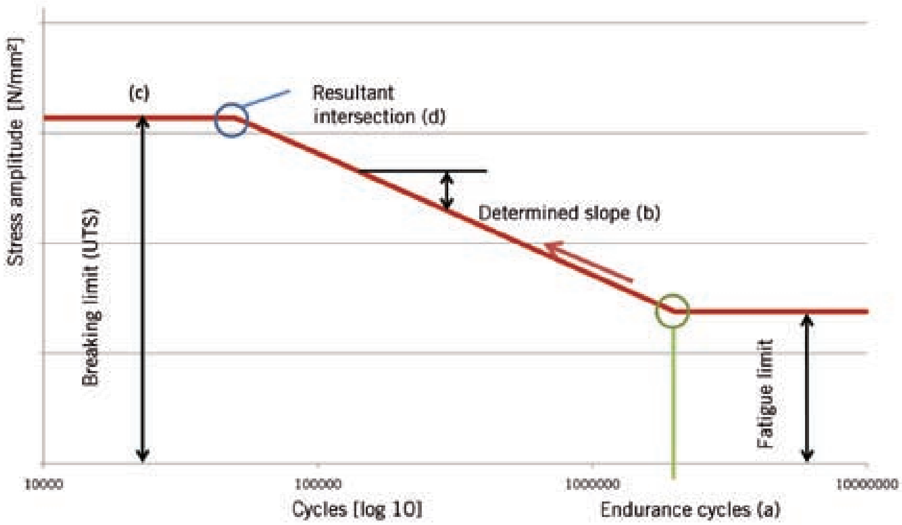
Loading conditions III)
eastern average condition country road:
Damage portion from the lateral acceleration is further rising while the portions from the vertical hits are declining. The



❸ Frequency distribution of the load changes regarding different road conditions



❹ Stress distribution of the trailer – v.Mises [N/mm²]



❺ Principle Woehler curve according to the verification of field observations and simulation

matter of this behavior can be that roads have more turnings and different driving styles as well.

Loading conditions IV) eastern bad condition country road: The medium vertical hits which are of frequent occurrence are the most damaging under these conditions. But also the medium lateral accelerations are much more damaging than with the other road conditions. The estimated endurance is significantly reduced under these conditions.

In reality a vehicle will not run just in one of these conditions all its life. Therefore, depending on the specific route, a mix of these four loading conditions shall be used. An additional data collection on a fleet of trucks is planned which should help to get more information and knowledge from these new conditions.

SUMMARY

The strain which is being exposed from the road to the chassis is much higher on bad road conditions in eastern Europe compared to good road conditions (for example Austrian highways), in frequency as well as in the amplitude spectra. These cognitions are shown through implemented measuring turns where structural loads have been measured through applied sensors, which act as input values for a specific development method from Kässbohrer Transport Technik for durability prognosis.

Beside the record of the load collectives during measuring terms a stress-cycle diagram (Woehler curve) was developed with the help of theoretical attempts in combination with field observations. That makes it possible to achieve a very realistic lifetime determination.

This prediction gives the chance to accomplish specific weak point analysis, to develop (weight)optimized constructions, optimized chassis suspension set up and layout criteria for the usage according to the road condition and in this manner to optimize the product quality.

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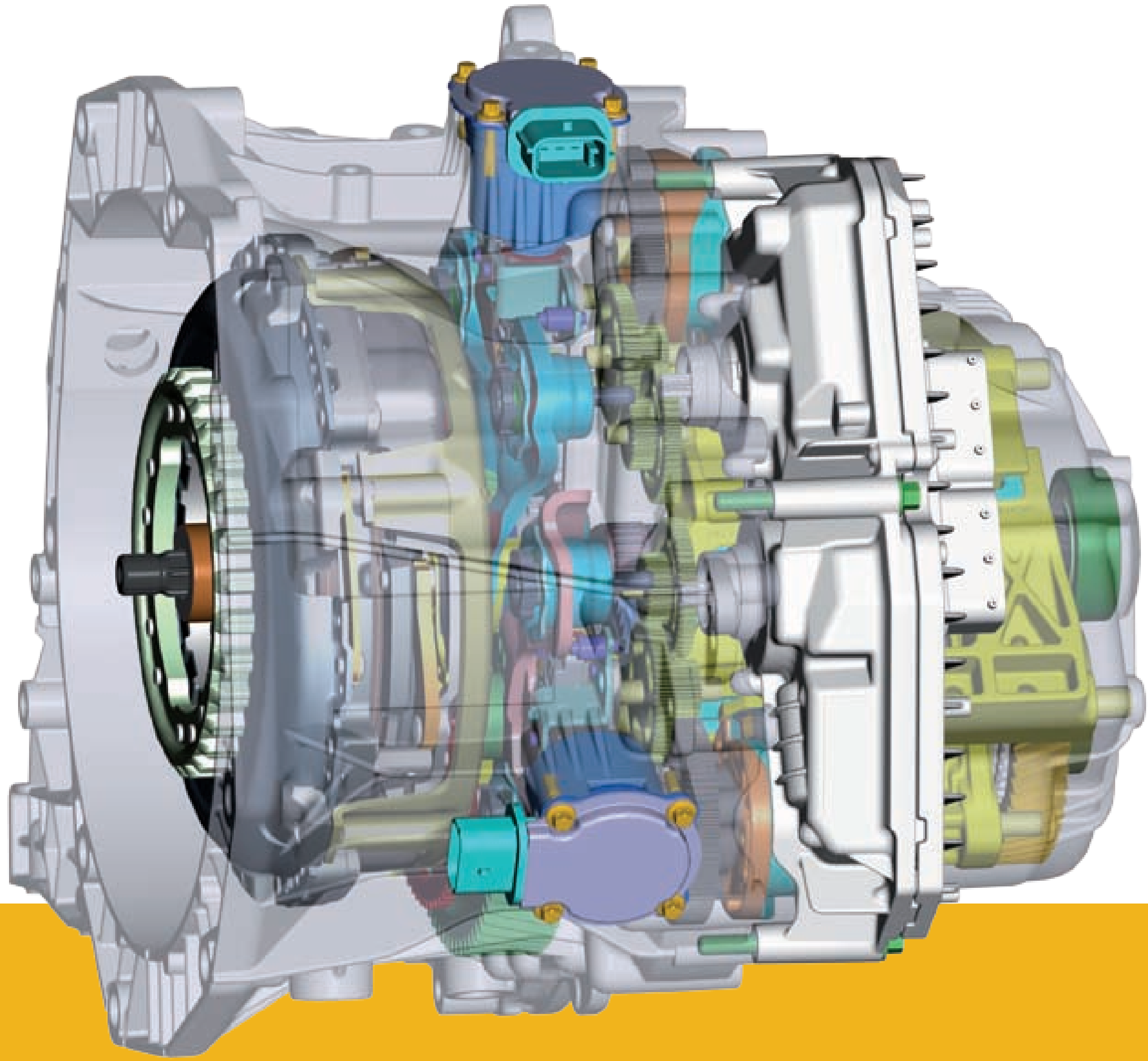

DUAL CLUTCH TRANSMISSION WITH DRY CLUTCH AND ELECTRO-MECHANICAL ACTUATION

The existing PowerShift transmission portfolio by Getrag with wet dual clutch and electronically controlled hydraulic actuation is expanded by a new version with dual dry clutch and electro-mechanical actuation. The range of applications sums up front-wheel drive, all-wheel drive, inline drive and longitudinal transaxle but now the new small transmission of the type 6DCT250 for front-wheel drives. The main characteristics of the new transmission, developed for worldwide applications with a torque capacity of up to 280 Nm, are the clutch and shift actuation with very low auxiliary power need and an integrated mechatronic actuator module.

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❶ Gearset with one dependency (4/6) and gears 1 to 6 plus reverse gear (R)

ADVANTAGES COMBINED

The dual clutch transmission (DCT) technology enables the car manufacturer to implement different vehicle characteristics by simple modification of calibration using the same hardware. Variable launch control is an important advantage for the system integration to achieve driveability differentiation with best balance between performance and fuel efficiency.

With the introduction of Getrag PowerShift transmissions to the market, in a wide bundle of front-wheel drive (FWD, 6DCT450), all-wheel drive (AWD, 6DCT470), inline drive (7DCI600) and longitudinal transaxle applications (7DCL750) from 150 to 750 Nm input torque, the advantages of manual and automatic transmissions are combined in an optimal manner. This leads to high efficiency with low CO_2 emissions linked with comfortable or sporty shifts without torque decrease [1-5]. In the second generation with dry clutches and fully new electro-mechanical actuation for clutches and shift actuation now the 6DCT250 is developed for further improvement of fuel economy by minimizing drag torque losses compared to wet clutches and by keeping out any hydraulic pump [6, 7, 10].

The Getrag PowerShift transmission portfolio perfectly satisfies the central challenge of offering the customer an optimised power train with improved fuel economy, low CO_2 emissions, and excellent driving comfort combined with fun to drive.

LAYOUT OF THE 6DCT250 TRANSMISSION

The dual clutch transmission 6DCT250 for FWD application, see Title Figure, has a three-shaft design with odd gears served by a first clutch, reverse and even gears by a second clutch. The dual dry clutch consists of two single plate clutches in parallel arrangement which can be used both for launch and shifting.

The gearset is designed in two versions with one dependency (4/6), ❶, or two (3/5, 4/6) dependencies. The single dependency version provides exceptional ratio flexibility in an efficient package length. In more demanding package applications, the double dependency can be offered with an approximately 10 mm length reduction, but some associated reduction in ratio flexibility. The overall design of the gearset, synchroniser and shifting system stays common.

SHIFT SYSTEM

For shifting the gears there are two e-motors implemented which rotate two shift drums by intermediate gears. The shift drums are identical and each of them has one groove, which activate the shift fork movement, ❷. Even for the design with two dependencies the shift drum stays identical.

By using the shift drum technology for each transmission part no locking device is required to prevent engaging two gears in the same transmission path by failure, it is inherent in the groove. The transmission performance with two shift drums is characterised by:

- high shift flexibility
- dynamic and comfortable clutch-to-clutch rock cycling (1 – R) without need for synchronizer engagement or disengagement
- all shifts are torque supported, some of them with the use of intermediate gears.

The axial positioning of the sleeves is done by detent springs and balls in the

❷ Inner shift system with the parts intermediate gears, shift drums, linkages and shift forks

❸ Dual dry clutch with dual mass flywheel

synchronizer system for getting low drag torque of disengaged gears for good fuel economy of the powertrain independent from tolerance chain.

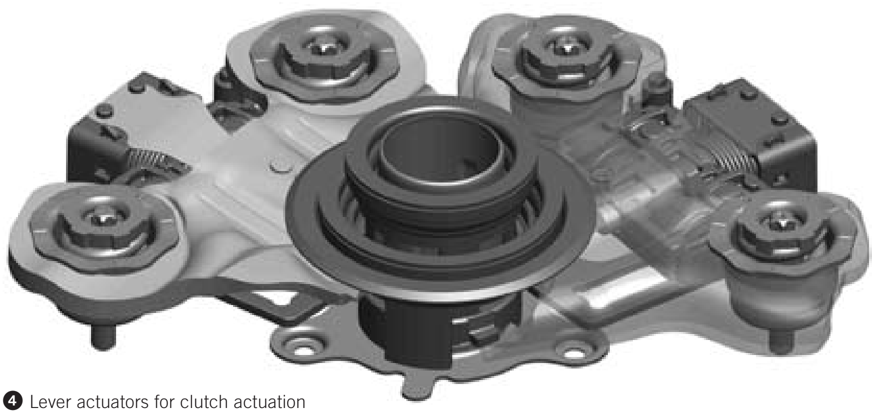
DUAL DRY CLUTCH

The dual clutch has three pressure plates which are positioned on the hollow shaft of the transmission with a ball bearing. The torque transfer is done via two single plate clutch disks in a parallel arrangement for the two transmission paths. All clutches are normally open systems for avoiding any tie up occurrence even in control malfunction situations. They have an integrated travel controlled mechanical wear adjustment system to keep the necessary actuator travel in a small range, ❸.

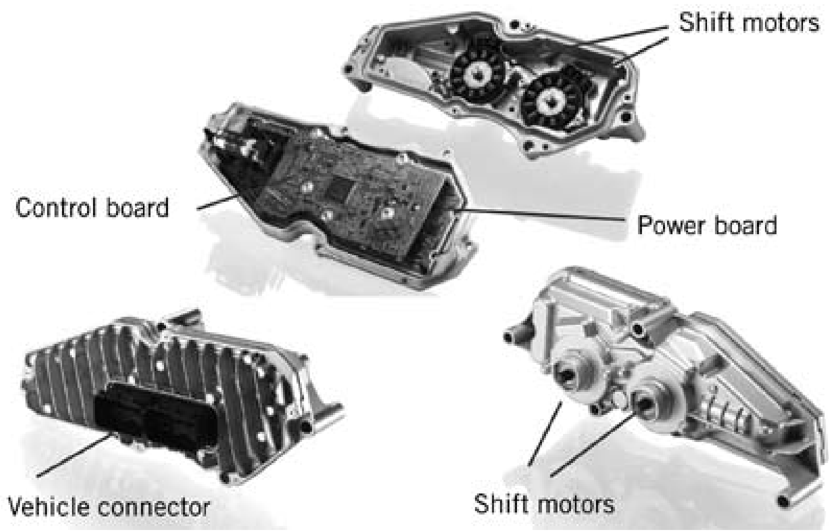
The dry dual clutch does not need any oil flow for lubrication and cooling and there is no need for a transmission oil cooler and oil pipes in the vehicle which saves costs on parts and vehicle assembly. Additionally, the drag torque is lower compared to wet dual clutches which is a contributor for low CO_2 emissions reached with this transmission concept.

The clutch system is available in a version with dual mass flywheel (DMF) for torsional isolation between engine and clutch and in another version with a direct cardanic connection of engine flywheel to clutch with dampers in the disks. The cardanic type clutch system is designed for applications with low engine excitations, for example naturally aspirated gasoline engines. The isolation of engine irregularity is done by the dampers in the output disks with support of clutch slip control depending on the driving situation. The clutch is designed for a very compact powertrain package and requires only a flexplate to connect to the engine. The total thermal mass is 6.2 kg. The DMF clutch system can even cover applications with higher engine excitations such as charged diesel and turbo charged direct injection gasoline engines, also with three cylinders. The total thermal mass is 6.9 kg.

To avoid any hydraulic pump losses and achieve very low mean power consumption for the whole actuation system, both shift actuation and the clutch actuation is fully done by e-motors. The clutch actuation is performed by a fully new developed lever actuator [8]. The electric motor drives a ball bearing spindle which connects to a trolley. For lowest possible actuation energy requirements the system is designed in the way, that the trolley only has to move the lever pilot point to generate the actuation travel for the clutch. On the other end of the lever, ❹, there are two preloaded springs for each lever system which store the actuation energy to maintain a low e-motor torque level for



❹ Lever actuators for clutch actuation



❺ Components of the mechatronic actuator module (MAM) with control and power board shift motors and vehicle connector

the actuation forces needed at the lever spring finger tips for high clutch torques.

With this electro-mechanic actuation system a high dynamic of clutch engagement and disengagement fully independent from shift actuation is reached which even can be used for intermediate clutching for synchroniser protection at high differential speeds. On the other hand it needs only low amounts of energy by making obsolete any pump drive and it delivers best performance for sailing, start-stop and hybrid systems independent from engine running.

ELECTRICAL SYSTEM AND SOFTWARE

For the electronic control and the electromechanical actuation system the following major development goals were defined:

: actuation system fully attached to the transmission to guarantee a plug and play transmission system for the integration into the vehicle with only one electrical interface to the customer
: minimization of the electrical power consumption for actuation to guarantee the CO_2 emissions reduction targets
: common part strategy of the cost intensive automation components for different applications of the world market
: serviceability to support maintenance.

The central control unit which comprises of the transmission control unit (TCU) and the two shift motors is named mechatronic actuator module (MAM), ❺. The integrated shift motors are brushless DC motors with low inertia for high shift dynamic. The MAM is a consequential optimisation of the well-known Getrag shift actuator technology for automated manual transmissions (AMT).

The MAM with its ambient air cooled aluminium housing is located on the cold side of the powertrain. Compared to integrated mechatronic modules which are installed inside the transmission this leads to much lower ambient temperature conditions. All the design work and material choice considers the life time requirements of the customers to guarantee the best reliability.

The electrical components were developed under the premise of a common part strategy for different applications around the world. Nevertheless flexibility is given to fulfil the customer specific requirements. The common part strategy led to a standardised TCU and shift motors for the dry dual clutch transmission. This ensures high quality and an optimised cost situation for further applications.

The model based software is developed by Getrag inhouse with high modularity. For different applications the calibration can be adopted according to the individual requirements for getting best customer satisfaction with respect to optimised balance between comfort, fuel economy and fun to drive.

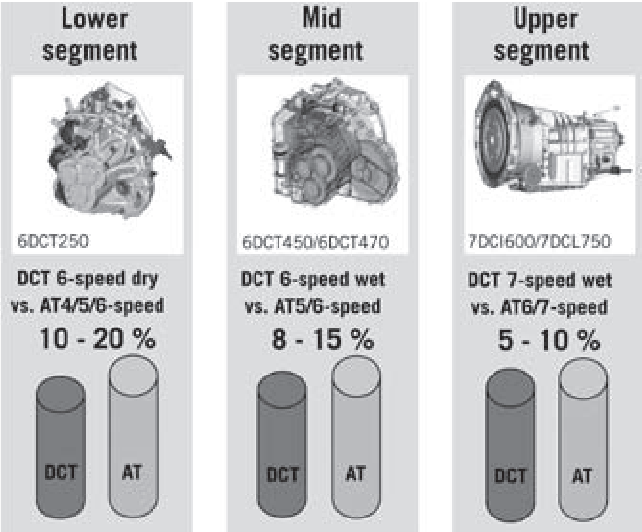
MODULARITY CONCEPT

Following the approach of a product for the world market, various versions of the transmission for a wide spread of applications in different combinations are available:

: FWD and AWD
: center distance of 183, 188, 197 and 205 mm
: gearset design with one (4/6) or two (3/5, 4/6) dependencies
: dual clutch systems with torsional damping by either DMF or damped disks with slip control.

For new applications usually the transmission housings need to be adapted. The ratios of the gearset and final drive are flexible and can be optimized to customer demands. The software receives commonly a new calibration to optimise the driving feeling for each vehicle application.

All other parts are carry-over parts from base variants mentioned before. This comprises the majority of the value of the transmission especially with



❻ Fuel consumption of the PowerShift transmission family compared to torque converter automatic transmissions for the lower, mid and upper vehicle segment

regard to the cost relevant automation components.

FUEL CONSUMPTION

DCTs combine strategic clutch control for brilliant dynamics with excellent comfort and low fuel consumption. The main development target of Getrag PowerShift transmissions is better fuel consumption over any other automatic transmission concept. The 6DCT250 compared to the state of the art planetary automatic transmissions with torque converter shows up to 20 % improvement of fuel consumption, ❻. It combines the dry clutch for low drag torques with an efficiency optimised gearset and electromechanical actuation for clutch and shift system without the need for any hydraulic pump drive.

The electro-mechanical actuator concept considers the superior CO_2 reduction targets by implementation of a new developed clutch lever actuator where actuation energy is only send between preloaded springs and the clutch pressure plates by electrical driven movement of the pivot point and with brushless DC-motors of low inertia for shift and clutch actuation.

In the practical Getrag driving cycle which includes urban, interurban and high speed (wide open throttle) parts, the average electrical power consumption for the clutch and shift actuation system (without microcontroller) is only 15 W. This is one of the main contributors to the CO_2 emission reductions gained with the 6DCT250 compared to all other automated transmissions. This equates to a current consumption of less than 1.2 A.

HYBRIDISATION CAPABILITY

The 6DCT250 dual clutch transmission offers superior adaptability for vehicle hybridisation relative to virtually all other automatic transmission alternatives. The primary elements which result in this advantage include:

: As a result of superior ratio flexibility, gear ratios can be tailored to take advantage of electric traction motor launch or boost.
: Unlike conventional A/T alternatives, the single or dual hybrid electric traction motor(s) can be directly coupled to either set of mechanical geared shafts, or multiplexed between the two geared shafts. Of course, it can also support integration of motor at the transmission input if desired.
: Electric traction motor sizing can usually be reduced when applied to Getrag PowerShift transmissions relative to other auto transmission alternatives.
: Since there are no onboard hydraulic systems, 6DCT250 can accommodate engine start/stop functions without any additive hardware required, such as electric pumps or hydraulic storage devices [9]. This has already been successfully achieved in demonstrator vehicles at Getrag.

INDUSTRIALISATION AND OUTLOOK

The design of the PowerShift 6DCT250 dual clutch transmission by Getrag focused on maximised flexibility in combination with a common part strategy. During the simultaneous development process for different applications and customers the unification of the system requirement specification and a strong change control process were central tools. To be a world market product requires world-wide industrialisation/production facilities. The launches for European and North American applications have already started in Bari (Italy) and in the new plant Irapuato (Mexico).

Common and flexible production strategy enables short-term project realisation. Therefore production and assembly lines, including end of line testing, are 100 % carry over and completely interchangeable between both facilities.

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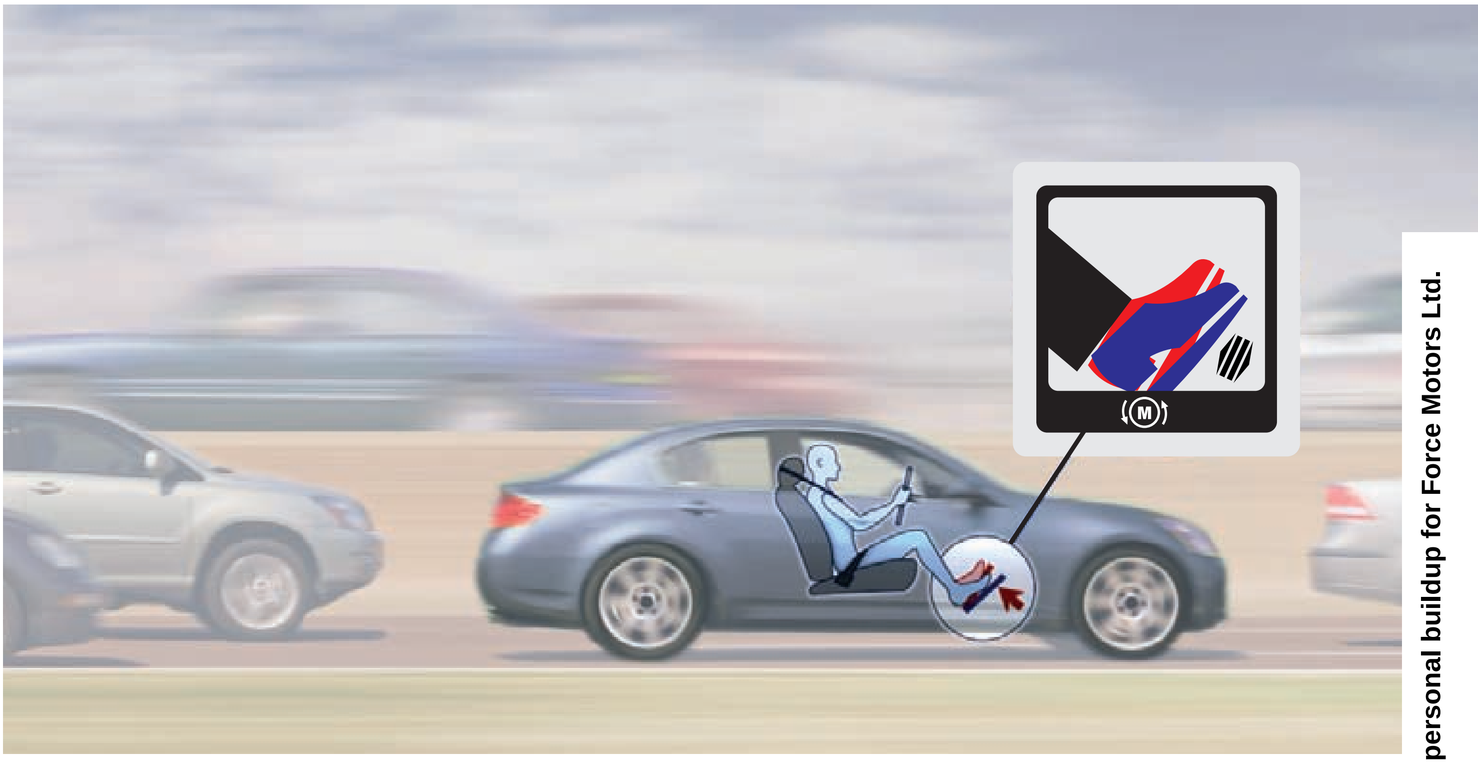

HAPTIC INTERFACES

Drivers must often decide themselves how they react to the flood of signals and informations in and around the car, which information has priority in the respective situation and what reaction it requires. Driver assistance systems are therefore a suitable means for reducing the load on drivers. At the same time, communication with the driver must be structured so that it does not become a stress factor itself. It should therefore be possible to follow requests to take action intuitively. If the communication channels between humans and machines are considered, then the visual channel is largely exhausted. On the acoustic channel the driver tends to have more capacity free, however audible feedback has a possible disadvantage. It is not discreet, because it cannot be perceived exclusively by the driver.

On the other hand, haptic feedback is directed exclusively at the driver and is immediately perceived – in contrast to optical gearshift indication. Based on this, the Chassis Electronics segment of the Chassis & Safety Division at Continental has developed the accelerator pedal further to a haptic interface in cooperation with Continental Engineering Services. This is an obvious choice, as the accelerator pedal exerts a major influence on the longitudinal vehicle dynamics. Two studies, one of them with 24 test subjects, have meanwhile confirmed that the new Accelerator Force Feedback Pedal (AFFP) has a broad range of application possibilities and the haptic signals are intuitively followed by drivers. Specific signals as information, for example on the optimum gearshift point, have resulted in average fuel consumption savings of over 7 %. With a modular design and scalability, vehicle manufacturers (OEMs) can realize the desired function in individual cases with the AFFP. The haptic signals range from a constant pedal counterforce to vibration to a subtle double-clicking.

ACCELERATOR FORCE FEEDBACK PEDAL

The AFFP has a modular design: The haptic feedback is generated by an active module, which can be applied to standing and hanging pedals. In the active mode, the actuator, a motor spring (which also produces the necessary redundancy to the pedal spring), a sensor for detecting the motor position, a temperature sensor and a compact engine controller (Electronic Control Unit, ECU) for actuation are integrated. The ECU of the AFFP is connected to the engine controller and other control units via the CAN bus and receives information from there as the basis for controlling the actuator.

The brushless actuator optimized to produce a high torque transmits its power in the release direction (= release of the accelerator) to the pedal rod with a solid shaft. The position of the motor is traced by the motor spring even without a power supply. A diagnostic function monitors the motor's state. Depending on the purpose (required signal level), the active module can be equipped with single or two-row actuators with various outputs. With single-row actuators, torques between 0.3 and 1.4 Nm can be produced, and with two-row actuators up to 2.2 Nm is possible. This scaling range also affects the weight. In the highest output class the actuator weighs approximately 500 g (18 oz.). While the first AFFP system generation from 2007, which is already in series production, weighed a total of approximately 1.9 kg (4.2 lbs), the current, second AFFP generation is at < 1.2 kg (2.6 lbs), ❶. With the third system generation currently in development,

ACTIVE ACCELERATOR PEDAL AS INTERFACE TO DRIVER

More safety at the wheel, improved fuel efficiency and the following decrease in CO_2 emissions as well as greater driving comfort cannot be realized without the driver's cooperation. As the highest control level, human beings play a decisive roll in deciding whether or not the technical potential of a vehicle is fully utilized. To optimize the cooperation between human beings and machines, Continental has developed an active accelerator pedal that supports the driver with haptic feedback.

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it will probably be possible to achieve a weight of 800 to 900 g (1.8 to 2 lbs) through increased integration and new materials. Prototypes of the second AFFP system generation are currently being tested in vehicles at various OEMs.

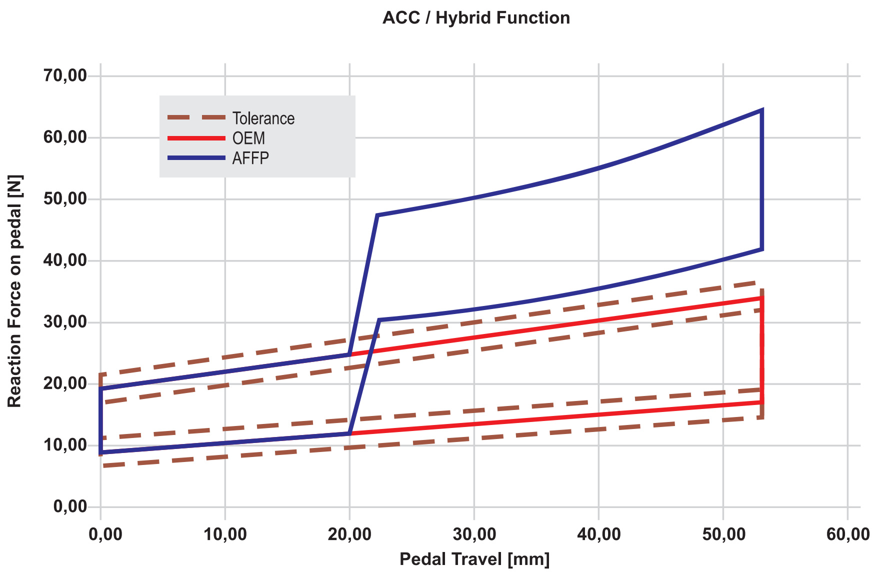
In this second system generation a driver pin is used to transmit force to the pedal. The pin is designed to be decoupled from the pedal rod. There is neither a positive nor a non-positive connection of the two components. As a result, the actuator is not capable of generating a torque in the wrong direction, and therefore cannot accelerate. Consequently, the active module has no negative effect on the safety of the normal drive-by-wire throttle function. When activated, the actuator carries out highly dynamic movements. Due to the direct transmission of force without gearing, the signal can be clearly recognized by the driver's foot.

❷ shows an example of a possible force-path curve which can, for example, be used for an intelligent cruise control function in which the edges can be variably specified depending on the distance to the vehicle traveling ahead. As an alternative, use in hybrid vehicles is also possible. During purely electric operation, the pedal can be set to be soft, while a somewhat harder characteristic then signals the transition to operation with an internal combustion engine.

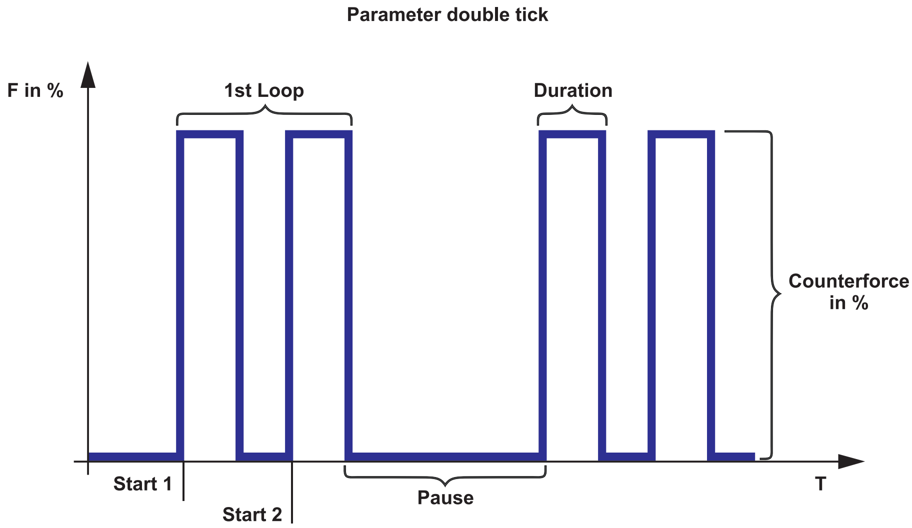
EXAMPLE OF USE FOR INCREASED SAFETY

An insufficient safety distance is one of the three most common causes of accidents (in addition to incorrect speed and distraction of the driver). An Adaptive Cruise Control (ACC) system can at first guide the driver to the proper safety distance using a characteristic curve as shown in ②. If the ACC detects a situation

❶ The generations of the Accelerator Force Feedback Pedal: Generations I to III (from left to right)



❷ The gearless direct drive provides for a dynamic, clear signal shape



❸ Double-tick cycle as indication of the optimal point in time for shifting

in which the driver approaches the vehicle driving ahead in a critical manner, the ACC can send a signal to the AFFP ECU. The actuator then, for example, generates a vibration which points out the danger to the driver directly on the control element relevant in the situation.

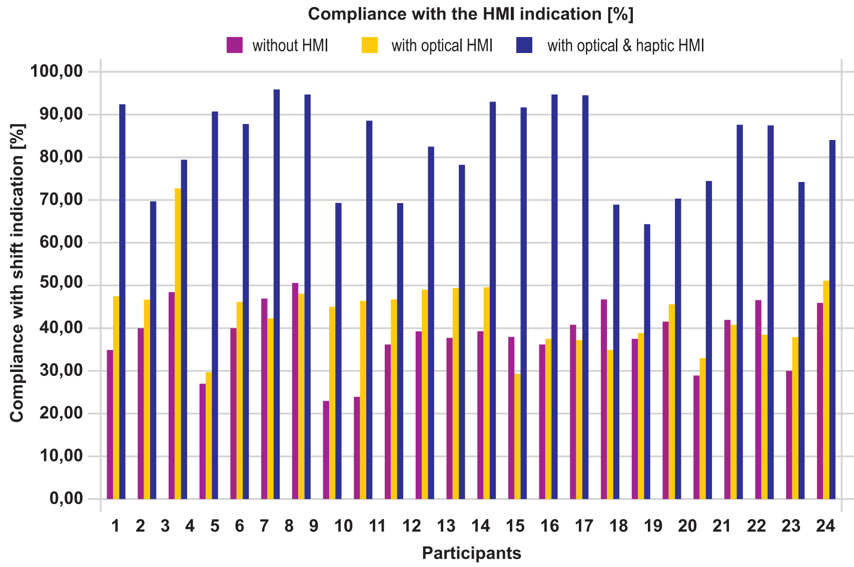
FUEL CONSUMPTION

An early consideration during the AFFP development was to indicate the ideal point in time for shifting to the driver so that the engine is operated as often as possible in the optimal speed range. To examine the acceptance and effect of this kind of system, Continental conducted a test subject study in cooperation with Continental Engineering Services and the Technical University in Munich, Germany in 2009. For this purpose, a test vehicle was constructed based on a BMW 530i (E60). The installed AFFP has a passive characteristic curve equivalent to the original pedals. A dSPACE MicroAutobox (MABX) served as the central control unit for the test. This control unit reads the original signals of the engine control unit, which include the control of the optical gearshift indication, and controls the haptic indication in the AFFP via CAN.

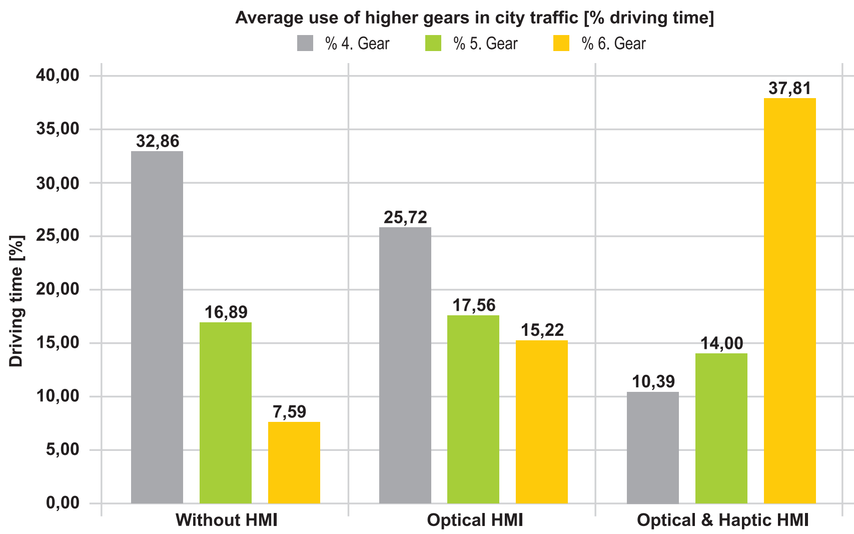
For example, a double tick, as shown in ❸, signals the point in time at which the next-highest or the next-lowest gear should be selected to the driver. If the driver fails to react, then the cycle is repeated twice after a short break each time, then it is canceled. The 24 test subjects of the study drove the same route around the Munich Ring three times – once without the shift-point indication or consumption information on the instrument cluster (IC), once with only the original visual shift-point indication in the IC and once with the visual shift-point indication and haptic indication with AFFP. As a result, 72 data records were available for the evaluation. Approximately 50 measured values with a scanning rate of 10 ms were recorded per measurement over the entire route. Variable additional loads, such as the AC compressor and lighting, were also recorded, however proved to be statistically insignificant.

It became apparent that little attention was paid to the visual shift-point indication, while the haptic indication triggered considerably more shifting within the defined reaction time of 10 seconds. All 24 test subjects followed the visual and haptic indication to a considerably higher degree than the purely visual one, ❹. Compared to the purely visual shifting indication, the mean speed before upshifting dropped on average by 450 rpm, and even by more than 500 rpm compared to driving without a shifting indication. With regard to the shifting behavior, it was possible to exactly reverse the lack of utilization of the upper gears, ❺. When averaged for all test subjects, the consumption dropped from 8.8 liters to 8.1 liters (improved from 26.7 mpg to 29 mpg) and resulted in savings of 7.7 % for the combination of a visual and a haptic shift-point indication. The acceptance of the haptic pedal also covered in this study was very good among the test subjects.

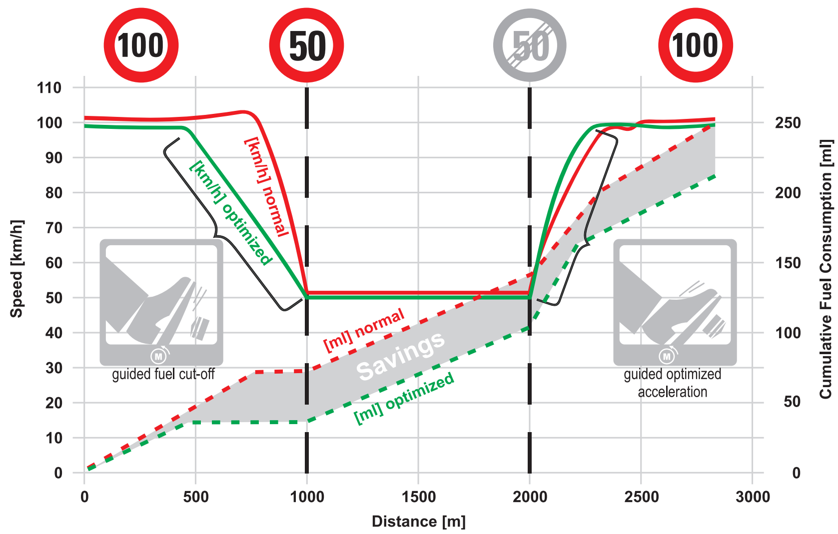
Further potential for energy-efficient driving is also offered by the interaction between the vehicle and the environmental sensors in order to guide the driver to overrun fuel cutoff as often as possible. To do this, information on the flow of traffic or the routing of the road must be available. Sensor technologies like radar, GPS-based navigation systems with their map material or future technologies like Car-2-X



❹ Effect of the type of indication on the shifting behavior of all test subjects



❺ The haptic signal leads to a considerably greater use of the upper gears



❻ Guided overrun fuel cutoff and guided acceleration reduce fuel consumption

communication can be used for this purpose. The energy efficiency of this kind of „managed deceleration" has been shown in a study with the Institute for Product Development of the University of Karlsruhe (IPEK). In the study conducted in 2008/2009, example driving maneuvers on level routes were analyzed with regard to their relevance for fuel consumption, then simulated and the simulation results were verified in the vehicle on a roller dynamometer. ❻ shows typical results of the IPEK study: With a temporary speed limit (driving through a town) the fuel consumption can be noticeably reduced by the AFFP signal. Here 80 % of the reduction results from the guided overrun fuel cutoff, while the remaining 20 % can be achieved through guided, ideal acceleration.

POSSIBLE USES: TODAY AND TOMORROW

Basically, the use of the AFFP can be very freely structured. The signal forms and intensities of the pedal feedback can be selected within broad limits. As a result, the pedal is an adaptable interface with which an OEM has a great deal of freedom with regard to both the supported function and the type of feedback required for it. This manufacturer-specific implementation can be carried out with various focuses. The target directions can be increased safety, more comfort (reduced load on the driver) and decreased fuel consumption. Due to its adaptability, it is even possible with the AFFP to store the driver's individual preference for a type of signal on the memory chip of the key.

If the AFFP is integrated in an electronic horizon with GPS and navigation data, then drivers can react suitably early on to traffic situations which they may not even be able to visually perceive yet – for example, a red light behind a curve. Conversely, drivers can be subtly guided in the longitudinal dynamics so that they use the green light phase in city traffic. Safety and optimized fuel consumption merge even further within Car-2-X strategies. Vehicles can inform each other on their approach to intersections so that, for example, privileged vehicles and practical coasting are pointed out to the drivers.

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ATZ PEER REVIEW
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

IMPRINT

ATZ WORLDWIDE

www.ATZonline.com

Founded 1898 as „Der Motorwagen"

Organ of the VDI-Gesellschaft Fahrzeug- und Verkehrstechnik (FVT)

Organ of the Forschungsvereinigung Automobiltechnik e. V. (FAT) and of the Normenausschuss Kraftfahrzeuge (FAKRA) in the DIN Deutsches Institut für Normung e. V.

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04 | 2010 _ April 2010 _ Volume 112

Springer Automotive Media | Springer Fachmedien Wiesbaden GmbH

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AD PRICES
Price List No. 53 (10/2009)

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SUBSCRIPTION CONDITIONS
The eMagazine appears 11 times a year at an annual subscription rate of 269 €. Special rate for students on proof of status in the form of current registration certificate 124 €. Special rate for VDI/ÖVK/VKS members on proof of status in the form of current member certificate 208 €. Special rate for studying VDI members on proof of status in the form of current registration and member certificate 89 €. The subscription can be cancelled in written form at any time with effect from the next available issue.

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INFLUENCING VEHICLE DYNAMICS BY MEANS OF CONTROLLED AIR SPRING DAMPERS

To achieve improved comfort as well as good handling characteristics when tuning the chassis suspension, the Technische Universität Darmstadt and the Vibracoustic GmbH & Co. KG have researched the use of controlled air spring dampers. Virtual road trials show that the course control by means of adaptive air spring dampers allows a decisive improvement in self-steering behaviour towards more agile handling and simultaneously ensures the stability.

THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN ATZ.
ATZ
PEER REVIEW
RECEIVED 2009-10-30
REVIEWED 2009-11-02
ACCEPTED 2010-01-18
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

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1 INTRODUCTION

The conflict between handling and comfort needs constant re-solving when tuning the chassis suspension. One possible solution is presented by the use of an adaptive pneumatic strut which combines energy storage (spring) and dissipation (damper). An air spring damper (LFD) makes a ratio of stiffness by factor 26 possible. Vehicle dynamics can be positively influenced by directly and proportionately supporting the roll momentum at front and rear axle. Technische Universität (TU) Darmstadt and Vibracoustic are currently developing control strategies for activation of LFD in order to achieve an optimum level of adjustment between vehicle dynamics and comfort.

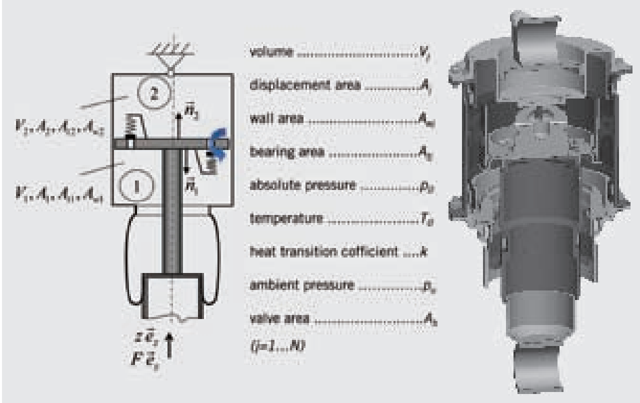
2 AIR SPRING DAMPER

In previous years various air spring damper prototypes have been developed for use in passenger cars and commercial vehicles. The goal of these developments was to improve the vehicle comfort while maintaining at least the same handling properties as in series vehicles with conventional damping systems.

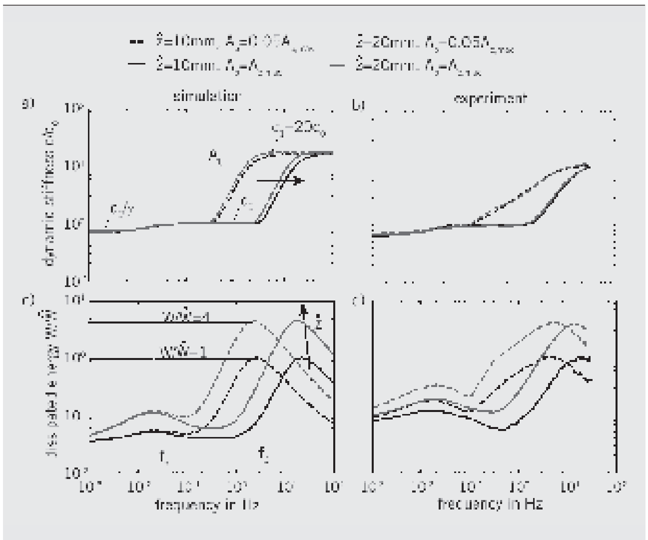
Every design of a LFD consists of at least two volumes of air that are separated by a piston, ❶ left. During the compression stroke the total volume is reduced, so that the enclosed air is compressed and represents a spring. In addition, a difference in pressure arises between the upper and lower volume which leads to a throttle flow through the valve – in this way energy storage and dissipation effects are linked. An improvement in comfort can be achieved by means of the LFD design: the sealing of the piston is carried out by air spring sleeves, as they are known from air suspension systems, ① right. Thus Coulomb friction of sliding seals which occurs with conventional hydraulic dampers is markedly reduced. The result is an improvement in continuous-travel comfort. For small excitations this advantage is reduced by an increasing harshness. For LFD an air supply and more available space than in conventional suspension is needed.

The dynamic behaviour of an LFD is described by means of a physical model with two ordinary differential equations for every volume, described in [2], [8]. The influence of excitation amplitude and frequency on the dynamic behaviour is shown in ❷. By means of computation and experiment the characteristics of stiffness and dissipated energy with respect to frequency are determined.

In ② a) and b) three levels of stiffness become apparent which are separated by the cut-off frequencies f_γ and f_0. The frequency f_γ describes the transition between isothermal and isentropic state. Typically f_γ ranges between 0.01 Hz and 0.1 Hz. By dimensional analysis in [7] it is shown that f_γ is proportional to



❶ Scheme (left) and design (right) of a two-chamber LFD (N=2)



❷ a) to d) Stiffness and dissipated energy of a two-chamber LFD for two excitation amplitudes und two valve areas (left: computation, right: experiment)

EQ. 1
$$f_\gamma \sim \frac{\lambda}{c_p \rho_0} \frac{1}{L^2}$$

where λ is the heat conductivity, c_p is the heat capacity at constant pressure and ρ_0 the inertial density of air as well as the typical length $L := V/A$ of the LFD which expresses the ratio of total volume and surface. The tuning frequency f_0 ranges between the stiffness levels c_0 and c_1. At this frequency the dissipated energy of the LFD reaches a maximum which is proportional to

EQ. 2
$$\hat{W} \sim p_0 V \hat{z}^2 / L^2$$

[2], with the inertial pressure p_0, the total volume V and the excitation amplitude $\hat{z}$. The quadratic dependence on the dissipated energy is also shown in ② c) and d). The next sections will show that the stiffness ratio between c_1/c_0 depends only on the geometry.

In the frequency range $f_\gamma < f < f_0$ the adiabatic stiffness c_0 appears. In this case the air mass flow between the chambers can be neglected. The result is a soft air spring with a volume that is represented by the sum of the chamber volumes. For small excitation amplitudes $\hat{z}$ the stiffness c_0 is given:

EQ. 3	$c_0 := \frac{dF}{dz}\Big\vert_{z=0} = \gamma p_0 \frac{AA_T}{V} + (p_0 - p_u)\frac{dA_T}{dz}\Big\vert_{z=0}, \; for \; f_\gamma < f < f_0.$

Thereby, $\gamma = 1.4$ is the adiabatic exponent, p_0 is the absolute pressure in initial position and p_u is the ambient pressure. The total volume V, the bearing surface A_T and the displacement surface A are defined as follows:

EQ. 4	$V := \sum_{j=1}^{N} V_j,$ $A_T := F_0/(p_0 - p_u) = \sum_{j=1}^{N} A_{Tj}\vec{n}_j \cdot \vec{e}_z \; and$ $A := \sum_{j=1}^{N} A_j = \sum_{j=1}^{N} \frac{dV_j}{dz}.$

N describes the number of chambers and F_0 the LFD force in initial position. With a cylindrical piston contour $dA_T/dz = 0$ the stiffness of Eq. 3 becomes

EQ. 5	$c_0 = \gamma p_0 \frac{AA_T}{V}.$

At high frequencies $f >> f_0$ the limit in stiffness is c_1 which corresponds to the parallel connection of two air springs. For small excitations and a cylindrical piston contour the upper stiffness c_1 results as:

EQ. 6	$c_1 = \gamma p_0 \sum_{j=1}^{N} \frac{A_j A_{Tj}}{V_j}, \; for \; f >> f_0.$

In this equation V_j designates the volumes, A_{Tj} the bearing areas, and A_j the displacement areas of the particular volumes in initial position. Air spring dampers achieve a ratio of upper to lower stiffness c_1/c_0 of up to 26 for a three-chamber LFD and up to 20 for a two-chamber LFD [7]. The stiffness ratio depends only on the geometry, for example the stiffness ratio for a two-chamber LFD is:

EQ. 7	$\frac{c_1}{c_0} \approx \frac{V}{A^2}\left(\frac{A_1^2}{V_1} + \frac{A_2^2}{V_2}\right) = 4 \ldots 20.$

For dimensioning a LFD two requirements should be considered. First, the LFD has to carry the static load of the vehicle:

EQ. 8	$mg \sim (p_0 - p_u)A_T \approx p_0 A_T, \; for \; p_0 >> p_u.$

In this case, m represents the quarter mass of the vehicle and g the constant of gravity. The system pressure is given by the maximum pressure p_0 of the air supply or the burst pressure of the component, hence the bearing area A_T is determined. Secondly, the vertical vehicle eigenfrequency $f = 0.8 \ldots 1.2$ Hz – depending on the vehicle manufacturer – is given. With Eq. 5, 8 and $A \approx A_T$ the eigenfrequency

EQ. 9	$f^2 \sim \frac{c_0}{m} = \gamma \frac{g}{L}$

depends only on the typical length L of the LFD [7]. Surprisingly, the pressure has no influence on the dynamic behaviour. In analogy to the mechanic pendular the eigenfrequency is defined by the typical length, thus the ratio of volume to area is fixed. The result is a narrow parameter range for dimensioning a LFD.

If the valve in the LFD is designed as a continuous adaptive valve the transition from lower to the upper level of stiffness c_0 and c_1 as well as the corresponding maximum dissipated energy can be shifted within the frequency range. By means of dimensional analysis it is shown in [2] that the tuning frequency f_0 for maximum energy dissipation is proportional to the valve area A_b:

GL. 10	$f_0 \sim \sqrt{RT_0}\,\frac{A_b}{V}\,fn\left(\gamma, \frac{V}{\hat{z}A}\right).$

R represents the ideal gas constant and T_0 the mean absolute temperature of the component. By Eq. 10 it is possible to adapt the tuning frequency f_0 on the excitation frequency by tuning the valve, ② a) and b). Hence, it is possible to adapt the stiffness between the lower and upper level which will be used for vehicle dynamics in this article. There is only a small influence of the temperature on the typical frequency. Increasing the temperature from -10° C to 23° C increases the typical frequency for 6 percent on a higher frequency [8].

Considering these properties we have been persuaded that the adapting of dynamic stiffness of the LFD and consequently the adapting of roll stiffness could improve the vehicle dynamics. The potential will be demonstrated in the following passages by means of simulations with a five-body vehicle model.

3 USED FIVE-BODY VEHICLE MODEL

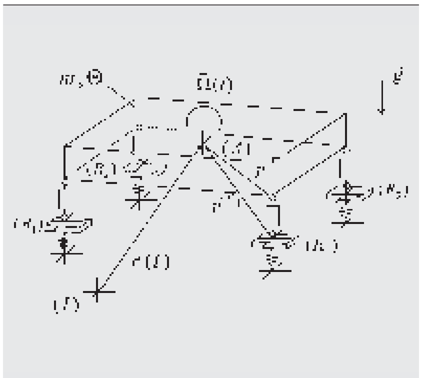
The vehicle model presented here was developed at the Chair of Fluid Systems Technology at TU Darmstadt in order to accomplish virtual road trials for an evaluation of model based and efficient LFD controllers. The use of a self-developed vehicle model has the advantage that the source code is open and expandable for new components.

The five-body vehicle model in ❸ bases on the dynamic equations for the rigid bodies, that is the wheels and the vehicle-body. Each body is characterised by its mass m and inertia tensor Θ. In Eq. 11 the conservation of momentum for the vehicle body in the inertial frame (I) and in Eq. 12 the conservation of angular momentum in the vehicle-fixed frame (A) are presented in index notation.

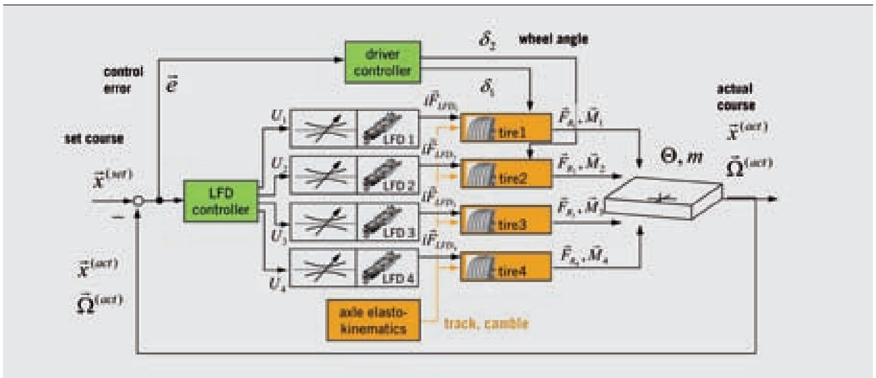
EQ. 11	$m\,\frac{d^2 r_j^{(I)}}{dt^2} = \sum K_j^{(I)}$
EQ. 12	$\dot{\Omega}_k^{(A)}\theta_{lk}^{(A)} + \varepsilon_{ljk}\Omega_j^{(A)}\Omega_p^{(A)}\theta_{kp}^{(A)} = \sum M_l^{(A)}$

The translational and rotational degrees of freedom of the vehicle body are described by the vectors $\vec{r}$ and $\vec{\Omega}$. The terms on the right hand side in Eq. 11 and 12 denote the sum of forces and momenta that affect the vehicle body.

The physical LFD model mentioned earlier was used for modelling the struts. Each wheel has three rotational and one translational degree of freedom in vertical direction. The axial-elasto-kinematics for camber and toe angles was determined by a multi-body simulation (MBS). The quasi-static results are stored by means of characteristic diagrams. In accordance with standard practice, jounce travel curves were generated while being exposed simultaneously to tire lateral forces. Further kinematic values such as position of the axis of roll were also determined using the MBS



❸ Physical five-body vehicle model (axle not depicted)



❹ Influencing the vehicle dynamics by means of adaptive LFD

model. The non-linear tire behaviour was modeled by Burckhardt's tire model which is described in [3]. In order to achieve this three-dimensional characteristic diagrams for the tires are implemented in the model which calculates the resulting coefficient of friction via wheel load and wheel slip. The product of coefficient of friction and wheel load gives the resulting horizontal tire force. Required data for the parameterization of the model such as masses, geometric data and bushing stiffness were made available by a vehicle manufacturer.

4 INFLUENCING VEHICLE DYNAMICS BY CONTROLLED AIR SPRING DAMPERS

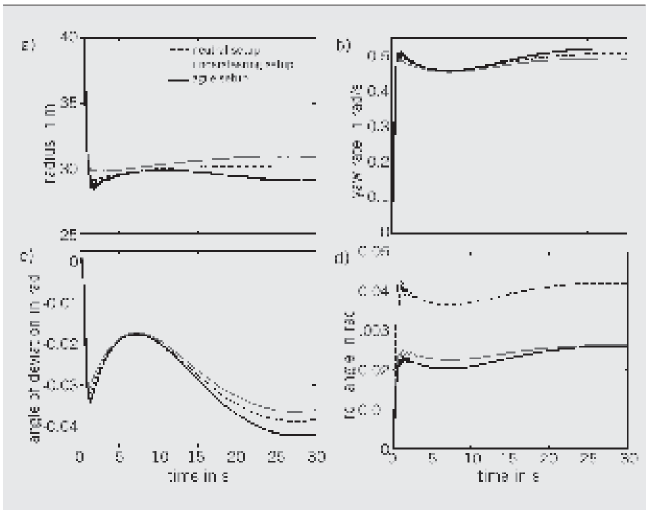
With the aid of the available stiffness ratio of the LFD it is possible to adapt the roll stiffness of the front and rear axle. Assuming that the additional elasticity of the axle can be neglected, the resulting roll stiffness of the struts is

EQ. 13
$$c_{LFD} \approx \frac{i^2}{2} c s^2,$$

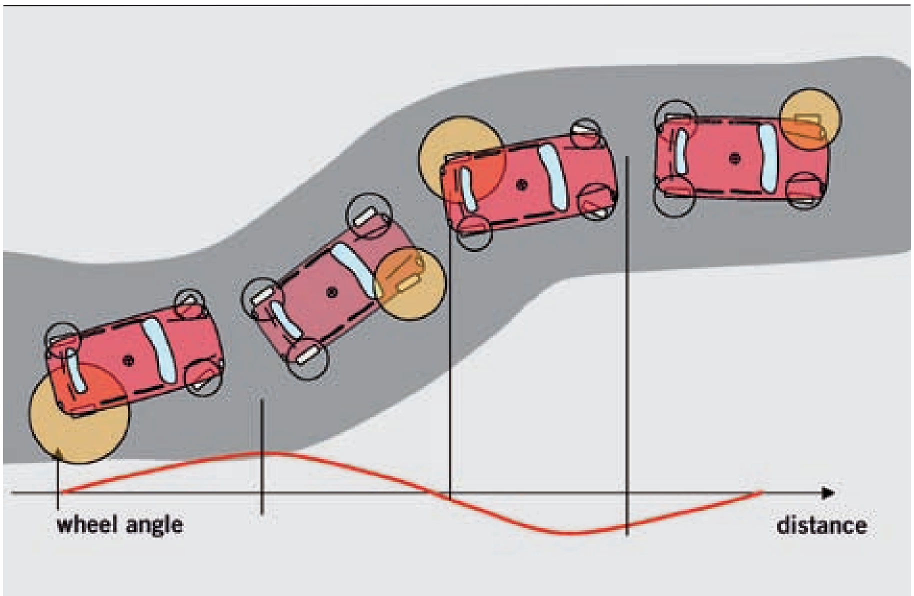
with the axle ratio i, the strut stiffness c and the wheel track s. If the ratio of maximum to minimum roll stiffness is established a ratio of 26 is also possible. Based on ❹ the influence of adaptive roll stiffness on the vehicle dynamics will be shown. In addition to constructive arrangements, adapting the roll stiffness is one possibility to influence the vehicle dynamics.

Two closed loops are depicted here. A driver controller brings about an alignment between the set course and the actual course by means of adjusting the steering angle. The LFD supports this process by intervening the vehicle dynamics by means of controlling the valves. By means of proportionate support of the roll momentum between the front and rear axle the handling properties can be switched between agile and understeering. It is made use of the degressive dependency of lateral tire force and wheel load as described in [6]. Thus the increase in the lateral force on the outer wheel with high wheel load is less than the decrease on the inner wheel with low wheel load. In total the axle with greater wheel load difference experiences a loss in cornering grip and a corresponding yaw moment arises. This effect is noticeable from lateral accelerations of 0.4g upwards. For showing the potential of adapting the LFD a steering angle step was investigated. This step is represented by a ramp with a rising time of 0.5 s and a height of 0.1 rad at a vehicle velocity of 15 m/s.

The results in ❺ show a delay time of 1 s after the manoeuvre starts. Here the simulated results are identical regardless of the chosen LFD setup. This delay time is typical for an adaptive system which is caused by the inertia of the vehicle. Thus an additional yaw moment can become apparent when a corresponding roll angle, ⑤ d), and a necessary wheel load difference are available. All graphs in ⑤ show a characteristic maximum at the time of 1 s and an asymptote for large times. The maximum is caused by a rising stiffness of the LFD which is a result of the compression and the increased temperature. The asymptote is caused by a relaxation which affects a decreasing LFD stiffness. For unsteady maneuvers the results up to 5 s are relevant. Dependent on the LFD setup ⑤ a) shows a spreading in curve radius of up to 2 m. The yaw rate in ⑤ b) could be increased for 5 % which means a higher agility. Adapting the LFD valves can also decrease the angle of deviation for 10 %, ⑤ c). Finally, ⑤ d) shows that a rising roll stiffness at front or rear axle leads to a significantly reduction of the roll angle (> 40 %).



❺ Results of a steering angle step with a) radius, b) yaw rate, c) angle of deviation and d) roll angle bases on neutral, understeering and agile LFD setup; lateral acceleration at steady state approximately 7.5 m/s²



❻ Schematic diagram of wheel load distribution influenced by a LFD controller while lane-changing

For demonstrating the control of the LFD in ❻ a vehicle is depicted in process of carrying out a lane-changing manoeuvre. The intervention in vehicle dynamics by means of dividing the roll moment between front and rear axle is likewise depicted. For this purpose the wheel loads are represented in form of circles, where the wheel load is proportionate to the radius of the circle. The decision at which time the adaptive valves are activated is made by a LFD controller.

Basically the controller adjusts the valves that way to achieve an agile handling. The related yaw moment is achieved by means of a high level of roll stiffness and a greater difference in wheel load on the rear axle respectively. In case of exceeding stability criteria, a correction to understeering takes place. For this purpose the roll stiffness of the front axle is increased. To identify a critical situation, the controller compares the actual yaw rate with a critical yaw rate. The critical yaw rate is determined by the lateral acceleration a_q

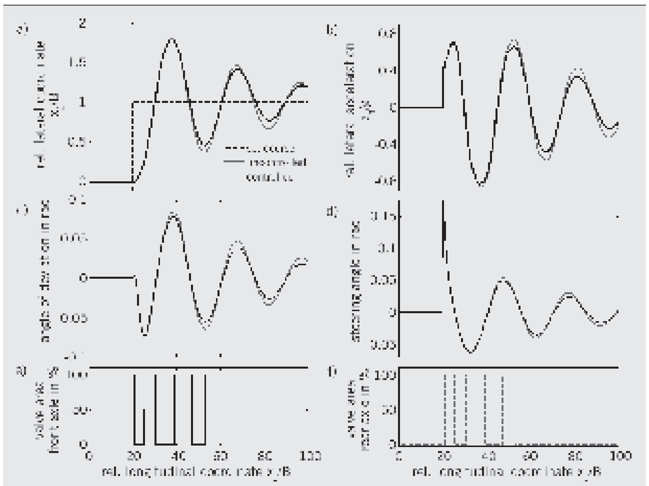
EQ. 14	$a_q = \frac{\upsilon^2}{R}$

and the curve radius R

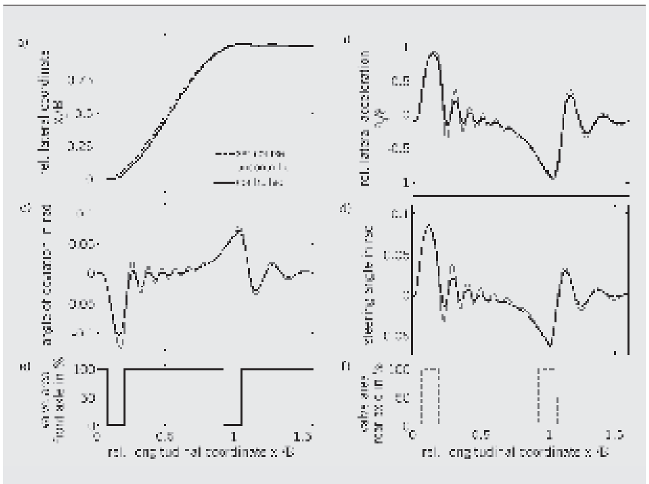
EQ. 15	$R = \frac{\upsilon}{(\dot{\alpha} + \dot{\beta})}$

with the vehicle velocity υ, the yaw rate $\dot{\alpha}$ and the angular rate of deviation $\dot{\beta}$. Insert Eq. 15 in 14 and replace a_q with $k\mu_{max}g$ the critical yaw rate becomes

EQ. 16	$\dot{\alpha}_{krit} = \frac{k\mu_{max}g}{\upsilon}.$



❼ a) to f) Single lane change with vehicle trajectory, lateral acceleration, angle of deviation, steering angle and valve positions of the LFD (width of lane change B = 2 m, velocity 24 m/s)



❽ a) to f) Half cycle of a cosine (set course) with vehicle trajectory, lateral acceleration, angle of deviation, steering angle and valve positions of the LFD (width B = 250 m, velocity 22 m/s)

Here $k = 0 \dots 1$ denotes the sensitivity of the controller and μ_{max} the maximum value of friction between tire and road. Furthermore, it was assumed that the angular rate of deviation can be neglected, whereas $\dot{\beta} << a_q/\upsilon$ is valid.

5 EVALUATE THE LFD CONTROLLER BY MEANS OF VIRTUAL ROAD TRIALS

In order to test the LFD controller virtual road trials were undertaken on a lane-changing manoeuvre which trajectory based on step function. The aim of control is defined as the minimal offset to the set course. The driver controller keeps the vehicle velocity constant during the manoeuvre.

In ❼ a) the vehicle's trajectories with and without LFD control and the set course are depicted. At the beginning there are identical trajectories of controlled and uncontrolled vehicle. For the controlled vehicle, the offset from the set course is seen to be less after the first half of the manoeuvre. In total, the controlled vehicle reaches a 15 % smaller integral offset than the uncontrolled vehicle, that is 96 m² versus 113 m². Furthermore, the LFD control leads to a decreasing lateral acceleration of 1.5 m/s², ⑦ b). Additionally, the magnitudes of the angle of deviation are significantly degreasing which leads to an improved vehicle handling, ⑦ c). The controlled vehicle shows in ⑦ d) a lesser demand for steering angle. Finally, the behaviour of the front and rear axle LFD valves are shown in ⑦ e) and f). Before the manoeuvre starts the controller switches to agile in order to affect an agile handling. At the manoeuvre start the driver reacts by a fast rising steering angle up to approximately 0.17 rad at the wheel, ⑦ d). The vehicle responds with a rising yaw rate and exceeds the critical yaw rate which is calculated by the controller. The controller reacts by switching the LFD valves to understeering.

On principle the single lane change manoeuvre exhibits a small durability. In the following the LFD controller will be evaluated at a manoeuvre with larger durability. Therefore a half cycle of a cosine is used as set course which dimensions orientate on a real road.

Both the controlled vehicle as well as the uncontrolled vehicle are able to pass the manoeuvre with small differences to the set course, ❽ a). However, the controlled vehicle shows advan-tages concerning stability and agile handling. This positive behaviour can be shown in the more damped oscillations of lateral acceleration and angle of deviation, ⑧ b) and c). A significantly improved handling can be established by the decreased demand for steering angle in ⑧ d).

6 SUMMARY AND OUTLOOK

The previous sections have shown a potential to influence the vehicle dynamics by means of controlled LFD. In summary, the course control by means of adaptive LFD offers a very good opportunity to make a decisive improvement in self-steering behaviour towards more agile handling and simultaneously ensures the stability.

In further investigations the influence of the LFD control on comfort and safety has to be analysed. It is expected that increasing the roll stiffness will cause a loss in comfort and damping. Therefore, the LFD control will be tested in combination with real road profiles. Furthermore, the interaction of the LFD controller with electronic stability control (for example ESP) will have to be investigated.

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ELECTRICITY FROM EXHAUSTS – DEVELOPMENT OF THERMOELECTRIC GENERATORS FOR USE IN VEHICLES

In the course of increasing demands to reduce fuel consumption and CO_2 emissions of vehicles with combustion engines, we will have to make use of hitherto unused energy flows. It is well known that as much as a third of input energy is converted to exhaust heat, so it is clear that work needs to be done to increase the utilisation of this energy. Through the application of thermoelectric generators (TEG), in the future a proportion of the exergy carried by the exhaust gases that would previously have been lost will be recovered as electrical energy. The German Aerospace Center (DLR) has developed TEG for use in vehicles for doing this, and, resulting from collaboration with the BMW Group, 200 W of electrical power in a test vehicle was achieved in a demonstration. Given the use of future materials, performances of up to 600 W can be expected, yielding a usage potential of 5 %. The current research focus is aimed at improving the manufacturing technique with a view to future standard production applications.

THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN ATZ.
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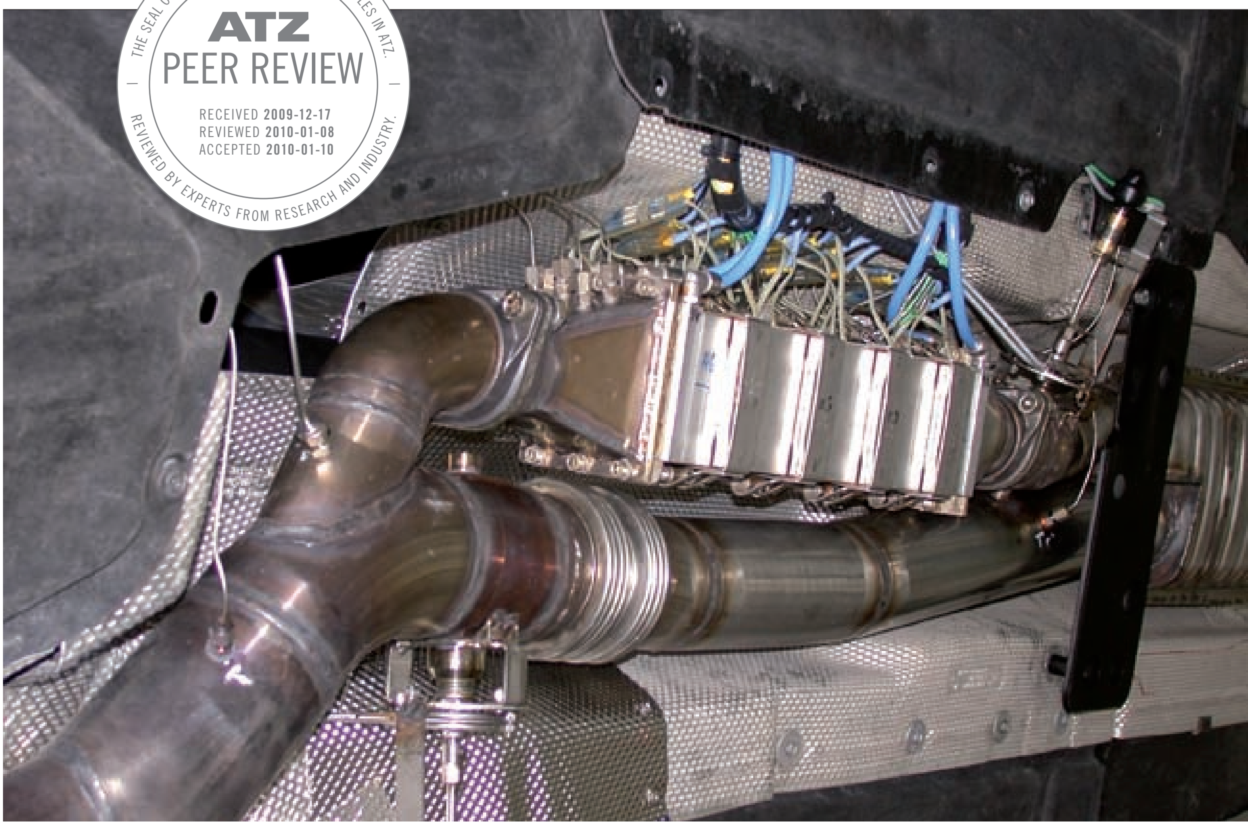
PEER REVIEW
RECEIVED 2009-12-17
REVIEWED 2010-01-08
ACCEPTED 2010-01-10
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.



1 INTRODUCTION

The ongoing challenge of reducing CO_2 emissions has forced the automotive industry to search for novel technical solutions. Thermoelectric energy conversion represents an interesting possibility in this regard [1]. This article describes the development of thermoelectric generators (TEG) for use in vehicles and how these are being integrated into the vehicle architecture.

Several institutes at the German Aerospace Center (DLR) are conducting research into thermoelectric energy conversion technologies. The Institute of Vehicle Concepts in Stuttgart is both coordinating and participating in the Secondary Energy Use in Vehicles project. This institute is responsible for defining requirements from the vehicle perspective as well as developing prototypes and integrating these into the vehicle architecture. The Institute of Materials Research in Cologne is contributing by analysing, developing and characterising thermoelectric materials and modules, while the Institute of Technical Thermodynamics is investigating the thermomechanical stresses experienced by the systems. Working in collaboration with the BMW Group and the Institute of Materials Research, the Institute of Vehicle Concepts has successfully integrated a prototype thermoelectric generator into a vehicle of the type BMW 535i US [2].

Some practical applications of thermal electricity are characterised in part by an ability to convert thermal energy into electrical energy without the need for any mechanical moving parts. Use in space has proven this type of technology to be particularly long-lived, making it an ideal candidate to examine for potential use in vehicles.

Following analysis of the requirements for in-vehicle applications, several generations of TEG prototypes have been developed, constructed and tested over the last few years. The aim of this research is to establish the effects of thermoelectric energy conversion on the whole vehicle. The calculational potential for reducing fuel consumption is in the order of a few percent [2].

The DLR in Cologne has test benches for determining thermoelectric properties, which are complemented by a hot gas test bench and climatic chassis dynamometer in Stuttgart. Research vehicles are also available that can be equipped with various components and used to carry out measurements.

2 FUNDAMENTALS

2.1 THERMOELECTRIC ENERGY CONVERSION

The conversion of thermal energy to electrical energy is based on an effect discovered by Thomas Seebeck in 1821 [3]. This states that an electric voltage will be created when a difference in temperature occurs on either side of a pair of conductors made from different materials $\Delta T = T_h - T_c$. Conversely, the conductor pair can also serve as a heat pump if electrical current is passed through it (Peltier effect).

The physical fundamentals and mathematical equations underlying these effects have already been described in detail in an article in MTZ [2]. Here it is only necessary to revisit some of the most important relationships for use in practice. The power generated thermoelectrically is defined as:

EQ. 1

$$P_{el} = \eta_{TE} \cdot \dot{Q}$$

Here, $\dot{Q}$ represents the heat flowing into the heated side, η_{TE} the efficiency of the thermoelectric material and $T_h - T_c$ the difference in temperature. For the maximum performance $P_{el\ max}$, the efficiency can be expressed as [4]:

EQ. 2

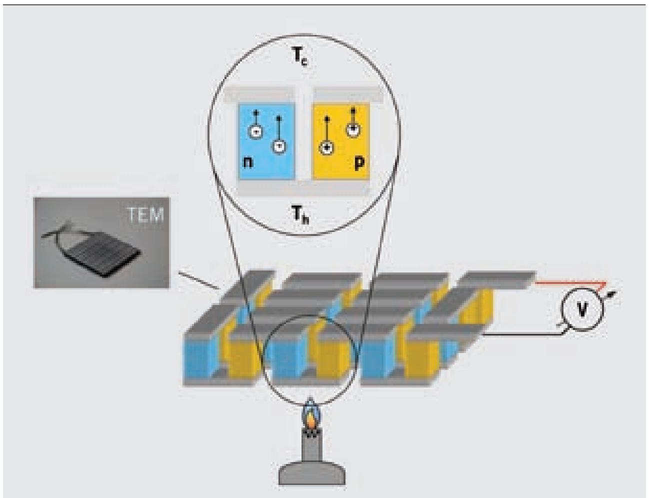
$$\eta_{TE}\,(@P_{el\ max}) = \underbrace{\frac{T_h - T_c}{T_h}}_{\eta_c} \cdot \frac{1}{\frac{4}{ZT} + 2 - \frac{1}{2} \cdot \frac{T_h - T_c}{T_h}}$$

ZT is a dimensionless figure of merit, which is included to characterise thermoelectric materials according to their properties. It is clear from these equations that electrical power increases with increasing heat flow, higher *ZT* values and increasing temperature difference.

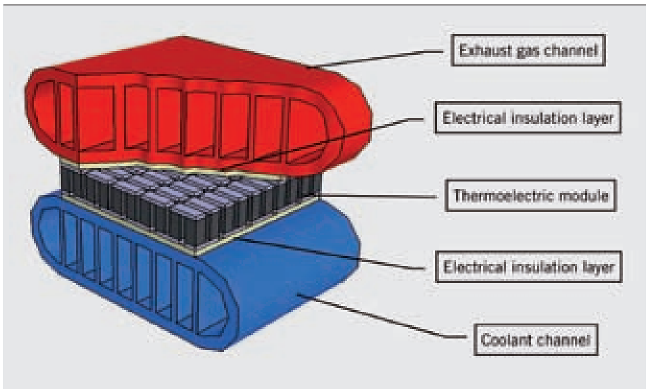
The *ZT* values' progress is showing a great deal of promise. Whereas the materials that have been used traditionally, such as bismuth telluride (Bi_2Te_3) and lead telluride (PbTe), achieve *ZT* values of around 1, with newer, more cost-effective materials such as silicides, half-Heusler materials and, most importantly, skutterudite, *ZT* values of 1.3 are now realistic. From a technical perspective, the thermoelectric materials are formed into flaky thermoelectric modules (TEM), which contain a large number of n- and p-doped thermocouples connected alternately in series, ❶.

2.2 VEHICLE BOUNDARY CONDITIONS

As explained above, the performance of a TEG depends to a large extent on the temperature difference and the flow of heat into the



❶ Principle illustration of a thermoelectric module (TEM) and its doped semiconductor sections [5]



❷ Functional layer of a thermoelectric generator with heat exchangers for exhaust gas and coolant, thermoelectric module (TEM) and electrical insulation

device, which means that installation locations with a high exergy potential should be given preference. For this reason, research activities concentrating on automotive applications tend to focus on integration of TEGs into the exhaust system. However, the vehicle boundary conditions that dominate here also place heavy demands on the TEG. In addition to experiencing high thermal loads, the components are exposed to corrosive effects due to aggressive exhaust gas condensate on the inside and salt solutions on the outside. Any development work must also consider the impacts, shock loads and oscillation that occur in a vehicle exhaust system, but that must not negatively affect the TEG.

3 EFFECTS ON THE ENTIRE VEHICLE SYSTEM

One major challenge lies in answering the question of what effects installing a TEG has on the vehicle as a whole. On the one hand, the vehicle weight will increase, on the other hand, the part of the heat flow which is not converted to electrical energy has to be released to the environment. The back pressure caused in the exhaust system must be minimised to avoid negatively affecting the gas exchange process and charge cycle work. Moreover, the process of heat extraction must not impair the operation of downstream exhaust gas treatment equipment. Additional components are also required to feed electrical power into the vehicle electrical system.

There are two main options for transferring thermal energy to the environment. The existing cooling system can absorb the additional heat flow and deliver this to the outside environment, provided that it is not working at maximum capacity. Problems should only be encountered in very rare situations, such as full engine load on a hot summer‘s day. In such cases, exhaust gas mass flows can be diverted around the TEG using a bypass mechanism. Increasing the surface area of the radiator would result in an increase in air resistance and should be avoided if possible.

The increased rate at which the cooling circuit is heated is an advantage. The heat introduced from the exhaust gases reduces the engine warm-up time, which in turn reduces engine friction and can improve comfort as a result of the passenger compartment heating up more quickly.

When selecting the installation position in the exhaust system, care must be taken to ensure that warming up of the exhaust gas treatment components is not impaired. However, various strategies can be an option. One option is to install the TEG behind those systems, but this has the disadvantage that there is much less exergy available at this point, meaning that the performance of the TEG is reduced. The alternative, with the TEG fitted in front of the exhaust gas treatment components, is to implement appropriate measures to ensure that the exhaust gas treatment system is able to start operating quickly. This could be achieved by using a bypass system. Beyond that, the TEMs can be run in “Peltier” mode to boost the heating of the TEG.

The construction of the heat exchanger makes high demands to equally dispensing the heat. The TEM must be exposed to the maximum permissible temperature. Where possible, none of them should be used at a differing operating point since this individual one affects the efficiency of the whole generator. A construction of the heat exchanger which can be described as ideal will thus not be valid for each mass flow rate. The electronic connection of individual module groups can be supportive if it compensates for uneven temperature distribution by individually adapting the module impedances. As done in the area of converter techniques in photovoltaics, DC/DC converters are to be applied here, which adapt the load resistance – corresponding to the proportion of current drawn from the vehicle electrical system – to the instantaneous internal electrical resistance of the modules.

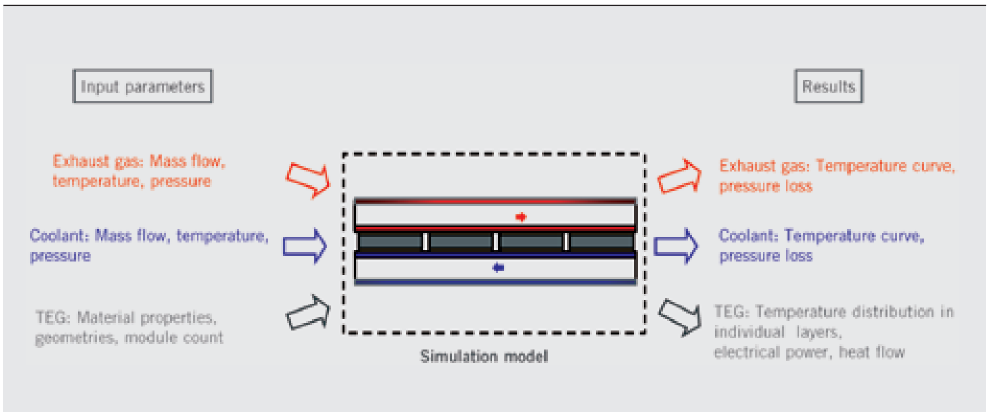
4 DEVELOPMENT OF THERMOELECTRIC GENERATORS FOR USE IN VEHICLES

4.1 TEG SIMULATION MODELS

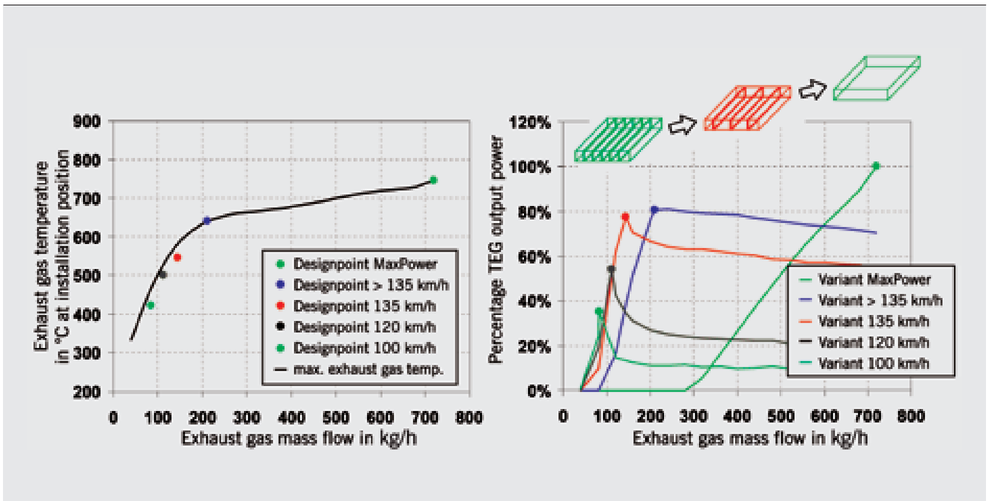
In order to allow the prediction of the expected electrical power and exhaust back pressure created, every development begins with the computation of parameters using thermo-technical simulations. DLR uses the EES software [6] package to create a discretised cell model of the TEG, which makes use of existing geometric symmetries and models a functional TEG layer, ❷. This is comprised of an exhaust gas heat exchanger in one half and a coolant heat exchanger in the other, separated by a TEM layer with corresponding insulation layers. The construction is divided into numerous individual cells and a system of equations is established for each cell to describe the energy balance, the material properties and heat transport mechanisms. The individual equation systems are combined so that each interacts with the neighbouring cells.

The empirical material data for the TEM determined by the Institute of Materials Research is also entered into the model. Thermodynamic boundary conditions for mass flow, temperature and pressure are then specified for the exhaust gas and coolant, allowing the complete system of equations to be iteratively solved. The outputs from this process are temperature curves, the heat flow transferred to the cooling system, and electrical output of the TEG. The back pressures caused by heat-exchanging structures in the fluid flows are also calculated using the corresponding correlations from the geometric relations and the Reynold’s number. ❸ shows a summarisation of the input parameters and result values of the TEG calculation model. If the TEG is to comprise multiple functional layers, the model is accordingly adapted by scaling.

When modelling the heat-exchanging fins there are two factors that have a decisive influence. Firstly, it is necessary to make sure that there is the maximum possible temperature difference across the TEM. With the Bi_2Te_3 modules available today, care needs to be



❸ Input parameters and result values of the TEG calculation model

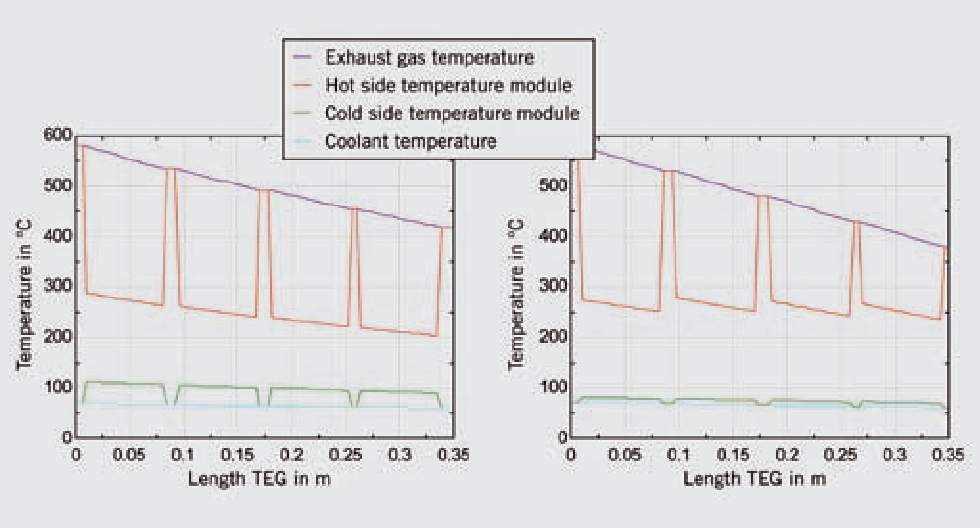


❹ Position of the design points in the exhaust gas temperature/exhaust gas mass flow map (left); simulated TEG power curves for TEG designs optimised for various load cases (design points) (right)

taken to ensure that the maximum temperature on the heated side of the module does not exceed 250 °C, as otherwise the module may be damaged. Secondly, the fins need to be designed in such a way as to ensure that the exhaust back pressure does not increase too dramatically. Because the fins are rigid and cannot be adapted to the highly-dynamic thermodynamic boundary conditions of the exhaust gas while the TEG is in use, their construction needs to be optimised for a specific driving situation (design point). When high loads occur, a proportion of the exhaust mass flow must be diverted around the TEG in order to protect the TEM and prevent the back pressure building up too greatly.

❹ shows (on the left hand side) the positions of a range of design points based on exhaust mass flow and exhaust temperature at a potential installation position in a reference vehicle. The brown curve represents the change in maximum exhaust temperature with increasing mass flow. On the right hand side, ④ shows the electrical power of the corresponding TEG design for the same range of exhaust mass flows. The power curves illustrate a typical peaked shape. In the lower load range the curves rise steeply, before reaching maximum power at the design point. When the design point is passed, the mass flow through the TEG is reduced by opening the bypass, resulting in a reduction in the temperature difference and the output power of the TEG. In the example, it can be seen that there is a very promising curve for the 135 km/h, six-gear option: it shows good response characteristics in the low load range, high absolute power and only a moderate drop-off beyond the design point.

The simulation model can be used to show that there is an enormous potential in the design optimization of thermoelectric generators. ❺ shows the simulated temperature curves of exhaust gas, heated side of the module, cooled side of the module and coolant in longitudinal direction of the TEG for a conventional (left) and an optimized design (right). In the conventional design, a fairly steady decrease of the temperature of the heated side of the module in the direction of flow can be seen. This is due to the increasing heat loss caused by the heat exchanger. The temperatures for the cold sides of the module are approximately 40 to 50 °C above the coolant's temperature. The calculated maximum power for this variant with design point is at 195 W, the exhaust back pressure produced is at 23 mbar. The heat-exchanging fins of the exhaust gas- and coolant heat exchanger was subsequently optimized in an optimization loop. In so doing, the exhaust gas heat exchanger was provided with an increasing fin density in the direction of flow in order to counteract the heat-sided temperature decrease. The temperature curve displayed on the right side of ⑤ shows that the temperature of the heated side of the module can be kept at a high level. From the coolant perspective, the fins were also improved which made it possible to significantly



❺ Temperature curves in TEG longitudinal direction for exhaust gas, heated side of the module, cold side of the module and coolant; conventional design (left) and design with optimised heat exchange with regard to exhaust gas and coolant (right)

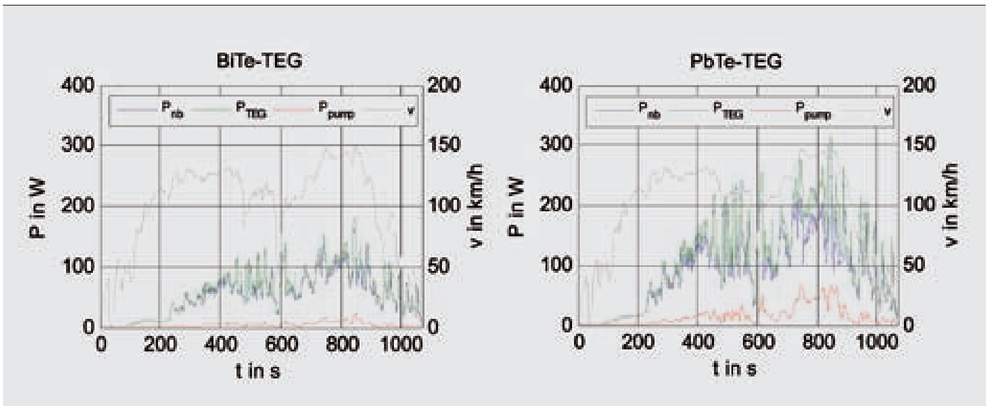
reduce the temperature of the cold side by 20 to 30 °C. When comparing the two temperature curves it can be clearly seen that the temperature difference connected to the module increases (difference between red and green curve) with the optimized design. This affects the calculated power which could be increased by 50 % to 300 W compared to the conventional design. Due to the fins' flow-optimized design, the exhaust back pressure only increases to 25 mbar.

In addition to the described stationary TEG model, there is a dynamic Modelica model [7] at the Institute of Vehicle Concepts which is able to simulate the transient behaviour of the TEG during the cycle. The left-hand graph in ❻ shows the calculated development of net useful power output for the 135 km/h design described above, using Bi_2Te_3 on the Artemis motorway cycle [8]. It s easy to identify the TEG behaviour during the warm-up phase; it should be noted that the underlying thermal masses have been derived from prototype geometries and that, at around 11 kg, the weight of the TEG is still considered relatively high. The maximum output achieved with this set-up is 180 W.

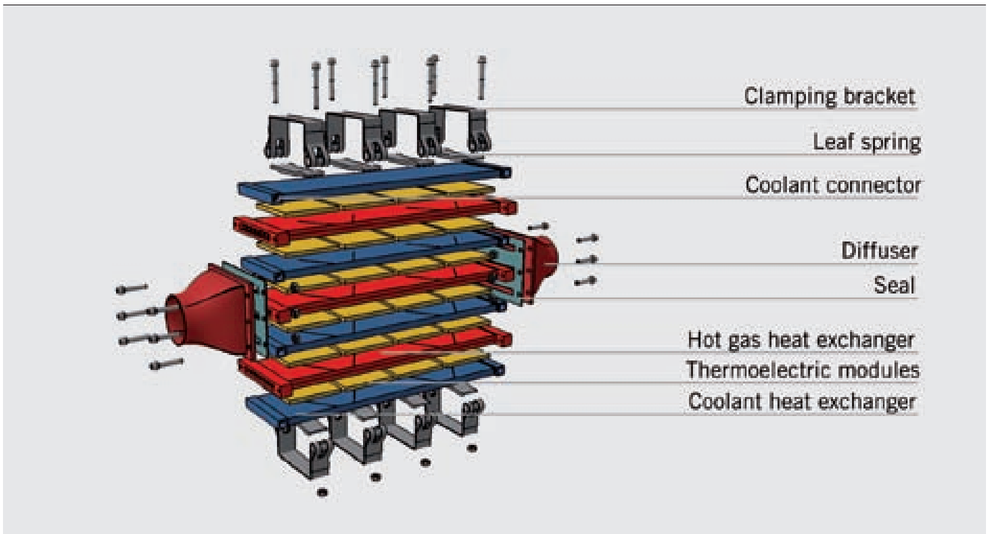
To highlight the potential offered by higher performance materials at mid-range temperatures, the right-hand graph in ⑥ shows the power curves for a TEG system with PbTe material data entered into the model. The structure of the heat exchanger was adapted to account for the fact that the lead telluride can operate at higher temperatures of up 450 °C. This allows the warm-up phase to be significantly shortened. A higher electrical power output of up to 300 W is also achieved during the cycle thanks to the higher heat flow transfer and the greater efficiency of PbTe. The influence on engine power of the exhaust back pressure caused by this system is currently being integrated into the model.

4.2 FURTHER DEVELOPMENT STEPS

The further development process takes the most promising variants to a more advanced stage of design. This involves finding compact, lightweight, production-oriented solutions for the various sub-functions of the TEG, taking into account the available packaging space in each case. Development at this stage also makes use of appropriate simulation programs. Finite element software is used to perform calculations to ensure the stability of the mechanical and thermo-mechanical components and to investigate the flow and temperature distribution within the TEG. In addition, the results are verified by investigating the sub-components experimentally, making use of the comprehensive range of test facilities available to the researchers. The end result of this design



❻ Calculated power curves in Artemis motorway cycle using various thermoelectric materials and heat exchanger designs adapted for those materials; the net power (blue) results from the TEG power (green) less the additionally generated power of the water pump (red) and the radiator fan (converted to engine power); the power curves illustrate the use of bismuth telluride (left) and lead telluride (right) as thermoelectric material [9]



❼ CAD exploded view of a TEG using stacked construction with 24 Bi_2Te_3 modules [2]

process is a CAD model of the TEG as shown in ❼ and the corresponding production drawings.

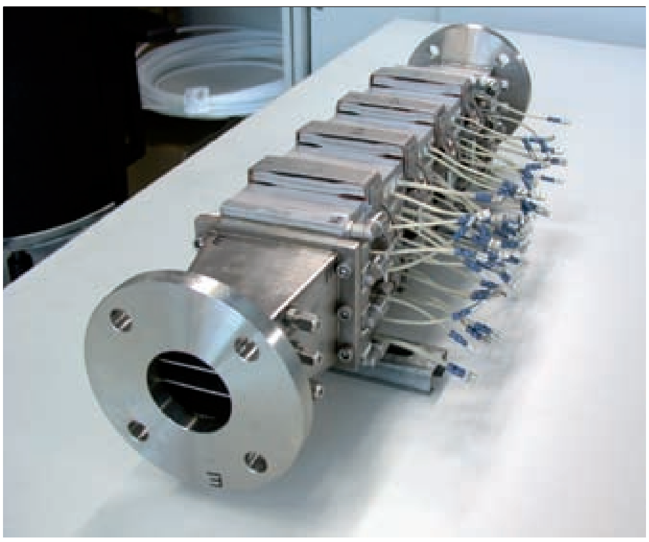
In the past, standard components have been used wherever possible in the construction of the TEG. Special parts could largely be manufactured in the DLR's own workshop. Previous DLR prototypes have given preference to the use of thermoelectric converter modules of type HZ-20 manufactured by Hi-Z [10]. These modules have an efficiency of up to 3.5 % and are characterised by high robustness and long-term stability. ❽ shows a built-up TEG functional model with flange-mounted test-bench diffusers.

4.3 VEHICLE INTEGRATION

As mentioned above, a TEG prototype developed in collaboration with the BMW Group has been integrated into a test vehicle. The DLR was responsible for characterising the base vehicle, implementing a large proportion of the technical hardware modifications, and finally testing the fundamental functionality of the vehicle on the climatic chassis dynamometer. The characterisation of the base vehicle on the chassis dynamometer provided the opportunity to gain experience with the vehicle and verify the exhaust gas temperatures used as boundary conditions in the design. The subsequent planning and conversion phase saw the development and implementation of solutions optimised in respect of the packaging space available for each of the exhaust system, cooling circuit and measuring system. After incorporating the TEG into the vehicle architecture, ❾, the functionality of the TEG was analysed on the climatic chassis dynamometer at DLR.

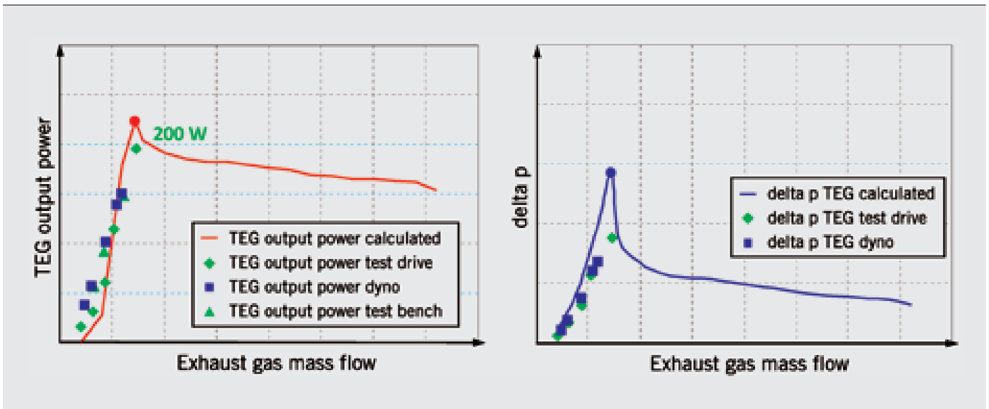
4.4 EXPERIMENTAL TESTING OF THE TEG

Testing of the TEG makes use of a 200 kW gas burner, a cooling system for controlling the temperature of the cold side of the TEG, and all the necessary measuring and data acquisition equipment. In addition to this, test vehicles equipped with the TEG prototypes can also be tested on the climatic chassis dynamometer.



❽ TEG functional model with flange-mounted test-bench diffusers

❾ Vehicle integration of a TEG functional model



❿ Comparison of simulated and measured power output (left) and exhaust gas back pressure (right)

❿ shows a comparison between the results from the developmental simulations and the experimental results obtained on the test bench and in the test vehicle for the 135 km/h variant discussed above. The power output corresponds very well to the simulations and has a maximum of 200 W. The exhaust back pressure produced was found to have significantly lower values than expected. It was only possible to reach the 135 km/h design point during test drives by BMW Group on a test track. At that stage in the development the hot gas test bench was unable to provide sufficient heating power, while on the chassis dynamometer the temperatures in the TEG cooling system rose too high due to insufficient airflow onto the radiator.

5 CONCLUSION AND OUTLOOK

The utilisation of exhaust gas enthalpy through use of thermoelectric generators (TEG) improves the energy efficiency of vehicles with an internal combustion engine. Through the integration of thermoelectric modules in the exhaust system and energy architecture of a BMW 535i US, the German Aerospace Center (DLR) succeeded in producing 200 W of electrical power with a TEG efficiency of 2.0 % on a research demonstrator vehicle. The future focus of development will be on improving efficiency using new and/or improved thermoelectric materials, the optimisation of heat transfer behaviour, and additional vehicle integration in other promising thermodynamic and operational areas.

An additional ongoing research focus area at the DLR has the goal of improving the manufacturing technique for the generators with a view to future standard production applications. The existing functional models are produced using frictional joining techniques and are therefore heavy and very laborious to assemble. As such, an assembly technique is being developed that uses joining techniques – preferably bonding techniques – that are suitable for mass production and therefore able to meet the stringent technical demands and cost restrictions on the system.

Given the thermoelectric materials currently available with *ZT* values of 0.85, the vehicle manufacturer expects to achieve an electrical power output of 600 W and fuel savings of 5 % during motorway use [2]. By further increasing the *ZT* values it should be possible to exceed even this figure, provided that the power generated above this level is also required and can be utilised by the electrical equipment.

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