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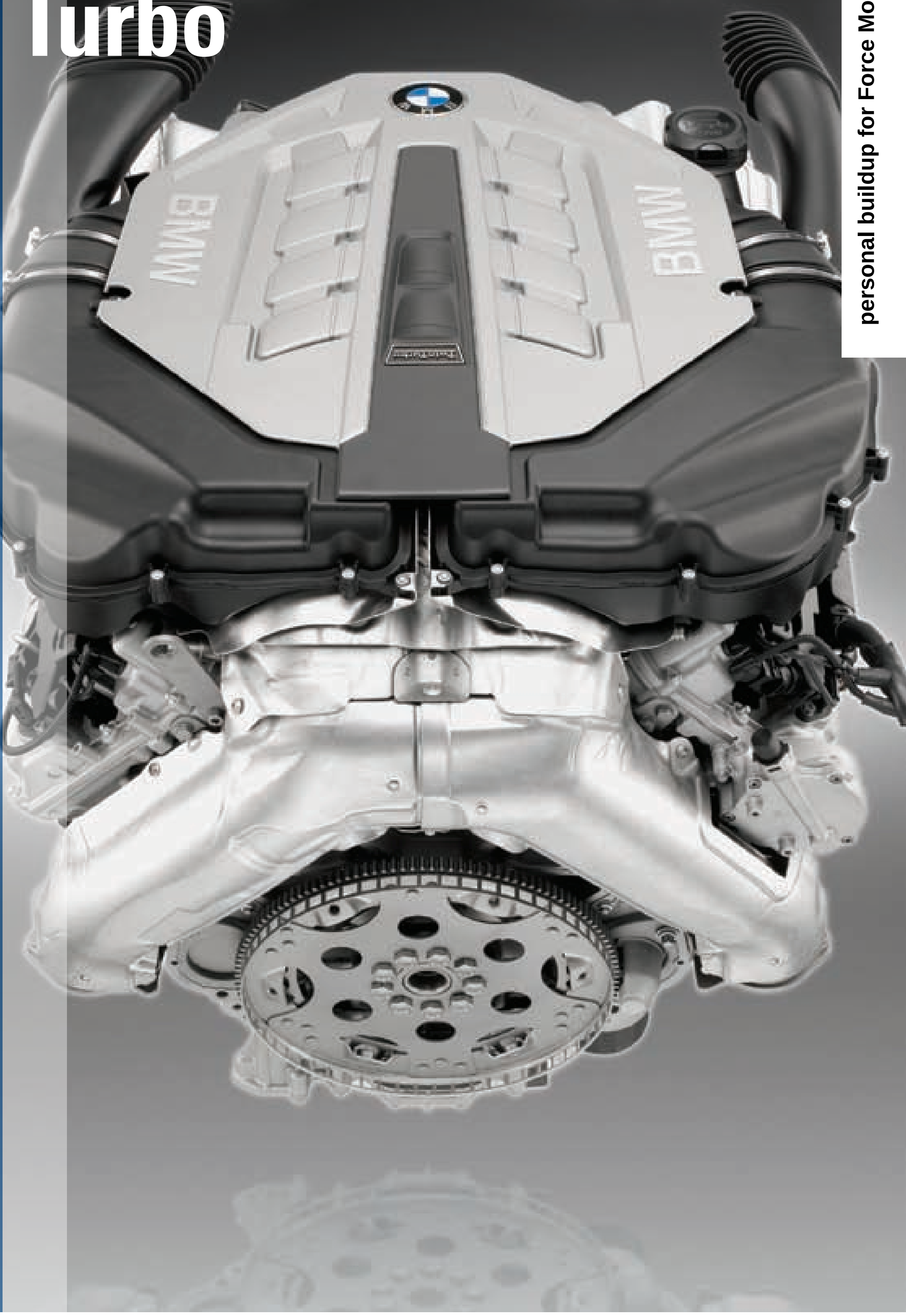

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COVER STORY

BMW V8 Gasoline Engine with Twin Turbo

4
BMW has developed a completely new turbocharged V8 **Gasoline Engine**. Using rotated cylinder heads, the hot side of which faces inwards into the engine Vee, the development has produced package benefits which enable the engine to be fitted in all relevant models without any major adaptation work and also enable the various drive derivatives to be made adaptable.

COVER STORY

DEVELOPMENT

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In Praise of Value Added

Dear Reader,

When visiting companies in recent years, I always liked to ask: "Do you manufacture component X yourself?" I was often given long and sometimes long-winded answers instead of a self-confident "Yes". The situation was quite different, however, at the Swiss sensor manufacturer Kistler, whose value-added depth extends so far that they even grow their own piezo crystals in a bunker-like room. This healthy self-confidence, cultivated over decades, explains what makes this company the number 1 in dynamic measuring technology. And what companies have to do themselves to maintain this position. And where cooperation is the better strategy.

What much bigger companies can learn from medium-sized enterprises like this is that there is nothing wrong with value-added depth. On average, the margins of industrial products with a great value-added depth are much higher than those of products for which manufacturers only carry out the final assembly. Or even simply stick a label on the product, thus making them nothing more than middlemen for their own products.

If a strategy with a high value-added depth is to be successful, it needs talented, experienced and committed development engineers who can maintain the technological lead. Conversely, a company with a low value-added depth does not need any factories in high-wage countries like Germany or Switzerland. And neither does it need large development departments. For that reason, the value-added strategy of your company should be of vital interest to you as a developer.

And if you happen to be a top executive in a multinational company in which assets or employees are seen as a burden, perhaps you might like to put your highly paid advisors on hold for a while and take some time off to pay a visit to a successful medium-sized company.

Johannes Winterhagen
Winterthur, 26 September 2008

Johannes Winterhagen
Editor-in-Chief

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Cooperation with the STG, Schiffbautechnische Gesellschaft e. V., Hamburg, in the area of ship drives by combustion engines

The New BMW V8 Gasoline Engine with Twin Turbo

Part 1 – Design Features

The new BMW V8 gasoline engine with two turbochargers in engine-mounted arrangement in the engine Vee is the latest chapter in the "BMW Efficient Dynamics" story. Development staff have been able to demonstrate the vision of "Sheer Driving Pleasure through efficient dynamics" by the excellent transient characteristics, the extremely agile behaviour in intermediate acceleration and the good levels of fuel consumption without compromising on performance.

1 Introduction

In 2001, the new generation of BMW V8 gasoline engines made their debut as 3.6 l and 4.4 l variants with variable valve lift Valvetronic and variable geometry inlet manifold [1,2]. After being revised in 2005 [3] and an increase in cubic capacity to 4.0 l and 4.8 l, they now deliver up to 270 kW and 490 Nm. They are currently still being fitted in models of the X5, 5 and 6 Series. Volume production use in the X6 in May 2008 and in the new 7 Series in September 2008 sees the launch of another fully redeveloped BMW V8 gasoline engine. The technology deployed in this engine is the next step after that introduced in in-line engines [4,5]. Development of this engine focused on favourable fuel consumption, superior development of power with outstanding vehicle response and scope for use around the world. Using rotated cylinder heads, the hot side of which faces inwards into the engine Vee, the development has produced package benefits which enable the engine to be fitted in all relevant models without any major adaptation work and also enable the various drive derivatives to be made adaptable. Last but not least, it had to be possible for the unit to be fitted in all relevant models without major adaptation work and for the various drive derivatives to be adaptable.

2 Design and Objectives

In the initial phase, a number of concepts were examined. The classic way of increasing performance, i.e. increasing cubic capacity, was rejected on package, weight and consumption-related grounds. Even the popular approach for gasoline engines of increasing speed did not satisfy the need for an everyday superior drive in the vehicle class being targeted. Only by using exhaust turbocharging could the torque and power potential required be tapped into. This approach paved the way for other power adaptations by varying the periphery while the reduced cubic capacity also delivers an extra consumption benefit. When combined with BMW's direct fuel injection, this concept ultimately proved to be the efficient dynamics solution opted for. The first designs planned for the classic V8 with external turbochargers. But it soon became clear that the need for all-wheel capability, integration of the turbochargers and air duct, free travel of the steering spindle and usability in various models was not possible with this design. A paradigm shift was needed to achieve this. The approach taken was to rotate the 'hot' and 'cold' sides of the cylinder head, **Figure 1**, a solution that brought with it clear functional benefits. To achieve this, the relatively large space in the engine Vee, which with the

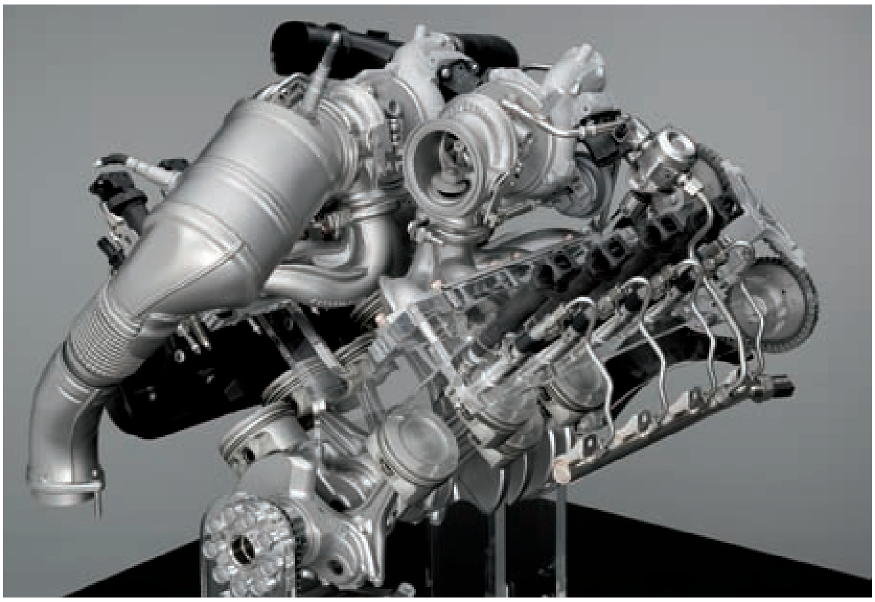

Figure 1: Position of exhaust turbo charger and engine-mounted catalytic converter in the engine Vee

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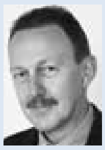

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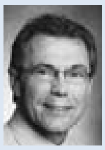

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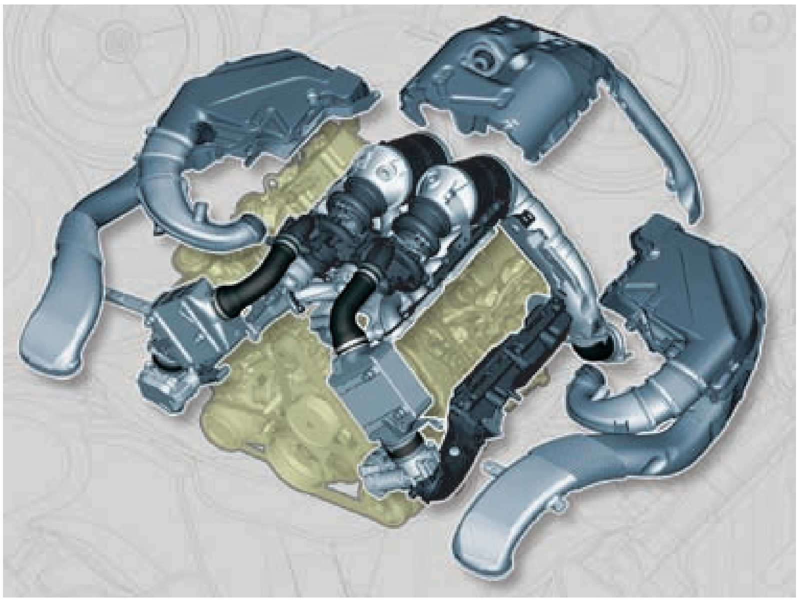

Figure 2: Intake duct and upper thermal protection

turbocharged engine only featured a small-volume suction unit, was used to house the charger and close coupled catalytic converters. This allowed the identical manifolds and turbochargers used for both banks of cylinders to be designed to be extremely compact and with optimum flow. This in turn enabled a universal and compact suction unit to be integrated on the outside of the engine to satisfy various package requirements relating to steering spindle geometries and front axle drives in the various models without the need for individual adaptations.

Numerous calculations on the thermal balance and flow in the engine compartment proved the concept to be promising with an appropriate thermal insulation plate design and specific air duct to dissipate heat, **Figure 2**. At the same time this concept allowed the package-intensive wheel arch insulation, a standard feature with external turbochargers, to be dispensed with. There was also no longer any need for high-quality materials to be used to protect all the components fitted there. The design featuring an inward rotation of the hot side of the cylinder heads (HSI) saw the birth of the first gasoline engine for cars with exhaust turbocharger in the engine Vee (LIV = turborcharger in Vee).

3 Description of the Engine

The **Table** contains a summary of the key technical data.

3.1 Cylinder Crankcase

The Alusil solid aluminium block was produced using closed deck technology for greater rigidity. Together with screwing the cylinder head down into the base plate of the cylinder housing, this ensures little distortion of the cylinder sleeves which were honed and laid bare. The deep skirt cylinder crankcase has a double main bearing screw connection. This is attached to the additional side panel using threaded supporting bushes and screws to absorb the huge cross forces originating in the crank drive. The cylinder pipes which are cast together are cooled in the hot area through drilled ducts. Thrust bearings 1 to 3 have a lengthwise bore to reduce losses from ventilation. It was possible for the engine block to be shorted. The space needed for the front axle drive's passage was created by directly driving the oil pump on the output side with the sprocket wheel integrally moulded on at the rear end of the crankshaft. The oil component of the blow-by gases was also reduced using oil return bores extending below the oil level and separate gas channels. The piston spray nozzles only open at 1.5 bar and higher using pressure control. They therefore allow a sufficiently large volumetric flow of oil to be maintained for Vanos adjustment in hot idling. The converter is screwed down to the flywheel at six points at an angle of 30° to the horizontal through an opening in the converter bell, offering a practical solution to gearbox changes. Intensive use of FEM, particularly in the design of the main bearing screw connection, cylinder pipe rigidity, rigidity of the gearbox flange and the unit connection, enabled a light yet high-rigidity design to be produced.

Table: Key technical data of the BMW V8 Twin Turbo

Parameter	Unit	BMW V8 Twin Turbo
Engine displacement	cm^3	4395
Firing order		1-5-4-8-6-3-7-2
Power at specified engine speed	kW 1/min	300 5500 - 6800
Torque at specified engine speed	Nm 1/min	600 1750 - 4500
Speed range	1/min	550 - 6800
Stroke / Bore	mm	88.3 / 89
Compression ratio		10.0 : 1
DIN-weight, incl. exhaust man. and tc	kg	228
Engine length / height / width	mm	L = 628; H = 753; W = 758
Cylinder offset	mm	98
Main bearing diameter	mm	65
Con-rod bearing diameter	mm	54
Con-rod length	mm	138.5
Valve diameter (Inlet / Exhaust)	mm	33 / 29
Valve angle (Inlet / Exhaust)	°	16.5 ° / 17.5 °
Admissible fuels	ROZ	91 - 98
Emission regulation		EU5 and ULEVII

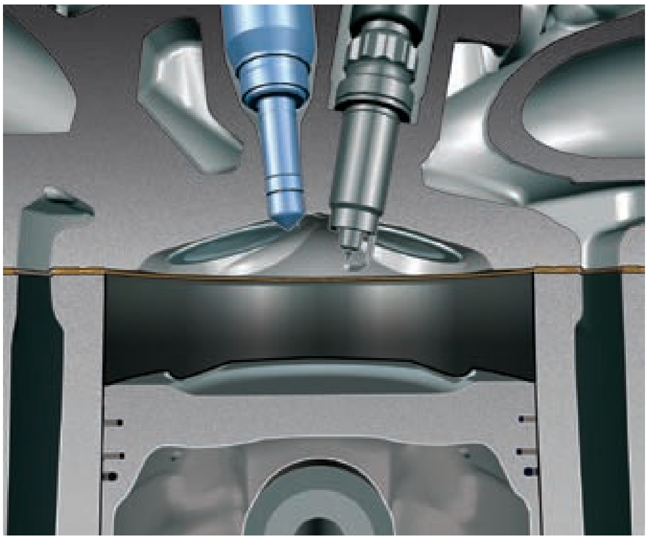

Figure 3: Central injector and spark plug system in the combustion chamber

3.2 Crank Drive

For ease of assembly the robust, iron coated aluminium piston and its less jagged surface has the same design for both cylinder banks. It also features a shape-optimised pressure ring in a hard anodised groove. The cracked forged con rods with trapezoidal piston pin lug have a three-material sputter bearing shell on the rod side, while three-material shells are fitted on the cover side and two-material shells in the main bearings. Oblique reciprocating pin holes were produced in the forged and twisted crankshaft with inductively hardened running faces in order to reduce weight. Inertial forces of the 1st and 2nd order are fully compensated for, as in the previous engine.

3.3 Cylinder Head

The main objectives of redesigning the cylinder head were to adopt the main design concept of the injector/spark plug arrangement from existing in-line engines and to enable a robust overall concept for high power levels. Furthermore the turbocharger in Vee (LIV) concept and cylinder head were to be integrated in the existing production line to cut costs. The central arrangement of the injector and spark plug in the combustion chamber characterises both the cylinder head's basic geometry and the combustion process, **Figure 3**. Combined with the optimised inlet duct, this produces very good fuel management with short burning periods for optimum efficiency. The increased pressure and temperature load required intensive theoretical sizing of the cylinder head structure and coolant supply in the early concept phase, especially as a result of rotating the flow of coolant from the inlet side to the outlet side. The solution found is characterised by high rigidity with good resistance to fatigue and offers potential for further increases in power. Reducing the cylinder head screw distance to 90 mm, the three-layered metal carrier seal and M11 screw connection greatly contribute to the robustness of the overall concept. The moderate total valve angle of 34° also permits unlimited access to the screws when the valve drive is prefitted. Positioning the high pressure gasoline pumps above the outlet camshafts required a new pressure die cast component. As well as supporting the pumps, this could also be used to mount the camshaft and high pressure pump tappet. The vacuum pump driven by the inlet camshaft is positioned on bank 1 on the output side. The unfinished cylinder head parts are cast in the BMW foundry using the gravitational procedure in double die casts. The aluminium pressure die cast cylinder head covers seal the oil chamber using a rubber profile gasket. They support the cyclone oil separators and form the base for screwing down various peripheral parts.

3.4 Valve and Chain Drive

The thermally joined camshafts feature forged cams, a cast flange for the chain gears and a sinter sensor wheel for recording the Vanos signals. The lever roller and cams are supplied with oil for cooling and lubrication by a directional oil spray hole in the calotte of the roller cam follower. It proved possible for the stroke to be reduced from that in the naturally aspirated engine. An extra three-way cam is fitted on the outlet side to drive the high pressure pumps. The newly developed bush joint tooth chains feature a high resistance to wear and are also quiet as the polygon effect is avoided in the sprocket wheel, **Figure 4**. The chain tensioner is fitted in the chain groove to save space and the sodium-filled outlet valves have a shaft chromium plating. The valve opening times were kept extremely short to prevent unwanted high residual gas content resulting from overflows during the irregular ignition sequence of the V8. Cam phasing was also used to prevent differences between the cylinder filling and residual gas content. The double Vanos typically used by BMW to continuously adjust the camshaft control times permits a high

Figure 4: Chain drive with double Vanos

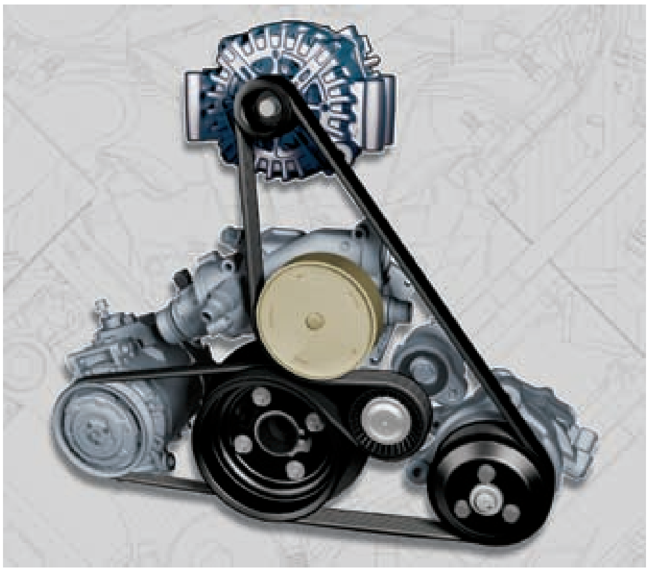

Figure 5: Primary and secondary belt drive

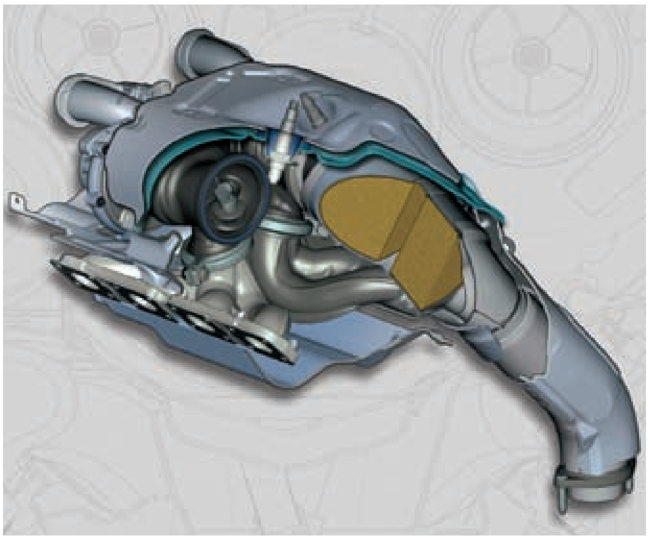

Figure 6: Cooling air duct around exhaust turbocharger and catalytic converter

residual gas element in partial load operations. This allows full use to be made of the consumption benefit achieved from reduced throttle losses when the throttle flap is wide open. Furthermore, the exhaust enthalpy flow and therefore turbocharger speed can be increased by rinsing thoroughly in a manner specific for direct fuel injection, resulting in excellent response even at low speeds.

3.5 Oil Circuit

The six-chamber oil pump functions using the pendulum slidegate principle seen in the six-cylinder engines of BMW and the BMW M5 engine. The volumetric flow control only pumps the amount of oil required by the engine in the current operating mode and so helps cut CO_2 emissions. The air intake is adapted to the corresponding vehicle-specific oil pan and screwed onto the standard oil pump housing.

The oil cooler thermostat and the oil filter element, made from synthetic fleece, are integrated in the oil pan. There are upper and lower sections of the oil pan for each series, made from aluminium pressure die casting and optimised in terms of strength and acoustics. A two-part oil scraper ensures especially low oil foaming levels even in extreme driving situations. The oil level sensor integrated in the bottom section of the oil pan meant that the oil dipstick could be dispensed with.

3.6 Crankcase Ventilation

This is derived from the six-cylinder engine and is also used as a modular component in future engine variants. Depending on type of operation (naturally aspirated or turbocharged), the blow-by gases are introduced to the cyclone separator either upstream or downstream of the turbocharger. Crankcase ventilation ensures a vacuum of around 50 mbar in the crankcase.

3.7 Fuel System

The High Precision Injection system is a 2nd generation direct fuel injection system with central piezo injectors. The maximum system pressure of 200 bar is provided bank by bank by single-plunger fuel pumps. When combined with the spark system, also located centrally, this system and its accurate fuel mixture generation provide the essential basis for the engine power attained, spontaneous charger response, compliance with EU5/ULEV II limit values and low fuel consumption. A knitted wire ring isolates the injector and cylinder head acoustically while a PTFE ring at the foot of the injector produces the seal to combustion chamber pressure. Line management in the fuel system is optimised to minimal line distances. Since the system pressure places high demands on the system's gas seal integrity, rails and lines are produced from stainless steel and the screw connections are 'hard', i.e. feature no seal.

3.8 Belt Drive

The main seven-ribbed drive for the power steering pump with optional variants for active steering and ARS also drives the air-cooled 210 A generator fitted in the engine Vee and the mechanical water pump, **Figure 5**. Dispensing with a reversing roller, which increases losses, the belt drive is only tensioned using one tensioning roller. By using a profile-free pulley for the water pump drive, some belt wear can be transferred to the tips of the belt ribs for the full service life. A patented drainage system on the crankshaft and power steering pump pulleys diverts water that ends up between the belt and pulley when spray levels are high or when driving through water. When using a four-ribbed auxiliary drive for the refrigerant compressor, both a tensioning roller and other assembly aids can be dispensed with and the patented, new revolver tensioning system used instead. The belt drive is driven by the primary side of the silencer. This avoids the stimulation of torsional vibration from the secondary side, which reduces service life and is usually used for this function.

4 Periphery

4.1 Intake Duct and Suction Unit

A multiple-flow intake duct with engine-mounted intake silencers was produced to optimise the package. The design of the intake silencer's upper section and the air hose allow for adaptation to the various package and acoustic requirements of the X6 and 7 Series. The overall arrangement, combined with integration of the ATL in the engine Vee, make minimal pressure losses possible on the suction and pressure sides. The charge air coolers screwed onto the face end of the cylinder heads are connected to the plastic suction units, standard for all engine applications, on the outside of the engine via the throttle bodies.

4.2 Exhaust Manifold and Turbocharger

Positioning the two identical 'Twin Turbo' exhaust turbochargers in the engine Vee allows the entire charging system and its periphery to be arranged in a very compact way, **Figure 6**. The mono-scroll turbochargers developed especially for the engine are controlled by a vacuum-controlled waste gate and integrated electrical diverter valves. They are characterised by high levels of compressor and turbine efficiency. The cast manifolds, designed for optimum flow, have a 4-in-2 combination for optimised ignition sequence and are also identical for the two banks of cylinders. Together with the turbochargers and counter pressure-optimised engine-mounted catalytic converters, they form the basis for excellent response and superior power and torque values.

4.3 Catalytic Converters

Pre and main catalytic converters are positioned just behind the turbines in a shared element. The very compact arrangement of the hot end combined with the continuous exhaust sensors ensures that the activation temperature is quickly reached. Strict EU5 and ULEV II emissions legislation can therefore be satisfied right from the start of volume production and there is no need for a secondary air system. Corrugated pipes integrated in the catalytic converter outlet pipes ensure acoustic and thermo-mechanical isolation.

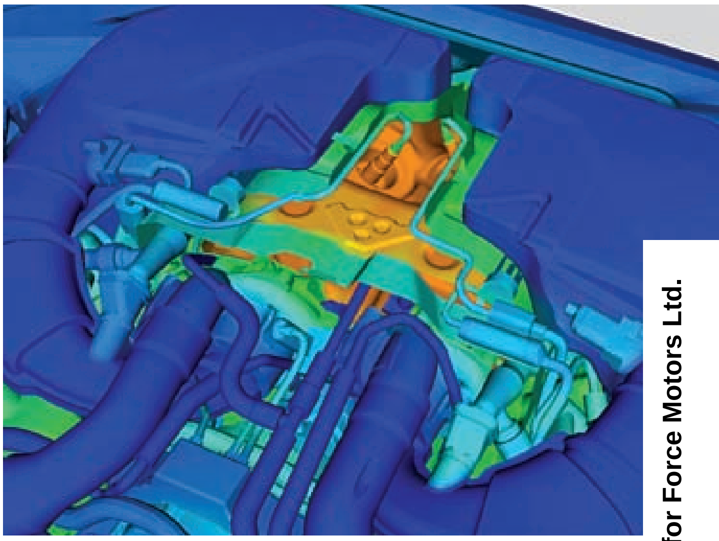

Figure 7: Simulation of component temperatures

4.4 Thermal Protection Measures

A lot of attention was of course devoted to the issue of temperatures in the engine Vee from the very start. Simulations of varying levels of detail were used to optimise flow characteristics and sheet metal geometries, and differentiated calculations were performed of the component temperatures for the various operating conditions, **Figure 7**. As a result, the exhaust manifold, turbine, catalytic converter and rear exhaust pipe are surrounded by a system of highly efficient, three-layered thermal protection sheets made from aluminised sheet metal with a layer of soft material in between. This firstly produces specific guidance of air flow through the engine Vee and secondly shields off radiant heat. During driving operation, excess pressure builds up due to back pressure and the fans in front of the engine while the buried flow in the undercarriage ensures a vacuum to provide a permanent and efficient flow of air through the engine compartment, **Figure 8**. Component temperatures are even kept within permissible limits when reheating the power train (shut down when hot) as a result of the fans continuing to run for an appropriate amount of time.

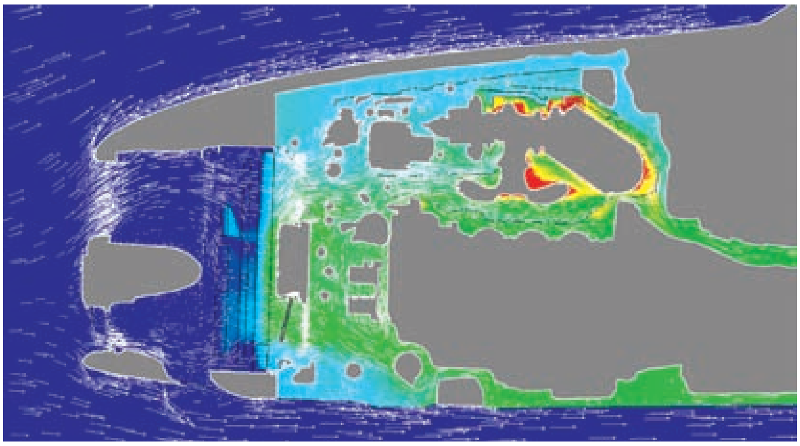

Figure 8: Simulation of air flow in front end

4.5 Coolant Circuit

4.5.1 Engine

By consistently integrating all coolant channels in the crankcase, most external coolant lines could be dispensed with in the main engine. Cross-section optimisations and concentrating the flow of coolant to structures and surfaces purposeful for heat exchange has resulted in a reduction in bypass coolant volumes of 42 % over the predecessor and a considerably shorter warm-up phase. Downstream of the coolant pump in the engine Vee, the coolant flow is led to the rear as a double channel right next to

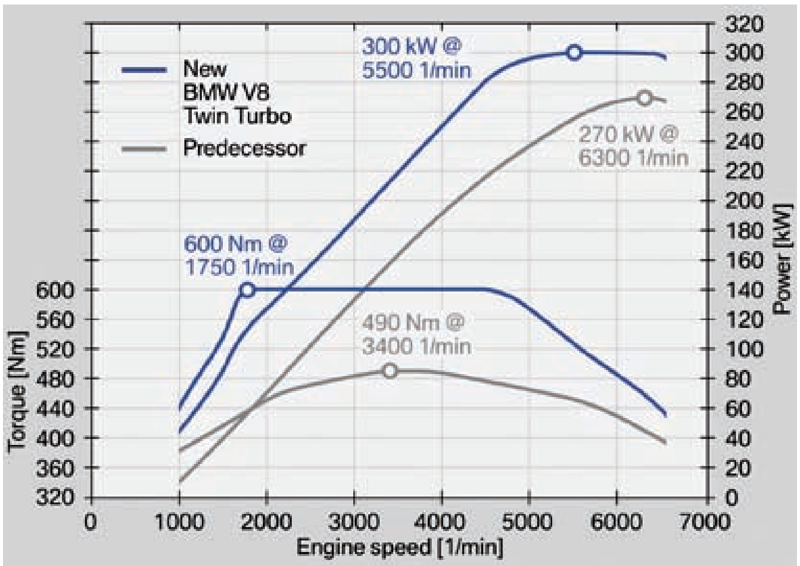

Figure 9: Power and torque levels compared with previous engine

the main oil advance channel running in the opposite direction. This design thermally links both media and has a positive impact on engine oil temperature. Once the coolant flow has been split between the left and right banks of cylinders, it passes through both banks diagonally (from the outside rear to the inside front). This method produces identical cooling routes in the individual cylinder units and therefore optimum cooling over the entire speed range. An extra electrical water pump is used to cool the turbocharger thrust bearings.

4.5.2 Charged Air Cooling

The compact space available in the vehicles and the water and oil cooling surfaces needed do not allow the compressed air to be cooled directly, which is why indirect charged air cooling is used. Indirect charged air cooling requires great attention to detail as a high charged air temperature impacts directly on the need for early ignition and therefore on consumption. This separate low-temperature coolant circuit is driven by its own electrical water pump. The two air/water heat exchangers are secured to the fronts of the cylinder heads.

4.6 External Cooling

To ensure optimum recooling, in the cooling module the heat exchanger for the low-temperature circuit of the charged air cooling was positioned in the first cooling level, upstream of the air conditioning system's condenser. The coolant cooler for the engine circuit, the gearbox oil/water heat exchanger, which is cooled by an extra supercooling route in the coolant cooler, and the power assisted steering cooler complete the cooling module. The cooler's efficiency is raised by a priming 850 W electric fan. An engine oil cooler is positioned in the right-hand wheelhouse in both series. The variant of the X6 designed for hot climates has an extra coolant cooler in the left-hand wheelhouse, while a second engine oil cooler is fitted here in the BMW 7 Series.

4.7 Exhaust System

When designing the pipework layout of the multi-flow exhaust system in the vehicle's undercarriage, attention was paid to producing pipes with optimised flow and the biggest pipe diameter possible. Exhaust flaps controlled using DME functions are positioned in the outlet pipes of the rear silencer. Together with the inner structure of the rear silencer in the X6, these produce a powerful V8 sound when accelerating as well as outstandingly acceptable acoustic levels during constant travel. In the application for the BMW 7 Series, excellently unobtrusive acoustics, matched to the comfort-based overall needs of the vehicle, are achieved with two individual rear silencers.

4.8 Engine Control

The MSD85 is a further development of the MSD81, familiar from the six-cylinder turbo engines. A tricore processor lies at the heart of the control unit hardware. Communication with the vehicle is handled using two CAN buses, a LIN and a BSD bus and in the BMW 7 Series' onboard supply system also using the rapid FlexRay. A four-bank system is used to activate the piezo injectors and ignition is controlled by the conventional ignition limit stages in the control unit. A housing with water cooling was developed to fit the control unit in the engine compartment of the BMW 7 Series and an air-cooled variant for the X6 control unit box.

5 Power, Torque and Emissions

As of 1750/min, the new 4.4 l engine has a torque of 600 Nm, **Figure 9**. As of around 4500/min, the torque is reduced to limit the engine's rating should speed increase yet further. Despite increasing engine power by around 15 %, it has been possible to reduce fuel consumption slightly from 12.6 l/100km to 12.5 l/100km. The engine fitted in a ‚global' variant satisfies both EU5 and ULEV II limits.

Read the second part of this article with a detailed outline of the functional characteristics in MTZ 12/2008.

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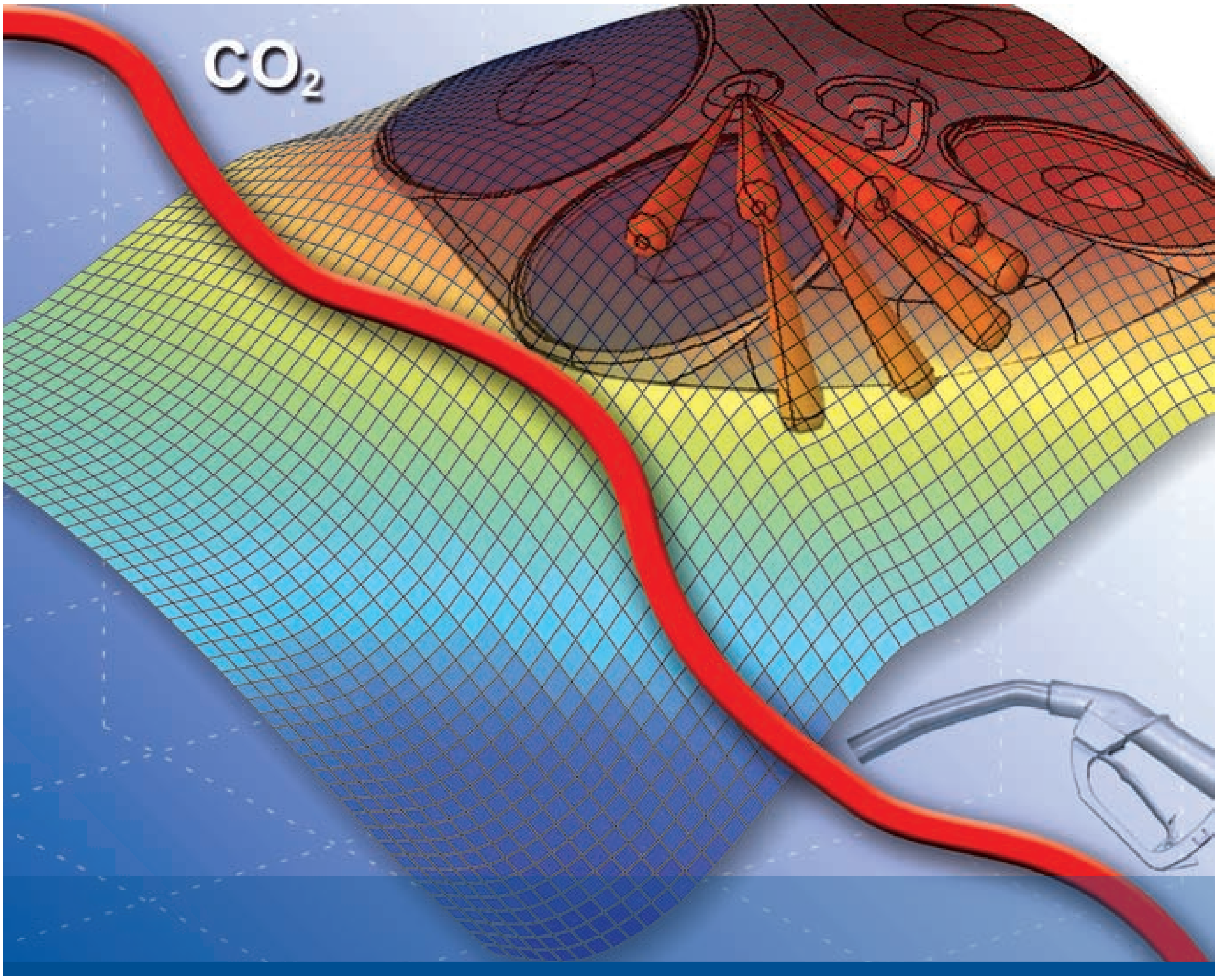

Simulation in Engine Development
On the Way to Virtual Application

The development of low-emission vehicles requires integrated tools and methods for describing a virtual vehicle and a virtual application. Here the detailed modeling of the combustion process, as it has been further developed by Ricardo, is decisive. This article describes a new tool for calculating the energy release under changing operating conditions. In addition, validation results for stratified charge operation of a turbocharged direct-injection gasoline engine with a jet-directed combustion process.

1 Introduction

Growing complexity, high innovation levels, short development cycles and the necessity of additional CO_2 reduction in future vehicle drives [1] hold the risk of weak points not being discovered until the final application in the vehicle. However, then they can often only be eliminated on schedule with a massive use of resources. With the early, continuous use of simulation tools up to and including the virtual vehicle with virtual application, the functions of individual subsystems and the entire engine and vehicle can be secured even without hardware.

Sequential development should therefore be replaced with a process in which the simulation results and measured values are treated equally from the start. These are then used to enable the effects a change in one or several setting parameters or a component version will have on the system behavior to be calculated at any time, **Figure 1**. The development of suitable integrated tools and methods [2] all the way up to a fully integrated, continuous process is currently proceeding at full speed, **Figure 2**. With the introduction of a real-time-capable, crank-angle-resolved simulation model for the transient engine behavior (Wave RT) [3], a tool is meanwhile available for the prediction of consumption, emissions and handling within specified, customer-relevant driving cycles. This tool enables sufficiently precise statements and, with its short calculation times, also permits service life effects to be taken into account. A central part of this kind of 1D-CFD tool is combustion modeling, which utilizes simple models for conventional gasoline and diesel-engine combustion processes, like the one according to Vibe. However, new combustion processes, like HCCI or stratified charge, have not been depicted with sufficient accuracy up until now. This is primarily true for the energy release under changed operating parameters.

2 Combustion Simulation

Combustion modeling within a fast cycle simulation must be simple and re-

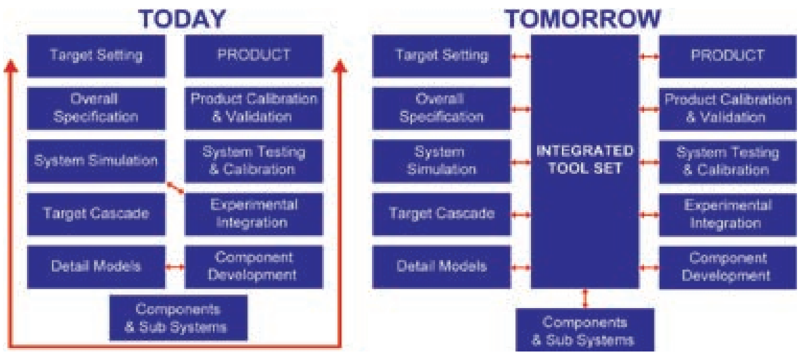

Figure 1: Sequential, integrated procedure of the development process

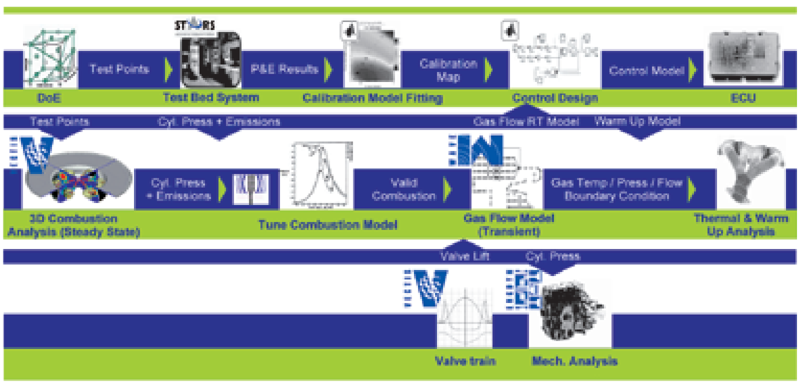

Figure 2: Integrated toolbox on the way to virtual application

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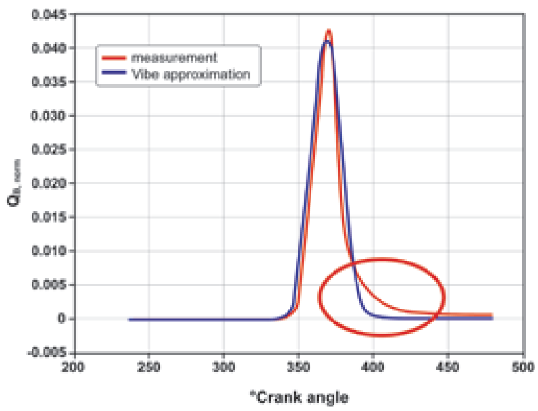
Table: Technical data of the one-cylinder research engine

Bore x Stroke	mm	86 x 86
Injector		8-hole injector, central position
Spark plug		Standard J-type
Recess		Asymmetrical, flat recess
Air conditioning		External; pressure, temperature, moisture
Oil and water conditioning		External
Exhaust gas recirculation		External, conditionable
Fuel pressure	bar	Max. 200, externally generated
Exhaust back pressure		Can be adjusted with flaps

quire little computing time, however it must also describe the energy release with sufficient accuracy. Only in this way can the effects of operating parameters and components, such as the effect on the maximum combustion temperature or the exhaust enthalpy, be precisely simulated. A new model should also be based on existing simulation models so that its use is easy to implement. This requirement is met by the substitute combustion curves according to Vibe which, as simple and multiple Vibe functions, are part of many programs for working process and 1D fluid calculations. However, these functions based on kinetic reaction approaches do not describe the fuel turnover in new gasoline and diesel-engine combustion processes with sufficient precision and cannot detect how the energy release changes under changing operating conditions.



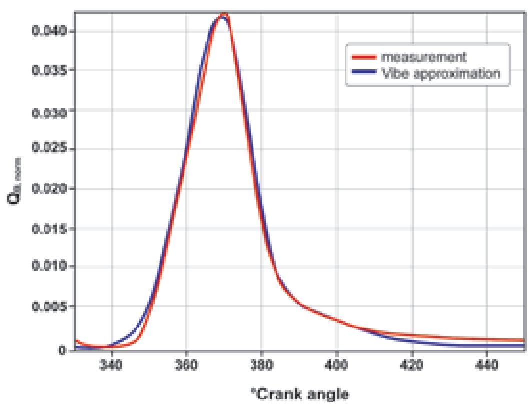


Figure 3: Substitute combustion curve for stratified charge operation – top: a simple Vibe function; bottom: a superposition of two Vibe functions

2.1 Model Generation

For the development of a new model for stratified charge operation of a direct-injection gasoline engine with jet-guided combustion, systematic variations of relevant influencing parameters were carried out on a one-cylinder research engine, **Table**. By fully indexing of the engine, it was possible by means of a thermodynamic analysis to calculate not only the combustion behavior, but also the charge cycle, and with it among other things the residual gas content at the point in time of closing of the intake valve. Combustion with lean charge stratification was also characterized:

Aided by the local high turbulences of the high-pressure injection shortly before TDC, the premixed charge cloud near the spark plug burns very quickly. However, local leaning-out occurs at the edges of the mixture cloud, which leads to slow burn-out when the front of the flame arrives. In the past, this type of combustion with two conversion phases could not be depicted with sufficient precision with a single Vibe function. The premixed combustion or the slow burn-out can be depicted well with a parametric Vibe function. However, especially at higher loads, the superposition of both processes results in a significant deviation of the calculated from the actual energy release during the slow burn-out [4]. **Figure 3** (top) shows the fuzziness of the process with a single Vibe function. A new, direct linking of two Vibe functions now enables a significant increase in the modeling quality, which precisely depicts both the maximum energy release, the 50 % conversion point, the 10-90 % combustion duration and the slow burn-out, Figure 3 (bottom).

2.2 Stratified Combustion

The new combustion model for charge stratification links two Vibe functions with the optimization factors α, β and γ for the specific combustion process as follows:

$\varphi_{BB2} = \alpha \cdot \varphi_{BB1}$
$\varphi_{BD2} = \beta \cdot \varphi_{BD1}$
$m_2 = \gamma$

The factors α, β and γ are determined with stochastic process models from a large number of standardized combustion curves of different engines with a similar combustion process. They are de-

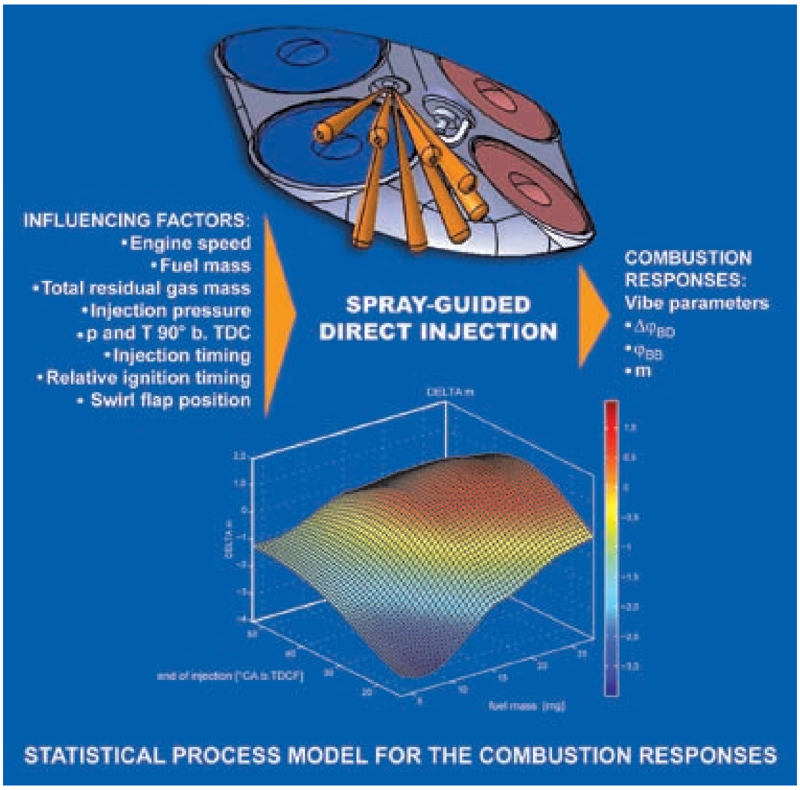


Figure 4: Influencing parameters and target variables of the stochastic process model

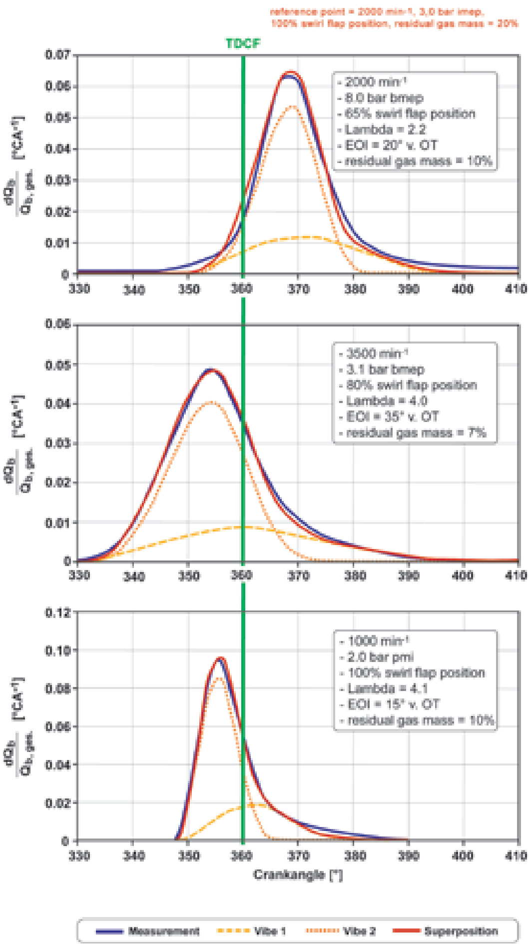


Figure 5: Comparison of a model prediction and the actual combustion curve under changed operating conditions

pendent in each case on important geometric factors like stroke, bore, deviation of the injector position from the combustion chamber center, spacing between the injector and spark plug and the combustion bowl size and depth. For the charge stratification in the one-cylinder research engine, this resulted in the values $\alpha = 1$, $\beta = 2$ and $\gamma = 1.5$.

2.3 Energy Release

To detect the change in the energy release under changed operating conditions, an empirical model based on statistical methods was developed. It continues the tradition of the models from Woschni/Anisits, Barba, Csallner, Witt and Theisen. However, in contrast to those, the characteristic Vibe parameters start of combustion (φ_{BB}), combustion duration (φ_{BD}) and the form factor (m) for a fixed degree of application a are described with stochastic process models in dependence on the relevant influencing parameters, **Figure 4**.

Like polynomial models or neuronal networks, stochastic process models [5] are mathematical methods for statistical test planning (Design of Experiment, DoE). They can be used for the direct mathematical description of target variables like fuel consumption, emissions or Vibe parameters in dependence on relevant influencing variables like the point of injection, exhaust gas recirculation rate and boost pressure without having to take the detailed physical background into account.

Stochastic process models are principally based on an extended Kringing method, which can predict a realistic result for unknown operating by taking the spatial variance within a test plan into account through interpolation of surrounding measured values states. Process models of this kind primarily enable an extrapolation in the near field of the test plan. The condition for using these models is a stochastic distribution of the influencing parameters and the target variables. Then correlation functions of the type expected result = mean value + systematic term + error describe the relationship between the target and the influencing variables.

To describe the change in the energy release under changed operating conditions, the changes in the factors start of

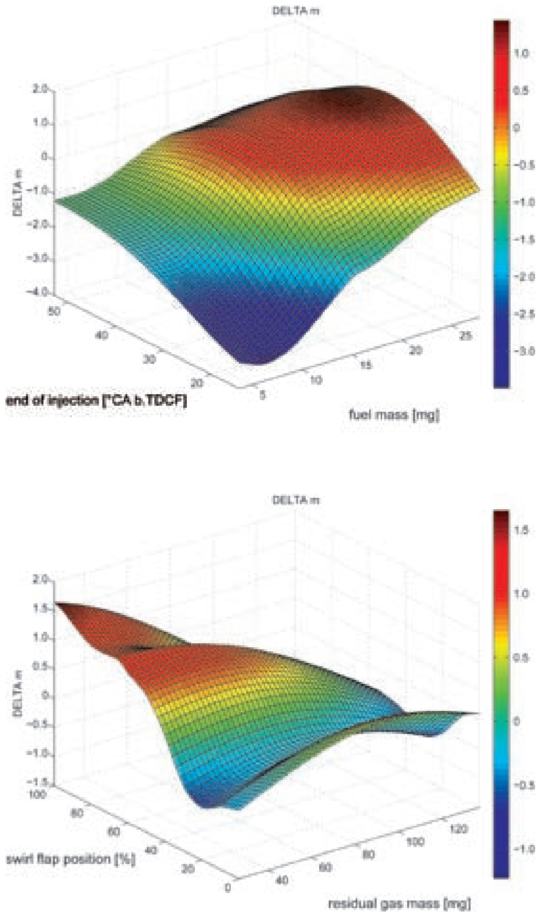


Figure 6: Modeling of the change in the form factor m, partial model

combustion 1(φ_{BB1}), combustion duration 1(φ_{BD1}) and form factor 1 (m_1) must be recorded in dependence on the selected influencing parameters after linking two Vibe functions. These factors shown in Figure 4 are determined with a sensitivity analysis on the test bench. Here the pressure and temperature at 90° before ignition TDC were selected in view of the simulation.

To enable the model to be applied to a similar engine, influencing parameters and target values are based on a measured reference point. This makes it possible, for example, to also take changes in the geometry or the spray into account by replacing the reference point, and the model created can be applied to a new engine. The combination of the start of combustion 1(φ_{BB1}), combustion duration 1(φ_{BD1}) and form factor 1 (m_1) cannot be clearly determined for a given combustion curve, which can result in inconsistencies during modeling. Therefore, a smoothed stochastic process model is determined for the form factor m_1 following approximation of the Vibe parameters from the measured combustion curves and linking of the two Vibe functions. In a second step the Vibe parameters are calculated again for all measured combustion curves. In the process, the modeled m1 is specified and the linking of the two Vibe functions is recorded. For the newly calculated parameters start of combustion and combustion duration, a smoothed, stochastic process model is also created. It clearly describes the changes in the superpositioned Vibe functions in dependence on the selected influencing parameters by the specified linking and the three models for start of combustion 1(φ_{BB1}), combustion duration 1(φ_{BD1}) and form factor 1 (m_1), **Figure 5**.

3 Model Validation

If actual combustion curves not included in the modeling process are compared with the prediction by the newly developed model, then it becomes apparent that it can predict the effects of changed parameters. **Figure 6** shows different variations of the operating parameters in each case based on the same reference point. Despite a significant change in major engine setting parameters, and therefore in the application, the model approximates the curve of the application very well to the measured values. The new model of the directly linked Vibe functions with stochastic process models for the depiction of the changed energy release is therefore suitable for precisely and quickly predicting the combustion. By selecting superpositioned Vibe functions as the basis, it can easily be implemented in the existing working process calculation and simulation programs.

4 Summary

The new empirical substitute combustion curve model for fast description of the energy release expands the virtual methods for new combustion processes. As a result, the target conformity with regard to fuel consumption, emissions and drivability during legally specified and customer-relevant driving cycles can already be studied in an early development phase of future vehicles.

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NVH Target Value Definition

From Cylinder Pressure right through to the Driver's Ear

The definition of NVH target values for a powertrain is a pre-condition for realising the desired vehicle noise and vibration behaviour. Such target values are no longer only defined for noise and vibration levels of an engine but in respect to the vehicle sound quality and interior noise. FEV develops proprietary methods, scatter bands and criteria for customer-specific definition of target values. The article presents current approaches and criteria.

1 Introduction

The development of an engine always starts with the definition of target values. Noise, vibration and harshness is only one cluster of aspects to be taken into account alongside others such as engine power and torque, fuel consumption, exhaust gas emissions, manufacturability and – above all – costs. If the engineer is given sufficient freedom and allowed to concentrate on acoustic priorities, it is possible these days to design an engine with excellent sound qualities. **Figure 1** is an example showing the actual noise levels of the 2.7 l V6 diesel engine launched in 2004 for the Jaguar S-Type in comparison to the scatter band by FEV [1]. Nowadays, when developing an engine, other aspects such as reduction of CO_2 emissions tend to be given a higher priority. What is important for the development process is a clear prioritisation and a correspondingly adapted definition of target values. The engine with the lowest production costs will never be the one offering best fuel economy, highest output and lowest noise levels at the same time.

In an engine development process, target values will be at least defined for engine noise levels for example at the rated operating point and at idling speed. In addition there are various noise level and sound quality target values for engines which need to be defined in different degrees of detail during the development process. At the end of the day, however, the buyer of the vehicle will not be particularly interested in these values; what matters is the interior and exterior sound quality of the car. The target cascading on system-, sub-system- and component-level, based on the client's request, is still and will be also in the future a challenge for engineers. This article discusses current approaches and criteria.

2 NVH Target Values

The acoustic of the full vehicle is determined by the acoustic performance of a couple of systems in levels and the corresponding interaction. In **Figure 2** they are illustrated exemplarily. The powertrain (Level 2 in Figure 2) is part of the vehicle (Level 1 in Figure 2), composed of an engine and a gearbox (Level 3 in Figure 2), which then consists of individual components etc. In the final analysis, the effect of most components can be divided into excitation and transfer of noise and vibration (Level 6 in Figure 2). However, there are interdependencies between the different components and between the different levels. Therefore, a traffic-light system is used here to indicate non-critical (green) or critical (red) conditions with amber indicating that a certain constellation might become critical. For this assessment empirical values, databases and scatter bands as well as some new approaches for analysis are used.

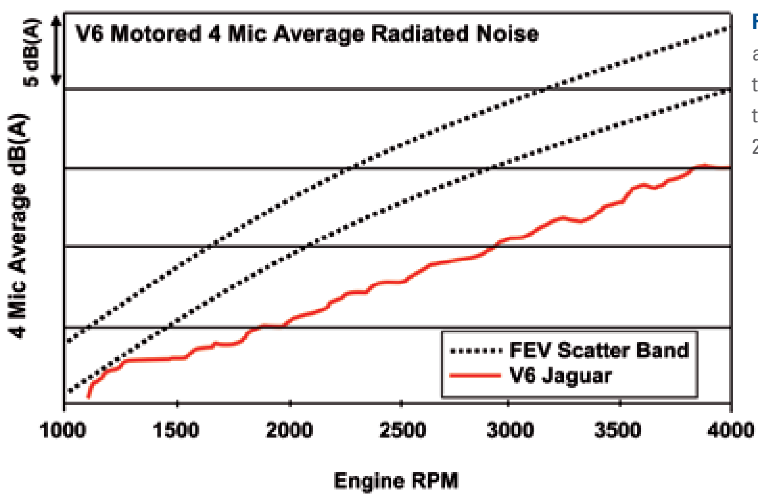

Figure 1: Overall level powertrain noise of the Jaguar V6 2,7 l DI engine

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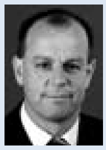

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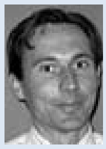

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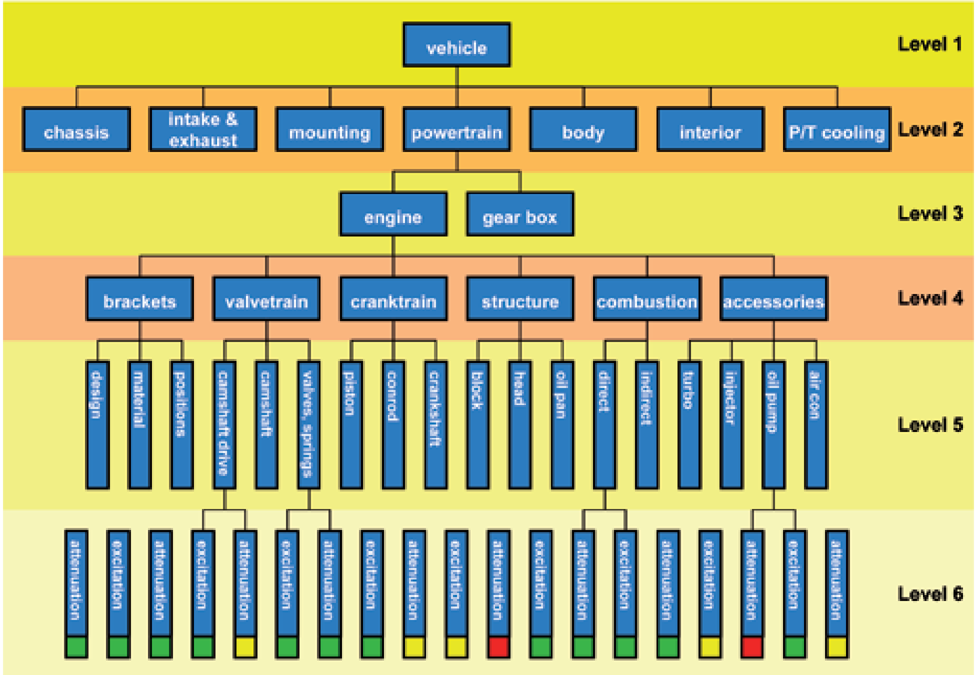

Figure 2: NVH targets from the vehicle-level to component-level

2.1 NVH Target Values for the Vehicle

One target for the vehicle is to comply with legal requirements, the pass-by noise. However, more significant for the development process are customer expectations. The target value for the total vehicle is defined on the basis of the marketing department specification and benchmark data like scatter bands. Characteristic variables would be noise level or articulation index. But also subjective evaluations are used for the vehicle. In this context it has to be mentioned, that the use of objective metrics, which are able to describe the subjective perception much better than the conventional metrics, becomes more and more popular. As examples parameter like Ticking, Knocking and Dynamic should be mentioned here.

To break down the overall vehicle targets into the Level 2 systems (e.g. power-train and body) the „Vehicle Interior Noise Simulation“ (VINS) is always used these days. It is possible to accurately establish the source of interior noise by analysing the different airborne-, mount- and body transfer functions [2]. By making a distinction between noise sources and transfer functions, different causes of noise can be linked to individual vehicle systems; it is then possible to derive target values for individual systems from the target values for the entire vehicle. Typical target values would be local stiffness, structure-borne and air-borne transfer functions for the body, mount stiffness and intake and exhaust system orifice noise.

A combination of this methodology with CAE-based data is also possible. Measured engine excitations can for example be combined with body transfer functions stemming from FE analyses to check on the body target values before the body is available as an actual component.

The vehicle-related methodology to arrive at system target values (Level 2 in Figure 2) is the same as the one used to determine engine noise target values (described above). Not only is the method used for power-train-induced interior noise, it has also been expanded to cover rolling noise (Chassis VINS) and vehicle exterior noise (VENS).

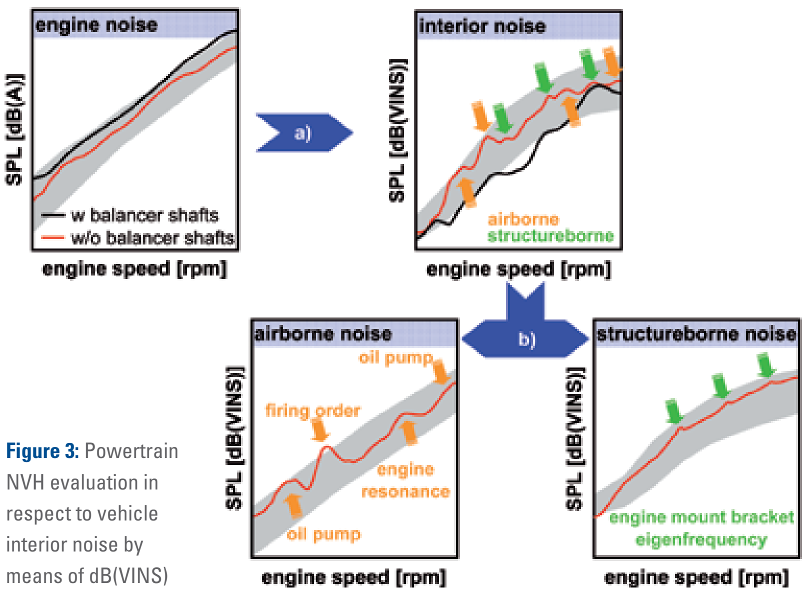

Figure 3: Powertrain NVH evaluation in respect to vehicle interior noise by means of dB(VINS)

The target values have to be challenged continuously throughout the entire development process in line with the maturity level of vehicle development so as to avoid exaggerated acoustic requirements and thus disadvantages with regard to costs, weight and package.

2.2 NVH Target Values for the Powertrain

Today noise and vibration levels are typical target values for the global assessment of an engine, even if they do not allow for an unfailing statement regarding the expected vehicle interior noise quality. For example: the noise level of an engine with balancer shafts is a few decibels higher than that of an engine without such shafts, **Figure 3**, but if you investigate the vibration excitation in line with the firing order, the balancer shafts clearly turn out to be advantageous. The question now is whether balancer shafts are needed or not.

To provide an answer, FEV uses a standardised interior noise simulation tool called dB(VINS). The typical frequency dependent A-weighting is replaced by the word "VINS", because the transfer behaviour of the vehicle is considered in the evaluation. The noise and vibration measurements on the engine are then – in line with the established VINS of the real vehicle – combined to form the interior noise level value. The result and the answer to the above mentioned question is presented in the Figure 3 on the right. In this example the balancer shafts clearly improve the vehicle interior noise, whereas just looking on the radiated powertrain noise, the balancer shafts have a negative influence.

Moreover, analysis of the individual excitations allows us to establish what actually determines interior noise levels. It is possible to distinguish between air-borne and structure-borne components of the excitation. Significant findings can then be traced back to individual engine sides and/or engine mount brackets. In this example there is a clear explanation for the interior noise curve peaks, and they can be allocated to specific components. It is now possible to specifically and effectively optimise the interior sound pattern.

For the evaluation of an engine the engineer will nowadays increasingly focus on objective parameters to describe relevant disturbing noise. This is particularly true in cases of diesel knock, injector ticking and turbocharger and timing drive whine.

3 Combustion-induced Noise Excitation

At FEV combustion-induced noise excitation on the basis of measured cylinder pressure is analysed by calculating the combustion sound level (CSL). For this calculation the excitation is broken down into four different causes of noise. The individual noise components are determined through the combination of excitation-producing variables together with the related average structure evaluation functions at the beginning of development or engine-specific structure evaluation functions as they become known. This results in the engine noise level CSL. The CSL, in combination with the body transfer behaviour, will then provide the vehicle interior noise level „CSL(VINS)“, **Figure 4**.

Already during the development of a new combustion process on a single-cylinder test engine this computation will, for example, provide the engineer with an idea of whether the excitation due to combustion pressure development could later become critical for engine and interior noise levels and how much of a problem this would pose. Any calibration combination can be evaluated and analysed. It becomes possible to evaluate the combustion noise excitation (Level 6 in Figure 2) with a view to its relevance for the noise level inside the vehicle (Level 1 in Figure 2). However, in order achieve the desired sound quality inside the vehicle it is not only necessary to limit the combustion-induced noise excitation to target values but also the transfer behaviour of both engine and body.

4 Engine Structure Transfer Patterns

Besides excitation and noise radiation, structure transfer behaviour is an important element in the noise generation chain and should therefore be at the centre of the engineer's attention at the very beginning of the development process. In this context the NVH target value definition needs to be broken down in a virtual environment into values for individual components since any development process will start with the design of individual components [3].

Structure transfer patterns are investigated on the basis of FEM (Finite Element Method). In addition, FEV has developed „DIRA“ (Dynamic Impact Response Analysis) to quickly analyse and evaluate housing structures. The structure-borne surface velocities of individual components, which are a measure for noise radiation, are calculated on the basis of Forced Response Computations and characteristic excitations. The resultant structure transfer pattern is then compared to

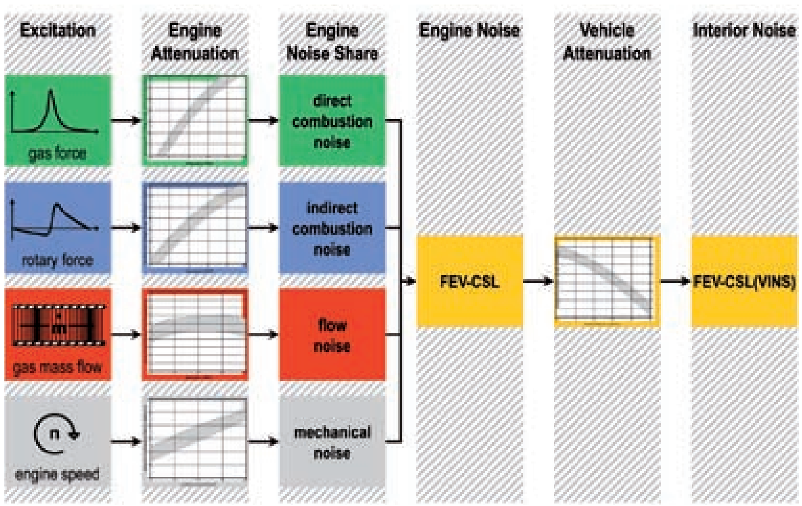

Figure 4: From cylinder pressure to powertrain radiated noise via CSL (Combustion Sound Level) up to the vehicle interior noise dB(Vins)

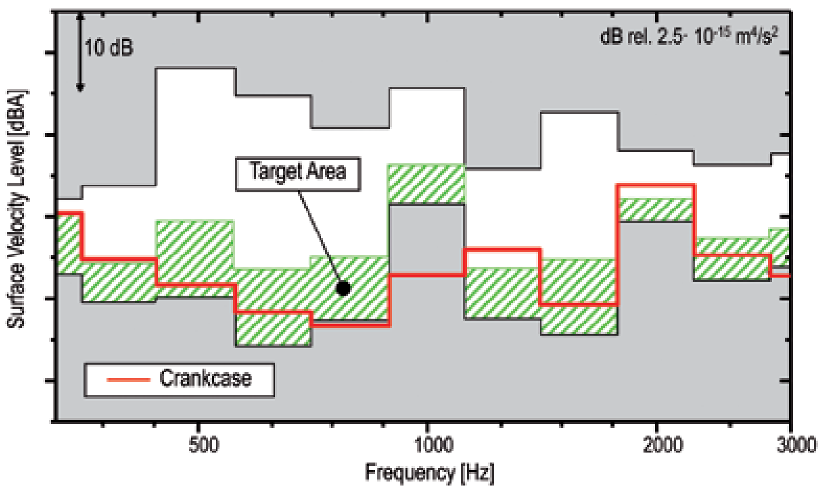

Figure 5: Target definition of the dynamic structure

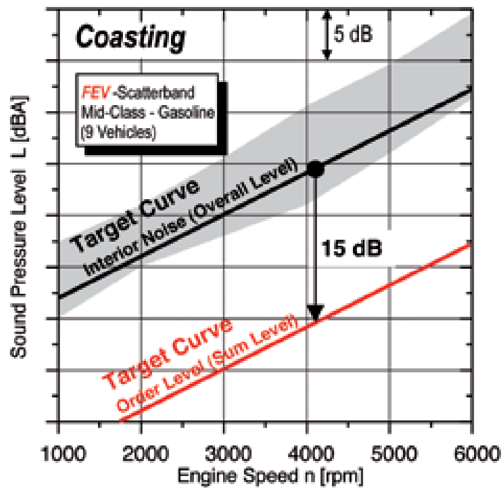

Figure 6: Interior noise relevant transmission target definition

target values derived from state-of-the-art structures, Figure 5. This enables targeted adaptation of the structure transfer behaviour of individual components with a view to frequency content and structure-borne noise levels. Alongside the evaluation of structure-borne surface velocities, characteristic eigenfrequencies will be judged similarly.

In line with the development process the evaluation of individual components will be followed by a final analysis of the total powertrain. The evaluation is again based on the state-of-the-art target values. Excitation functions derived from Multi Body Simulation (MBS) computation or test bench measurements will be used for Forced Response Computations resulting in realistic structure-borne surface velocity levels. On this basis Boundary Element Method (BEM) and FEV's proprietary virtual vehicle interior noise simulation „V-VINS“ even make it possible to calculate interior noise levels and compare them to auditory results.

5 Transmission Gearbox

In the following section the target value is defined for an experimental investigation of individual (sub-)systems using the example of a transmission gearbox. The gearbox, pre-optimised with FEM and MBS analyses, will be acoustically analysed in the next development step and evaluated on a components test rig. The target values to be defined for the components test rig need to take into account the impact exerted by the engine and the effects of the vehicle transfer functions for the engineer to be able to specifically optimise the gearbox with respect to vehicle interior sound quality.

The dB(VINS) evaluation is used for this purpose as well. The gearbox interior noise share, is then correlated with the total level of vehicle interior noise to identify critical gearbox noise. For example, tonal gearbox noise is correlated with a target curve for the overall state-of-the-art value for vehicle interior noise. A frequently used criterion is the minimum distance of 15 dB between the overall vehicle interior noise level and the interior noise share of the gear meshing orders of the transmission, Figure 6. If the order level exceeds the target curve, engagement geometries and shaft or housing stiffness require optimisation.

6 Summary

NVH target values for an engine should be defined to ensure that the vehicle will later meet all requirements concerning interior and exterior sound qualities. However, the noise level measured at the engine test bench, which is partly defined as a central target value, often is of somewhat lesser significance for the vehicle. A more vehicle-specific target value definition promises reduced development time and cost savings in the overall development process and, at the same time, acoustically more favourable and more cost-efficient vehicles. For this purpose, FEV develops methods, scatter bands and criteria that allow such a target value definition “from cylinder pressure right through to the driver's ear” to be carried out in a customer-specific fashion. Current approaches and criteria have been discussed in this article. Methods such as FEM/MBS, CSL and VINS are combined so as to arrive at conclusions and limit values for the design of individual components and their contribution to the entire vehicle's noise and vibration behaviour. Today this is possible for a few, though not all, systems and components; it will probably never be possible to fully assess all components during the development process. However, the development of vehicle-specific target value definitions will certainly continue over the next few years.

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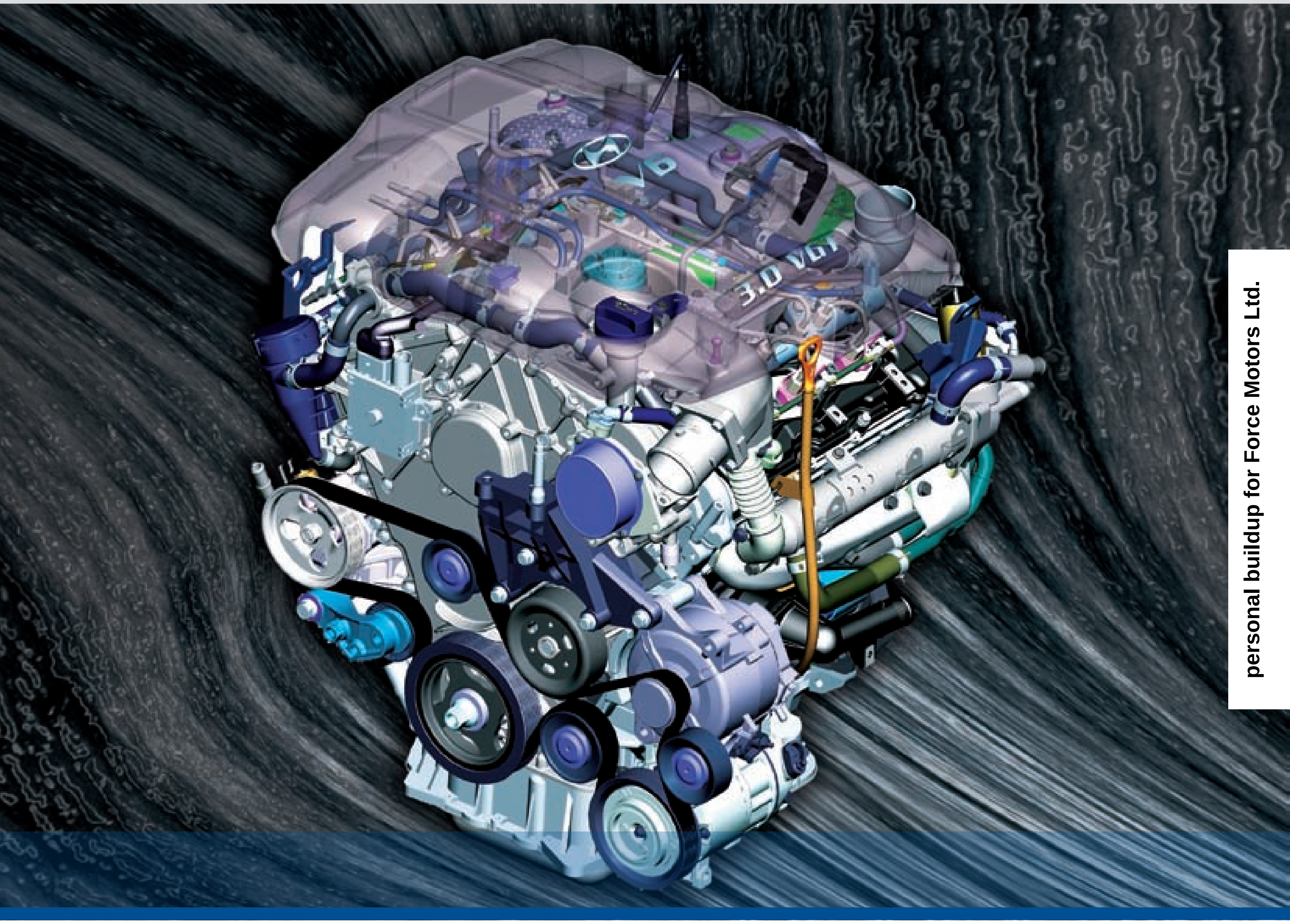

New V6 3.0 l Diesel Engine
for Hyundai/Kia's SUVs

To strengthen the appeal of Hyundai/Kia's premium SUVs, a new V6 3 l diesel engine has been developed. Its combustion system provides both high performance for driving comfort and potential for Euro 4 emission standard. This engine provides a foundation for coping with the next generation Euro 5 emission standards by adopting advanced combustion and after-treatment systems while development work has already started on the application of urea injection NO_x after-treatment system in order to meet the US Tier 2 Bin 5 emission standard.

1 Introduction

Since the first modern high speed direct injection diesel engines for passenger vehicles were introduced in the 90s, the line-up and applications have been continuously expanded. In the early days of passenger diesel engines their displacement volumes were mostly around the 2 l mark that is, 0.5 l per cylinder mainly due to several technical issues.

But the evolution of fuel injection systems has made it feasible to produce a smaller injection hole with a higher conical angle and to arrange multiple holes with an even distribution, while the technical progress of turbochargers also allows higher efficiency with a smaller wheel size. These are the main factors which have driven the expansion of the passenger diesel line-up from very small three-cylinder 0.8 l engines to eight-cylinder 4.8 l engines.

1.1 Development Concept

For most OEMs, six-cylinder engines with a displacement volume in the vicinity of 3 l are their flagship engines installed in large SUVs and luxury cars and demonstrate their technical sophistication by adopting state-of-the-art technologies. The main development concept is:

- deliver competitive power output
- provide competitive torque over a wide engine speed range for excellent drivability
- optimised noise, vibration, harshness (NVH) characteristics
- competitive fuel consumption.

The engine was developed for two different vehicles namely the Hyundai iX55 and Kia Mohave. For engine mounting, these vehicles have different layouts: The iX55 engine is mounted in a transverse direction while in the case of the Mohave the engine is longitudinally mounted. This leads to distinctive power output and torque characteristics due to a slight variation in their air intake systems and torque capacities of their transmissions.

1.2 Engine Specifications

Torque and power output are slightly different for Mohave and iX55 due to the aforementioned differences in engine mounting and breathing system layout. In the Mohave, the V6 3 l engine delivers a maximum power output of 184 kW at 3800 rpm and maximum torque of 540 Nm through a wide engine speed ranging from 1800 rpm through 2750 rpm. Although power output is slightly reduced for iX55, this engine generates 176 kW of maximum power at 3800 rpm and provides nearly flat 451 Nm of torque from 1750 rpm to 3500 rpm, **Table**.

A piezo common rail diesel injection system with a maximum system pressure of 1600 bar was implemented together with a variable geometry turbocharger controlled by an electronic actuator. The selection of compacted graphite iron (CGI) over conventional cast iron for the main block proved to be effective in increasing power output and reducing weight thanks to CGI's superior tensile and fatigue strength. The compression ratio was adjusted to 17.3, which was the optimal compromise between two contradictory requirements: low emissions versus acceptable startability and combustion stability under cold conditions. An instantaneous start system (ISS) was applied to guarantee good starting under cold conditions. ISS accelerates the heat-

Table: Engine specifications

Engine Type		V-Type six-cylinder
Valve Train		four-valve DOHC
Displacement Volume	cc	2959
Power	kW / rpm	184 / 3800
Torque	Nm / rpm	540 / 1800 ~ 2750
FIE System		Bosch CRI 3.0 Piezo
Aftertreatment System		DOC (DPF as Option)
Emission		Euro 4

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up of glow plugs and optimises power control for glowing.

Although this engine demonstrates sufficient potential to comply with Euro 4 emission regulations using an oxidation catalyst only, a Diesel Particular Filter (DPF) will be provided as optional package thereby further reducing particulate matter to the levels lower than Euro 5 emission standard.

2 Main Features of Engine Design and Calibration

2.1 Main Features of Engine Hardware

In addition to competitive power output, fuel consumption and NVH characteristics, several technical features were adopted to guarantee sufficient potential for compliance with Euro 4 emission standards, **Figure 1**.

The combination of piezo common rail system and electronically actuated variable geometry turbocharger supported by a refined calibration strategy were the main source of the engine potential. A cooled Exhaust Gas Recirculation (EGR) system controlled by an electronic EGR valve generally accepted as standard emission package for Euro 4 regulation was adopted. A variable swirl system was used for improving emission potential at low speed and low load operating conditions.

Other technical features included CGI cylinder block that was beneficial for improved structural strength and compact design. The engine adopted a timing chain system for reliability and bed plate structure for enhanced NVH characteristics. An ISS glow system ensured good startability and combustion stability under cold condition through a faster heat up and optimised temperature control during the post-glowing period.

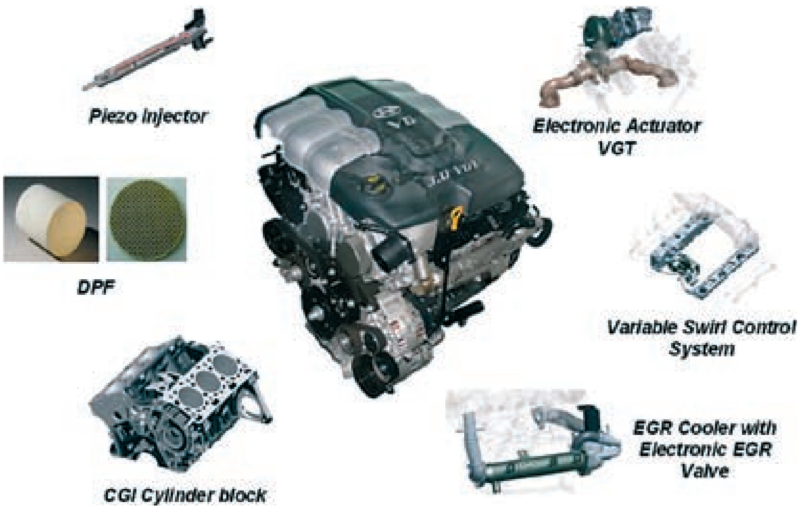


Figure 1: Main technical features of V6 3 l CRDi engine

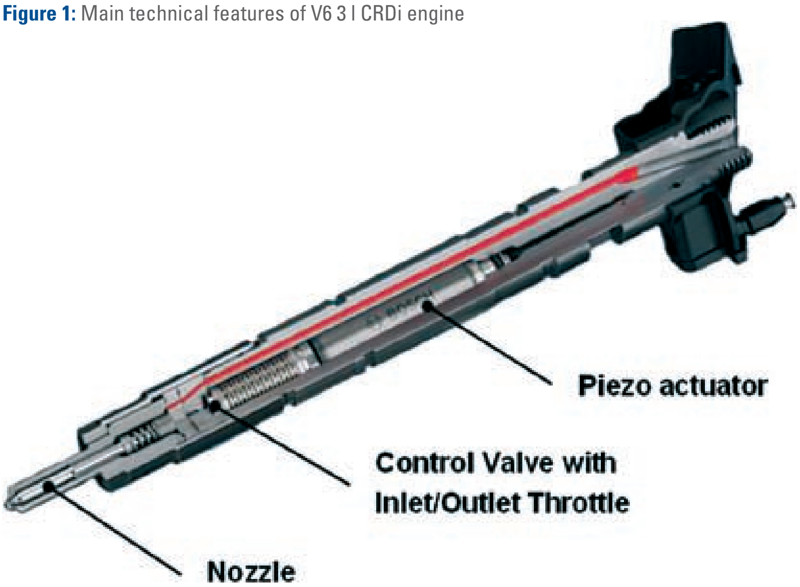


Figure 2: Piezo Common Rail injector

2.2 FIE System and Calibration Strategy

A piezo common rail system with 1600 bar capability was proven to have far more control precision and flexibility over a solenoid system by enabling more precise control of small injection amounts and allowing shorter separation between adjacent injections, **Figure 2**. These characteristics were effective in achieving higher emission and acoustic potential by precisely controlling the pilot injection amount and separation which have a strong impact on smoke and acoustic behaviour. An elaborated injection control strategy was implemented to fully utilise the potential of the fuel injection system according to operating conditions. The optimisation of numerous control parameters for the injection amount and timing was done by Design of Experiment (DoE) technology from the viewpoint of emission, fuel consumption and acoustics.

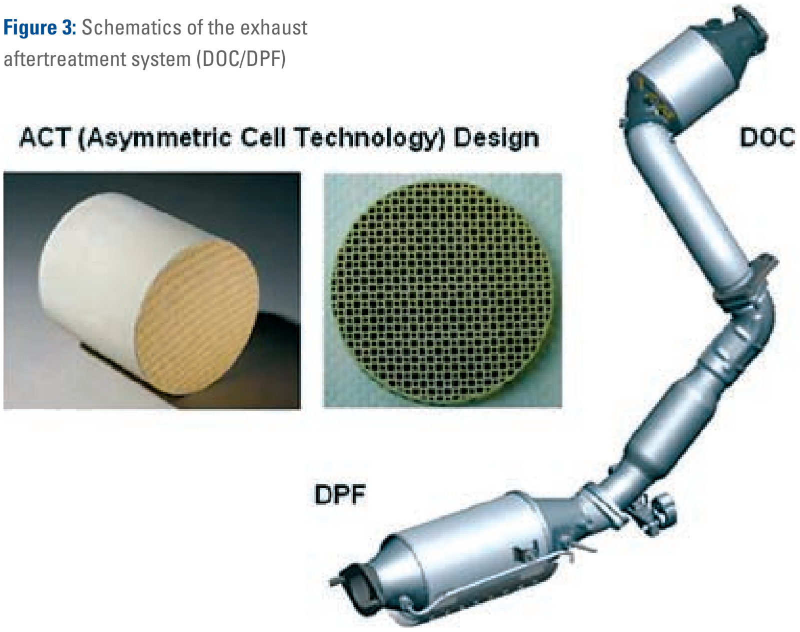
The focus of the injection control strategy was to secure enough emission potential while keeping a competitive acoustic level, and this strategy was reflected in the application of double pilot injection to a wide operating range covering most of the emission cycle. The degradation of smoke and fuel consumption due to double pilot injection was compensated by an elaborated calibration of other parameters. The operating range of single pilot injection was also extended within the system reliability limit.

Pilot injection coordination was another important parameter to be calibrated to achieve optimal compromise among emission, fuel consumption and acoustics. Double pilot injection was applied to a wider operating range for the lower gear group where the combustion noise is more apparent due to lower vehicle speed, while the range was reduced for the higher gear group where the combustion noise is concealed by tire or wind noise.

2.3 Aftertreatment System

Although the V6 3 l engine mated to an automatic gearbox has demonstrated

Figure 3: Schematics of the exhaust aftertreatment system (DOC/DPF)



sufficient potential for Euro 4 emission standard using a diesel oxidation catalyst (DOC) only, a diesel particulate filter is provided as an optional package and further reduction of particulate matter (PM) below 5 mg/km is possible.

Dura Trap Aluminium-Titanate substrate supplied by Corning and featuring improved asymmetric cell technology (ACT) design was adopted to increase the ash loading capacity. It was coated with precious metal to enhance the continuously regenerating trap (CRT) effect thereby reducing trapped soot under normal driving conditions.

The DOC and DPF were canned separately and the DOC was located at a close coupled position just downstream from the turbocharger to improve carbon monoxide (CO) and hydrocarbon (HC) emissions resulting in a shortened light off time while the DPF was located at an underfloor position, **Figure 3**.

Intensive efforts were focused on calibrating the active regeneration of the DPF and estimating the trapped soot mass inside the DPF. Parameters related to DPF regeneration were optimised from the view point of system reliability and regeneration efficiency. The target regeneration temperature was carefully selected considering regeneration efficiency and maximum temperature reached under severe conditions where the operating condition is changed to idle or overrun conditions during the regeneration process.

Estimation of the trapped soot mass was based on the pressure difference across the DPF substrate. But simulation-based soot mass estimation was also applied to operating conditions where the deviation between actual and estimated soot mass was excessive. The deviation was augmented with the exhaust gas flow rate, encountered below a certain threshold value, and a similar deviation was also observed once the CRT or thermal regeneration caused by occurrences of high exhaust gas temperature and for such operating conditions simulation-based soot mass estimation was applied.

To assure safe and efficient regeneration with minimised deterioration of fuel consumption and oil dilution, regeneration coordinator was carefully calibrated. To prevent the failure of the DPF due to excessively high regeneration temperatures mostly caused by too much soot loading, several parameters except pressure difference-based soot mass were considered as the trigger for regeneration. These parameters included mileage, engine running time and fuel consumption and the regeneration was triggered once the threshold for any of these parameters was reached.

For compliance with the Euro 5 emission standard, a combined DOC and DPF in a single can located at a close coupled position is under development. This will achieve an enhanced CRT effect and more favourable regeneration calibration.

2.4 Turbocharger

A variable geometry turbocharger was implemented to ensure the specific power output potential (as high as 62 kW/l) and an electronic actuator was one of the key technologies for achieving enhanced re-

Figure 4: Variable geometry turbocharger with electronic actuator

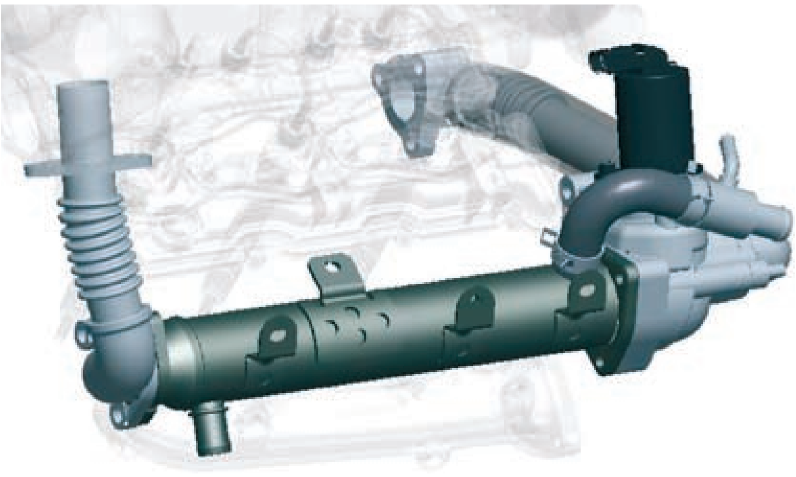


Figure 5: Electronically controlled cooled EGR system

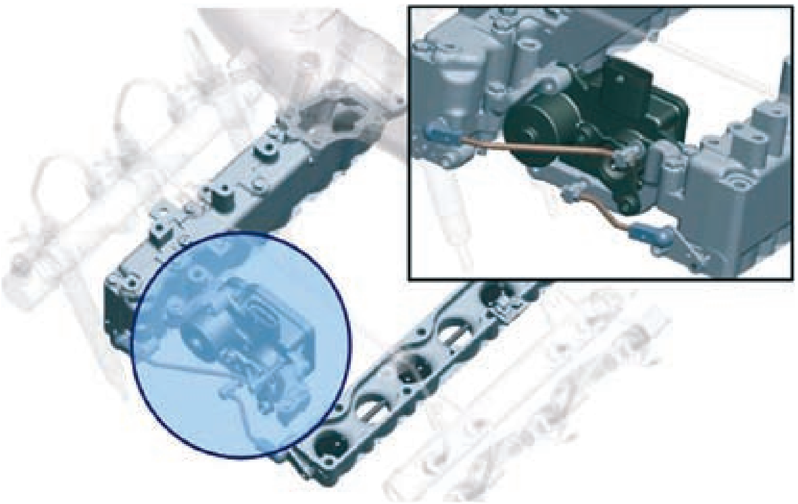


Figure 6: Variable swirl control system

sponse and control accuracy, **Figure 4**. Compared with a conventional vacuum actuator system the electronic actuator provided distinctive advantages in response and control accuracy. It has demonstrated more accurate boost pressure control and faster boost pressure build-up characteristics under transient operating conditions. In addition, the electronic actuator also reduced the hysteresis in the boost pressure defined as the gap in boost pressure trace between vane opening and closing. All these improvements have contributed for better vehicle performance and reduced emission scattering.

2.5 Exhaust Gas Recirculation System

A cooled EGR system controlled by a linear solenoid type EGR valve, **Figure 5**, considered as a standard package for Euro 4 emission compliance, was also used and the EGR valve was located upstream of the cooler to prevent fouling. The shell and a tube-type EGR cooler were installed and the surface of the inner tube was corrugated to enhance the efficiency of the heat exchange between the exhaust gas and cooling water. The location and shape of the EGR port where the exhaust gas was introduced in the stream of fresh air was optimised within the limits of the package layout using three-dimensional flow analysis.

By-pass functionality is not mandatory for the Euro 4 emission standard because the EGR requirement is not critical from the viewpoint of CO and HC emissions. But for Euro 5 system by-pass functionality will be implemented to cope with CO and HC especially under cold conditions. The total system will be investigated from the viewpoint of pursuing the most cost effective system and considering the most relevant systems such as the EGR system, precious metal loading of the catalyst and degradation of fuel consumption for compensating emission penalty.

2.6 Variable Swirl System

A variable swirl control system was an effective measure in improving emission potential by reducing PM especially at low speeds and low load operating conditions where effective swirl and air density were insufficient for efficient combustion. This system was also beneficial for increasing power output through an optimised intake port design. With variable swirl system the required in-bowl swirl level from the emission point of view could be reached with a rather lower port swirl level. And reduced port swirl requirement in most cases provided favourable boundary conditions in the port design toward higher flow coefficient that is clearly beneficial for higher power output.

Although the variable swirl system really worked for reducing emission, it caused a fuel consumption penalty mainly due to the increased pumping loss. The operating range and closing angle of the swirl control valve were carefully calibrated and verified to enhance the emission potential with minimal penalties in fuel consumption. Swirl control valves were installed in the helical ports on both banks and these valves were operated simultaneously by one electric motor through a dual linkage system, **Figure 6**.

2.7 CGI Cylinder Block

Compacted graphite iron material was used for the cylinder block, which was beneficial in improving the performance potential and reducing weight due to its enhanced properties over conventional cast iron, **Figure 7**. Since this material demonstrated far higher tensile and fatigue strength than cast iron, power output could be improved by applying higher peak firing pressure and weight reduction was also possible by the compact design while retaining the same stress relevant boundary conditions. The bed plate and four-bolts per journal structure were also introduced to enhance the structural

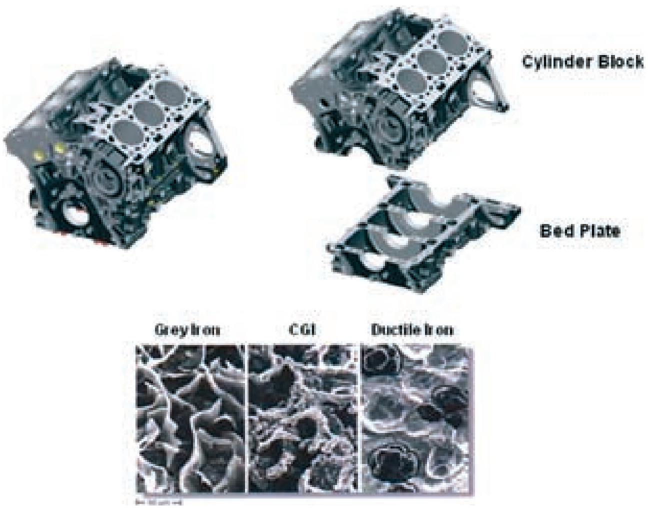


Figure 7: CGI cylinder block and bed plate structure

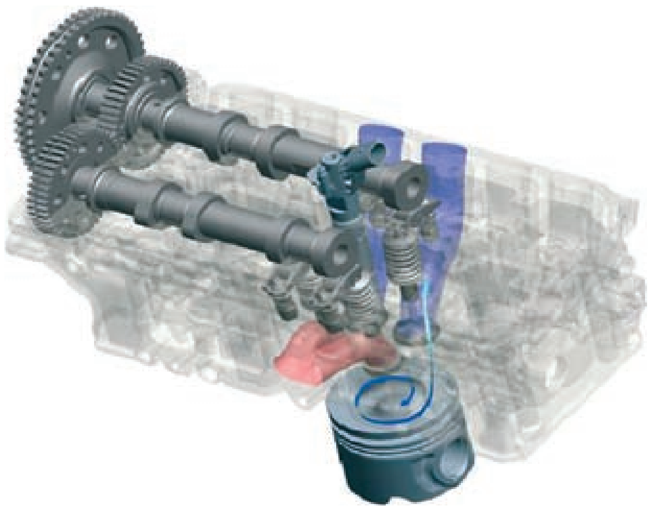


Figure 8: Port and combustion system layout

strength and acoustic characteristics of the cylinder block, and intensive structural analysis was performed to increase the natural frequency.

2.8 Port and Combustion System

Parallel port design was adopted where the connecting line between the intake valve centres was parallel to the centre line of the cylinder head since this port configuration was beneficial for structural strength and cooling of the cylinder head. For full utilisation of the high pressure capability of the piezo common rail fuel injection system, a greater emphasis was placed on a high flow coefficient rather than high port swirl capability and this parallel port configuration fit this requirement much better than a conventional skewed port configuration.

Intake ports adopted a single-helical and single-tangential configuration and the variable swirl control valve was located in the helical port. The seat swirl chamfer was introduced to enhance the port swirl during the initial stage of the intake valve opening and this was proven to be effective in improving emission characteristics.

The combustion bowl shape and spray patterns including number of holes and spray angle were matched with intake port swirl from the viewpoint of performance and emission potential. The wide bowl shape was matched with a relatively big spray angle to balance the high and low end torque and utilise the improved potential of the injection system, **Figure 8**.

3 Engine and Vehicle Performance

3.1 Engine Performance

Based on the aforementioned design features and refined calibration, the S diesel engine provides impressive power output, emission and fuel consumption potential for Hyundai/Kia's SUVs iX55 and Mohave. For Mohave, it delivers 184 kW at 3800 rpm corresponding to the specific power output of 62 kW/l, which is one of the highest in the competitive set. And it also generates a maximum torque of 540 Nm that is equivalent to the brake mean effective pressure (BMEP) of 23 bars through a wide engine speed ranging from 1800 rpm to 2750 rpm.

Balance between maximum power output and low end torque was another point on which combustion hardware system matching and elaborated calibration work were focused. In parallel with the highly competitive maximum power output, BMEP at 1000 rpm reached 10.4 bars and more than 80 % of maximum torque was reached at 1500 rpm with acceptable smoke level, **Figure 9**. Most of all, almost flat torque was stretched from 1800 rpm to 2750 rpm. This balance between power output and low end torque together with widely stretched maximum torque were the main source of good drivability realised with both iX55 and Mohave.

3.2 Vehicle Performance

Spurred by the aforementioned power output combined with a refined selection of gear step ratios offered by the

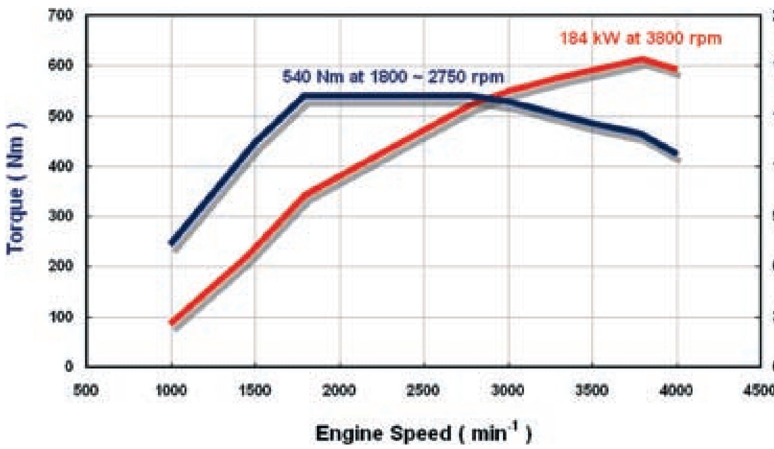


Figure 9: Power output and torque

matched six-speed automatic gear box, impressive drivability and fuel consumption and NVH characteristics were achieved. For Mohave elapsed time for acceleration from standing to 100 km/h and from 80 km/h to 120 km/h are 9.7 s and 6.0 s respectively, which are highly competitive against other SUVs with similar displacement engines, **Figure 10**.

4 NVH Optimisation

Much effort was also focused on NVH development. In parallel with the refined calibration of the injection parameters, the engine hardware was optimised for competitive NVH characteristics. The main sources of noise emissions were identified and appropriate measures were implemented to improve the overall acoustic behaviour.

To optimise the structure and NVH characteristics, the main components of the powertrain were subjected to structural analysis. Based on this modal analysis, reinforcement was undertaken by applying and re-locating ribs to the engine and gearbox components such as the chain cover, block, head and oil pan and by this process of structural optimisation the bending stiffness of the powertrain was increased by 27 %.

The compatible dynamic stiffness of the powertrain was achieved by introducing a bed plate structure and bell shaped matching plane between the engine and the gear box combined with improved acoustic behaviour of the chain system, oil pan and other components, and competitive sound pressure levels were measured, **Figure 11**.

5 Summary

Hyundai/Kia has demonstrated its competence in the development of passenger diesel engines by launching the V6 3 l engine mounted in both companies' premium SUVs. This engine has proved itself to be competitive by delivering 184 kW maximum power output and 540 Nm maximum torque through wide engine speed range for Kia Mohave. Although this engine has demonstrated enough potential against Euro 4 emission standard only with oxidation catalyst particulate filter system will be provided as optional package to cope with the requirements from the market.

6 Outlook

This engine will be the basis for the development of next generation engine aiming Euro 5 emission standard and it is expected that relative small improvements in combustion and aftertreatment systems will be enough for this target. The improvements in combustion systems will include reduced compression ratio with appropriate countermeasure against degradation in cold startability and HC, CO emissions, enhanced fuel injection system with higher system pressure and increased nozzle hole numbers and EGR cooler with enhanced cooling capacity and by-pass functionality.

The aftertreatment system for Euro 5 emission standard will keep the same strategy as Euro 4, but the position of combined DOC and DPF will be located at close coupled position for favourable temperature and regeneration condition. Active NO_x aftertreatment system equipped with urea injecting SCR systems is currently under development for coping with US Tier 2 Bin 5 emission standard. ■

Figure 10: Vehicle performance of the Mohave

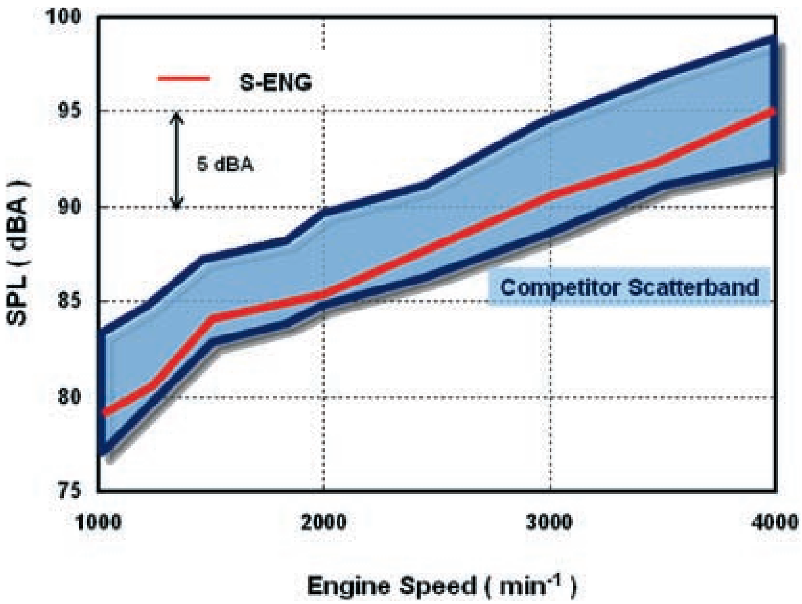


Figure 11: Sound pressure level (full load)

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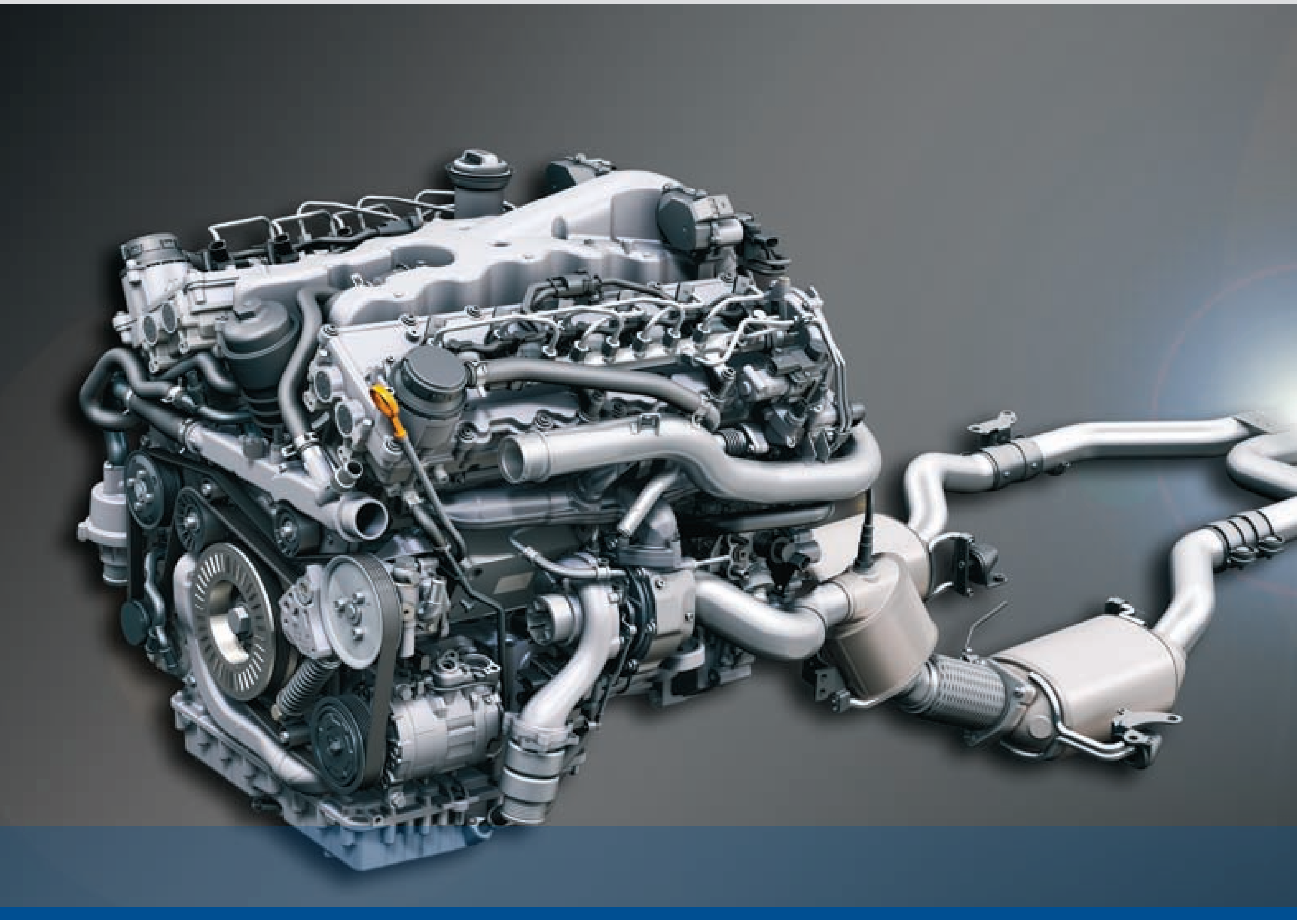

The Audi 6.0 l V12 TDI

Part 2 – Thermodynamics, Application and Exhaust Treatment

This second part of this paper sets out the focus of development in the fields of thermodynamics, application and exhaust treatment of the new Audi 6.0 l V12 TDI engine for the Audi Q7. The first part in the MTZ 10/2008 dealt with the design and mechanical construction for this powerful twelve-cylinder car engine. The new V12 diesel engine delivers a maximum power output of 368 kW at 3750 rpm and the extremely high torque of 1000 Nm in the engine speed range from 1750 to 3250 rpm. The engine is employed in the Audi Q7 in conjunction with a six-speed automatic gearbox and quattro drive. The complete vehicle was developed by quattro GmbH.

1 Introduction

Audi is able to call upon a wealth of experience with its familiar V6 and V8 TDI engines, several generations of which have been employed in a variety of VW Group models [2, 3]. The development of a V12 can be seen as rounding off the top end of the V-TDI engine range and as a production spin-off from the successful TDI racing engine fitted in the R10 [4]. The development process was able to utilize many synergies with the Audi V-configuration engine family.

The challenge in developing the V12 TDI engine was to combine a number of highly ambitious goals: to attain the lowest emissions; the best possible fuel economy; the best acoustics; a maximum torque of 1000 Nm across a very wide engine speed range; and not least, a nominal power output of 368 kW. Pleasingly, those ambitious goals were achieved in the existing production Audi Q7 without major modifications to the vehicle.

2 Combustion Process

The combustion process selected for the V12 engine is of course closely based on that applied in the smaller V6 and V8 engines, which represent a very high level of development [5]. The design features two intake ports per cylinder, of which one can be closed steplessly by means of flaps built into the inlet module in order to generate higher swirl in the partial load range for better emissions. In the new three-part cylinder head design the intake and exhaust ports are on the bottom deck, and so necessarily run somewhat flatter than on the V6 or V8 engine with the conventional head design. The ports were carefully developed to ensure that the same level of swirl is attained in the partial load range as on the V8 engine, while at the same time providing for high flow rates under full load.

Relative to the V6 and V8 TDI engine, the compression ratio was reduced, finally to 16.0. This compression ratio proved to be the lowest possible compromise between power demand, emissions, and cold-starting capability. The main dimensions of the selected piston bowl are shown in **Figure 1**. With a maximum diameter of 52 mm, it has a turbulence ring diameter of 48 mm. The recess depth is 15.1 mm.

The combustion chamber described was combined with an eight-jet injector nozzle, representing the best compromise between full load power and emissions. The resultant combustion process delivers outstanding EGR compatibility as a result of high partial load swirl, and thus minimal emissions. The low particulates level ensures long DPF regeneration intervals.

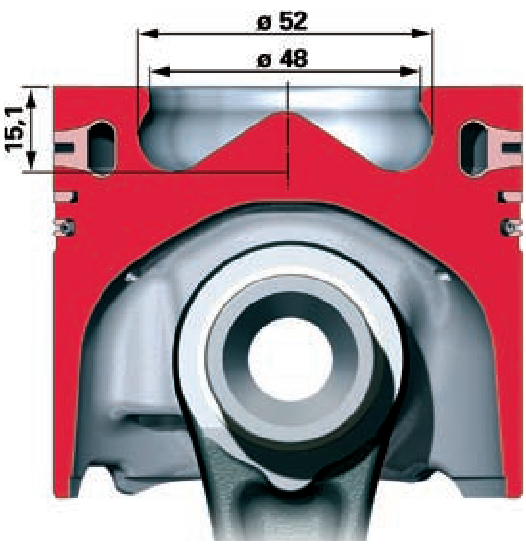

Figure 1: Main dimensions of the piston bowl

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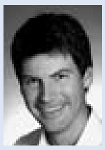

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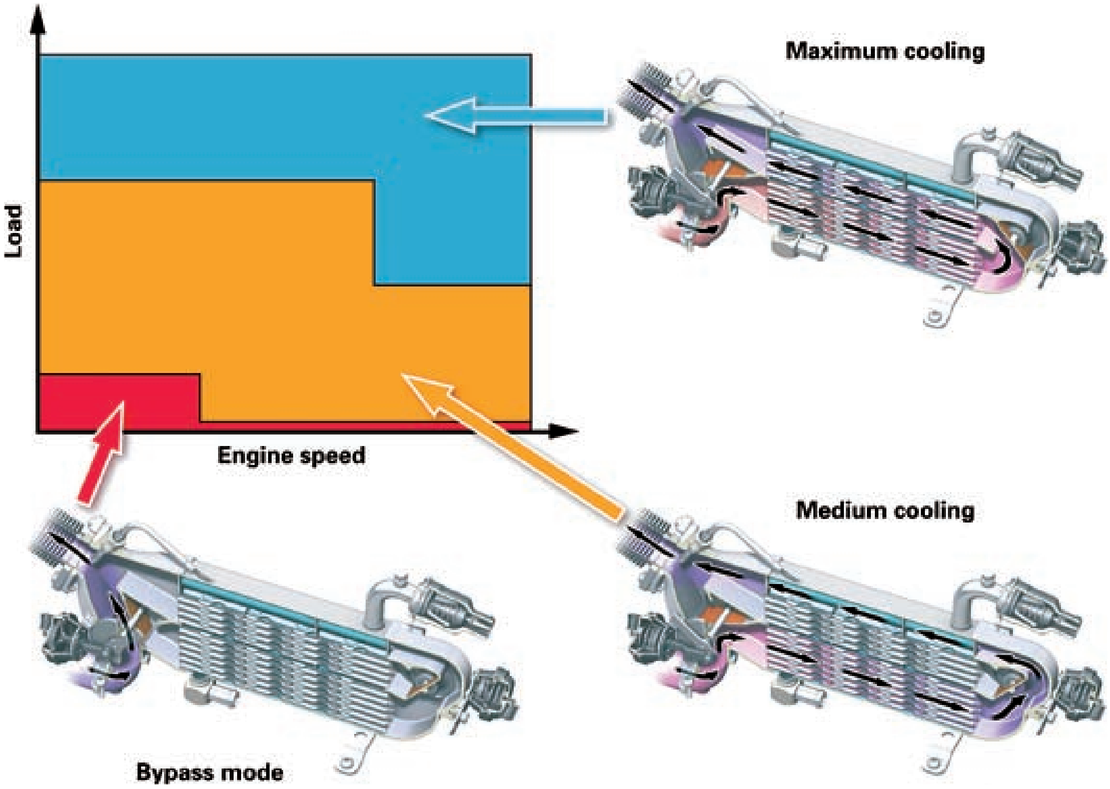


Figure 2: EGR cooling stages

3 Exhaust Gas Recirculation

The three-stage cooling of the recirculated exhaust gases (EGR) described in the first part of this paper enables the cooling power to be optimized in the various operating states of the V12 TDI engine on the road, **Figure 2**. When the engine is cold and under extremely low loads, when it is vital, to minimize emissions and consumption, to warm the engine up to operating temperature as soon as possible and attain the catalytic converter light-off temperature at an early stage, no cooling of the EGR takes place and the cooler is bypassed by opening the bypass valve.

When the warmed-up engine is operating under low load, the bypass valve is closed. The exhaust gas flows through only part of the cooler matrix under the control of the cooling valve, and so is moderately cooled. Under higher loads the cooling valve is switched and maximum exhaust gas cooling is employed. The temperature of the recirculated gas is recorded by a sensor downstream of the EGR cooler.

In addition, particular attention was paid in designing the EGR system to ensure that the recirculated exhaust gas was distributed uniformly across the two cylinder banks and the individual cylinders. For this reason, the design of the EGR port into the inlet module downstream of the throttle valve was optimized at an early stage by means of CFD calculations.

Figure 3 shows, by way of example, the very good mixing of exhaust gas with fresh air inside the intake manifold for a partial load operating point at an engine speed of 1350 rpm and the resultant even distribution of the EGR across the intake ports of all twelve cylinders. The downstream improvement in homogenization of the inlet gas can be seen, starting from introduction of the EGR for both banks, here shown in red, into the fresh air, shown in blue.

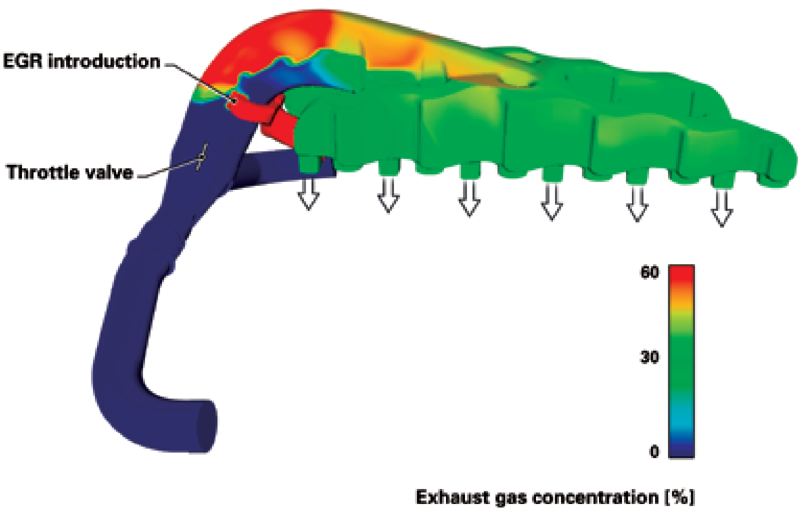
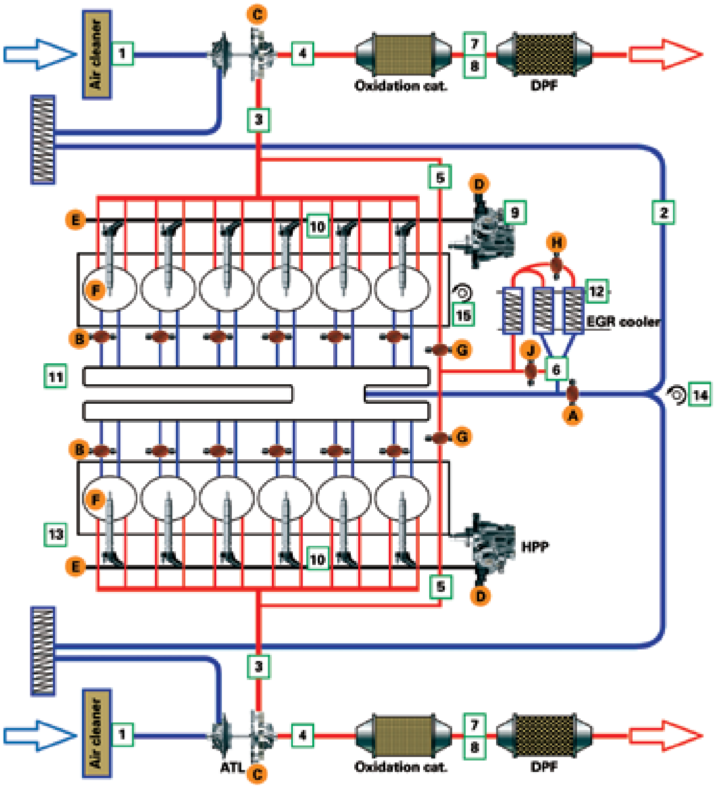


Figure 3: EGR uniform distribution



Sensor	Description	Amount
1	HFM = Hot-film mass sensor (for air flow)	2
2	Boost pressure / boost temperture combined	1
3	Exhaust gas temperature before turbine	2
4	Air/fuel-ratio sensor	2
5	Exhaust gas pressure before turbine	2
6	Exhaust gas temperature after EGR cooler	1
7	Exhaust gas temperature before particulates filter	2
8	Relative exhaust gas pressure before particulates filter	2
9	Fuel temperature	1
10	Fuel pressure	2
11	Coolant temperature	1
12	Coolant temperature before EGR cooler	1
13	Engine oil temperature	1
14	Engine speed	1
15	Camshaft phase	1

Actuator	Description	Amount
A	Throttle valve	1
B	Swirl control valve	2
C	Variable turbine gemoetry	2
D	High-pressure pump with throttled inlet	2
E	Pressure control valve	2
F	Injector	12
G	EGR valve	2
H	EGR cooler control valve	1
J	EGR cooler baypass valve	1

Figure 4: Engine schematic – sensors and actuators

4 Fuel Injection System

The third-generation common-rail fuel injection system, featuring a maximum rail pressure of 2000 bar, represents the latest state of development by Bosch. The fuel path is broken down as follows: Starting from the tank with its in-tank pump, with a pre-delivery pressure of 1.3 bar, the fuel is transported from the filter into the gear pumps, flanged on to the two high-pressure pumps, and then into the high-pressure pumps themselves. Each of the two pistons of the high-pressure pumps transports the fuel by way of a dedicated high-pressure line into the rail of the respective cylinder bank. The piezo inline injectors are connected by a high-pressure line with an inside diameter of 3 mm to the rails. There is no bank-to-bank connection of the rails. Consequently, pressure control is separate for the left and right sides, with independent control circuits for each bank comprising the high-pressure pump intake restrictor (MProp), the pressure regulating valve (PRV) on the rail and the rail pressure sensor.

A Bosch inline piezo injector is used. With a view to the emissions, the flow rate of the eight-hole nozzle with conically flow-optimized spray holes was kept relatively low. The nominal flow rate is 700 cm^3/60s at 100 bar and 0.7 mm needle lift. The disadvantage of the low flow rate in the upper full load range is compensated by the high injection pressure of up to 2000 bar.

5 Engine Management

Finding the optimum engine management system for the V12 TDI engine posed a major challenge for Audi and Bosch. The engine management system comprises two identical EDC17 engine control units (CP24) in a master/slave configuration, communicating via an internal CAN separate from the drive train CAN.

The key feature of the EDC17 family of control units is a 32-bit Metis processor operating at 150 MHz, with a 136 KB internal RAM and 2 MB internal flash memory. The external flash memory holds a further 2 MB. In terms of the application, this engine management system thus provides a storage capacity corresponding in this case to an application depth of more than 28,000 labels.

Each of the control units operates the six injectors of a cylinder bank and controls the cylinder bank rail pressure. For future applications, such as $DeNO_x$ systems based on selective catalytic reduction (SCR) or cylinder-pressure-based combustion control, this generation of control unit already features the appropriate input and output circuits.

6 Sensors and Actuators

The selected master/slave configuration offers the advantage of being able to distribute sensor and actuator connections across two control units. Nevertheless, the number of sensors and actuators used should be kept as small as possible for a number of reasons. The number of pins which can be accommodated on the control unit connectors is limited, and the cost of the sensors and actuators and of the associated circuits in the control unit and the wiring must also be considered. Moreover, all elements must be incorporated into the diagnostic system (EOBD), meaning further application costs. **Figure 4** shows a schematic view of the air and exhaust gas paths indicating

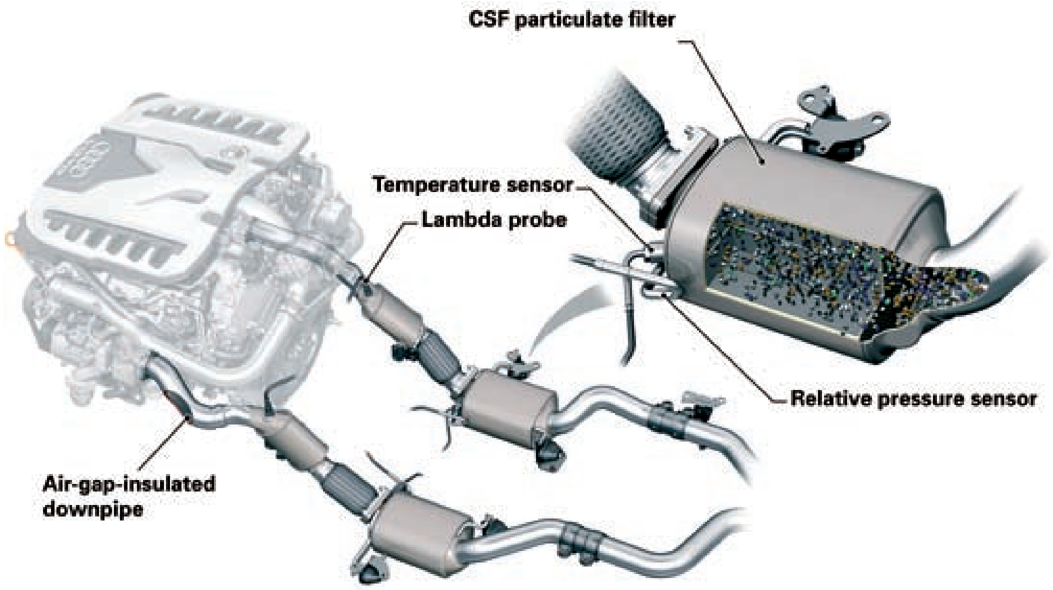


Figure 5: Exhaust system with DPF

the close-coupled sensors connected to the engine control units.

Keeping numbers to the essential minimum subject to the above premises, there is still need for a large number of sensors. One of the reasons for this is the dual-flow design of the intake and exhaust system. Another is the fact that the various models computed by the control units demand that a minimum number of input parameters be recorded so that variables not directly measurable can be determined with sufficient accuracy.

No additional pressure sensor was installed downstream of the throttle valve. Instead, a substitute value is determined from the throttle valve position and charge pressure sensor for inclusion in the filling model. As on the V8 TDI engine, in order to avoid turbocharger overspeed – such as at high altitudes – a model is employed which determines the current turbocharger speed from the air mass flows, the ambient pressure and temperature, the charge pressure and the charge air temperature. With regard to recording the engine speed, the inductive sensor used in the V8 TDI engine was replaced by a Hall sensor/multi-pole wheel system which delivers significance advantages in terms of accuracy and tolerances.

To protect the variable turbine geometry (VTG) against excessively high temperatures, a temperature sensor (T3) is fitted in the inlet of the exhaust turbines, and is used by a control system to provide the maximum engine output under all conditions – such as severe heat or high altitude – without exceeding the maximum permissible temperature of 830 °C. For the first time p3 pressure sensors are also used at this point as input variables for the air model. Figure 4 also shows the close-coupled actuators of the V12 TDI engine. An innovation relative to the V8 TDI engine is the EGR cooling valve which – as described – enables full cooling or only a portion of it to be applied, depending on load, engine speed and temperature.

7 Application

A new feature is the large number of models applied in the application process. Model-based control systems permit much faster response to changes in operating points and so deliver significant advantages in dynamic operation, both in terms of response to load demand and in terms of avoiding emission spikes. The fill and charge models (ASMod – Air System Model, MCC – Model based Engine Charge Control) register and regulate the interaction of the charge pressure control, EGR valve and throttle valve to provide operating-point-dependent control of the EGR rate. This greatly simplifies the application.

In parallel with the models, various adaptations and learning functions are executed in the control units during driving, for example to ensure a high accuracy of the injection quantity to ensure that emissions criteria are met over the lifetime of the vehicle. The many degrees of freedom offered by the injection system were balanced iteratively, based on the experience gathered from the V8 TDI engine. Automated processes on the engine test rig alone do not cover all relevant criteria. The engine sound, in particular, which ultimately can only be assessed inside the vehicle, necessitates recursions so as to provide optimum balance the rail pressure, the timing of the pilot injection, main injection, and the post-injection phases.

8 Exhaust Treatment

In designing the exhaust system, attention was paid to minimizing heat loss in order to rapidly reach the light-off temperature of the oxidation cat. This is all the more important in view of the fact that the specific loads under which the V12 TDI engine is operated in the MVEG test are very low. In view of the high targeted power output and considerations of fuel economy, efforts were also focussed on minimizing the exhaust backpressure.

The schematic in **Figure 5** shows the dual exhaust system of the V12 TDI engine. The exhaust gas is transported separately for each bank through the

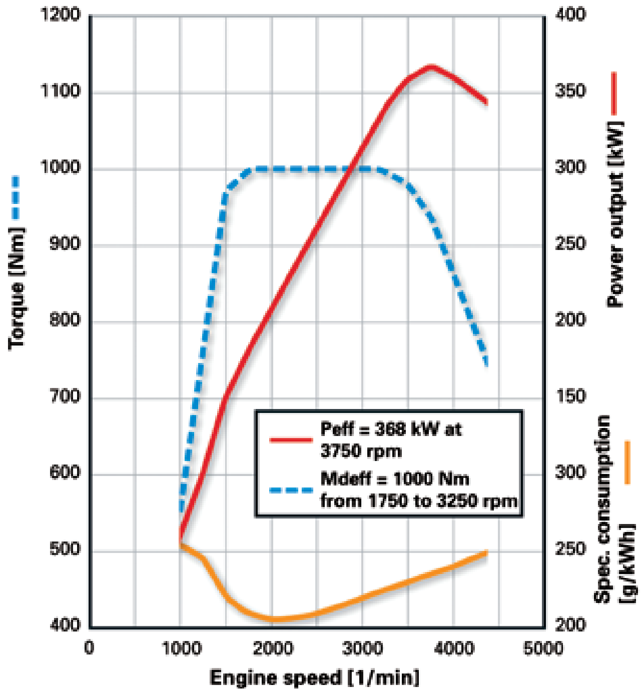


Figure 6: Full load

tained only by a small number of single-stage-charged four- and six-cylinder engines. The V12 TDI engine delivers the maximum torque of 1000 Nm over a very wide engine speed range from 1750 to 3250 rpm. 1000 Nm corresponds to an outstanding effective mean pressure of 21.2 bar, though there is plenty of potential for even higher mean pressures. The maximum absolute charge pressure is 2.7 bar. The maximum peak cylinder pressure was kept to a moderate 165 bar. **Figure 6** shows the impressive full load curve of this unit, embodying what is the world's highest power output and torque delivered by a passenger car engine.

The resultant specific fuel consumption of just 235 g/kWh at the nominal power point must likewise be classed as excellent. As already indicated by the full load figures, the entire consumption map of the V12 TDI engine is highly economical, **Figure 7**. At the optimum point a figure of 204 g/kWh is attained. The range with a specific consumption below 240 g/kWh is very broad, as in the case of all Audi V-configuration TDI engines.

air-gap-insulated manifolds with an integrated EGR tap to the VTG exhaust turbine. On emerging from the turbine and the air gap insulated downpipes, the exhaust gas is routed to the close-coupled platinum-coated oxidation catalysts. The downstream underfloor particulate filters with SiC substrate are catalytically platinum/palladium-coated (CSF: catalytic soot filters) which separate particles almost entirely. The filter volumes were designed for lifetime maintenance-free operation. The loading status of the diesel particulate filters (DPF) is monitored by two different models:

- a physical model incorporating the measured relative pressure upstream of the DPF
- a simulation model incorporating the stationary and dynamic soot input and the continuous soot burn-off by oxidation with NO_2.

Linking together these two models and the resultant regeneration strategy featuring up to five map-dependent injections (two pilot injections, one main injection and a maximum of two post-injection cycles) has already proved its worth in the development of the V8 TDI engine. Depending on the driving profile, regeneration intervals of well over 1,000 km are attained.

9 Engine Results

The highly ambitious power output target of 368 kW – particularly in combination with the targeted minimal exhaust emissions – corresponds to a specific power output of 62 kW/l. In view of the limited space available – even in the Audi Q7 – which restricts the cross-section of the intake air and exhaust gas routing, the size of the charge air coolers etc., this is a performance per litre previously at-

10 Vehicle

The V12 TDI engine accelerates the Audi Q7 from 0 to 100 km/h in just 5.5 seconds, taking the SUV to sports car levels of performance, **Table 1**. The top speed is electronically limited to 250 km/h. The elasticity figures achieved are excellent

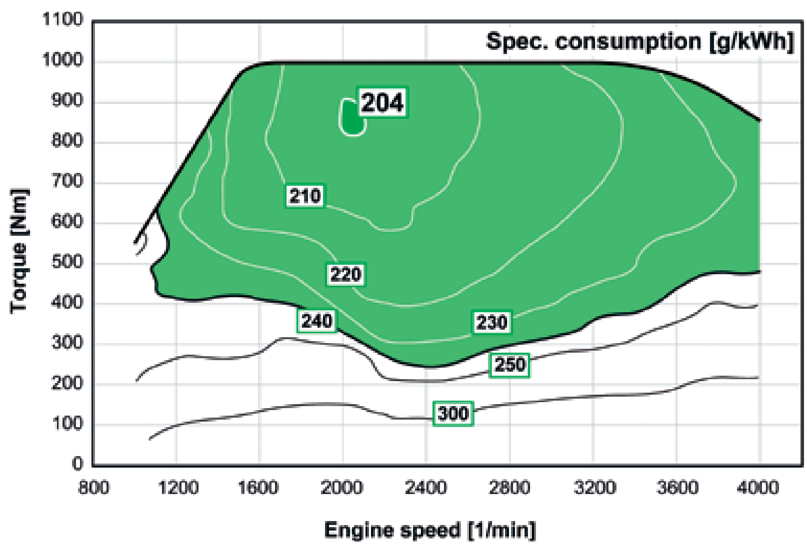


Figure 7: Fuel consumption map

Table 1: Mileage/fuel consumption

Engine		368 kW/1000 Nm
Acceleration 0 – 100 km/h	[sec]	5.5
Acceleration 0 – 200 km/h	[sec]	20.1
Maximum speed (governed)	[km/h]	250
Elasticity 60 – 120 km/h (4th gear)	[sec]	6.2
Fuel consumption	[l/100 km]	
MVEG, urban		14.8
MVEG, extra-urban		9.3
MVEG, combined		11.3
CO_2 emissions	[g/km]	
MVEG, combined		298

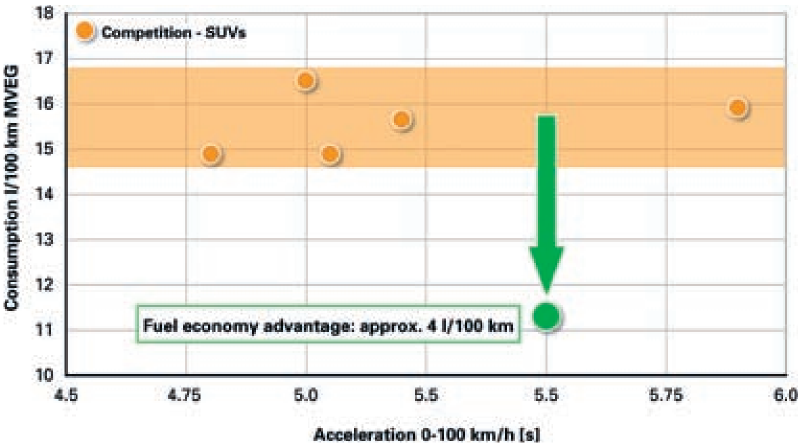


Figure 8: SUV comparison – acceleration versus fuel consumption

Table 2: Emissions

Emissions MVEG		
HC	[mg/km]	28
NO_x	[mg/km]	247
NO_x + HC	[mg/km]	275
CO	[mg/km]	209
PM	[mg/km]	1

for an SUV. All of this means the V12 TDI engine is supremely assured in real driving conditions.

Despite this, the model's fuel consumption in the MVEG test comes out at just 11.3 litres per 100 km, corresponding to a CO_2 emission rating of 298 g/km. Comparing the Audi Q7 V12 TDI engine with high-powered competitor SUVs in terms of the fine balance between performance and fuel economy, the unique position of this model becomes strongly evident, **Figure 8**. **Table 2** sets out the emission figures attained in the MVEG roller test. This vehicle complies with the EU4 emissions standard, and can be approved to EU5 Cat.C as soon as the law allows.

11 Summary

The new Audi V12 TDI engine is the world's first and only V12 diesel engine in a production car. The concerted implementation of Audi's many years of diesel expertise and the utilization of synergies with existing production engines and with engines deployed in the motorsport sphere enabled the company to develop a truly extraordinary engine.

The deployment of state-of-the-art fuel injection technology made it possible to combine supremely assured performance, exemplary smoothness and exciting engine sound with minimal emissions and outstanding fuel economy within the high-end SUV segment. The Audi V12 TDI engine also offers the potential to keep pace with even more stringent future challenges in terms of power, fuel economy and conformance to emissions standards.

Above and beyond purely demonstrating technical feasibility, the Q7 V12 TDI engine offers customers in the SUV segment a truly supreme driving experience and extraordinary driving enjoyment. And after all: the wishes and needs of those customers are the focus of all Audi's engine development activities.

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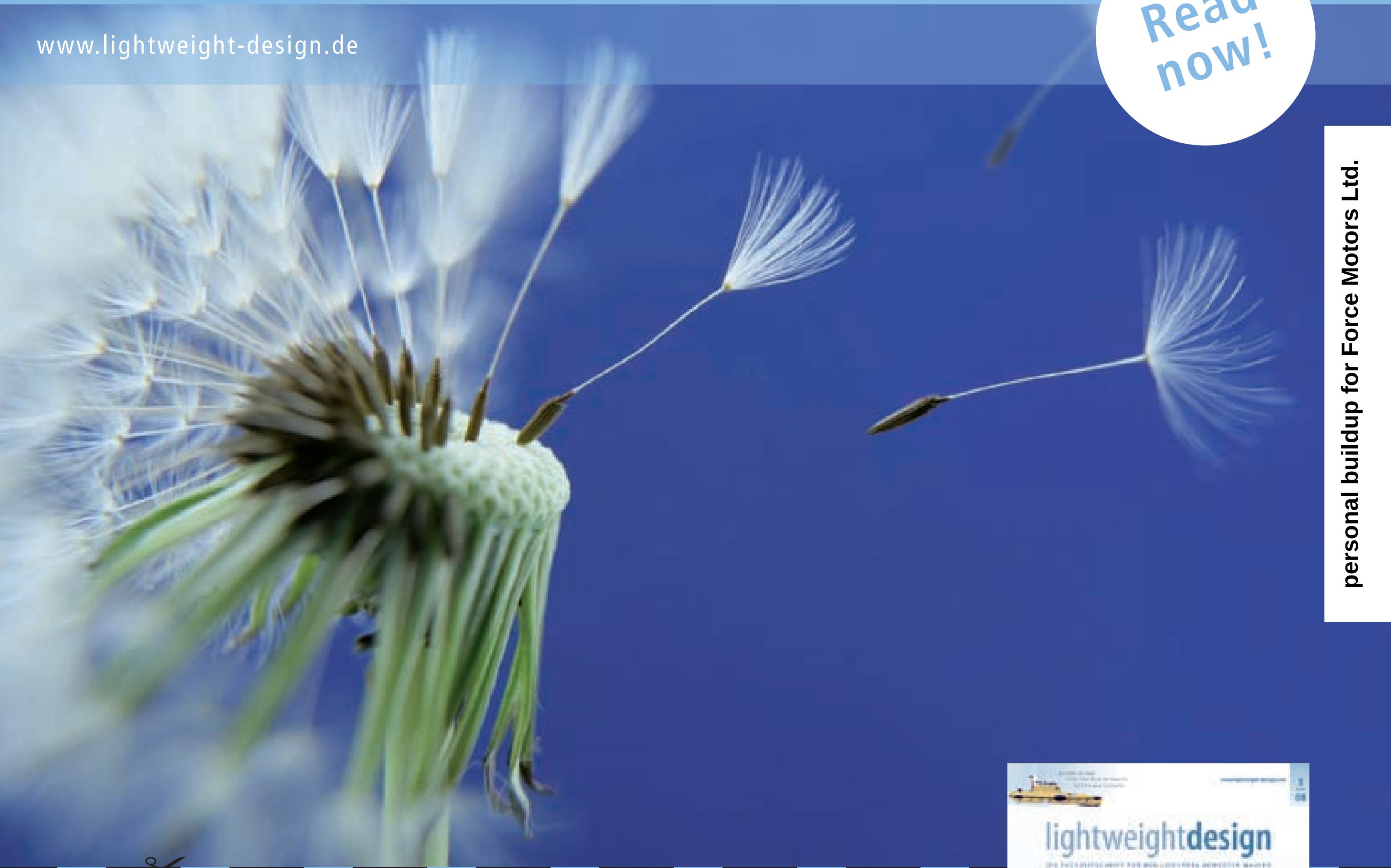

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Oil Mist Separators Integrated in Polyamide Cylinder Head Covers

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1 Introduction

Driven by the pressure difference between the combustion chamber and inner engine cavities of diesel or spark-ignition engines, so called blow-by-gases can bypass piston rings and valve guides, **Figure 1**. To avoid engine damage by high internal pressures, efficient crankcase ventilation needs to be installed. On its way from the combustion chamber towards the ventilation, blow-by-gases form a vapour with oil droplets of different sizes. Oil droplets need to be separated from this mist efficiently before blow-by-gas can be re-directed into the air intake system. **Figure 2** shows an oil mist droplet spectrum with oil volumetric fraction by particle class and distribution curve over particle diameter for a four cylinder 2.0 l TDI-Engine. The mean value of the particle diameter is near 1 µm. These small diameter particles need to be removed efficiently from the gas without exceeding crankcase pressure boundaries due to inner flow resistance of the oil separator [1]. Oil separator systems of miscellaneous dimensions and configurations have been in use for years. With diminishing available space in the engine compartment, separators have to combine high efficiencies with small external dimensions. In order to meet this objective use of integrated plastics oil separators is recommended. Their production by injection moulding, together with automated joining technologies such as friction welding, allows an efficient and economical manufacture of complex geometrical shapes. Due to the wide use of polyamide cylinder head covers in engines it is possible to save cost and space at the same time by integrating oil separator functions. To integrate both parts, solutions must be found which match all complex and sometimes conflicting requirements for cylinder head covers and oil separators, **Figure 3**. As vehicle manufacturers strive to minimize engine oil consumption, there is a strong demand for more efficient oil separators. A synchronisation of the individual components' development phases is required to utilise the given development period fully. The function of the "integrated oil separator" component has to be assured before testing the complete system on an engine test rig in order to avoid a late and therefore costly redesign.

2 Development Process

To reach series-production readiness of a cylinder head cover system a multistage process of development is used, **Figure 4**. It can be divided into three columns. The left column covers the prototype development of the oil separator. The right column contains the cylinder head cover prototype development. Both join in the middle area as the combined system enters pilot series. During both prototype phases design changes can be implemented easily. If the combined system is in pilot series redesigns can become both costly and time-consuming. Therefore it is fundamental to ensure the components meet functional requirements before leaving the prototype phase, and that they are tested as a system. With growing computational capabilities, Computer Aided Engineering (CAE) has become a strong tool for part development and testing. Starting with a requirements specification, prototypes are designed as a computer model. By using Computational Fluid Dynamics (CFD) it is possible to predict separators' characteristics in a virtual test. Different designs can be examined in detail and the best concept can be identified. If the computer model fulfils all specifications, a functional prototype can be produced by rapid prototyping and tested on an oil separator test-bed. Thus separator performance can be verified in advance of production of the cylinder head cover or the full engine build. The development process of oil separator and cylinder head cover can be managed in parallel. Both components are merged in a single part just before mould tool manufacturing. After injection moulding trials the prototype must pass all required test rig and engine tests. The development process concludes with a system of cylinder head cover with integrated oil separator tested for series-production readiness according to engine producer specifications.

2.1 Development Phases

The development process can be divided into four main phases:

- construction design process
- calculation validation
- test-bed validation
- engine test validation.

2.1.1 Construction Design Process

From process engineering many solutions are known to separate a liquid droplet fraction from a gas flow. Considering the possibility to integrate the separator in a cylinder head cover, a limited number of suitable processes remain. In passenger cars three main concepts have been established. These are inertia driven solutions or filtration with a mechanical operating principle and separators with electrical principle. Concepts which can operate without external energy and maintenance over lifetime are used most commonly. In addition, the separator design should be based on simple structures which can be efficiently produced by the injection moulding process. This explains the widespread use of baffle and cyclone separators in cylinder head covers. To separate oil droplets of varying sizes in different regions and to avoid high loading of limited areas, multi-stage separator systems are used. The first stage separates big droplets from the gas flow and prevents splashed oil from entering the separator. A second stage removes small diameter particles. Separated oil drains through a valve or siphon system back into the engine, the purified blow-by-gas flows into the air intake system via a pressure control valve.

2.1.2 Calculation Validation

All separators designed during the construction process need to be evaluated at an early point in time to identify and to modify or to discard concepts not in line with the specification. Fluid mechanics calculation methods are used for this purpose. For simple shaped types such as a cyclone separator, analytical approaches can be sufficient as they allow an estimation of pressure loss and separation efficiency [2]. An accurate prediction of the interaction of combined separator stages, as is common in most integrated oil separators, is rarely possible. These structures

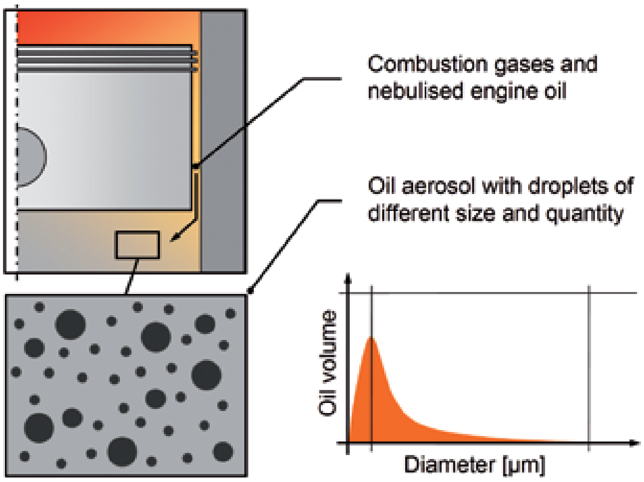

Figure 1: Origin of blow-by-gas

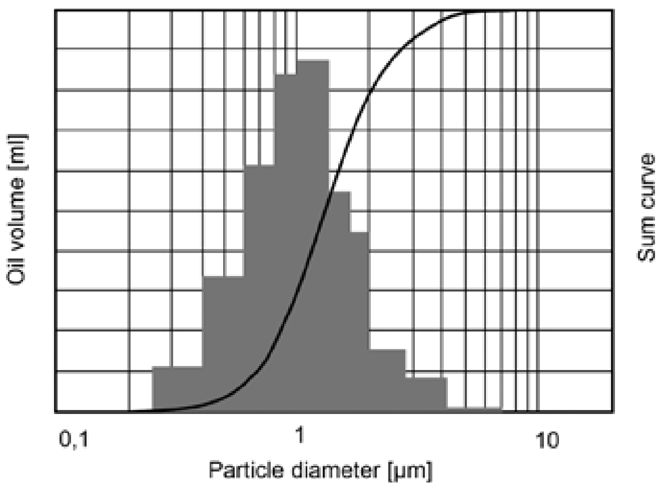

Figure 2: Volumetric distribution of oil mist particles in crank case

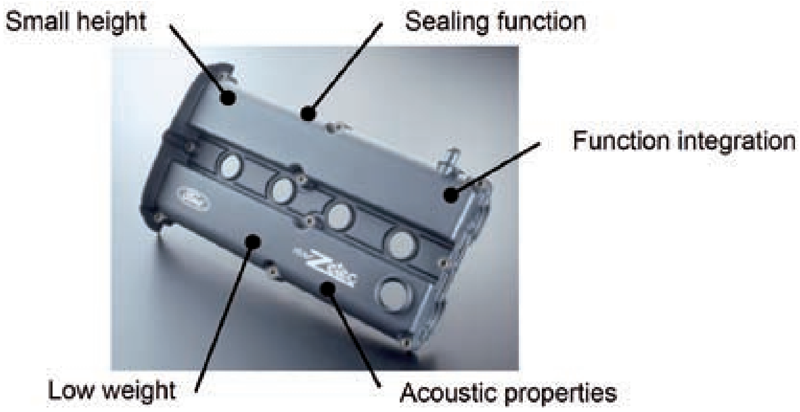

Figure 3: Extract of requirements for cylinder head covers

require more advanced methods of calculation. Integrated in the CAE process, computational fluid dynamics provides results for pressure losses and separation efficiencies of complex virtual prototypes. In addition, CFD can visualise gas flows in detail as flow velocities, path lines or oil droplet impact locations on separator walls. Furthermore, CFD allows identifying vortex structures introducing drag without enhancing total separation efficiency. CFD offers an in-depth insight in the flow and improves the understanding of the complete separator system.

2.1.3 Testbed Validation

To verify the calculated characteristics of an oil separator prototype, test-bed trials are essential. This prototype needs to include not only the separator itself but all flow or function relevant features found in the cylinder head cover, for example, inlet geometry, oil drains and outlet towards the air intake system. A fast and cost effective way to produce a temperature and oil resistant part is to use polyamide laser sintering. Test-bed evaluation allows simulation of all important engine operation points defined by the blow-by-gas volumetric flow, its' pressure and temperature. The test-bed gas flow contains oil droplets of the relevant di-

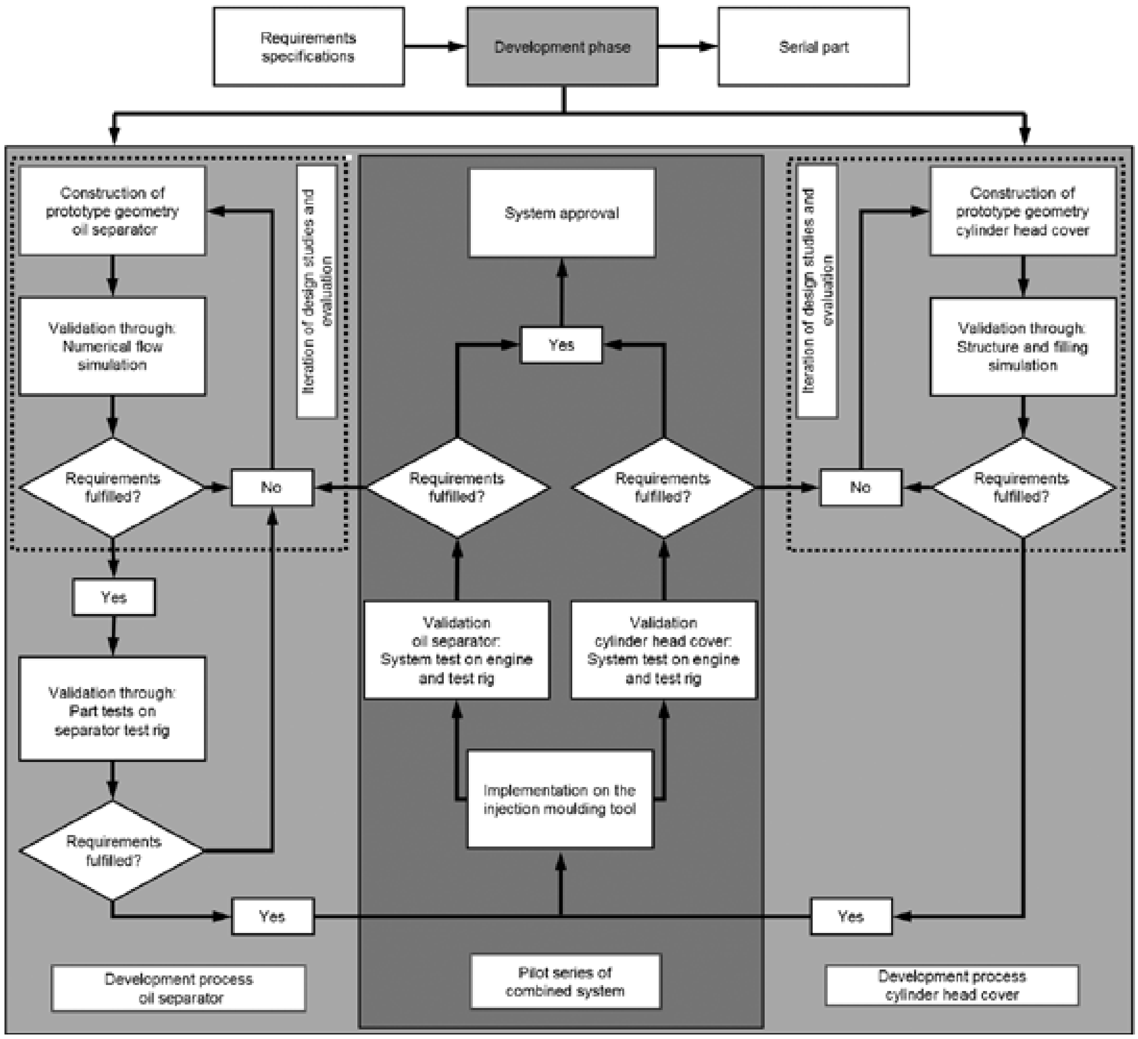

Figure 4: Development process of a cylinder head cover with integrated oil separation

ameter spectrum. Although the main focus is to validate calculated separation efficiency and pressure loss, contamination tests and tests of robustness against "sloshed" oil are also possible. In particular, separation efficiency tests require high standard measurement instrumentation. Gravimetric measurement methods can be applied when total oil output needs to be rated, but results show a high statistical spread. For detailed studies of separation efficiency depending on particle diameter, optical particle counters such as laser aerosol spectrometers show considerable advantages in accuracy.

2.1.4 Engine Test Validation

The final and decisive barrier for a cylinder head cover with integrated oil mist separator is the complete system test on an engine test rig at different engine speeds and load levels. In spite of all previous trials, it is essential to verify separation efficiency and pressure loss on the operating engine. The main criteria are effects such as crankcase pulsating pressure in dependency of engine ignition rate and varying flow conditions at the separator inlet, for example, due to rotating camshafts.

3 Summary

Integrating different functions in a single plastic assembly and combining all components shortly before mould tool manufacturing requires a structured and synchronised development process that allows full utilisation of the given development period fully. Computer-aided part design and testing has a key role in this process as it enables identification of non-conforming concepts and therefore to reduce the number and extent of testing.

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Driven by the pressure difference between the combustion chamber and inner engine cavities of diesel or spark-ignition engines, so called blow-by-gases can bypass piston rings and valve guides, **Figure 1**. To avoid engine damage by high internal pressures, efficient crankcase ventilation needs to be installed. On its way from the combustion chamber towards the ventilation, blow-by-gases form a vapour with oil droplets of different sizes. Oil droplets need to be separated from this mist efficiently before blow-by-gas can be re-directed into the air intake system. **Figure 2** shows an oil mist droplet spectrum with oil volumetric fraction by particle class and distribution curve over particle diameter for a four cylinder 2.0 l TDI-Engine. The mean value of the particle diameter is near 1 μm. These small diameter particles need to be removed efficiently from the gas without exceeding crankcase pressure boundaries due to inner flow resistance of the oil separator [1]. Oil separator systems of miscellaneous dimensions and configurations have been in use for years. With diminishing available space in the engine compartment, separators have to combine high efficiencies with small external dimensions. In order to meet this objective use of integrated plastics oil separators is recommended. Their production by injection moulding, together with automated joining technologies such as friction welding, allows an efficient and economical manufacture of complex geometrical shapes. Due to the wide use of polyamide cylinder head covers in engines it is possible to save cost and space at the same time by integrating oil separator functions. To integrate both parts, solutions must be found which match all complex and sometimes conflicting requirements for cylinder head covers and oil separators, **Figure 3**. As vehicle manufacturers strive to minimize engine oil consumption, there is a strong demand for more efficient oil separators. A synchronisation of the individual components' development phases is required to utilise the given development period fully. The function of the "integrated oil separator" component has to be assured before testing the complete system on an engine test rig in order to avoid a late and therefore costly redesign.

2 Development Process

To reach series-production readiness of a cylinder head cover system a multistage process of development is used, **Figure 4**. It can be divided into three columns. The left column covers the prototype development of the oil separator. The right column contains the cylinder head cover prototype development. Both join in the middle area as the combined system enters pilot series. During both prototype phases design changes can be implemented easily. If the combined system is in pilot series redesigns can become both costly and time-consuming. Therefore it is fundamental to ensure the components meet functional requirements before leaving the prototype phase, and that they are tested as a system. With growing computational capabilities, Computer Aided Engineering (CAE) has become a strong tool for part development and testing. Starting with a requirements specification, prototypes are designed as a computer model. By using Computational Fluid Dynamics (CFD) it is possible to predict separators' characteristics in a virtual test. Different designs can be examined in detail and the best concept can be identified. If the computer model fulfils all specifications, a functional prototype can be produced by rapid prototyping and tested on an oil separator test-bed. Thus separator performance can be verified in advance of production of the cylinder head cover or the full engine build. The development process of oil separator and cylinder head cover can be managed in parallel. Both components are merged in a single part just before mould tool manufacturing. After injection moulding trials the prototype must pass all required test rig and engine tests. The development process concludes with a system of cylinder head cover with integrated oil separator tested for series-production readiness according to engine producer specifications.

2.1 Development Phases

The development process can be divided into four main phases:
- construction design process
- calculation validation
- test-bed validation
- engine test validation.

2.1.1 Construction Design Process

From process engineering many solutions are known to separate a liquid droplet fraction from a gas flow. Considering the possibility to integrate the separator in a cylinder head cover, a limited number of suitable processes remain. In passenger cars three main concepts have been established. These are inertia driven solutions or filtration with a mechanical operating principle and separators with electrical principle. Concepts which can operate without external energy and maintenance over lifetime are used most commonly. In addition, the separator design should be based on simple structures which can be efficiently produced by the injection moulding process. This explains the widespread use of baffle and cyclone separators in cylinder head covers. To separate oil droplets of varying sizes in different regions and to avoid high loading of limited areas, multi-stage separator systems are used. The first stage separates big droplets from the gas flow and prevents splashed oil from entering the separator. A second stage removes small diameter particles. Separated oil drains through a valve or siphon system back into the engine, the purified blow-by-gas flows into the air intake system via a pressure control valve.

2.1.2 Calculation Validation

All separators designed during the construction process need to be evaluated at an early point in time to identify and to modify or to discard concepts not in line with the specification. Fluid mechanics calculation methods are used for this purpose. For simple shaped types such as a cyclone separator, analytical approaches can be sufficient as they allow an estimation of pressure loss and separation efficiency [2]. An accurate prediction of the interaction of combined separator stages, as is common in most integrated oil separators, is rarely possible. These structures

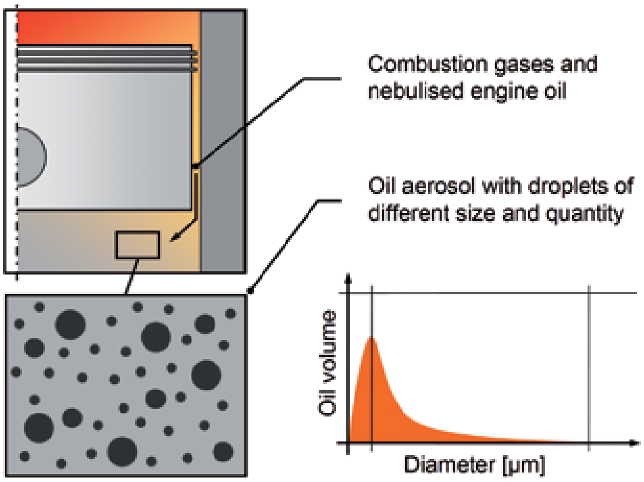

Figure 1: Origin of blow-by-gas

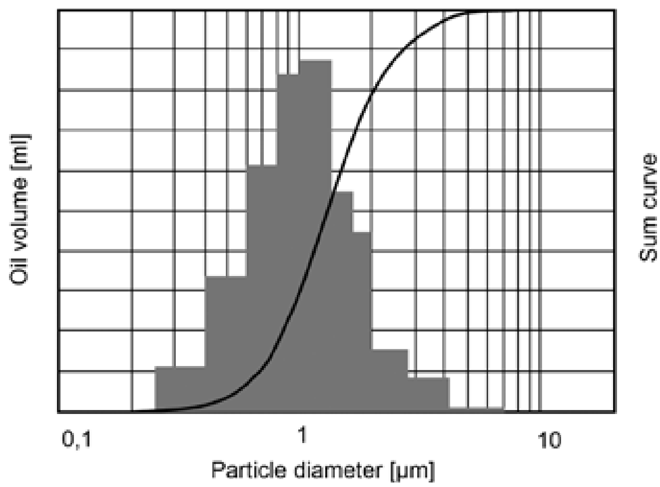

Figure 2: Volumetric distribution of oil mist particles in crank case

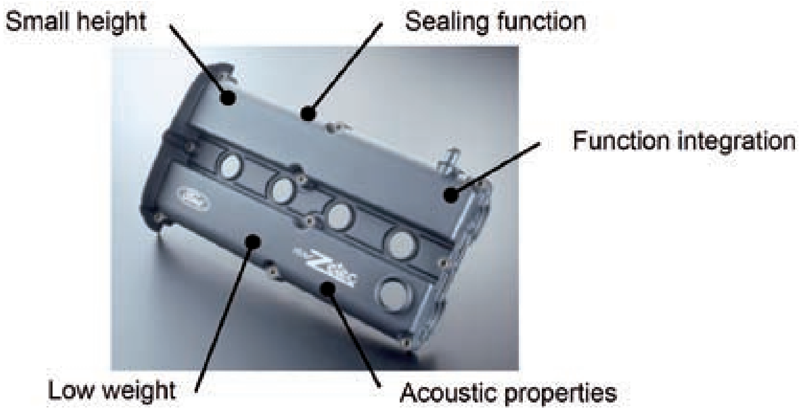

Figure 3: Extract of requirements for cylinder head covers

require more advanced methods of calculation. Integrated in the CAE process, computational fluid dynamics provides results for pressure losses and separation efficiencies of complex virtual prototypes.

In addition, CFD can visualise gas flows in detail as flow velocities, path lines or oil droplet impact locations on separator walls. Furthermore, CFD allows identifying vortex structures introducing drag without enhancing total separation efficiency. CFD offers an in-depth insight in the flow and improves the understanding of the complete separator system.

2.1.3 Testbed Validation

To verify the calculated characteristics of an oil separator prototype, test-bed trials are essential. This prototype needs to include not only the separator itself but all flow or function relevant features found in the cylinder head cover, for example, inlet geometry, oil drains and outlet towards the air intake system. A fast and cost effective way to produce a temperature and oil resistant part is to use polyamide laser sintering. Test-bed evaluation allows simulation of all important engine operation points defined by the blow-by-gas volumetric flow, its' pressure and temperature. The test-bed gas flow contains oil droplets of the relevant di-

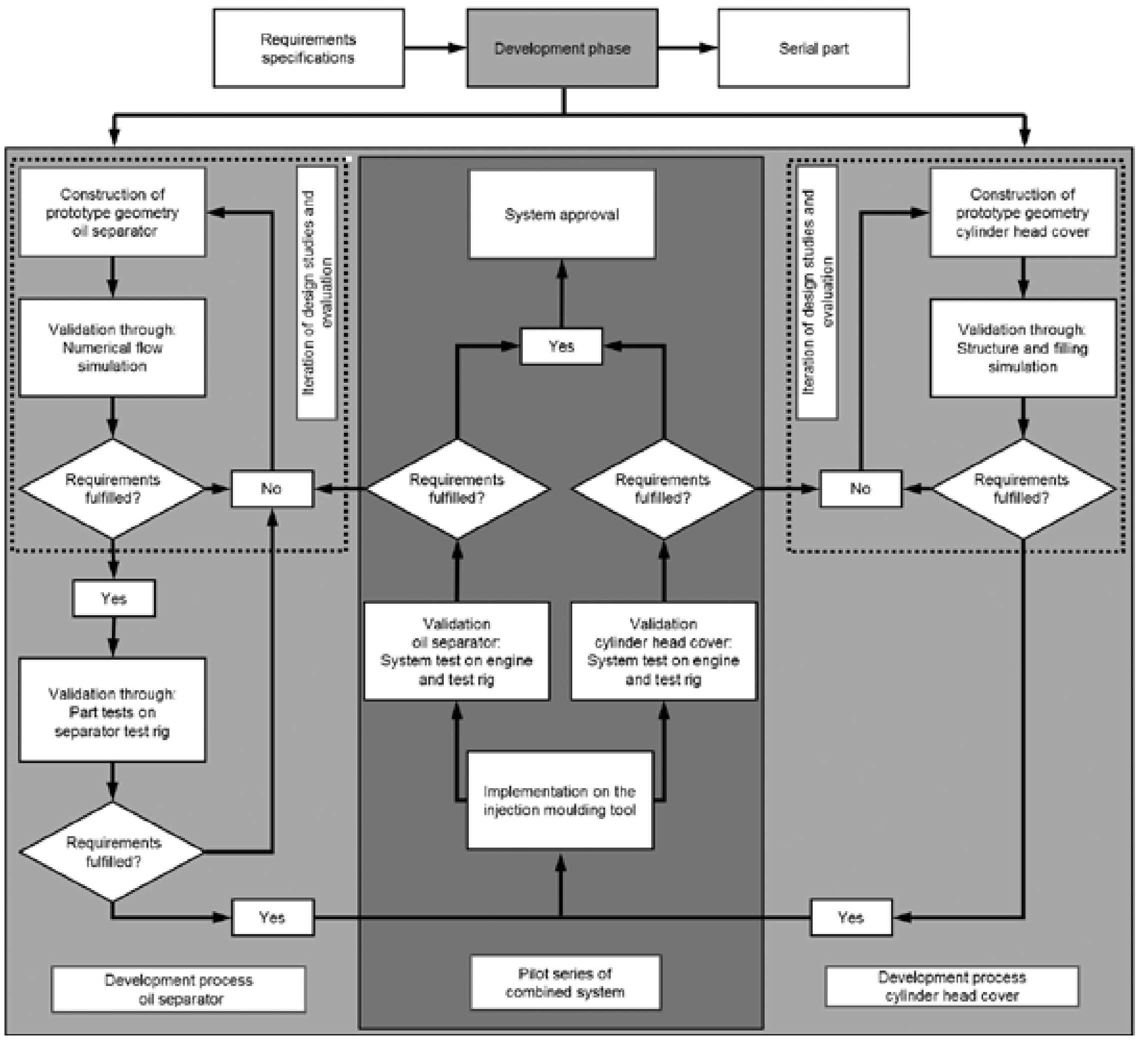

Figure 4: Development process of a cylinder head cover with integrated oil separation

ameter spectrum. Although the main focus is to validate calculated separation efficiency and pressure loss, contamination tests and tests of robustness against "sloshed" oil are also possible. In particular, separation efficiency tests require high standard measurement instrumentation. Gravimetric measurement methods can be applied when total oil output needs to be rated, but results show a high statistical spread. For detailed studies of separation efficiency depending on particle diameter, optical particle counters such as laser aerosol spectrometers show considerable advantages in accuracy.

2.1.4 Engine Test Validation

The final and decisive barrier for a cylinder head cover with integrated oil mist separator is the complete system test on an engine test rig at different engine speeds and load levels. In spite of all previous trials, it is essential to verify separation efficiency and pressure loss on the operating engine. The main criteria are effects such as crankcase pulsating pressure in dependency of engine ignition rate and varying flow conditions at the separator inlet, for example, due to rotating camshafts.

3 Summary

Integrating different functions in a single plastic assembly and combining all components shortly before mould tool manufacturing requires a structured and synchronised development process that allows full utilisation of the given development period fully. Computer-aided part design and testing has a key role in this process as it enables identification of non-conforming concepts and therefore to reduce the number and extent of testing.

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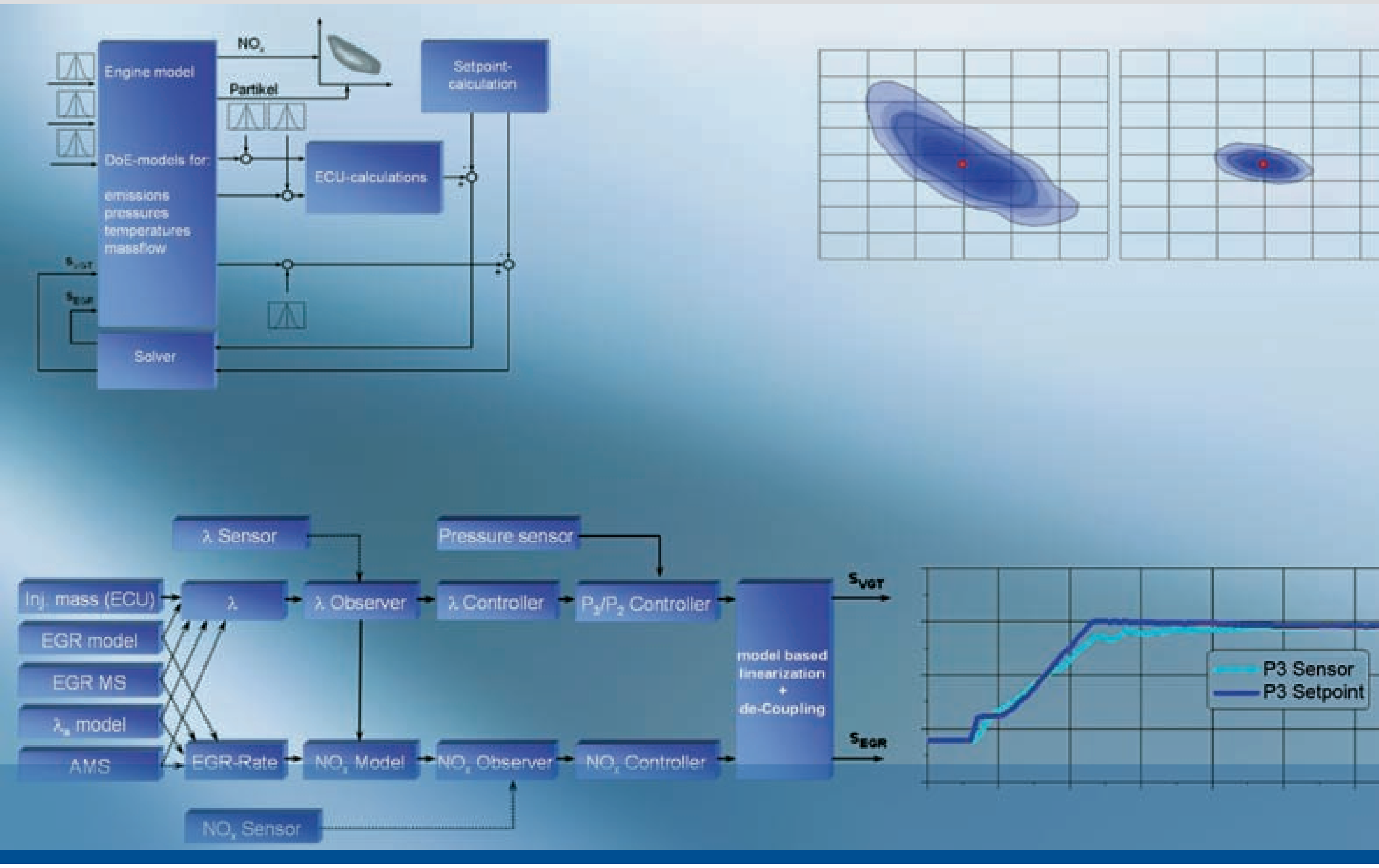

More Stringent Requirements for Air Path Control in HD Engines

Goal of new developments regarding air-path control of heavy-duty engines is to set stationary emissions in a stable way independent from boundary conditions and to allow fast control during transients. This article by RWTH Aachen and FEV presents methods to assess control accuracy for stationary and dynamic operation and describes an innovative control structure which allows a compromise between engine responsiveness and emission behaviour by calibration.

1 Introduction

The requirements for advanced commercial vehicle and industrial engines are increasing due to further tightening of emission limit legislation in all areas of application. Because of the drastically reduced exhaust emissions rates throughout the entire engine map area, higher exhaust gas recirculation rates must be considered at increasing levels of supercharging, while avoiding disadvantages in fuel consumption and continuously improving the transient behaviour.

The transient emissions of diesel engines are for the most part the result of the widely differing dynamic cylinder charge conditions, for example, inadequate exhaust gas recirculation rates or insufficient boost pressure at increased exhaust back pressure. Current design criteria with respect to the stationary requirements that are to be met thus follow a certain overcompensation strategy. This is due to the variance that is to be expected as part of the transient test, having adverse effects when it comes to fuel consumption and, in the future, probably also with respect to the regeneration interval and the operational safety of the particulate filter in real-life daily operation.

In this context, new intelligent control structures on the air path side can have clear advantages with regard to fuel consumption and reduction of production variance. This article describes the fundamental dependencies and explains the main features of a new advanced control concept.

2 Legal Parameters and Required NO_x Levels

Reduction of nitrogen oxides, while simultaneously achieving the lowest possible particulate levels, is the reason behind the increased use of cooled exhaust gas recirculation in combination with high-pressure supercharging and high-pressure direct injection in commercial vehicle engines. As shown in **Figure 1**, a nitrogen oxide limit of approximately 1.6 g/kWh must be reached in the US to reliably be within the legal limits of the currently enforced emission norm, US07. The engines that are currently serially produced meet the required limits using cooled, external exhaust gas recirculation in combination with a particulate filter. The slightly less stringent Euro 5 limits are also met, in part, using cooled exhaust gas recirculation with an open or closed particulate filter. However, the standard technology in Europe is the active NO_x aftertreatment with SCR in combination with a low PM combustion system. For the upcoming Euro 6 norm level as well as for the emission norm US10, that has already

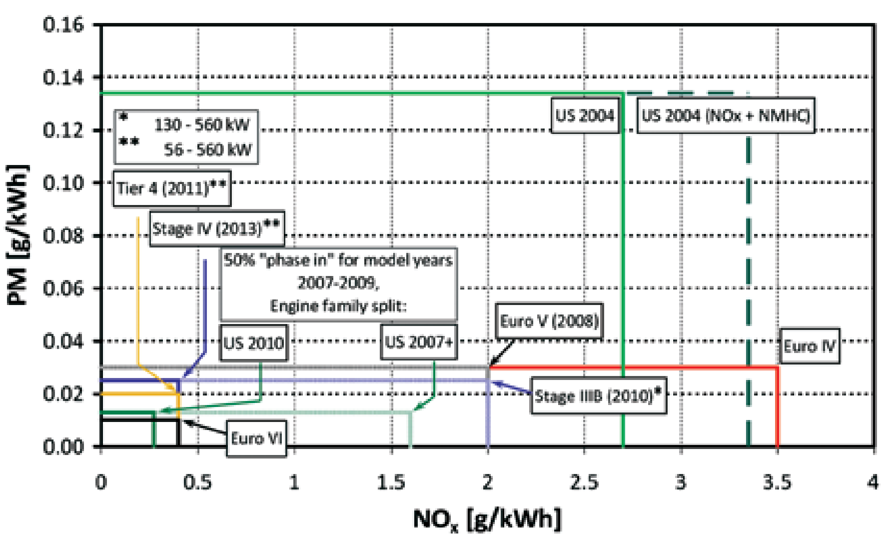

Figure 1: Certification limits for heavy-duty and non-road applications

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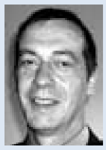

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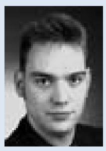

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been adopted, the nitrogen oxides must be lowered further by approximately 80 % in each case, when compared to the corresponding predecessor norm. For serial production, part of the reduction in NO_x is achieved through a further increase in exhaust gas recirculation. Additionally, a considerable nitrogen oxide reduction, down to values below 0.27 g/kWh (US10) or 0.4 g/kWh (Euro 6 proposal, see [3]), is achieved through the SCR system.

Further, Figure 1 shows that stage IIIb, which will be introduced in 2010 for industrial engines, is comparable to the Euro 5 standard for truck application. In this case cooled exhaust gas recirculation is the most effective in-engine technology. The following step to stage IV is once again comparable to the Euro 6 standard for commercial vehicle engines. That means, at that time the application of cooled exhaust gas recirculationin combination with active aftertratment for industrial engines can be expected.

Figure 2 shows the required exhaust gas recirculation rates in order to fulfill the different NO_x emission levels, solely with in-engine methods. To lower nitrogen oxides, it is necessary to reduce the oxygen concentration at the engine inlet [5]. The resulting trend is one of significantly higher exhaust gas recirculation rates and particularly lower relative air-fuel ratios with each progressive emission level. Simulations and single cylinder investigations show [1] that, with the appropriate combustion system and exhaust gas recirculation rates of more than 50 % shown in Figure 2, as well as an air-fuel ratio of below 1.3, the US10 NO_x limit value can, theoretically, be achieved even without any exhaust aftertreatment, by using extreme pressure injection systems. However, doubling the full load exhaust gas recirculation rates for serial production has to be regarded as critical from today's perspective with regard to fuel consumption, vehicle cooling system loads, and the required transient behaviour.

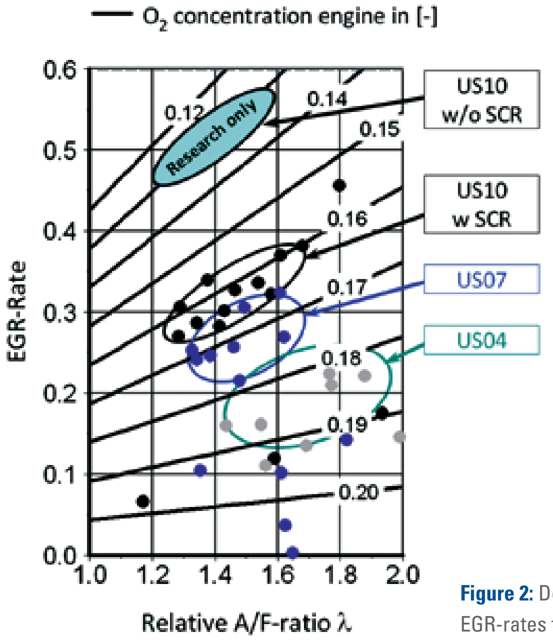

Figure 2: Development of required EGR-rates for US certification levels

The significance of the air path parameters, oxygen concentration and relative air-fuel ratio, shown in Figure 2, were analyzed repeatedly [5]. Their significance lies in the correlation to the actual target values of the air path regulation of diesel engines, the nitrogen oxide and soot emissions. The O_2 portion at the engine inlet that is relevant for the NO_x development can only be determined exactly, as shown in Figure 2, if the relative air-fuel ratio and the EGR rate are known. In order to control soot emissions, the exact relative air-fuel ratio must be known and taken into account in the air path control.

3 Influencing Emissions via the Air Path: Correlations

Figure 3 shows data with different combinations of VGT and EGR positions. A model of the emission changes can be created very easily with an exponential approach [5]. In Figure 3, the correlation

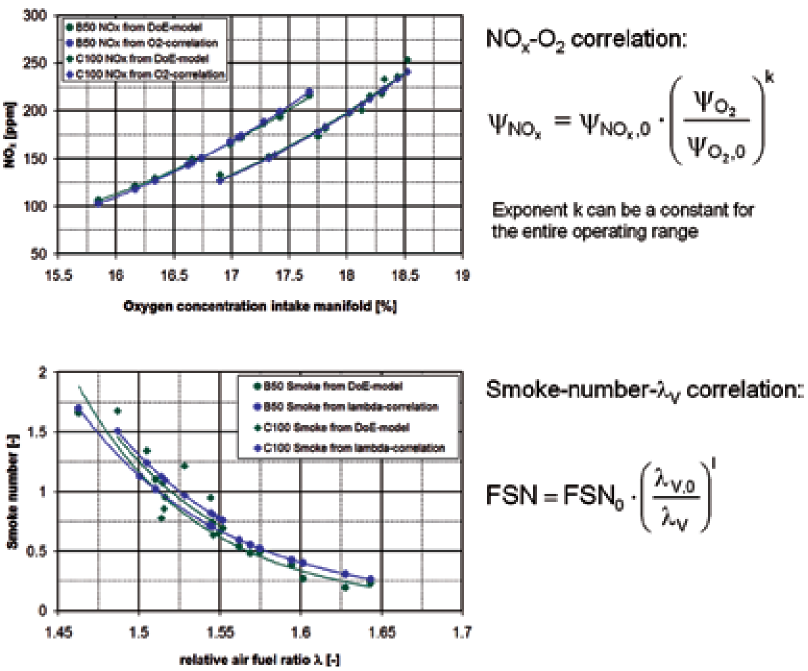

Figure 3: Quality of the correlation of NO_x and smoke

between O_2 and NO_x is very good as expected for a full load operating point and an operating point at 50 % load. The correlation between the combustion air-fuel ratio and the smoke number, which in turn can be converted, with the aid of correlations, to particulate matter emissions, is less favorable. As analyzed in [5], [4], there will always be deviations in the lambda soot correlation when the charge density varies due to the boost pressure. Especially at high load this must be considered, while at part load this is less significant. Thus in Figure 3 the lambda correlation gives better results in the BSO mode.

The map-based exponential models shown in Figure 3 have constant parameters through load and engine speed. Since the correlation parameters depend on load and engine speed, the models can be calibrated with little effort, and are, therefore, very well suited for the implementation in engine control units. Since control concepts were developed at FEV based on these very simple model approaches, other influences such as injection parameters or temperature influences were also investigated. As an example **Figure 4** shows the effect of the variation in charge air temperature on the accuracy of the identically parameterized O_2 based NO_x model in each case. The deviations of approximately +/- 10 ppm, with a variation of charge air temperature of +/- 8 °C, can be accepted to be in the range of the model accuray.

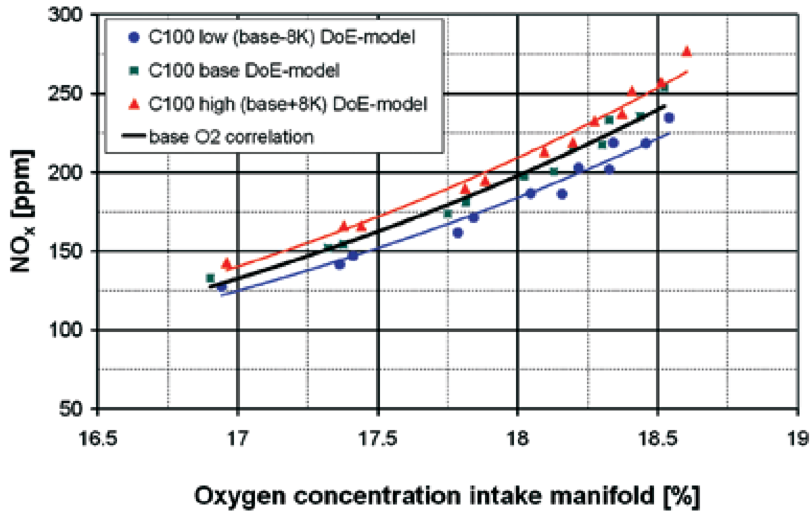

Figure 4: NO_x-O_2-correlation with varying temperature after intercooler

Emission models that are easy to calibrate are used in advanced commercial vehicle engine controls as a source of information in order to control not just exhaust aftertreatment, but also the air path. For example, for FEV's EGR control concept, the virtual NO_x signal is determined on the basis of the oxygen concentration in the engine inlet. The accuracy of the emission control then, basically, depends on the accuracy of the oxygen concentration that is determination at the engine inlet.

As shown in Figure 2, the current EGR rate as well as the current relative air-fuel ratio in the exhaust is needed. To determine the EGR rate, 2 of the 3 mass flows such as fresh air, EGR, and engine mass flow are required in the form of a measurement or model. For every 1 % error in EGR rate, there is approximately 10 % error in NO_x. As can be seen in Figure 2, for the given scale the lines run at an approximately 45 ° angle – in reality, the correlation between the NO_x deviation and the lambda deviation is similar to that of the relative air ratio. In other words, the exact air-fuel ratio must be known. With a very accurate air mass measurement and a very small number of errors, it might be possible to calculate the combustion air-fuel ratio with sufficient accuracy. However, from current point of view, a lambda sensor is needed to deliver the required accuracy.

4 Sensor System and Sensitivity Analysis

The sensor concepts that can be used for specific applications under given boundary conditions can differ widely. While suppliers of industrial engines, for instance, must cover the most diverse package conditions with just one sensor concept, vehicle manufacturers have potentially more degrees of freedom when it comes to packaging sensor systems, such as air mass flow sensors. Also, the level of complexity varies with the emissions legislation, requirements regarding stability of emission in case of exhaust aftertreatment, and engine properties such as the specific output.

The sensor system selection must, therefore, be specified individually and might subsequently change due to the ongoing development in this field. FEV uses a DoE-based statistical method in order to quantify the requirements, or the advantages of different combinations of sensors and models with respect to engine emission variances in serial production, already during the planning phase.

In the process, DoE models are used to create predictions for the relevant control and emission variables. These forecasts can be influenced by varying boundary conditions or falsified due to sensor variances. The controlled variables that are used in the end can be actual measurement values as well as calculated variables. An example of the latter is the virtual NO_x signal, based on the oxygen concentration, in the case of FEV's EGR control concept. With FEV's EGR control concept, various sensor combinations can be used to calculate the virtual NO_x signal.

By calculating a large number of different variants, it is possible to calculate the distribution of the emissions around the base point of the calibration in detail. This investigation permits a statistical analysis of the occurrence of deviations of a defined value. The emission variance depends on the boundary conditions (for example, load condition of the particulate filter), sensor accuracies (for example, accuracy of the EGR mass flow rate measurement), and the control structure selected (for example, sensor

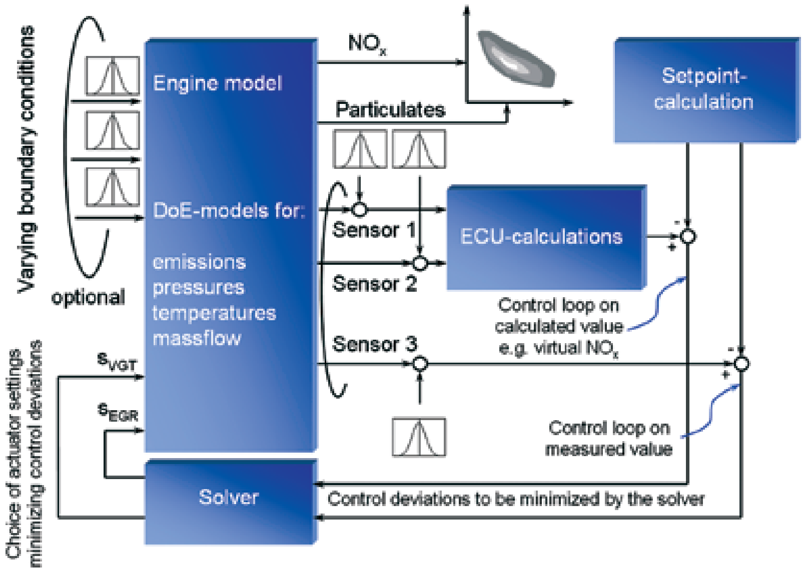

Figure 5: Structure of DoE-based sensitivity analysis

equipment for the computation of virtual controlled variables), **Figure 5**.

Figure 6 shows the variance of the smoke number and NO_x emissions in relationship to the control concept and the sensor system that is installed. The values shown here prove that 65 % and 90 % of all simulated systems can be found within their respective isolines. The variance of the sensor system is shown as the percentage relative to the nominal value in the base point. The percentage is set to 3 σ, in other words 997 of 1000 sensors are within the interval that is specified in percent. The variance of the boundary conditions (exhaust back pressure, rail pressure, injection timing) is the same in both cases. In the left figure, the system is regulated to a NO_x value from the correlation. However, the sensor concept has its weaknesses: An inaccurate engine mass model (total mass flow) makes it difficult to determine the EGR rate and, instead of an independent measurement, the air-fuel ratio of the ECU quantity and the calculated air volume is used. In contrast to this, the right side of Figure 6 shows the stationary accuracy of the reference concept with NO_x and λ sensor. There is a wide range of possible sensor combinations that can be used to realize the control concept as shown in Figure 9. The variances illustrated are two extreme cases. The method shown here can be used to determine the best sensor and control concept for a particular application. In general, we find that systems with a sensor in the exhaust path (lambda and also NO_x sensors) are able to meet stationary emissions with greater accuracy than designs where the sensor system is only used in the air path. The concept on the left in Figure 6, for instance, can be improved considerably by using a lambda sensor.

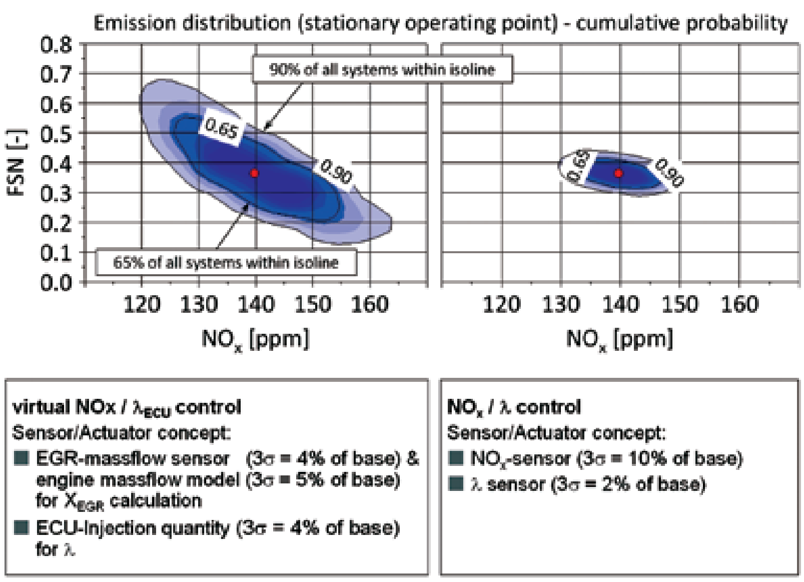

Figure 6: Effect of sensor choice on stationary control quality

5 Transient Emissions Behavior

In addition to the stationary emissions of the engine, that are relevant for cycles such as ESC or NTE legislation (USA), all commercial vehicle engines, and industrial engines as well in the future, must be emission tested in transient, highly-dynamic cycles. Transient engine operation means that thermodynamic equilibrium processes, implying pressure and temperature changes, take place in the air path. On the part of the engine control, only boost pressure and EGR rate can be influenced, in addition to the very rapid injection parameters (cycle to cycle).

Figure 7 shows the simulation of a load step to full load at 2000 rpm. The model that is used is kept very simple and is, thus, quick. In contrast to a 1D engine process simulation, it is real-time capable, and it is therefore possible to simulate entire driving cycles within reasonable CPU times. During a load step from lower partial load to full load, the boost pressure sets in with a clear delay, so that the injection quantity must be limited immediately after the increase in load in order to limit the air-fuel ratio and thus the soot emissions. It is, therefore, not possible to regulate the boost pressure to the stationary specified value directly after the load step. This effect is the main reason for the increase of transient emissions in diesel engines [2].

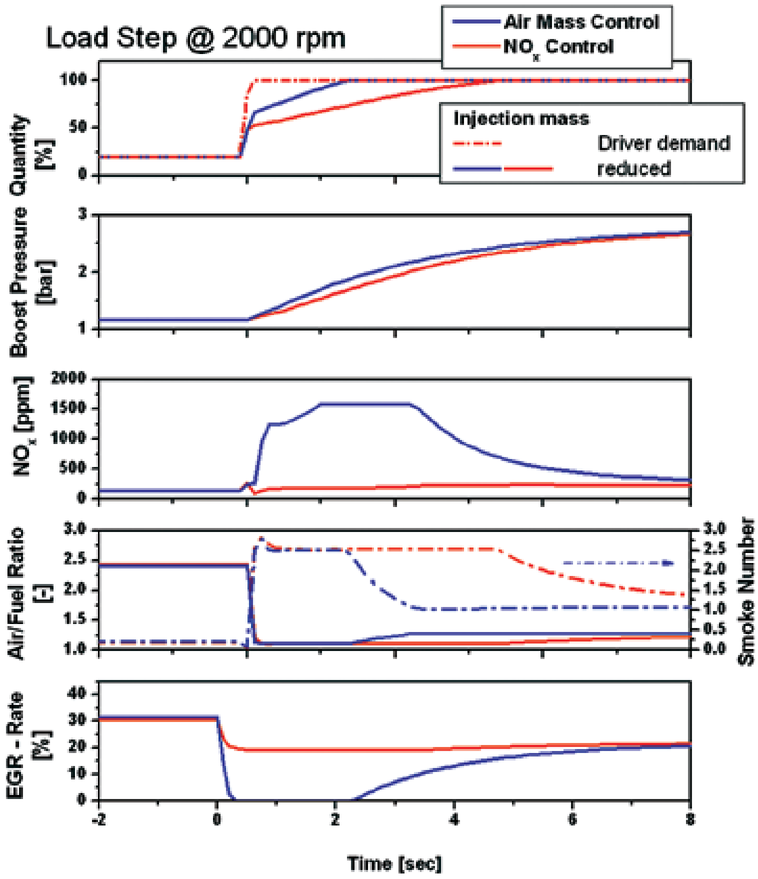

Figure 7: Comparison of air mass and NO$_x$-control during load step

transient nitrogen oxides levels become higher compared with stationary tuning. **Figure 8** shows the integral rise in transient nitrogen oxide and soot emissions. Without taking any measures on the injection path, the nitrogen oxides in the ETC increase by more than 50 % compared to the stationary emissions calculated through map interpolation. It would only be possible to meet the transient limit value, if the stationary nitrogen oxide level was reduced accordingly, as shown in the lower left part of Figure 8. However, this generally brings disadvantages in fuel consumption due to a basic calibration with too much NO$_x$, and shortens the regeneration intervals of the particulate filter due to higher stationary PM emissions.

FEV adopts the approach of maintaining the tuned stationary nitrogen oxide level, even under the transient conditions, by regulating and controlling the

As mentioned earlier, the boost pressure cannot be regulated immediately after the load step. The VGT is adjusted to a feasible extent or quick pressure build-up, or if no VGT is available, the wastegate is closed. However, in addition to the injection parameters (smoke limitation, transient adjustment of start of delivery and rail pressure), the emissions and the engine response behavior can be influenced to a large extent by the transient EGR rate control. As illustrated in Figure 7, the best engine response behavior can be achieved by switching off the EGR. Beside this, the transient smoke peak is reduced and shortened and thus the transient soot emissions are minimised.

This type of EGR rate control is achieved with an air mass control that is widely being used in passenger car engines. The disadvantage here is a substantial increase in nitrogen oxide emissions caused by deactivating the EGR. The transient increase in nitrogen oxides can not be fully compensated through the injection path, so that the

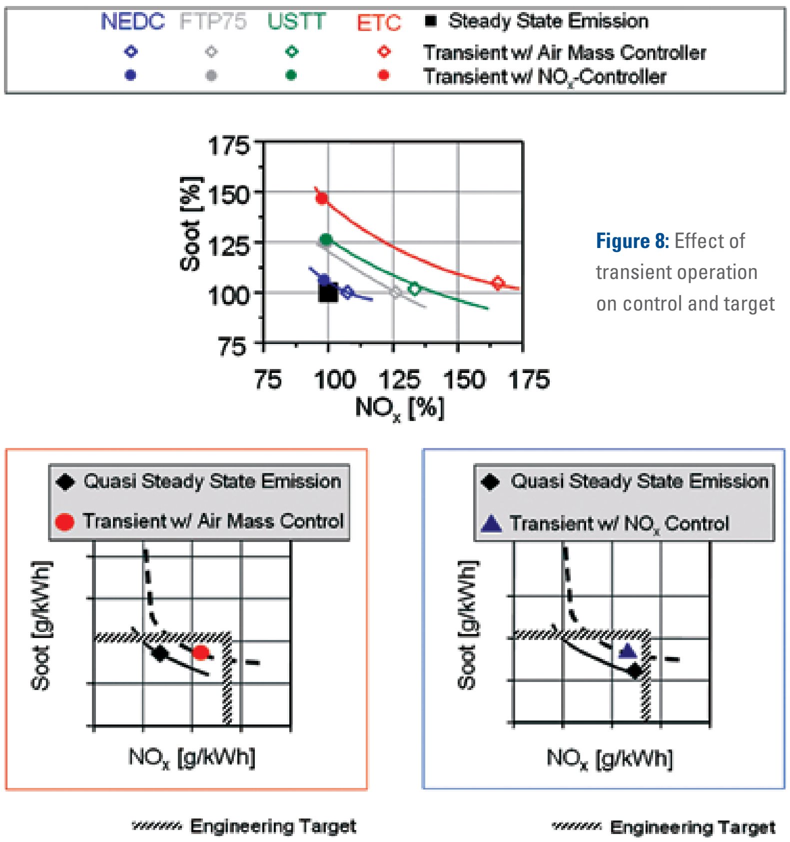

Figure 8: Effect of transient operation on control and target

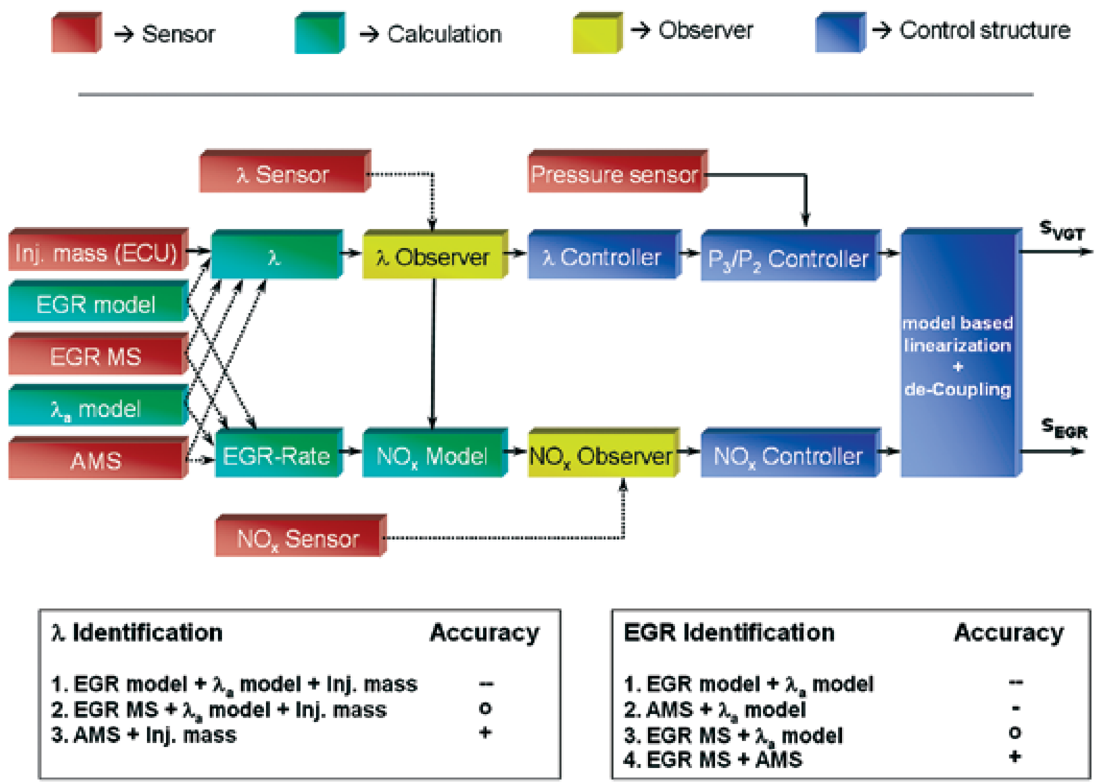

Figure 9: FEV's EGR-control structure

EGR and injection path accordingly (refer the lower left part of Figure 8). To do this, the EGR must be controlled as precisely as possible during the load step. By tuning the smoke limitation, injection strategy, and a quick NO_x regulation via the EGR path, the calibrated stationary nitrogen oxide level can also be maintained in the ETC. This permits stationary tuning of the engine to the higher nitrogen oxide level compared to the air mass control, and thus results in a reduction in fuel consumption and lower stationary PM emissions.

6 Air Path Control Concept

In general, an air path control concept consists of a combination of actuators, sensors, and intelligent software strategies. As illustrated before, the sensor concepts vary widely depending on the requirement of the manufacturer. It also has to be taken into account that the sensor development process is still ongoing. The software concept developed by FEV, shown in **Figure 9**, is highly flexible with regard to the most diverse scenarios of sensor combinations.

Furthermore, the emphasis is on the option of a complete double closed loop air path control (VGT and EGR). The systems can be controlled with VGT or wastegate using pressure signals (p2 or p3) or the relative air-fuel ratio through the first closed loop, and the nitrogen oxide concentration through the second closed loop (EGR). As shown in [5], it is not sufficient to directly control with the signal of an NO_x sensor that comes with a long time delay. In order to represent fast NO_x control, a virtual, model-based NO_x signal is required [5,6,7]. As shown in Figure 9, the virtual nitrogen oxide signal is calculated from the EGR rate and the air-fuel ratio. The application effort involved in the NO_x model is relatively low, since it operates with fixed parameters. As an option, the virtual NO_x signal can be calibrated with an NO_x sensor and, if necessary, the effort involved for the accuracy of the NO_x model can be reduced. In the same way, the lambda signal used for the VGT is calculated virtually, thus quickly, and, as an option, calibrated in stationary operation using a lambda sensor. The control shown in Figure 9 uses a model-based linearization of the actuators. Depending on the sensor system that is being used, the decoupling effort involved can be reduced significantly – decoupling filters as per [6] can also be used.

Figure 9 shows the possible combinations that can be used to determine the values of EGR rate and relative air-fuel ratio needed for the NO_x model. In order to be flexible in reacting to different requirements, the software concept illustrated here permits the use of any of the combinations of the computations shown, and, if the accuracy is not sufficient, to calibrate the lambda computation or the NO_x model with the corresponding sensor. Another option to increase the accuracy is to create an overdeterminate engine condition. In comparison to a calculation of the EGR rate with the air mass flow rate sensor and the EGR mass flow rate sensor, the accuracy of the system can be increased, if the information of a volumetric efficiency model is used. As a result, one would receive three pieces of information, which usually contradict one another physically, but which can be used to determine the statistically most probable condition of the engine. Proficient use of all of the information available has shown to lead to a better result than using the most accurate information alone.

Figure 10 shows an executed example for the application of a control concept on an FEV demonstrator. The diagram shows the regulation of the exhaust gas pressure and the (virtual) nitrogen oxide concentration during a load step. It shows that the exhaust back pressure control can follow the specified value very quickly. In this case the EGR mass flow rate sensor is relatively slow, which leads to a signal speed of only 300 to 500 ms for the virtual NO_x concentration. However, controlling to this signal still allows mobile tuning without NO_x overshoot. The NO_x sensor follows, as expected, with a considerable time delay and cannot be used as a feedback for transient control. If dynamic requirements are high, in particular at low rotational speeds and applications with a fixed geometry turbine, the software allows to calibrate for adequate response time a controlled nitrogen oxide overshoot.

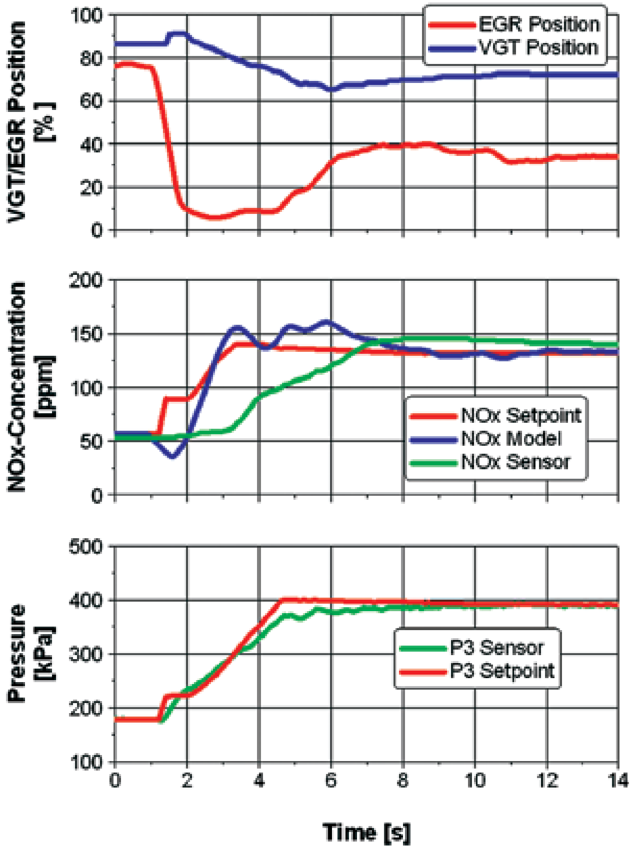

Figure 10: Load step with virtual NO_x control

personal buildup for Force Motors Ltd.

7 Conclusion

Reliable compliance with a lot more stringent exhaust emissions standards for future commercial vehicle and off-road engines, while at the same time meeting increasing dynamics requirements calls for new intelligent control structures and systematic evaluation standards for the assessment of concepts, in particular when it comes to the sensor system package that is required. By taking a close look at the stationary sensitivities of the system, a detailed assessment of the stationary accuracy that is to be expected is possible. The dynamic emissions and the response time are assessed based on an isolated consideration of the system dynamics including sensors and actuators. The FEV control concept illustrated here makes it possible to calibrate the required tuned compromise between emissions and transient response time as part of adequate dynamics functions, by generating specified NO_x values, without abandoning the controlled operation of the air path. This new innovative approach makes it possible to regulate the emissions emitted in a test cycle more quickly and in a more stable manner and to considerably minimize dynamic nitrogen oxide overshoot. Thus, the nitrogen oxide emissions in the stationary range can be increased by up to 50 % depending on the test cycle and the basic engine concept. As a result, fuel consumption advantages of up to 3 % can be achieved, both in the statutory test cycle as well as during the real-life vehicle operation. This is done, on the one hand, through efficiency improvements, and on the other hand, through significantly reduced regeneration intervals of the particulate filter.

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Combustion of the Alternative Marine Diesel Fuel LCO in Large Diesel Engines

Large diesel engines represent the heart of the ships, which transport worldwide about 80 % of the goods over the sea route these days. Regimentations of the IMO are planning drastic reductions of nitrogen oxide and sulfur oxide emission limitations from marine diesel engines. At the Laboratory of Engine and Combustion (ECO) of the Kyushu University in Fukuoka (Japan), experiments were carried out on a medium size, single cylinder, diesel engine with two-stroke technology in order to investigate the use of Light Cycle Oil (LCO) in large diesel engines with new combustion processes.

1 Introduction

Container ships, cargo ships and tankers are handling 80 % of the worldwide commodity flow. These ships are propelled by large diesel engines, which often suffer from high pollutant emission due to high temperature combustion of heavy fuel oils. The International Maritime Organization (IMO) has therefore introduced new amendments to the MARPOL Annex VI regulations, which aim besides the already existing regulations for sea water pollution from ships (MARPOL Annex I-V) the reduction of harmful emissions from ships [1, 2]. These tight regulations are mainly focusing on the sulfur dioxide and nitrogen dioxide emission from low- and medium-speed diesel engines. Large diesel engines provide the possibility to use heavy fuel oils, where residual oils from the distillation and cracking process during oil refining are burned.

Heavy fuel oils consist, besides long-chain alkanes, of highly aromatic compounds like naphtaline, anthracene as well as hydrocarbons containing sulfur and nitrogen compounds (asphaltine). The fuel sulfur content of heavy fuel oils can get up to 4.5 %, which reaches the current limitations for fuel sulfur content in heavy fuel oils.

Until 2010 the limitation will be reduced to 3.5 %, from 2020 even to 0.5 %. Tighter restrictions are effective in so called SO_x Emission Control Areas (SECAs), which also include the North Sea region, with a current maximum fuel sulfur content of 1.5 %. Further accentuation by the IMO within the SECAs is aiming a maximum fuel sulfur content of 1.0 % from 2010 and 0.1 % from 2015 [1]. The IMO plans in accordance to the Tier II NO_x regulations a decrease in NO_x emission limitation of 20 % from 2011 [2]. Due to these drastic changes in emission regulations for large diesel engines, the development of advanced and clean combustion processes as well as the application of alternative fuels is of exceptional interest.

2 Homogeneous Combustion Processes

The high temperatures of above 1800 K offer the energetic circumstances for the NO_x generating reaction mechanisms. A promising combustion process is the widely discussed homogeneous charge compression ignition (HCCI) concept [3, 4]. In comparison to conventional diesel combustion, which is based on a diffusive combustion after fuel injection near the top dead center (TDC), the early injection timing used in the HCCI combustion process results in the combustion of a premixed and lean fuel-air mixture.

Due to the fuel injection during compression stroke enough time for proper mixing of the fuel with the surrounding air in the combustion chamber is provided. The combustion without fuel-rich zones (premixed combustion) leads to decreased flame temperatures in the combustion chamber. Hence the chemical reactions that cause the formation of NO_x and soot are prevented. Recent research regarding the HCCI combustion process is mainly aiming on the automo-

Table 1: Properties of the fuel types MDO, LCO and HFO

Fuel type	Density at 15 °C	Cetane index	Aromaticity	Lower calorific value	Kinetic viscosity at 50 °C	Sulphur content
Unit	kg/m³	–	%	kJ/cm³	mm²/s	% (weight)
Marine Diesel Oil (MDO)	867	46	24	36.7	2.35	0.40
Light Cycle Oil (LCO)	934	23	79	38.8	2.45	0.18
Heavy Fuel Oil (HFO)	989	–	–	39.8	241	2.71

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Table 2: Technical specifications of the single-cylinder test engine

Characteristics	Value	Unit
Type of engine	diesel engine in two-stroke technology, with direct injection and supercharge	–
Power output	50	kW
Torque	1325	Nm
Bore	190	mm
Stroke	350	mm
Number of cylinders	1	–
Compression ratio	12	–
Mean piston speed	5.95	m/s
Exhaust valve timing	open 94° BBDC	–
	close 67° ABDC	
Scavenging port timing	open 48° BBDC	–
	close 48° ABDC	
Number of exhaust valves	1	–
Number of injection nozzles	2	–

bile industry; however the development of such combustion processes for ship diesel engines is of great importance regarding the introduction of strict emission limitations in the maritime sector.

3 Partially HCCI Combustion

Ship engines display the possibility of burning heavy fuel oils, which represent the bottom products of the distillation process. These residual oils mainly consist of long-chain hydrocarbons containing impurities such as sulfur, nitrogen and heavy metals. In order to reduce the high viscosity and the amount of impurities of such a fuel, it is common to mix the heavy fuel oil with other fuels (fuel blending).

One example is the blending of heavy fuel oil with low viscosity and low sulfur fuel. Typical fuels that are used for fuel blending are for example Light Cycle Oil (LCO) and Clarified Oil (CLO). LCO and CLO represent a residue from further fuel processing composed of vacuum distillation, desulfurization and catalytic cracking of the residual oil. **Table 1** shows the positive properties of LCO such as its low viscosity, high density and low fuel sulfur content; however linked to a high amount of aromatic compounds. Aromatic compounds generally result in a deterioration of the ignition properties of the fuel due to the higher activation energy required for radical formation compared to straight hydrocarbons. The poor ignition properties of LCO lead to long ignition delays, which limit the addition of LCO to the fuel mixture.

The Laboratory of Engine and Combustion (ECO) of the Kyushu University was therefore investigating new combustion processes aiming the application of LCO as fuel in ship diesel engines due to reasons such as its low sulfur content, the high availability and its low price on the fuel market [5, 6]. Main prospects of the research are the reduction of NO_x, SO_x and CO_2 emission in the maritime sector. Experiments on a medium size, single cylinder test engine investigate the suitability of LCO and MDO for partially HCCI combustion. MDO is a high quality ship diesel fuel that is widely used in marine vessels due to its good ignition and combustion properties. The use of LCO in gas turbines was investigated [6] however the results seem to be different due to the different methods of burning the fuel in internal combustion engines and gas turbines.

The principle of partially HCCI combustion [7] describes the division of the fuel injection into two parts (pre- and main injection) and has already been investigated [8, 9, 10]. The fuel-air mixture after pre-injection is burned according to the HCCI concept; the main injection leads to diffusive combustion assuring the burning of the complete fuel-air mixture in the combustion chamber.

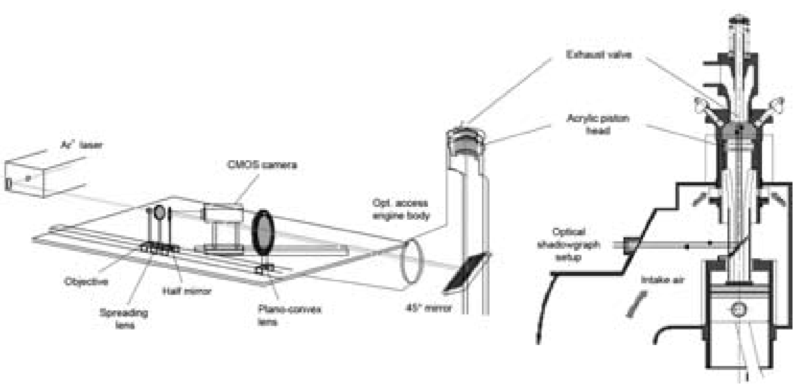

Figure 1: Optical measurement system for schlieren photography (left) and modified test engine for the visualization of combustion (right)

4 Experimental Setup and Conditions

In the following the experimental setup composed of test engine, injection system and optical measurement system for combustion visualization is presented. The test engine is a medium-speed, single cylinder, two-stroke diesel engine with 190 mm in bore and 350 mm in stroke. Further information can be obtained from **Table 2**.

The engine has been modified for the visualization of the combustion process using an optical measurement setup, **Figure 1**. The engine modifications are as followed: A window in the engine body enables the optical access to the engine.

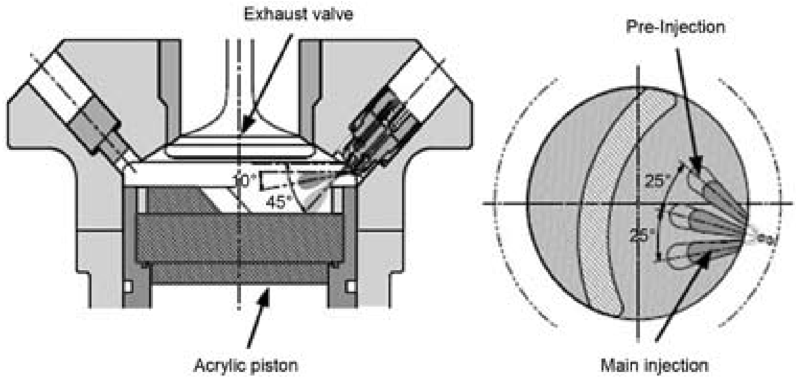

Figure 2: Schematic of fuel spray direction regarding pre- and main injection – cross section (left) and top view (right) of combustion chamber

Insight into the combustion chamber is realized using a mirror and an acrylic piston. The mirror is mounted in a 45° angle in the inside of the elongated piston. Due to the electronically controlled injection system and a specially ECO designed double needle injector, different fuel injection pressures and injection angles for pre- and main injection can be realized, preventing the fuel to impinge on the cylinder liner during early injection phase.

Due to the low temperature combustion during the HCCI process, the combustion occurs without the emission of light within the visible spectrum. For the purpose of combustion visualization an optical measurement setup using an Ar^+ laser as well as lenses, orifices and a high speed CMOS camera has been established.

The combustion in the combustion chamber is visualized using the schlieren method. The laser beam, which is spread and parallelized by the lenses, is refracted by the strong density gradients in the combustion chamber after passing the acrylic piston. It is then reflected on a mirroring surface on the bottom on the cylinder head and captured by the high speed CMOS camera after leaving the engine body.

The schlieren method visualizes the first derivative of the density field and is therefore providing useful information regarding propagation of pressure waves and flame front in the combustion chamber. Further in-cylinder pressure and temperature are measured by sensors for the investigation of heat release rates during combustion. The intake air pressure is accumulated by a compressor to a boost pressure of 0.26 MPa. In addition the intake air temperature is adjusted by an air heater, which is installed at the intake air duct. The engine rotation is assisted by an electronically controlled motor till a speed of 350 rpm. During pre-injection at -70 ° ATDC 0.3 cm^3 of fuel is injected through three nozzle holes with a nozzle diameter of 0.23 mm. The injection pressure for pre-injection is set to 40 MPa. The remaining fuel volume (0.33 cm^3) is injected during main injection at TDC through three nozzle holes with a nozzle diameter of 0.37 mm and an injection pressure of 75 MPa. Regarding pre-injection the fuel is injected with an angle of 45° and the main injection with an angle of 10 °, **Figure 2**. The fuel jets are leaving the nozzle with an angle of 25 ° to each other.

5 Results and Discussions

Figure 3 shows the pictures of the partially HCCI combustion of MDO and LCO visualized by the schlieren-system. After injection start at -70 ° ATDC, the fuel jets of MDO and LCO penetrating the combustion chamber can be seen at -66 ° ATDC. First differences can be determined from the evaporation behavior of MDO and LCO during jet penetration. At -60 ° ATDC the penetration of the MDO fuel jets can be clearly determined, whereas the fuel jets of LCO show already promoted mixture formation with the surrounding air resulting in a shorter penetration length. Consecutively the fuel is evaporated and mixed with the air in the combustion chamber. The air-fuel mixture becomes gradually transparent indicating the formation of a homogeneous mixture. The appearance of strong density gradients due to HCCI combustion could be obtained at -25 ° ATDC for the MDO case, whereas the ignition of the LCO-air mixture seems to appear at a later stage, at -15 ° ATDC.

From 5 ° to 20 ° ATDC the diffusive combustion due to main injection takes

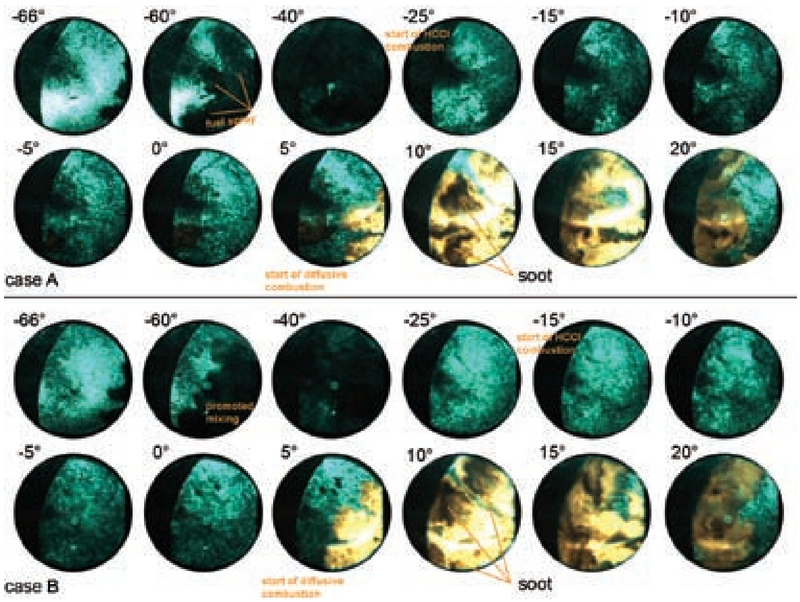

Figure 3: Schlieren pictures of partially HCCI combustions with the fuels MDO (case A) and LCO (case B)

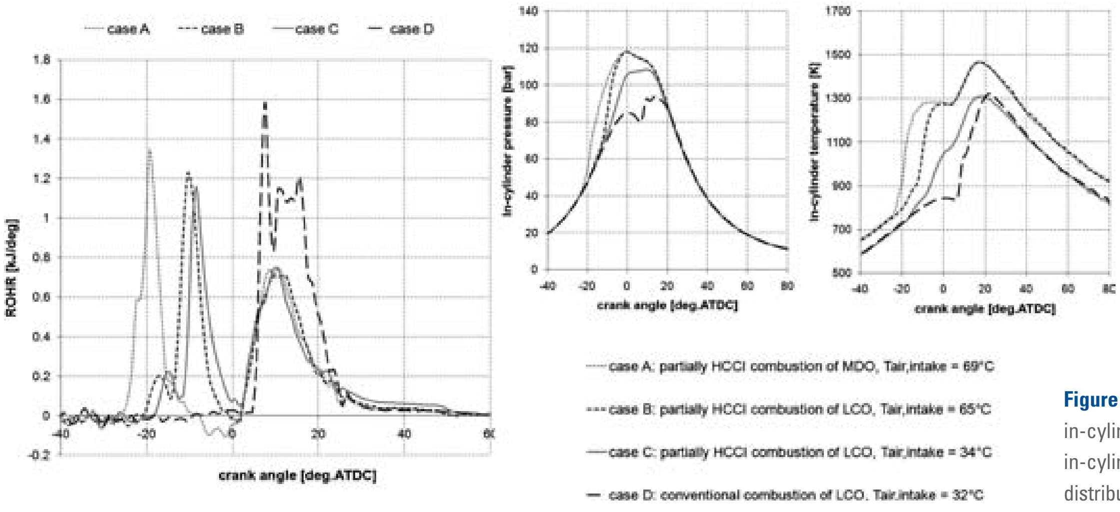

Figure 4: Heat release rates, in-cylinder pressure and in-cylinder temperature distribution for the cases A to D

place. The dark areas, which can be seen in the combustion pictures at for example 10 ° ATDC, display soot generating areas resulting from diffusive combustion [11]. Due to the absence of these dark areas during HCCI combustion, Figure 3, it can be assumed that no soot is formed during HCCI combustion. **Figure 4** shows the heat release rate as well as the in-cylinder pressure and in-cylinder temperature data obtained during the experiments.

A to D are the four cases, in which the effects of combustion process, fuel properties and intake air temperature on the ignition and combustion behavior have been investigated. Further the NO_x concentration in the exhaust gas of the cases C and D have been measured. **Table 3** represents the experimental conditions for the cases A to D.

The high quality MDO and the low quality LCO from the FCC-cracking process were used as fuels in this study. Figure 4 shows the expected heat release rate curve split up in the parts due to HCCI- and diffusive combustion. The rate of heat release during HCCI combustion displays the typical characteristics of HCCI combustion with a high pre-mixed peak and strong heat release gradients, which result in a sudden increase in in-cylinder pressure.

The differences in ignition behavior of MDO and LCO are consistent with the results obtained form the analysis of the schlieren pictures. MDO (case A) shows an early ignition timing at -25 ° ATDC. The strong pressure gradients in the combustion chamber followed by this early ignition lead to high complementary forces on the piston resulting in reduced thermal efficiency as well as in the danger of causing heavy damage to engine components. The application of LCO (case B) under the same experimental conditions, however, shows a delayed start of ignition at -15 ° ATDC, which can be referred to the high aromaticity of the fuel. LCO consists of classic aromatic single-ring systems (benzene) as well as a large amount of two-ring (naphtaline) and three-ring (anthracene) systems.

Table 3: Experimental conditions with the cases A to D

Case	Fuel	Combustion type	Intake air temperature in °C	Thermal efficiency in % in %
A	MDO	partially HCCI combustion	69	33.1
B	LCO	partially HCCI combustion	65	34.6
C	LCO	partially HCCI combustion	34	34.9
D	LCO	conventional diesel combustion	32	34.1

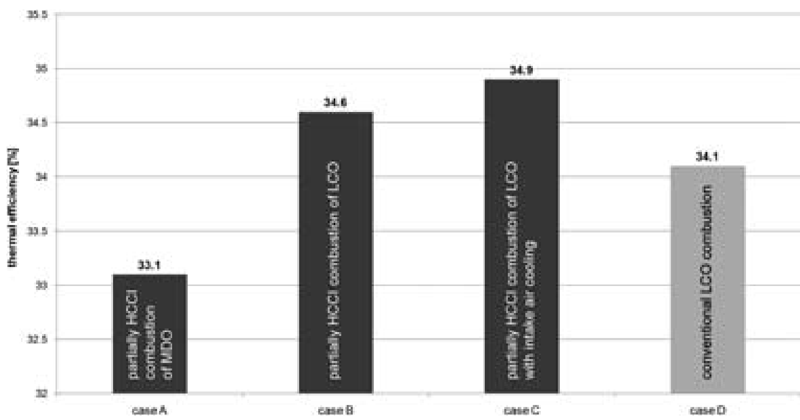

Figure 5: Thermal efficiency in the cases A to D

Therefore the activation energy needed in order to break up these structures is higher compared to straight hydrocarbons. Further shift of ignition start to TDC could be realized by lowering the intake air temperature (case C). Case D represents the heat release rate due to conventional diesel combustion of LCO with reduced intake air temperature. **Figure 5** shows the thermal efficiency of the different experiments.

The early ignition start of MDO and hence the high complementary forces on the piston result in the lowest thermal efficiency of 33.1 %. Due to the application of LCO the start of ignition is delayed significantly resulting in an increase of thermal efficiency of 1.5 to 34.6 %. Further gain in thermal efficiency to 34.9 % could be realized due to intake air cooling. **Figure 6** shows the NO_x concentration in the exhaust gas of the cases C and D. The NO_x concentration is increasing asymptotically due to mixing of the exhaust gas with the existing air in the exhaust pipe and is reaching its threshold value after 13 firing cycles. Due to the partially HCCI combustion of LCO the overall NO_x emission could be reduced of 10 % compared to the conventional diesel combustion of LCO.

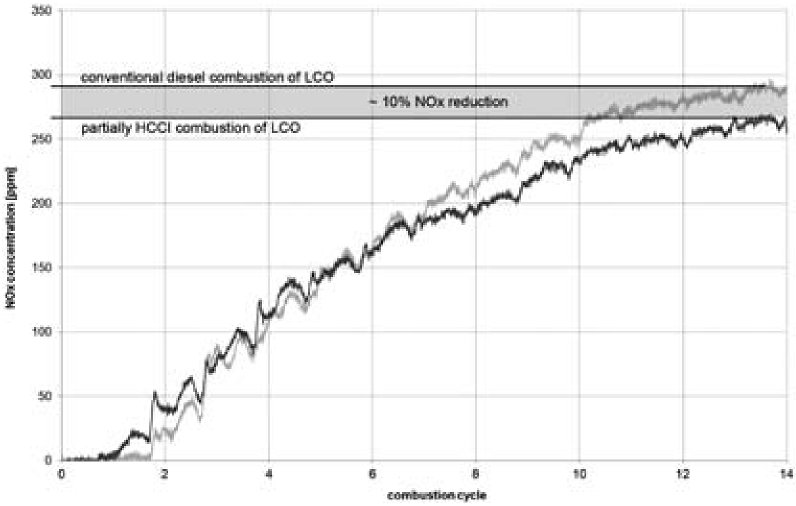

Figure 6: NO_x concentration with conventional and partially HCCI in the exhaust gas due to the number of combustion cycles

6 Conclusions and Future Prospects

Experiments at Kyushu University in Fukuoka (Japan) have been carried out using a single cylinder, two-stroke, diesel engine in order to investigate the application of low quality fuel with high aromaticity in state of the art combustion processes such as the partially HCCI combustion process. The HCCI combustion of the fuels Marine Diesel Oil (MDO) and Light Cycle Oil (LCO) was visualized according to the schlieren method using an Ar^+ laser.

The investigation points out the effects of fuel properties and intake air temperature on the ignition and burning behavior during partially HCCI combustion. Further, assumptions regarding soot and NO_x generation from partially HCCI combustion could be made. The experimental results can be summarized as followed: The use of MDO is likely to cause engine trouble due to too early ignition of the lean air-fuel mixture and is therefore not suitable for partially HCCI combustion despite its good combustion properties. However the bad ignition behavior of LCO due to its high aromaticity combined with its good evaporation properties turn out to be advantageous applied in HCCI engines [12]. The high potential of NO_x reduction due to HCCI combustion combined with the low fuel sulfur content of LCO display a promising approach for the reduction of pollutant emission from marine diesel engines. The analysis of the combustion images show a clear combustion without presence of soot generating areas indicating that almost no soot was formed during HCCI combustion of LCO. Further, the application of LCO in partially HCCI combustion and the reduction of the intake air temperature result in an increase in thermal efficiency of 1.8 to 34.9 %. The NO_x concentration in the exhaust gas could be reduced by 10 % due to partially HCCI combustion compared to conventional LCO combustion.

Continuous research is aiming the practical application of LCO in ship diesel engines as well as a theoretical approach to the combustion process using current CFD-simulation software. In order to simulate the combustion of high aromatic fuels such as LCO accurately, the development of advanced ignition and combustion models, which include the chemical and physical properties of the fuels, is required.

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