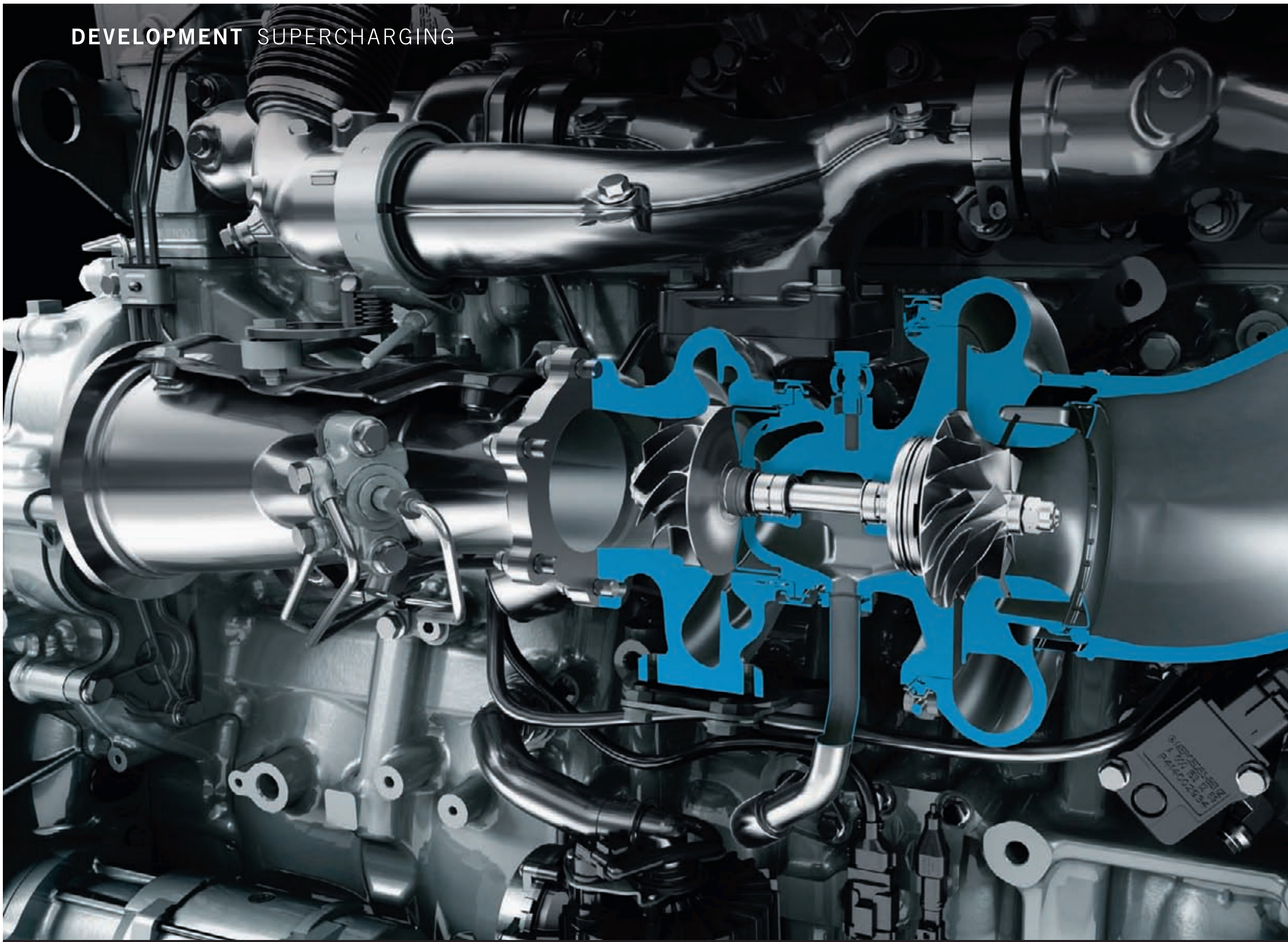
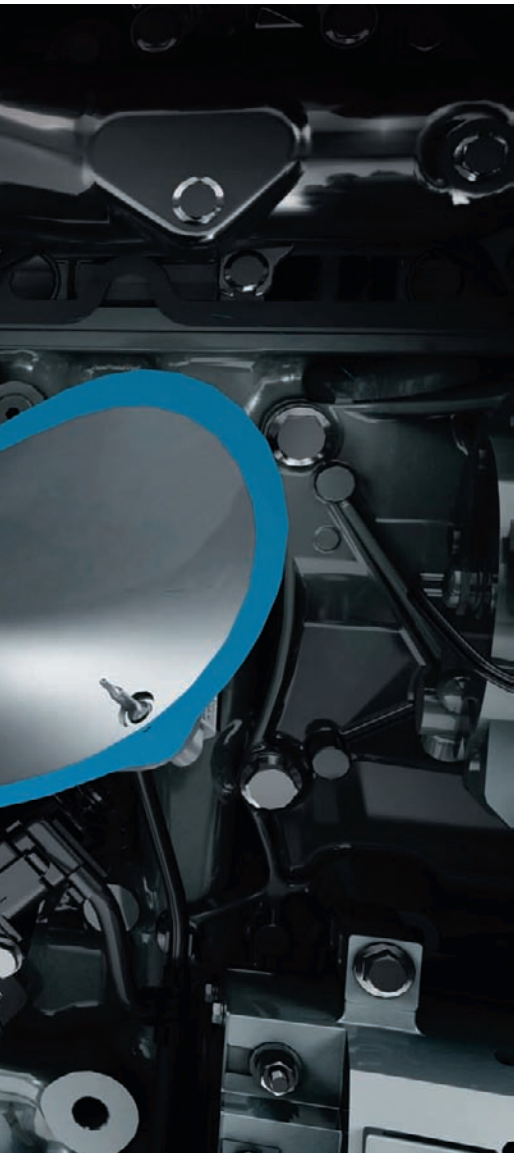


DEVELOPMENT OF AN EXHAUST-GAS TURBOCHARGER FOR HD DAIMLER CV ENGINES

The matching of a turbocharging system to the specific requirements of an entire engine with regard to fuel consumption, emissions and service life is a key element of engine development. This is the reason for taking the decision to initiate in-house turbocharger component development as the New Engine Generation (NEG) engine series of Daimler AG was being developed. The aim of this in-house turbocharger development is to produce the best possible turbocharging system for the entire engine with regard to fuel consumption, emissions and economy.



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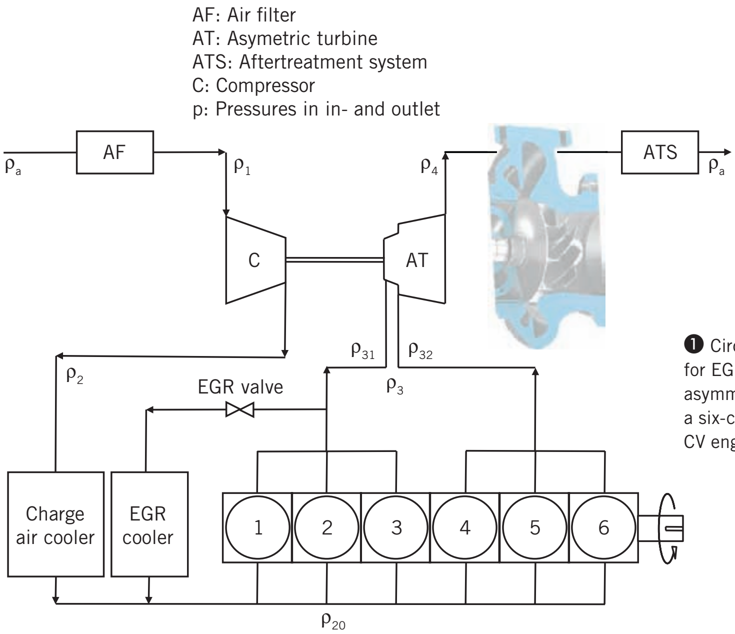
INTRODUCTION

In large-scale commercial vehicle production, the precompression of combustion air via exhaust-gas turbocharging in diesel engines has been with us since the 1960s. At the end of the 1970s, turbocharging technology also conquered the passenger car (PC) diesel engine sector. This trend can also be seen in gasoline engines in the past five years. This progression towards an almost complete market penetration clearly shows that the exhaust-gas turbocharger is the most efficient unit for combustion engine turbocharging. In the heavy-duty (HD) commercial vehicle (CV) engines of Daimler AG, the exhaust-gas turbocharger is also – via the use of an asymmetric twin-scroll turbine – used as an extremely efficient exhaust gas recirculation system (EGR). It is possible, thanks to the asymmetric turbine, to dispense with the alternative variable turbine geometry, which is also less advantageous with regard to fuel consumption, quality and economy. The advantages resulting from asymmetric turbocharging are used in the modern NEG commercial vehicle engines of Daimler AG in order to already be the benchmark with regard to fuel consumption and service life for future emissions legislation.

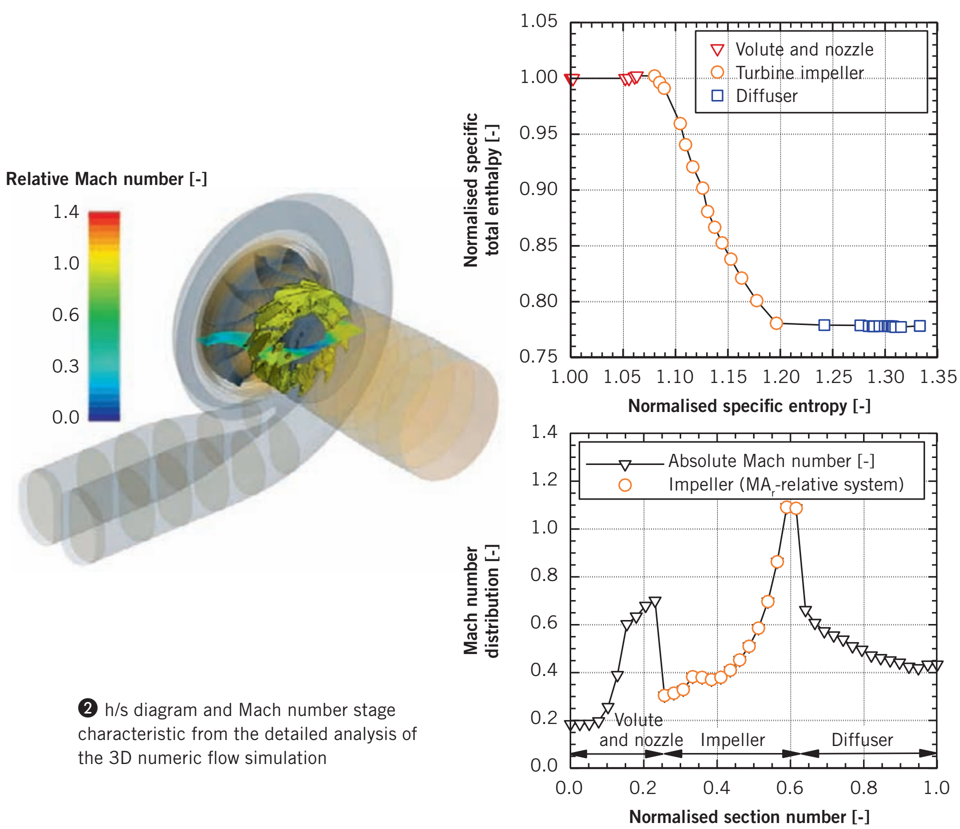
For achieving this target, it is essential to integrate the exhaust-gas turbocharger development into the entire engine development. A deep understanding of the interaction between the reciprocating piston engine and turbomachine is required in order to obtain the best performance from the combination of both units.
The major difficulty related to the custom matching of the exhaust-gas turbocharger to the engine is due to the fact that the turbocharger modular systems available on the market do not necessarily represent an optimum for the turbocharging system of the overall concept. This problem occurs, in particular, in the commercial vehicle sector. The low unit figures in comparison to the PC sector also mean that only a few turbocharger suppliers have included the CV sector in their product range.

The economic and technical evaluation of this situation led Daimler AG to the decision to develop its own exhaust-gas turbocharger for the HD CV engine series, which has started in 2006. Beginning in 2013, this exhaust-gas turbocharger, produced in the Mannheim plant, will be offered for the first time in the NEG engine internally called OM472 for the Freightliner Cascadia Evolution; the turbo-compound supercharging system will still also be offered [1]. The model change towards asymmetric turbocharger with the subsequent specific matching of the engine requirements has resulted in a 4 % fuel consumption advantage, alone due to the engine. Additional applications with an in-house developed exhaust-gas turbocharger will follow in the coming years based on the evolutions in the current engine series.

In the following sections, the necessary steps for the development of an exhaust-gas turbocharger are described. The basic turbocharging concept will be presented, the turbocharger design shown and the mechanical validation up to production standard described.



❶ Circuitry concept for EGR supply with an asymmetric turbine for a six-cylinder Daimler CV engine

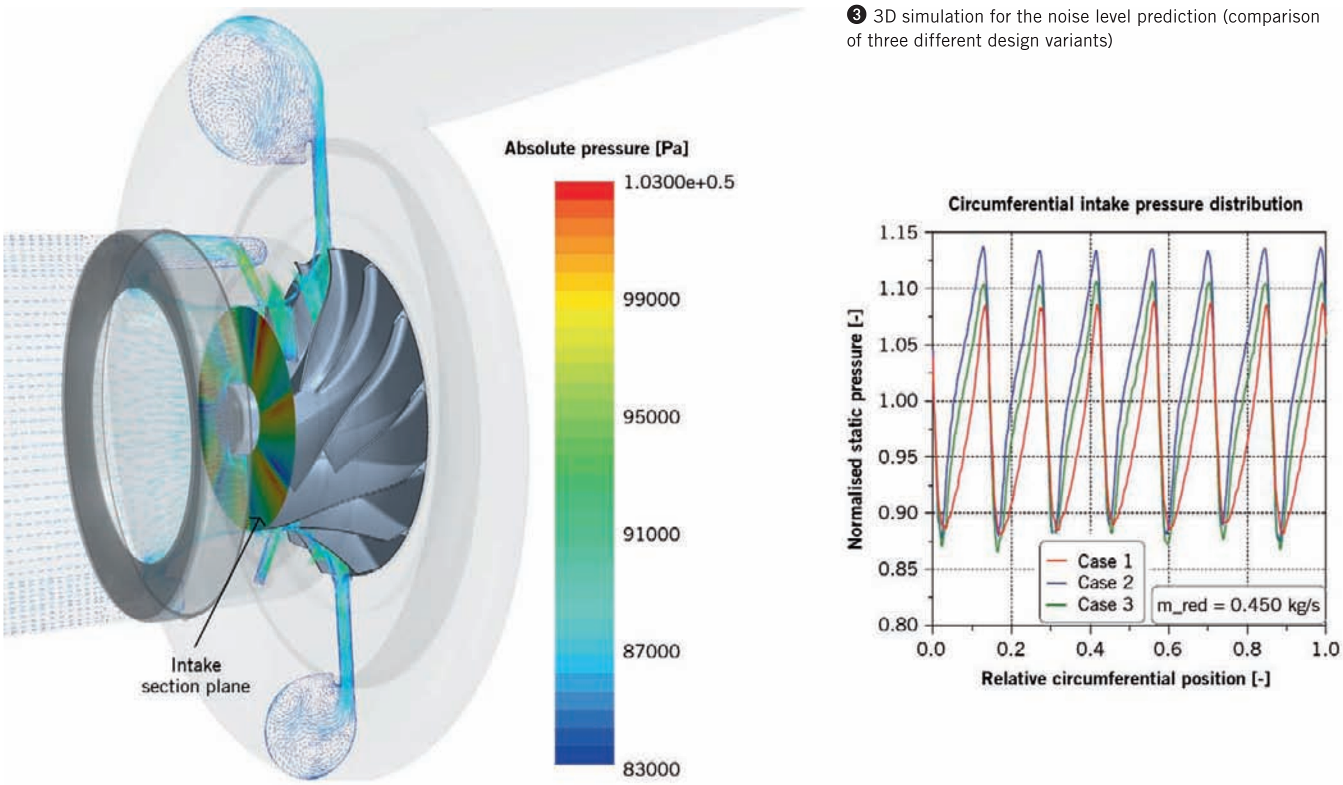


❷ h/s diagram and Mach number stage characteristic from the detailed analysis of the 3D numeric flow simulation

THE ASYMMETRIC TURBINE

For the six-cylinder in-line engines with between 10 and 15 l of displacement, Daimler is banking on the patented turbocharging concept with an asymmetric twin-scroll turbine due to the need to achieve exhaust-gas recirculation (EGR) rates of up to 35 % in full load, achieve high turbocharging efficiency (particularly in the main operating range) as well as ensure an exhaust-gas turbocharger operating service life of over 1 million km, while at the same time producing an extremely economic engine concept. The exhaust gas recirculation rate is mainly determined by the asymmetrical twin-scroll turbine size acting together with the engine. Here, one of the two turbine scrolls is sized such that an increased exhaust-gas flow backpressure effect is created. The exhaust gas can thus also be used for air supply purposes due to the cylinder group partitioning, ❶.

When the EGR valve is open, engine cylinders 1 to 3, ①, work as an EGR pump, which pumps part of the stream of exhaust gas to the engine intake through the recirculation channel prior to the turbine inlet through the EGR cooler on the air side using the generated pressure difference (p_{31}-p_{20}). The remaining exhaust gas of this cylinder group flows through the EGR scroll of the splited twin-scroll turbine. Cylinder group 4 to 6 is gas-tight in the turbine inlet and separated from the first cylinder group; all its exhaust gas passes through the larger scroll (p_{32}) at a generally lower pressure ratio p_{32}/p_4. The concept of the twin-scroll fixed geometry turbine thus allows you to master a high-pressure EGR transport with positive charging cycle (p_{20}-p_{3M} > 0; p_{3M} = average energetic turbine inlet pressure of both cylinder groups), whereby an engine fuel consumption reduction can be derived in comparison to other EGR turbocharging systems that do not offer any cylinder group partitioning. This concept was first introduced with the US EPA 2004 legislation in the OM460 CV engine. The outstanding field experience with this engine confirmed the robustness of turbocharging with asymmetric turbines.



❸ 3D simulation for the noise level prediction (comparison of three different design variants)

DESIGN

Both numeric and experimental procedures were used for designing the exhaust-gas turbocharger. While a large part of the aerodynamic and thermomechanical turbine design can nowadays be achieved in virtual prototypes based on existing simulation methods, a higher empirical effort is required when it comes to specifying the compressor and bearing geometries.

THERMODYNAMICS OF RADIAL TURBINES

During the design of the asymmetric turbine, the individual scroll size must be closely defined with the engine thermodynamics. Numerically simulated h/s and Mach number diagrams show the turbines areas with the highest losses and are used, ❷, for optimising turbine efficiency.

The radial turbine impeller is designed with the largest possible exit angle β from a casting perspective. This amounts to -57° and allows for high turbine efficiency, particularly in the part load operating range of the engine. For a further efficiency increase at part load, the target degree of reaction at full load is selected to that the maximum efficiency is moved in the direction of the partial load. This optimisation significantly depends on the engine application profile of the vehicle and thus represents a compromise between high subcomponent efficiencies and low fuel consumption. The consistent use of the complex numerical simulation processes was essential to design an asymmetric turbine having a peak mechanical efficiency $\eta_{T,mech}$ of almost 71 %.

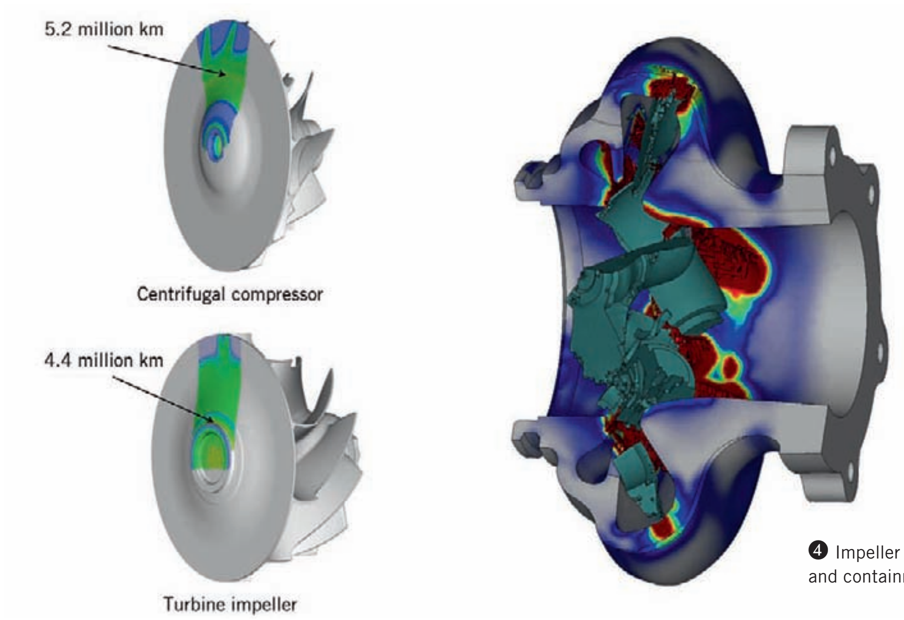
THERMODYNAMICS OF CENTRIFUGAL COMPRESSORS

A compression ratio of 4.2 is striven for the design of the single stage centrifugal compressor. A peak efficiency of $\eta_{C,is}$ = 80 % is reached as a compromise between map width and optimum efficiency. This is enabled by a numerically optimised back sweep blades with the accordingly required compressor blade wrap angle. In a second step the diffuser and the ported shroud casing treatment are optimised. The individual design parameters of the stage are set via experimental studies, and supported by means of numerical simulations, ❸.

A so-called noise reflector is implemented in order to decrease the compressor noise level as well as increase the comfort level in the cabin and fulfil future, more stringent noise emission legislation. For the noise reflector design, particular care is taken to ensure that the compressor efficiency and surge characteristics are not negatively affected. For this purpose, numerical methods have been developed that allow the localisation of the noise source. This noise reflector reduces the overall noise level by around 2 dB.

MECHANICAL STRENGTH

The blade thickness distribution of the turbine impeller is designed such that the resulting blade's natural frequency (eigenfrequency) is above the fifth order of the rotor speed. In this way vibration fractures due to excitations from asymmetric turbine scroll could be prevented. The same value is also used as design criteria for the compressor blades. To ensure the required impeller service lives, their fatigue limit is determined from the aggregate speed values of the customer-relevant test cycle, ❹. The development of this method includes the impeller stress calculation, the derivation of a damage matrix based on the



❹ Impeller service life simulation and containment simulations

material-specific Wöhler-curves and the statistical transferability of the test sample results and load profiles. The analytical optimisation of the impeller-back geometry and the use of strain hardening via ultrasonic shot peening are key factors for increasing the service life of the aluminium centrifugal compressor and the inconel turbine impeller. The balancing marks are positioned on the back of the impeller near the tip so that mechanical stress peaks are outside the balancing mark area.

To ensure the in-house safety requirements, the compressor and turbine housing must be able to withstand the bursting of the respective impeller. For this purpose, containment simulations are carried out to detect housing weak spots and eliminate these by improving design. The turbocharger is then released after an experimental confirmation of the simulation results.

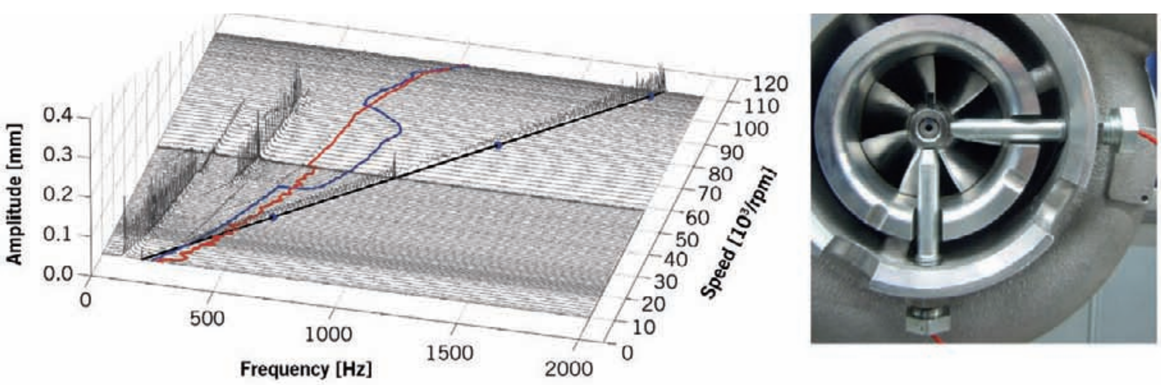
BEARINGS

The classic dual float bush radial bearings as well as a wedge-shaped axial bearing are used between the impellers. This combination is an extremely well-suited concept with regard to the friction loss, robustness vis-à-vis oil grade, imbalance and mechanical efficiency. The main dimensions are determined by means of a linearised vibrational mode analysis of the rotor assembly. The detailed geometry is then specified via non-linear ramp-up simulations of the overall system. The numerical simulation of the plain bearing system is continuously supported and validated in the development process via the measurement of the friction loss, structure-borne sound and shaft orbit offset in modularised prototype assembly systems.

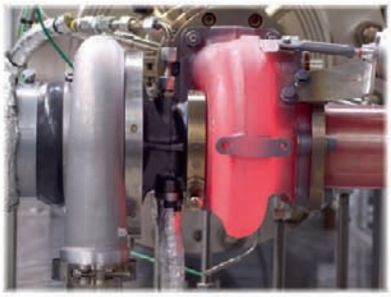
The rotor shaft orbits measured during a ramp-up are analysed using an Fast Fourier Transformation FFT and evaluated with regard to sub-synchronous oscillation rates as well as instability characteristics, ❺. The bearing damping is increased via suitable design measures until the bearing runs in a stable manner at maximum speed, at a reduced oil pressure of 1.5 bar and an oil supply temperature of 120 °C, taking into account all production and imbalance tolerances.

FUNCTIONAL VALIDATION

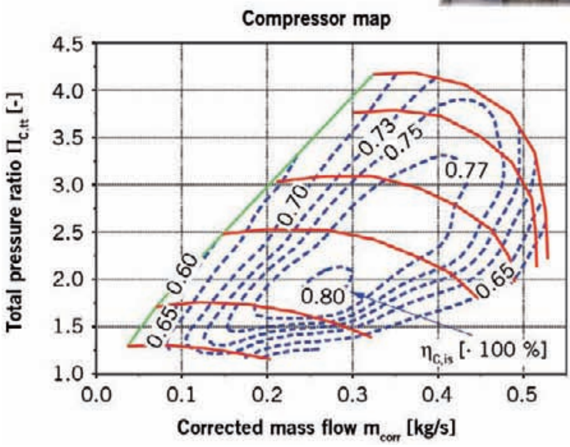
The aim of the functional validation is to verify the required turbocharging system performance and reliability via suitable endurance test programmes on the component, engine test stands and in the vehicle. For this purpose, the exhaust-gas turbocharger characteristic maps are inspected continuously on the gas test stand during the entire development phase, ❻. The characteristics must remain

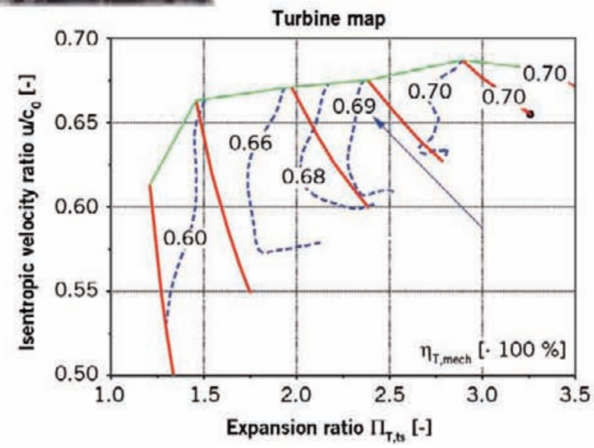


❺ Shaft orbit measurement including bearing bush speeds of the bearing system (red: speed turbine bushing; blue: speed charger bushing; black: speed charger)



❻ Compressor and turbine characteristic maps





constant between the individual sample statuses and after the endurance testing. To ensure the long-term quality of the rotor assembly, a centre section imbalance is striven, which does not change during hot operation. This is monitored in a large number of turbochargers, either via back measurements on balancing machines or via the imbalance measurement method during operation. It is thus ensured that the turbocharger operates at the oil supply limit of the engine properly. This is additionally validated by a 1000 h low oil pressure endurance test.

The required impeller lifetime is checked by specific accelerated tests on cold spin test stands as well as hot turbocharger and engine test benches. Compressor and turbine impellers are operated with the largest possible amplitude from minimum to maximum circumferential speed so that alternating stress is generated in the impellers due to the centrifugal force. The achieved number of cycles up to fatigue fracture reflects the service life and at the same time provides valuable data for comparing and calibrating the numeric structural simulation models. The impellers are optimised so that they achieve over 200,000 cycles at a given accelerated test on the hot turbocharger rig. Based on the numerical structural simulation, the number of cycles is converted into kilometres by adopting a customer-relevant cycle. Hot/cold accelerated tests are carried out on the component test stand, followed by 2000 h endurance tests on the multi-cylinder engine to check the thermo-mechanical strength, especially that of the turbocharger particularly hot parts.

Special solutions are developed that ensure the oil leak-tightness of the turbocharger, due to the typical long engine idling phases currently available in the American market. For this purpose, special accelerated tests, in which the pressure ratio is varied between the bearing housing and compressor back-side, are carried out on the gas test stand. The resulting geometry optimisation is subsequently confirmed by evaluation runs on the multi-cylinder engine. The active boost pressure control of the engine requires a very high number of wastegate valve movements. The actuator of the boost pressure regulating valve is developed so that at least 4 million cycles can be carried out without failure. Furthermore, the wear in the moving components is minimised by suitable material combinations.

Vehicle tests under extreme weather conditions such as coldness and hotness are essential for the final turbocharger validation. These include, for example, test drives in the Spanish Sierra Nevada at 2500 m above sea level at an ambient temperature of over 30 °C and test drives in the USA up to 4000 m altitude. Under these conditions, the compressor margin against surge and the maximum load temperature of the individual components are checked. In addition to Europe, further tests take place in Asia, America and Africa. In sum, over 9 million km as well as 150,000 h engine operating and at least 70,000 h gas test stand time were driven in order to release the turbocharger. This operating time was required in order to satisfy the high performance, safety and quality requirements of Daimler.

SUMMARY AND OUTLOOK

Future optimisations in the diesel engine are mainly possible if subcomponents such as the turbocharging system are specially adapted to the specific requirements of the engine via integrated turbocharger development. The in-house turbocharger development at Daimler, which started with the development of the NEG engine OM472, has produced the optimal turbocharging system for the use of that engine with regard to fuel consumption, service life and economic efficiency. This convincing end product will result in further applications in the future with an in-house developed exhaust-gas turbocharger adapted to the specific engine requirements.

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[1] Heil, B.; Schmid, W.; Teigeler, M.; Sladek, W.; Öing, H.; Arndt, S.; Melcher, S.: The New Daimler Heavy Commercial Vehicle Engine Series. In: MTZ (70) 2009, No. 1