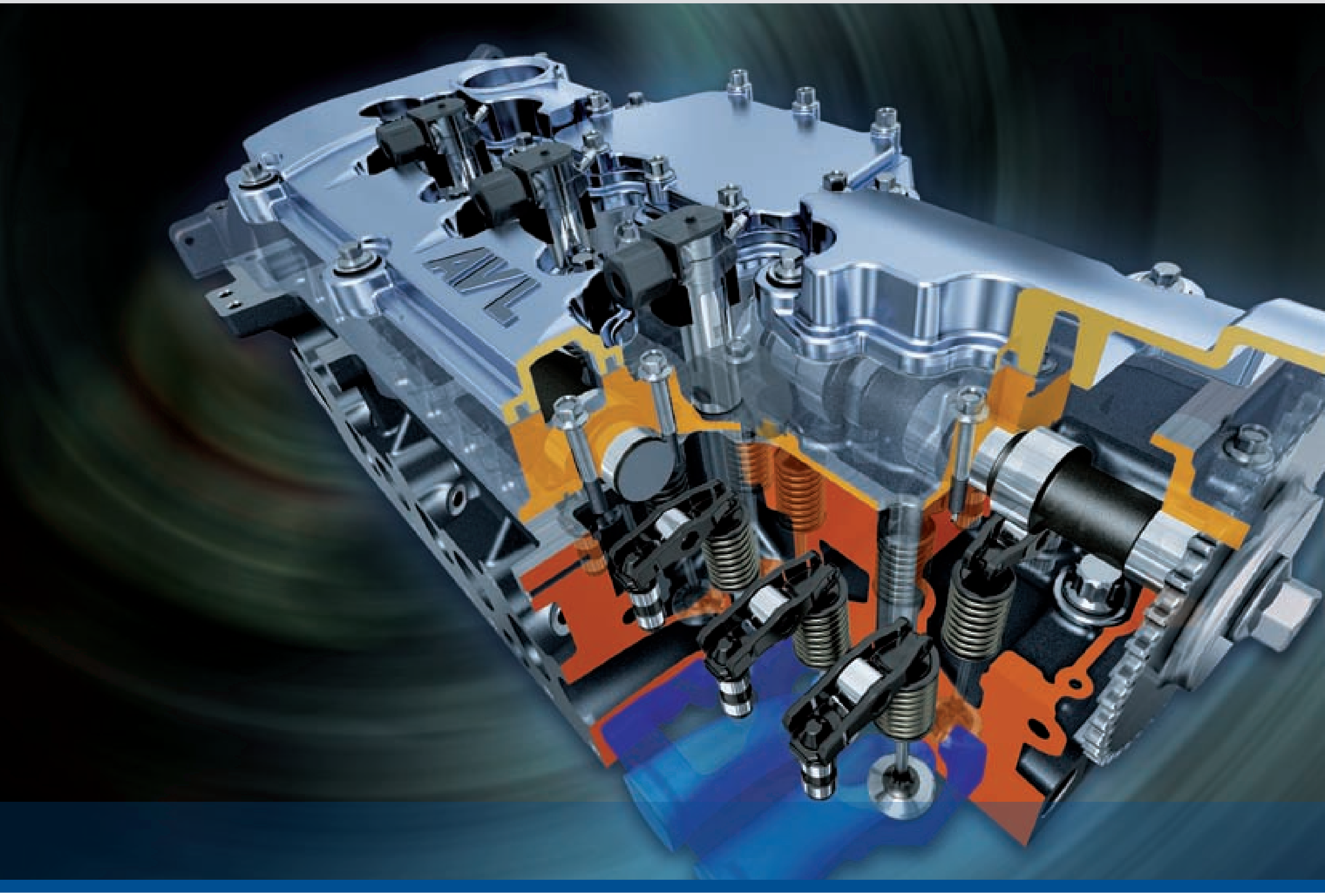


Highly Loaded Cylinder Heads in Diesel and Gasoline Engines
Trends and Potentials

Increasingly stringent emission regulations and higher demands with regard to performance and fuel consumption accelerate the introduction of new technologies and development procedures for today's diesel and gasoline engines. The resulting increased thermo-mechanical loads on the basic engine, in particular the cylinder head and the exhaust gas system, are especially challenging. This paper discusses future solutions from AVL for highly loaded cylinder heads within the context of the conflict „technology versus costs".

1 Introduction

The emissions and fuel economy driven development of passenger car diesel engines is leading to ever higher peak cylinder firing pressures. Design optimisation under consideration of the application limitations of potential component materials is described using the cylinder head as example. For the gasoline engine, the requirement to reduce fuel consumption is leading to a trend towards turbocharging with VTG technology and the integration of exhaust gas routing within the cylinder head. Design solutions addressing these developments are illustrated. In addition, this paper deals with the processes necessary to meet exacting scheduling and quality standards, and the intergrated application of methods implemented within AVL in the development of the basic engine. This Design Validation Process (DVP) constitutes a symbiosis between two approaches proven in practice, “AVL Frontloading” and “AVL Load Matrix”.

2 Cylinder Heads for Passenger Car Diesel Engines

2.1 Trends

Since its introduction in Europe the high-speed direct-injection diesel engine has had an ever increasing market acceptance and thus achieved a market share of more than 50 %. A worldwide interest in passenger car diesel engines, particularly in the US and Asia, exists due to the incessant increase of fuel costs, limited crude oil resources and the engine’s attractive driveability.

In the future the diesel engine must be able to fulfil stringent legislation at acceptable cost without losing its fuel consumption advantage in competition with gasoline engines. At the same time its power characteristics must be adapted in order to maintain its competitive position especially relative to turbocharged direct-injection gasoline engines. **Figure 1** shows the expected maximum specific power and torque for the basic diplacement classes.

In order to avoid an essential increase of cost disadvantage of the diesel compared to the gasoline engine, see also Figure 7, the number of engine families will tend to be reduced which in turn will result in a higher production volume. A concentration of technology which is expected to occur with it supports the ‘economy of scale’ approach. The technological challenge will be an increased performance range within the engine families. Bearing in mind that a performance increase is usually accompanied by a higher peak firing pressure shows the need to provide standard and premium versions with high power density on the basis of largely unmodified basic engines with minimum compromising. Particularly with regard to the tightened CO_2 legislation this is of major importance. The decision for the right way of attaining this goal will be dependent on the respective engine portfolio. The spread of the expected power and torque requirements will increase and especially so in the volume production/High-end area, ranging from about 50 to 75 kW/l, thus concerning primarily engines of capacities between two and three l.

Against this background in the following chapter the correlations between peak firing pressure, compression ratio, and peak temperatures of aluminum cylinders used will be discussed from which core strategies for the design, focussing on structural stiffness and adequate cooling systems, can be deduced. The simulation of thermomechanic fatigue due to thermomechanic alternating stress in the process is the greatest challenge on strength calculation. By means of an example the methods used by AVL will be described, that are not only centred

Authors

Dr. Helfried Sorger is Head of Design, Engineering and Technology Powertrain Systems at the AVL List GmbH in Graz/Austria.

Dipl.-Ing. Franz Zieher is Head of Analysis, Engineering and Technology Powertrain Systems at the AVL List GmbH in Graz/Austria.

Dr. Ulrich Sauerwein is Manager Mechanical Development, Engineering and Technology Powertrain Systems at the AVL List GmbH in Graz/Austria.

Dr. Wolfgang Schöffmann is Manager Engine Design, Engineering and Technology Powertrain Systems at the AVL List GmbH in Graz/Austria.

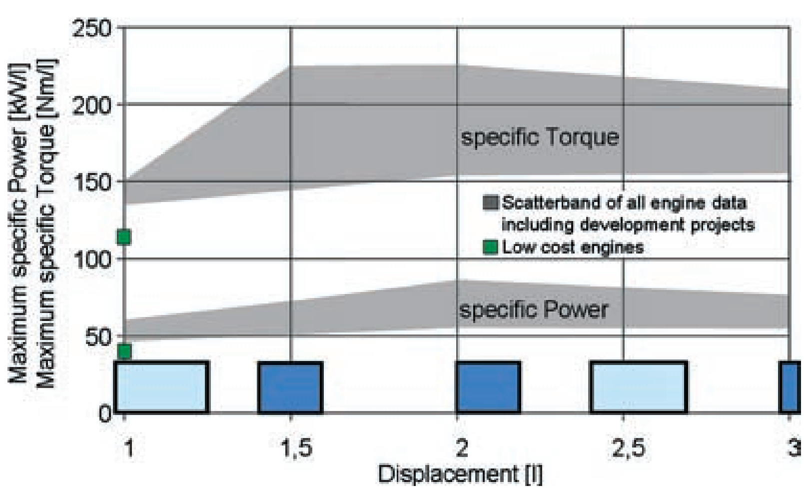


Figure 1: Specific performance for swept volumes

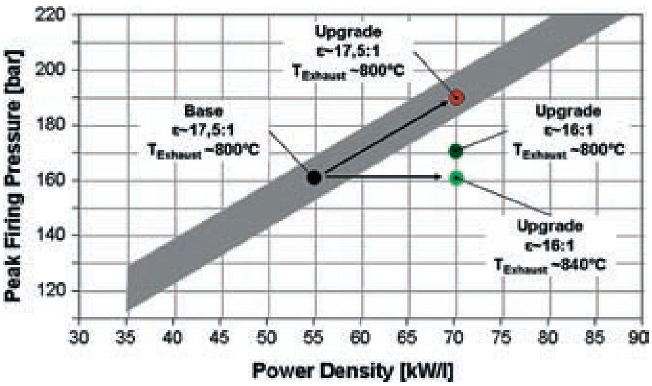


Figure 2: Effect of compression ratio and exhaust gas temperature on peak firing pressure

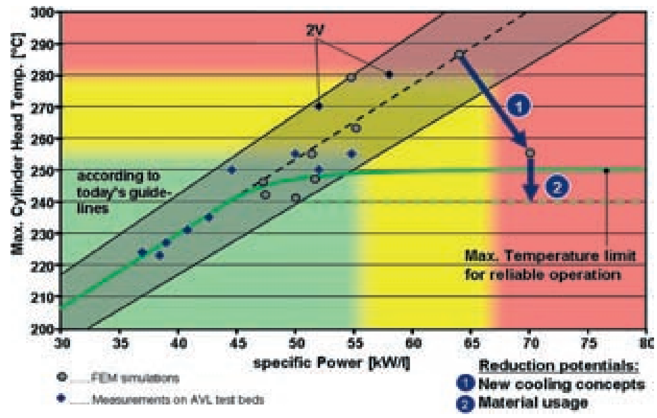


Figure 3: Cylinder head temperature vs. specific power

around HCF (High Cycle Fatigue) and TMF (Thermo Mechanical Fatigue) but also take into account stresses caused during the manufacturing process.

2.2 The Correlations Between Peak Firing Pressure, Compression Ratio, and Temperatures

Due to today's fuel injection technologies with ever rising injection pressures and variabilities as well as improved cold-starting aids the geometrical compression ratio can be reduced to an amount ranging from 15.5 to 15:1. **Figure 2** shows that an increasing power density results in the continuous increase of the cylinder peak firing pressure which functions as major parameter for the design of the basic engine. Apart from the cranktrain special attention has to be paid to the selection of the material to be used as well as the concept design of the crankcase [2,3] and the cylinder head.

In the face of emission scenarios to be met (Euro5, Euro6, US) and the demand of reduced CO_2-emissions (friction-optimised) the peak firing pressure requirements in the future are expected to range between 190 and 200 bar.

Another method to reduce peak firing pressures is to increase the exhaust-gas temperature. The example shows that at a constant peak firing pressure an increase of specific power from 55 kW/l to 70 kW/l can be achieved. As a matter of course, the increased demand on cold start and cold running, on the cylinder head peak temperature and the turbocharger have to be considered.

Figure 3 illustrates the relation between specific power and the resulting maximum cylinder head temperatures. It can be inferred from test bed measurements and simulations that for a specific power of >55 to 60 kW/l with unmodified geometry, cooling strategy, and material the temperature of 4-valve cylinder heads rises in a linear way, resulting in temperatures >280 °C, which are not permissible. Adequate cooling concepts on the cylinder head, however, can reduce this peak temperature by 25 to 30 °C. Furthermore, optimized casting methods (e.g. dual alloy casting), systematic machining process steps (e.g. HIP process), or the use of high-grade alloys allow an additional reduction of 8 to 10 °C.

2.3 Core Strategies for the Design of Highly Loaded Cylinder Heads

The investigations carried out by AVL result in the following design features for highly loaded cylinder heads:
- Parallel valve pattern with short ports to provide symmetrical stiffness
- Moderate valve sizes in order to increase the cooling flow between the valves
- Closed profile structures to ensure uniform stiffness
- Introduction of a conical intermediate deck for extremely high peak firing pressures
- Possibility of thermal expansion of the valve bridges, e.g. by the use of saw cut
- Precision cooling for the valve bridges located close to the fire deck
- Cross-flow cooling for uniform cylinder temperatures

Figure 4 shows a 4V aluminum high-performance cylinder head, designed according to the above mentioned design strategies, with a single piece water core. Apart from its compact design with regard to friction reduction it is equipped with a roller finger follower with hydraulic valve-lift adjustment, a concept used in almost every model of today's diesel engines. Possible implementations for the crossflow cooling are either with a single piece or a dual piece water core, the single piece one being favoured due to functional and cost reasons.

In conclusion it is found that, taking into acount all of the mentioned design properties, specific power of about 70 kW/l at a maximum peak firing pressure of 190 bar is possible, whereby a single piece water core is adequate to meet the peak temperature of about 250 °C for the AlSi aluminum alloy used.

2.4 HCF and TMF Simulation Methodologies

The performance limit of today's high-speed diesel engines is defined by the thermomechanic load and the resulting thermomechanic fatigue (TMF). For the assemblies piston, cylinder head, exhaust manifold and exhaust-gas turbine different modes of damage occur due to the different materials used and the specific load conditions. On the cylinder head the superposition of high-frequent combustion pressure stress and low-frequent thermal alternating stress causes also a superposition of High Cycle Fatigue (HCF) and Thermo-mechanical Fatigue (TMF).

AVL has developed and standardly uses simulation methodologies [10-13] that allow to predict the lifetime and fatigue of the assemblies mentioned and thus permits the thermomechanic optimization in terms of the Design Validation Process already during the virtual development phase, see chapter 3. In the following the method will be described by means of the example of a passenger car diesel engine cylinder head and is used in an analagous manner for heavy duty cylinder heads, pistons, exhaust manifolds and exhaust-gas turbine housings. The simulation represents the defined test run in a realistic way and by the following steps:

- thermal shock test cycle in agreement with AVL Load Matrix results
- CFD-calculation of the combustion and gas exchange processes for critical engine conditions in the test cycle to enable the determination of the local heat transfer
- CFD-calculation of the coolant flow for the critical engine operating points and derivation of the local heat-transfer coefficient
- FEM calculation for the temporally variable component temperatures for at least two test cycles, taking into account local boiling effects (AVL/BDL boiling model)
- cyclic FEM calculation of stress and elongation conditions caused during assembly and the thermal load over several test cycles using viscoplastic material models
- prediction of the number of test cycles until the occurrence of cracks based on the occurring stress-elongation-hysteresis and experimentally determined material damage parameters.

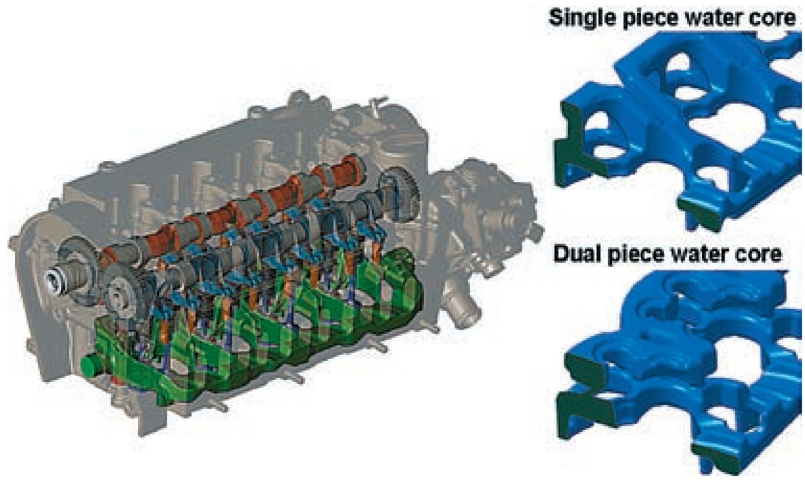


Figure 4: Crossflow cooling, single piece vs. dual piece water core

In order to find materials suitable for the thermomechanic alternating stress as it occurs in the test cycles AVL has developed special material models describing the ageing behaviour for aluminum alloys and the asymmetry for cast-iron materials under tensile and compressive loads. The necessary model parameters are determined by specific thermomechanic sample tests via the operating-temperature range. Damage prediction thereby is based on the stress-elongation-hysteresis in stable conditions, i.e. after corresponding ageing at operating-temperatures.

By means of the example of a cylinder head fire deck **Figure 5** shows the calculation of the test cycles to the point of cracking. Corresponding to the description above several test cycles are simulated with a holding period of about 30 hours. The holding period was introduced to represent the overaging of the material during the entire period of the test cycle. The yield stress is considerably reduced and the stress-elongation-hysteresis strongly increases. This hysteresis loop is used as the basis to predict the number of cycles until crack initiation.

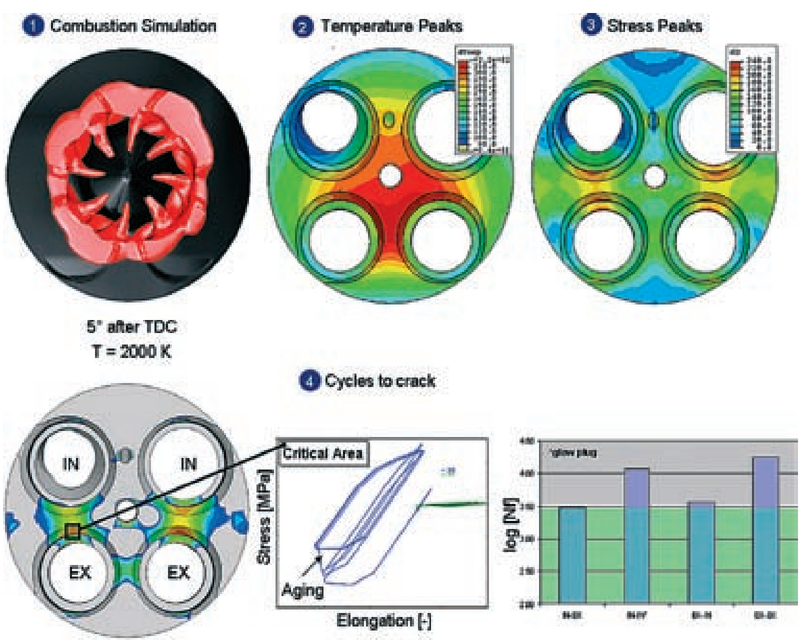


Figure 5: Crack-Simulation

2.5 Stresses Caused During the Manufacturing Process

By now the consideration of stresses caused during the manufacturing process is indispensable for the virtual development phase of highly loaded systems to determine the respective strength requirements.

In the manufacturing process of an aluminum cylinder head residual stresses emerge during the quenching phase that are only slightly reduced by heat treatment. Due to removal of material during machining a redistribution of residual stresses occurs. Additionally, permanent stresses can be caused by the machining process itself. At the end of the quenching phase the structure is fine-grained, resulting in a very low yield stress. Thus, the material characteristics also have to be recorded for the material condition following the quenching phase to be used in the simulation model. **Figure 6** illustrates the sequence of the implemented simulation steps.

Measurements often show residual stresses of 70 MPa and above. The occurrence of such residual stresses at areas with high operational stress can result in an overstress and thus in a premature failing of the component. Consequently by now the quenching process is conducted on air ('air quenching') by which the internal stresses can be significantly reduced. The disadvantages of this method are increased costs and the reduction of the fatigue-strength value by 10 %, which in turn results in a reduction of the overall load capacity of the cylinder head.

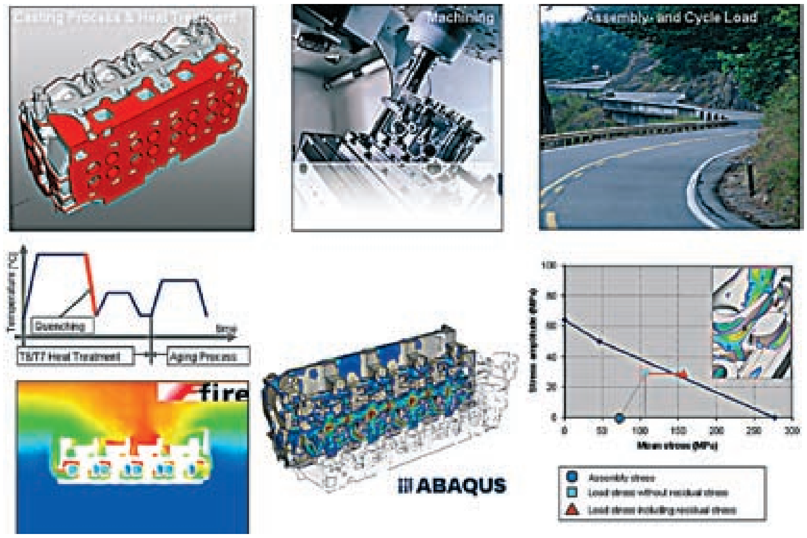


Figure 6: Stress simulation according the manufacturing process

3 Cylinder Heads and Turbocharging for Passenger Car Gasoline Engines

3.1 Trends

Until only a few years ago turbocharging for gasoline engines was primarily used to increase the performance range of existing engines. In the course of further fuel consumption reduction via the shifting of operating points turbocharging for downsizing-/downspeeding concepts gains more and more importance. Thereby the realisation of high torque at lowest speed (Low-End-Torque), fast transient torque build-up, the reduction of fuel consumption at high loads, reduced enrichment demand at full load, and an improved part load efficiency [4,5] constitute the greatest challenges.

Various technologies exist to achieve the desired targets, ranging from concepts of supercharging via exhaust turbocharging with fixed turbine geometry to the application of variable turbine geometry (VTG) that can also be applied together [4]. The market introduction of the various technology packages is subject to an extensive cost pressure as the cost advantage relative to the turbocharged diesel engine is considerably affected. **Figure 7** shows a technology cost comparison of various gasoline engines with the diesel engine, suggesting the necessity not only of selecting the most cost-efficient technology mix for the respective intended application but also to look for possible cost savings at the basic engine. Approaches to solutions for these conflicting objectives are described in the following by means of the application of a variable cooled VTG-turbine as well as a cooled exhaust manifold.

3.2 Variable Cooled Turbine

In a first approach the required variability in the gas exchange for the gasoline engine can be provided by a fixed geometry turbine and a cam phaser. Beyond that, the use of a variable turbine geometry enables a further extension of the torque/ performance range. Thereby, the exhaust gas temperatures, that are significantly higher compared to those of the diesel engine, constitute a major challenge. Basically, the variable turbine geometry can also be used for inlet temperatures of >1000 °C. The resulting additional costs, however, restrain the application area of such high-temperature-VTG to vehicles of premium class. [5] The comparably cost-saving and mass-produced VTG-technology available for diesel engine applications, however, exhibits a strongly limited temperature range with about 820 °C maximum temperature and is thus not adequate for gasoline engines. By the use of optimized combustion processes designed for suitable exhaust gas temperatures and additional effective measures for exhaust gas

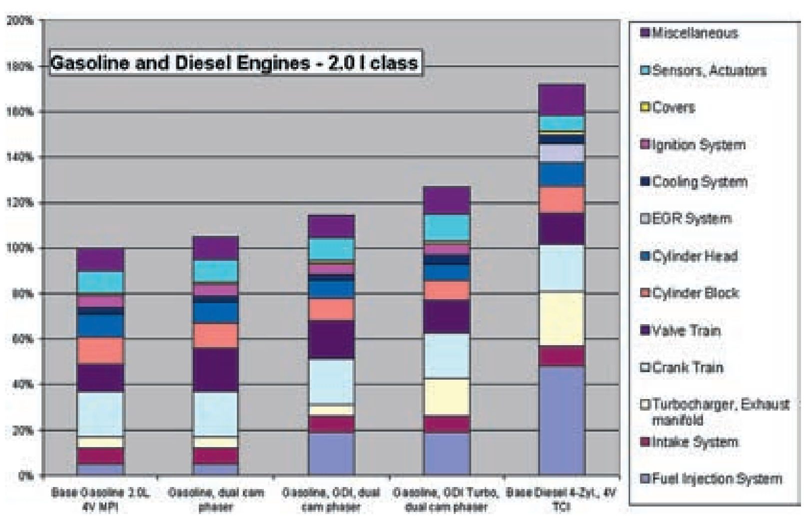


Figure 7: Technology cost comparison Gasoline and Diesel engine

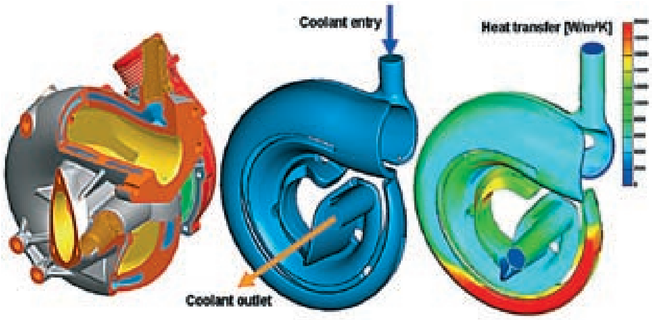


Figure 8: Water cooled aluminium VTG-Turbine

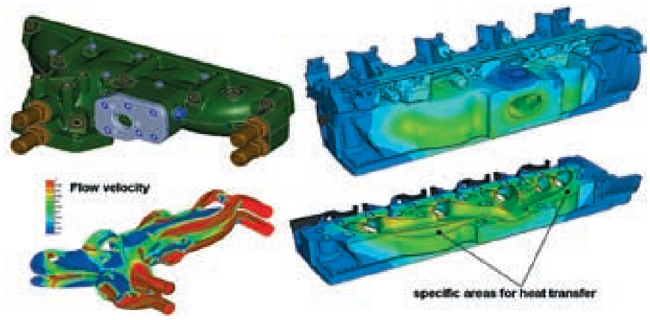


Figure 9: Cooled aluminum exhaust manifold

cooling, such as a cooled exhaust manifold or one that is intergrated into the cylinder head, the required low turbine inlet tempertures can be achieved while at the same time providing attractive performance and fuel consumption values, see 2.3. In the process the loss of enthalpy, which results from the exhaust gas cooling and is disadvantageous especially at low torque, is overcompensated by the beneficial VTG-characteristics providing an extremely high Low End Torque (pe=19 bar at 1000 rpm, 25 bar at 1500 rpm) combined with a high specific power (>80 kW/l) is achieved.

A further approach leads to water cooling of the turbine housing. **Figure 8** shows a monoscroll cooled aluminum VTG-turbine, the mounting of the rotor including lubrication and cooling having been adopted from the diesel-VTG practically without modification. During the design phase special attention was paid to the gas flow to secure a uniform heat transfer in addition to the observation of material peak temperatures and the avoidance of boiling areas. Hence an uneven expansion and resulting enlarged clearances between rotor and turbine which would decrease efficiency can be avoided.

3.3 Cooled Integrated Exhaust Manifold

A significant reduction of the exhaust gas temperature can also be achieved by direct cooling measures, such as the use of an exhaust manifold which is either cooled itself or integrated directly into the cylinder head. Apart from the resulting thermodynamic benefits several other advantages of the exhaust manifold integrated directly into the cylinder head occur, for example the resulting potential weight reduction and the lowering of costs due to the omission of sealing faces and other parts. **Figure 9**, on the left, shows a cooled exhaust manifold with integrated Waste-Gate-function, which has been used for basic investigations. The focus during the development phase was, analogously to the considerations concerning the cooled turbine, the optimization of the gas flow with regard to material peak temperatures and boiling ranges. The investigation led to the following main results:

- the maximum cooling demand must be determined depending on the nominal power output
- the heat transfer areas on the gas-side are the prime parameters for the additional cooling
- within the reliable heat transfer zone the cooling parameters play only a minor role (self-regulating system)
- the layout of the additional water jacket requires special attention to the surface area and to boiling effects.

Figure 9, on the right, illustrates the direct integration of the exhaust gas routing into the cylinder head, bringing in the experience gained in the basic investigations mentioned above.

As a negative effect a reduction of the turbine performance occurs due to the loss of enthalpy, which in turn results in a reduced torque build-up at low engine speed. Additionally, a significantly increased amount of heat flows into the cooling circuit, ranging between about 7 kW and 25 kW depending on the operating point and the cooling gradient (example for a 2l turocharged engine). Accordingly, the systematic consideration of the entire cooling circuit of a vehicle is of major importance. By means of **Figure 10** the

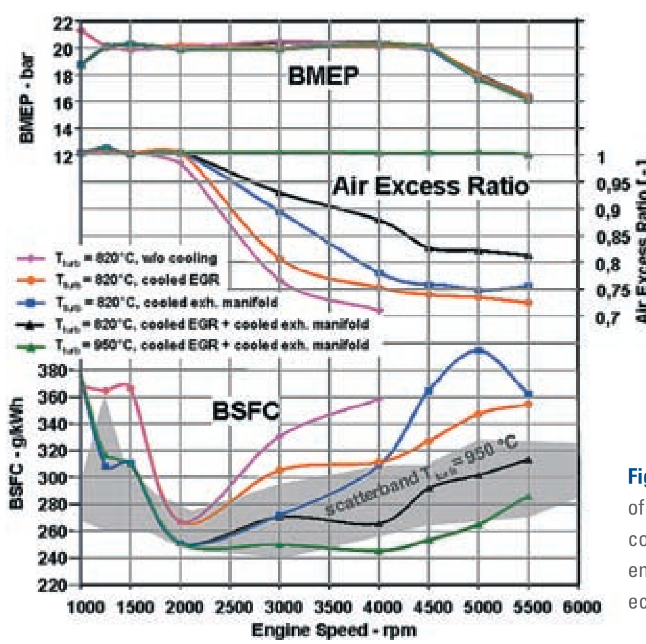


Figure 10: Influence of exhaust gas cooling on full load enrichment and fuel economy

efficiency of the exhaust gas cooling measures, described so far, can be assessed:

Without exhaust gas cooling a VTG-turbocharger designed for a specific power of about 80 kW/l can achieve a BMEP of more than 21 bar. In order to keep a turbine inlet temperature limit of 820 °C, increasing enrichment is required starting at 2000 rpm, limiting the reasonable full load operation to about 4000 rpm. By using a cooled exhaust gas recirculation system the enrichment demand is decreased especially at high engine speed and thus enables full load operation over the entire speed range. Although the loss of enthalpy due to low and average engine speed leads to a reduction of the maximum BMEP from 21.4 to 19 bar at 1000 rpm, by employing a VTG the BMEP level of >20 bar can be maintained by use of an exhaust gas cooling system at 1100 rpm and above. Respectively, a level of 25 bar can be realized at 1250 rpm and above, Figure 8. At high engine speed, however, the achievable cooling efficiency is not sufficient anymore. There, at a comparable air ratio, suitable BSFC figures can be achieved by employing a cooled exhaust gas recirculation system.

4 The AVL Design Validation Process

The realization of reduced development costs and times is significantly enhanced by the shifting from a hardware-based functional and durability development to a virtual development (AVL Frontloading). The main focus of the hardware-based development is changed insofar as 'development assignments' are becoming 'validation assignments'. The virtual development is based on approved methods that include function and durability objectives. This requires fundamental rethinking and involves great potential capacities but also risks. Potential risks arise mainly due to the fact that for a successful implmentenation all development partners involved (including system suppliers) must be closely interlinked throughout the development phase and integrated into each stage of the process. This approach has been systematically pursued within AVL for several years and is described by the AVL Design Validation Process (DVP).

DVP is based on comprehensive standardisations and methodology developments in the development areas design, simulation and testing:

– standardised assembly structure defined by technical specification manuals (TSM), system specification manuals (SSM), and technical part specification (TPS)
– standardised design methodology to enable a quick realisation of variants and to provide a direct interface for simulation [14,15]
– standardised CAE-Task database with more than 120 CAE tasks securing the achievement of the targeted values defined in the technical specification manuals
– AVL Load Matrix methodology [7,8] to determine the validation scope within DVP.

The coaction of the methodologies mentioned is mapped at the top of **Figure 11** and schematically shows the interaction of virtual and physical validation steps. One of the most important tasks is the validation of simulation results by test results, so that they can be used to enhance the simulation methodologies which are to be brought into a virtual optimization loop prior to the first hardware phase.

The execution of more than ten new engine developments a year permits AVL rapid processing of newly acquired knowledge, which can instantaneously be used for new developments and technology upgrades, respectively.

In the following, the integration of the patented AVL Load Matrix Process into DVP is described in some detail.

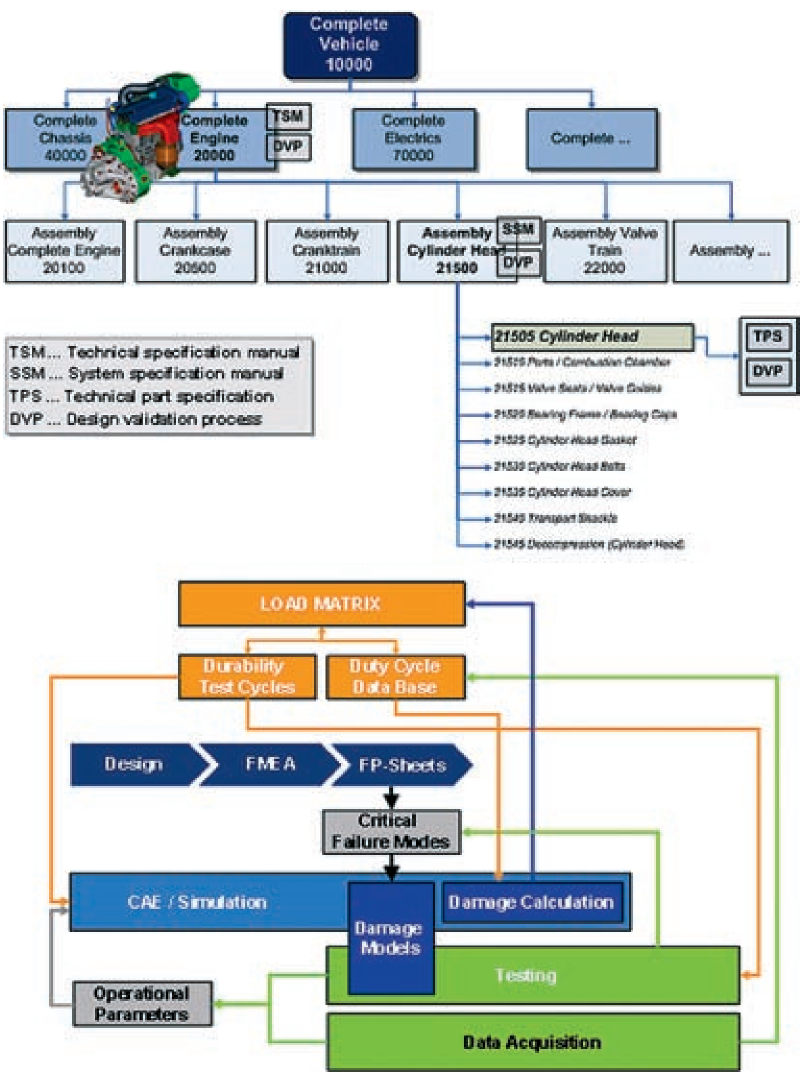
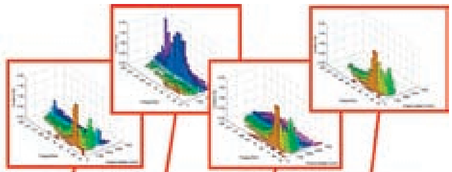


Figure 11: Design validation process (DVP) for systems, assemblies, and parts

Figure 12: Acceleration factors in view of TMF considering different duty cycles



Cylinder Head / Low Cycle Fatigue	Mixed	Highway	Rural	City
Test bed tests				
Cyclic Load Test	2,3	0,74	1,19	10,36
Rated Power Test	0,5	0,18	0,29	2,48
Thermal Shock Test	11,4	3,74	6,04	52,55
Piston & Cylinder Head Crack Test	32,3	10,6	17,1	148,86
Cold Start Part Load Test	1,2	0,4	0,65	5,69

5 AVL Load Matrix as Basis for DVP

Against the background of the introduction of new, partly complex technologies and the trend of increasing the load on the individual components, the validation process represents an essential factor within the product development. Along with the tightened and improved application of CAE-methodologies, described in extracts above, efficient validation processes are indispensable to meet customer demands concerning quality and reliabilty despite shortened development periods. In order to specify the Design Validation Plan (DVP) AVL uses the patented methodology 'AVL Load Matrix'. [7,8] Load-Matrix is a continuous, failure-mode-oriented process that effectively supports the identification of project-specific critical failure-modes and the product validation. Objective is the completion and optimization of Validation Plans as to the verification of function, reliability and durability.

While in the past primarily the optimization of test plans was in the foreground, DVP increasingly draws upon simulation results as well, at which the reliability of the results is of capital importance. At the bottom of figure 11 the interaction between simulation and testing taking place in the Design Validation Process is shown. The elements connecting the respective analytic and experimental methodologies are:

- Release requirements in the form of durability runs to be passed
- The definition of critical failure modes and the modelling of the related damage mechanisms
- The operating parameters for the engine sub-systems.

For the identification of project-specific critical failure modes the so called FP-Sheet Analysis (Failure Mode Influence Parameter) is applied, a process based on the FMEA-methodology, proceeding especially in view of actions to be performed by taking into account relevant impact parameters and suitable validation measures.

Concerning the validation of reliability and durability a central element within the AVL Load Matrix is the identification of failure-mode-specific acceleration factors of durability runs by the use of damage models. **Figure 12** shows acceleration factors of the AVL standard test bed durability runs, calculated under the specific boundary conditions of a diesel engine project, with regard to the failure mode TMF-cylinder head. The higher the acceleration factor the higher the damage rates that emerge during the observed durability run, or, conversely, the less the collective load of the observed application.

With established acceleration factors the equivalent mileage of each test run can be calculated in terms of the considered failure mode so that the verifiable reliability of the test program can be determined by the use of Weibull-models [9]. The key figures acceleration factor, verifiable durability and reliablity function as indicators for the project-specific assessment and optimization of test plans. Thereby, the objective is to achieve a maximum of verifiable reliability and durability at minimum test effort.

The test plans thus optimized are applied in testing and – as described in the preceding chapter – at the same time form the basis for the design phase, while, in agreement between simulation and development, the test cycles are determined that have to be passed for component releases.

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