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System and control concept for future turbocharger exhaust-gas recirculation systems for diesel passenger cars (2005)
Introduction

VTG turbochargers have been in use on diesel vehicles for some years now. Moreover, the cooled, high-pressure exhaust gas-recirculation valve (HP-EGR) is state of the art in today’s diesel applications in order to comply with emission regulations. On the HP-EGR, the exhaust gas is tapped in front of the turbine and routed directly into the intake manifold at the fresh-air end. In many applications, both the VTG turbocharger and the EGR valve are actuated pneumatically. However, there is an increasing trend towards replacing the pneumatic actuators by electrical final control elements. The advantages of electrical adjustment are its better target value-approach characteristics for the VTG blade position and the EGR valve position. Moreover, the availability of a vacuum pump is no longer guaranteed in all applications in future.

Figure 1 shows the limit values of previous, today’s and future particulate and NOx emissions both for the American automobile market and for the European automobile market, whereby the NOx and particulate emissions must be considered as the most critical pollutant emissions of a diesel vehicle. The US 07 regulations will become binding in the USA as of 2007, and introduction of the Euro-5 Standard is planned in Europe as of 2008, whereby there are still no definitive targets for the limit values for Euro-5. There are diesel vehicles today able to meet the Euro-4 emission regulations without an additional particulate filter (DPF). However, owing to the significant production in particulate emissions from Euro-4 to Euro-5 and the public and political pressure, carmakers face a major challenge of managing without additional exhaust-gas aftertreatment systems.

The NOx emissions increase tremendously if the combustion temperature in the cylinder and air-fuel ratio (lambda) increase, whereas particulate emission increases in rich operation. The occurrence of NOx and particulates is a local phenomenon. Consequently, the local lambda and the temperature in the cylinder are crucial to generation of these pollutants.
The known interrelationship between NO\textsubscript{x} emission and particulate emission means that changes in combustion towards more favorable NO\textsubscript{x} emissions involve a rise in particulate emission and vice versa. Emissions in transient situations, e.g. during acceleration, result in brief-duration peaks in particulate emission owing to inadequate fresh-air mass supply with increased injection quantity. During the acceleration phase, fresh-air mass flow does not follow quickly enough owing to the volumes between air filter and induction manifold. Moreover, the turbocharger also has to accelerate to higher speeds in order to meet the increased charge pressure demand.

**How does EGR influence combustion?**

Ideally, the exhaust gas can be assumed to be an inert gas. On diesel vehicles operating with high lambda, the recirculated exhaust gas still contains a slight oxygen concentration but this share is very low compared with the share of oxygen in the fresh air. Consequently, the entire oxygen concentration in the cylinder is reduced by replacing fresh air by recycled exhaust gas, leading to a reduction in NO\textsubscript{x} emissions. The specific thermal capacity of the exhaust gas is higher than that of the fresh air since water and CO\textsubscript{2} are contained. This results in a reduced combustion temperature and lower NO\textsubscript{x} emissions. Additionally cooling the recycled exhaust gas assists this effect. By contrast, the charge pressure must be raised in order to achieve the same lambda and in order to avoid increased particulate emission.
The situation described above leads to the crucial requirement to control the charge pressure $p_2$ and the fresh air mass flow $m_{\text{air}}$ as precisely and quickly as possible in order to avoid increased emissions in particular in the transient phases. Consequently, the control concept is a crucial factor for enhancing the emission behavior in the case of transient phases. Owing to the anticipated rapid rise in the use of particulate filters on diesel passenger cars, the focus is aimed chiefly at minimizing NO$_x$. Of course, the aim here is to reduce the scope of complex NO$_x$ aftertreatment or dispense with it entirely.

**System configuration**

A turbocharged, direct-injection series engine was used for the engine test stand tests and for simulation purposes. This series engine is equipped with an HP-EGR and pneumatic actuators both for the VTG turbocharger and for the EGR valve. The very first thing done was to examine this standard configuration on the test stand for its emission behavior in order to have a basis for comparison. In parallel with this, GT-Power simulations were conducted in order to be able to better assess the potential for emission reduction of various new system configurations. The best system for pollutant reduction was selected with the aid of these simulations and it was installed on the engine test stand. The newly defined system was first investigated at stationary operating points. Measurements were then conducted on the transient behavior of the system with the aid of abrupt load changes, firstly with a newly calibrated standard control concept simply extended with actuation of the new actuators. The influence on pollutant emission was investigated in further detail on the basis of these measurements. To conclude, it was planned to repeat these test stand runs with the newly developed control concept.

![Figure 2: New system configuration](image)

The system configuration shown in Figure 2 shows an extension of today’s conventional cooled HP-EGR concept. There is also an HP-EGR here but it is uncooled by comparison with the conventional HP-EGR. The newly added, second low-pressure exhaust-gas recirculation valve (LP-EGR) taps the exhaust gas downstream of the particulate filter and routes this part of the...
exhaust gas to the fresh air precisely upstream of the turbocharger’s compressor. An exhaust-gas valve is also required in order to increase the pressure gradient in the LP-EGR. With low exhaust-gas mass flow rates in the lower part-load ranges, the pressure gradient generated as the result of the exhaust gas system which follows is inadequate to route the exhaust gas through the LP-EGR.

The air-air charge cooler is replaced by a water-air cooler operated via an additional low-temperature water cooling circuit in order to reduce volumes. This modification makes it possible to fit the charge-air cooler nearer to the intake manifold instead of conventional positioning of the air-air charge cooler in the front area of the vehicle. A two-stage water-air cooler manufactured by the Behr company is integrated in the LP-EGR. The cooler cools using a conventional water cooling circuit in the first stage of the exhaust gas. A second stage follows in which the exhaust gas is cooled further with the aid of the additional low-temperature water cooling circuit in order to reach lower temperatures in the intake area of the compressor.

The LP-EGR makes possible both enhanced thorough mixing of the recirculating exhaust gas with the fresh air and better distribution over the cylinders. The novel type of cooling concept reduces the temperatures in the intake manifold which has a positive impact on reducing NOx emissions. Only the HP-EGR is used in the case of cold starts and in the case of transient phases, owing to the shorter reaction times. Consequently, no EGR cooler is foreseen in this recirculation system.

**Sensor and actuator concept**

**Actuator concept**

The decision to use electrical final control elements for almost all actuators relates to their enhanced control characteristic. The EGR valve (Figure 3a) consist of a DC motor and a position sensor. Since the valve features no other electronics, an external electronic circuit was used in test stand mode in order to approach to defined positions. This made it possible to adjust to the position simply by means of a PWM setpoint selector.

The only modification to the series VTG turbocharger also consists of substituting the pneumatic control box by an electrical servo drive. This actuator (Figure 3b) is a so-called “torque motor” which is also equipped only with a Hall position sensor system. Consequently, here as well, the external position control system is implemented either as in the vehicle by the engine electronics or on the test stand by a “hardware-in-the-loop” (HIL) system as is possible, for instance, with the ASCET system. A “Jacobs exhaust-gas brake valve”, as used in the Dodge RAM for instance, was used as the exhaust-gas throttle (Figure 3c). This is where an intelligent electrical servo
drive with internal electronic control and power circuitry is used for position control instead of the pneumatic series actuator. This electrical final control element is actuated by means of a PWM setpoint and returns a PWM return signal for contactless position acquisition. A pneumatic series valve was used for the HP-EGR (Figure 3d).

The throttle valve which is used in series has an electrical actuator. However, since the throttle valve would be open at the intake end at all operating points, it was dispensed with since it makes no sense to throttle both at the exhaust-gas end and at the fresh-air end. The pneumatic servo drive was also replaced by an electrical actuator for the test stand tests in order to be independent of a vacuum pump.

![Figure 3: Final control elements](image)

**Sensor concept**

One other important part of the control concept, in addition to the actuator concept, is the identification of the required sensors with the aim of using only the sensors really absolutely required, in order to avoid unnecessary costs. The result is the sensor concept shown in Figure 4.
Most of the sensors shown in Figure 4 are already used in the standard diesel application, such as the mass flow rate sensor, the sensor for $p_0$, $T_0$ (ambient conditions), $p_2$, $T_2$ (charge pressure and charge temperature) and $n_M$ (engine speed). Position feedback signals of the individual actuators ($s_{HP\_EGR}$, $s_{VTG}$, $s_{LP\_EGR}$, $\beta_{HP\_Th}$, $\beta_{LP\_Th}$) are, on a general basis, available in conjunction with electrical servo drives. These feedback signals are required for position control and for calculating the missing states of the system. The temperature sensor ($T_{EM}$) in the exhaust-gas manifold, $\lambda$, $T_\lambda$ (lambda and temperature upstream of DPF), $\Delta p_{DPF}$ (pressure drop across the DPF) and $NO_x$, $T_{NOx}$ (temperature downstream of the DPF) are required to be able to control regeneration of the DPF. The sensor for $\Delta p_{LP\_EGR}$ (pressure drop across the LP-EGR valve and the position of the LP-EGR valve help to estimate the mass flow rate through the valve. Measurement of the turbocharger speed of rotation ($n_{TC}$) allows overspeed of the turbocharger to be avoided and calculation of the mass flow rate, temperature and pressure upstream and downstream of the compressor. Since most sensors are already used in the standard control concept, only two additional sensors ($n_{TC}$, $\Delta p_{LP\_EGR}$) are required for the new system.

Additional sensors were used in the development phase on the engine test stand in order to investigate the system behavior in greater detail and to be able to detect possible errors in the control algorithm more quickly. Only the sensors shown in Figure 4 are planned for series launch.

**Development of the control concept**

The new controller is based on a model-based, predictive approach which actuates all the actuators presented at the same time. A model of the air path, with real-time capability, is used
for prediction. This model is a reduced version of an originally more detailed model that was used for pure “software-in-the-loop” (SIL) simulation for controller synthesis.

The model implemented in MATLAB/Simulink represents a zero-dimensional physical mapping of the engine. The engine model was developed in parallel with the stationary tests on the engine test stand. This parallel development process can also be seen in Figure 6. For this purpose, the measurements on the engine test stand with the standard controller (Figure 5) were compared with the simulation results in GT Power and used to assign parameters to the engine model. As further measurement data was received, the newly derived model of the air path was further adapted and optimized.

The Simulink model of the control system represents the basis of controller development since the model-based controller includes a reduced simulation model. Consequently, the simulation model was used to verify the controller approach and assign parameters to it in advance. This was then performed in a pure SIL simulation, as hinted at in Figure 7. The engine is replaced by the engine model in the simulation.

![Image](image.png)

**Figure 5:** Measurement results of the new VTG/EGR system (EGR distribution is shown here by way of example)
Mechanical construction of the system on the engine test stand

On the test stand, the newly designed controller controls the HP-EGR valve, the intake throttle valve, the LP-EGR valve, the exhaust-gas throttle valve and the VTG adjustment, see Figure 9. Since it was not a part of the project to develop a new engine timing system, the algorithms were
implemented in an HIL system in the bypass for the standard engine controller (ECU), whereby the development ECU which was made available still controls important functions such as injection. Using a so-called ETK interface of the ECU, it was possible to read in sensor signals and other required variables in the HIL system.

An ASCET ES1000 real-time system was used as the HIL system. The ASCET system on the one hand, receives sensor signals via the ETK interface and, on the other hand, provides additional input and output channels for the sensors not available in the standard ECU. The controller determined under Simulink was implemented in the ASCET system in order to take over all previously simulated controller functions on the test stand in real time. The actuators are controlled in this case by the D/A or PWM output drivers made available in the ASCET ES1000 system.

![Figure 8: Signal flow on the test stand](image)

Figure 8: Signal flow on the test stand
Figure 9: Final control elements actuated by the new BorgWarner algorithm

Engine model in Simulink

The engine model, \textit{Model\_E}, is based partially on physical interrelationships (e.g. for the manifold and other volumes) and partially on equations determined empirically (e.g. for the turbocharger and the cylinders). Figure 10 shows all modeled physical components and the most important variables via which the components interact.

Part models for volumes

All volumes are described by the balance equations for the gas mass (1), the EGR rate (2) and the energy (3), i.e. by the balances of the inflowing and outflowing gas mass flow rates. The entire air mass, assumed as an ideal gas, in the system, is concentrated in these volumes.

\[
\frac{d}{dt} m = \sum_i m' - \sum_i m'_{j} \tag{1}
\]

\[
\frac{d}{dt} r_k = \frac{1}{m} \sum_i [m'_i (r_{i_k} - r_k)] \quad \text{where} \quad r_k = \frac{m_k}{m} \tag{2}
\]

\[
\frac{d}{dt} T = \frac{1}{m} \left[ \frac{\dot{Q}}{c_p V} + \kappa \left[ \sum_i (m'_i T_i) - T \sum_j m'_{j} \right] - T \frac{d}{dt} m \right] \tag{3}
\]

Where:
the states of the relevant volume:

\[ m: \text{ mass of gas mixture} \]
\[ r_k: \text{ mass share of the gas component } k \text{ (e.g. EGR)} \]
\[ T: \text{ temperature} \]

the mass flow rates and energy flow rates

\[ m_i': \text{ inflowing mass flow rate} \]
\[ m_j': \text{ outflowing mass flow rate} \]
\[ Q: \text{ heat flow rate} \]

and:

\[ t: \text{ time} \]
\[ c_v^0: \text{ isochoric thermal capacity of the gas mixture} \]
\[ \kappa: \text{ isentropic exponent} \]
The mass flow rates between these volumes are assumed to be isothermal since this represents a more realistic description of the system (better than isentropic description). We shall take a throttle valve or valve between two pipes to illustrate the situation. This system can be viewed as adiabatic and with a stationary flow through the throttle valve. Both assumptions must now be applied precisely in this way to an isentropic system as well. We thus have the following for the balance equations:

\[
\frac{d}{dt} m = m_{\text{in}}' - m_{\text{out}}' \tag{4}
\]

\[
\frac{d}{dt} U = Q' + m_{\text{in}}' h_{\text{in}} - m_{\text{out}}' h_{\text{out}} \tag{5}
\]

The stationary process leads to:

a) \( \frac{d}{dt} m = 0 \), consequently \( m_{\text{in}}' = m_{\text{out}}' = m' = \text{const} \)

b) \( \frac{d}{dt} U = 0 \) for the internal energy

The following applies to an adiabatic system:

\( Q' = 0 \)

For the specific enthalpy, we thus obtain the following:

\( h_{\text{in}} = h_{\text{out}} = 0 \)

In the case of an isothermal process, the following applies to an ideal gas:

\( T_{\text{in}} = T_{\text{out}} \)

**Part model for the turbocharger**

The classic compressor characteristic map (see Figure 11) shows the pressure characteristic and the isentropic efficiency as a function of the reduced mass flow rate.
If we now know the pressure ratio and the turbocharger speed and wish to determine the reduced mass flow rate from these characteristic maps, we may obtain two or even more solutions owing to the shallow curves. In order to avoid such problems, the information of the two characteristic maps is combined, thus converting the characteristic maps. This leads to two new characteristic maps with which it is possible to find an iterative, unique solution to the above-described problem.

For the turbine, the situation is similar, but the influence of the VTG makes it somewhat more complex. The turbine characteristic maps (see Figure 12) show the reduced mass flow rate and the product of mechanical and isentropic efficiency as a function of the pressure ratio. Both curves have the parameters of reduced turbocharger speed and VTG position. It is primarily the lower curve that causes problems in evaluation owing to the highly fragmented curves. These difficulties can be remedied in a similar manner to the difficulties with the compressor characteristic maps. This obtains curve characteristics which can be evaluated better and in which the turbocharger speed no longer occurs as a parameter.
Part models for crankshaft drive and turbocharger shaft

The dynamic characteristics of the crankshaft drive and the turbocharger shaft are allowed for by the relevant moment of inertia.

Part model of the cylinder

The cylinder part model is partially physically based (i.e. modeled as a volume) and partially based on equations determined empirically, i.e. it is based on test stand measurements conducted on a series engine (with a series ECU). Consequently, information on EGR variations at the individual operating points was available for instance, at the time at which this model was generated.

For this reason, the model may lead to more or less high inaccuracies if the engine is operated at an EGR rate differing greatly from the series status.

The cylinder model is an RMS value model, i.e. its states are not modeled dependent on the crankshaft angle. The following states need to be calculated in each case: (a) the quantity of gas...
mixture inducted into the cylinders during a cycle, (b) the EGR rate, (c) the temperature of the exhaust gas emerging from the cylinder and (d) the engine output.

The following assumptions were made during development of this model:

1. Since the entire model is aimed at applicability in real time, more detailed physical modeling was not possible at this point.
2. The mass flow rate inducted by the cylinder is dependent more on its density than on the pressure in the intake manifold.
3. Fresh air and EGR in the intake manifold form an homogeneous mixture.
4. The quantity of exhaust-gas mass flow rate emerging is equal to the sum of the inducted gas mass and the injection fuel quantity (stationary process).
5. The exhaust-gas temperature and the engine output depend on
   (a) the charge of the cylinder and consequently, the density in the intake manifold and the engine speed,
   (b) the air-fuel ratio (lambda),
   (c) the EGR rate in the cylinder and
   (d) the injection instant (SOI). However, this was not considered here since no information was available from the measurements.

Figure 13 shows an example of such a calibration function (index 3 stands for the intake manifold here).
Figure 13: Example of an engine characteristic as determined from test stand measurements of a series engine
Figure 14 shows simulation results during an abrupt change of the VTG actuator. The engine was simulated here at operating point 3,000 rpm and 9 bar intermediate pressure.

![Simulation results](image)

**Figure 14**: Simulation results of the engine model during a simulated abrupt change of the VTG actuator

**Model-based predictive controller**

A model-based predictive controller (MNPC: Model-based Non-linear Predictive Controller) was developed for this application. It consists of an observer, a predictor and an optimization algorithm (see Figure 15).

The observer model, Model_S, uses all available sensor signals to calculate the non-measurable states of the system. On the basis of these states, the predictor, Model_G, predicts the future system behavior for N different combinations of actuator positions. The optimization algorithm (J) now uses a cost function to assess this prediction for each combination of actuator position. Both the observer (Model_S) and the predictor (Model_G) are model-based. Both originated from the above-described engine model (Model_E).
Observer model

On the basis of the sensor signals available, the observer model, \textit{Model\_S}, calculates all required states of the engine in real time. This model must thus calculate all engine states that are not directly available as sensor signals, e.g. the EGR rate in all volumes. These states made available in real time by \textit{Model\_S} then form the basis for subsequent prediction.

Predictor model

The predictor model, \textit{Model\_G}, does not use sensor signals since it calculates only into the future and, of course, no sensor signals are available for this yet. The optimization algorithm specifies new actuator positions by which \textit{Model\_G} is stimulated. This now simulates the anticipate engine behavior.

Air filter part model by way of example of model reduction

In order to clarify the special aspects in model reduction by the simulation model (\textit{Model\_E}) with respect to the observer (\textit{Model\_S}) or with respect to the predictor (\textit{Model\_G}), this procedure will be explained by way of example of the air filter model (volume V₁).

Air filter model in \textit{Model\_E}

The air filter model in \textit{Model\_E} uses the following input variables:

1. The ambient states,
2. The states of the lines between the LP-EGR valve and the air filter (mass flow rate, temperature and EGR rate), and
3. The mass flow rate through the compressor.

The fresh air mass flow rate is now calculated in this part model via the three balance equations (1) – (3). The output of this part model consists of internal states and the fresh air mass flow rate. The flow coefficient was calibrated from test stand measurements.

Air filter model of the observer (Model_S)

The observer model *Model_S* operates in real time. The air filter model must not calculate the fresh air mass flow rate in this case since this variable is available as a sensor signal. Balance equations (1) - (3) can be ignored here owing to the low dynamic response. As a consequence of this, only a stationary mixing process can be implemented here for the EGR rate and the mixing temperature. By comparing the measured and calculated fresh air mass flow rate, the flow coefficient can be calibrated for *Model_G*. The calculated total mass flow rate for the mixture of fresh air and EGR and the other states now form the initialization values for the following prediction.

Air filter model of the predictor (Model_G)

The air filter model in *Model_G* in turn must now calculate the inducted fresh air mass flow since, as already mentioned above, no sensor signals are available during the prediction. Since the model is to respond as exactly as possible to various actuator positions, the balance equations (1) - (3) can now no longer be ignored. Before starting the prediction, all integrators in *Model_G* must be reset externally to their initialization values.

Optimization algorithm

After prediction, the results of simulation are used by *Model_G* to determine the new actuator positions with which the given target values are best achieved by means of an optimization algorithm (*J*). Owing to the only limited hardware and software resources available in an ECU it is not possible to use a classic optimization algorithm at this point. The only practicable option is to conduct many predictions for many different actuator positions starting from the current state (made available by *Model_S*) and to decide between these, i.e. to decide which of the simulated actuator positions leads to a minimal cost function *J*. The optimization algorithm *J* is currently only capable of minimizing the cost function *J* as a function of two variables (*u₁* and *u₂*). For example, the position of the LP EGR valves and the VTG position can be used as the variables.
In order to establish a global optimum and not only a local optimum, it is necessary to allow for the entire range of the vector \( u = (u_1, u_2) \). The combinations \((u_1, u_2)\) form a characteristic map consisting of \( N \) points. A prediction cycle consisting of the following steps is now started for each of these \( N \) combinations \((u_1, u_2)\):

1. The predictor model \( \text{Model}_G \) is reset and initialized with the states from \( \text{Model}_S \).
2. The prediction is now conducted for the current combination \((u_1, u_2)\).
3. Certain integration steps of \( \text{Model}_G \) are conducted.
4. The results, i.e. the predicted states \( y_G \), the predicted control deviations \( dy = (y_{\text{set}} - y_{\text{actual}}) \)
   and the current actuator positions \( u_{\text{actual}} \) are saved.

The cost function \( J \) for all \( N \) predictions is now evaluated:

\[
J_n = \sum_{i=1}^{3} \left[ c_{y1} |y_1 - y_{1,\text{set}}|_{u,n} \cdots \\
       c_{y2} |y_2 - y_{2,\text{set}}|_{u,n} \cdots \\
       c_u |u - u_{\text{old}}|_{u,n} \cdots \right]
\]

The minimum of the cost function \( J \) now corresponds to the optimum, new combination of actuator positions \( u_{\text{new}} \).

A weighted mean value can now be formed in order to damp the system behavior in order to improve the stability of the algorithm, i.e. in order to avoid oscillations. Moreover, the \( N \) actuator positions can be determined using “windows”.

**SIL environment for developing the MNPC**

On the engine test stand, the MNPC is used in parallel with a series engine controller (see Figure 15). Certain functions of the ECU now need to be bypassed for implementation.

An SIL environment was set up for developing and pre-calibrating the MNPC. The engine and the ECU are replaced by the simulation model (\( \text{Model}_E \)) for this purpose (see Figure 16).
The actual MNPC, *Model_J*, operates with a sampling rate of 1 ms. In order to integrate this in the SIL environment it is now necessary to implement a simulation environment allowing various sampling rates in one model since *Model_E* requires a far lower sampling rate (in this case 5 µs) for reasons relating to accuracy and for reasons relating to numerical stability.

**Results of the SIL test of the MNPC**

Selection of the control variables has a major influence on the performance of this MNPC algorithm. The setpoints for the fresh air mass flow rate, the EGR rate and the charge pressure are taken from the series controller and are thus optimized for a system with pure HP-EGR. Consequently, they do not precisely match the engine modified by us with an additional LP-EGR line.

Compared with LP-EGR, the HP-EGR leads to higher temperatures in the intake manifold. So we require a higher charge pressure (owing to the thermodynamic state equations) with HP EGR in order to reach the same air mass flow and the same EGR rate. If the charge pressure is still taken from the series ECU with the modified system, the engine is operated with an increased charge pressure.
The fresh air and EGR mass flow rates are more important than the charge pressure for the actual cylinder charge process. This is also clear from the simulation results in Figures 17 and 18. In Figure 17, the fresh air mass flow rate and the charge pressure were used as control variables while, in the simulation shown in Figure 18, control was towards fresh air mass flow rate and EGR rate. These results clearly indicate that the control variables should be selected allowing for the modified engine hardware. This means that developing the MNPC is more difficult since no adequate setpoints for the control variables are available during the engine tests since the modified engine is operated together with a series ECU.
Modifications for implementing an LP-EGR

Recirculation of exhaust gas into the combustion chamber is a tried-and-tested measure for reducing nitrous oxides. Ultimately, the effects of this measure are based primarily on reduction in the peak temperature in the combustion chamber and reduction of the partial oxygen pressure in the cylinder. The potential of this measure increases with the rate and as low a temperature as possible of the recirculated exhaust gas. In principle, recirculation rates of 60 % and far more are possible, owing to the process, on the diesel engine as the result of the broad ignition limits.

To date, besides internal engine measures (valve overlap) – which provides practically no option for cooling and very minor options for control - the exhaust gas is tapped in the manifold upstream of the turbine of the turbocharger and admitted into the intake duct via a timed valve downstream of the compressor. This circuit arrangement is normally referred to as high-pressure-end exhaust-gas recirculation. In this case, it is frequently problematic to achieve an equal distribution of the recirculated exhaust gas over the individual cylinders – with the corresponding effects on the production of nitrous oxides. Better results and more advantages can be achieved by low-pressure-end exhaust-gas recirculation – tapping the exhaust gas downstream of the turbine and admitting the exhaust gas upstream of the compressor. The major potential – with increased system complexity – is the combination of high-pressure and low-pressure exhaust-gas recirculation.

Low-pressure-end exhaust-gas recirculation means that the exhaust gas needs to be routed through the compressor of the turbocharger. On the one hand, this increases the inlet temperature into the compressor, depending on the degree of recooling. This results in high thermal stressing on the components with regard to the high compressor pressure ratios necessary for the process – with corresponding high compressor outlet temperatures. The initially gaseous constituents in the exhaust gas cause corrosive attack on the components which is far greater if the actual temperature drops below the dew point as the result of wetting with acid media. In addition, uncombusted carbon particles and other constituents of the exhaust gas tend to cause very adhesive resinous components to be deposited on the compressor impeller, the diffuser and all downstream components. These cause far reduced efficiencies as they increase in thickness. From today’s point of view, relatively large fluid droplets impacting at high velocity and at an unfavorable angle on the inlet edges of the compressor blades represent the most critical stressing. This droplet strike relatively quickly leads to material fatigue, causing areas of the component surface to become brittle and ultimately break off, which quickly leads to destruction of the compressor impeller. Finally, bombardment with soot particles rounds off the extremely complex loading spectrum.
It is obvious that this increased loading cannot be accepted without taking additional measures on the components. The solution to the problem must also be based on a detailed analysis followed by an adaption of the overall “exhaust-gas recirculation, cooling and turbocharging” system.

Initial investigations indicate that droplet strike causes the most critical damage. It is necessary to avoid large droplets which are able to follow the gas stream only inadequately at all costs. This means that the cooler operated in the low-pressure recirculation section must be operated so that the actual temperature does not drop below the dew point. This also means that no liquid phase will occur in the low-pressure exhaust-gas recirculation section in standard operating states. Condensation can occur only and for the first time at the low-pressure exhaust-gas mixing point – depending on the thermodynamic state of the inducted air. Here, it is crucial that mixing is an aerodynamic process and occurs very close to the compressor inlet. Since condensation and droplet growth in particular are time-dependent processes, only short distances for droplet growth may be available upstream of compressor. If the critical components are arranged, where possible, so that condensate which may form on cold start is precipitated on the pipe walls without contacting the components susceptible to failure – and runs on the correspondingly aligned pipe walls into the exhaust duct and leaves the vehicle together with the exhaust gas, this operating state will not be able to supply any contribution relevant to failure either. If the corresponding components are protected by suitable coatings, it should be possible to accept temporary wetting and achieve overall operation without failure over the vehicle service life. These coatings also change the wettability of the surface so that the problem of accruing, resinous coatings can be at least greatly reduced, besides the coating’s excellent corrosion protection characteristics.
References


