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System Structure and Controller Concept for an Advanced Turbocharger/EGR System for a Turbocharged Passenger Car Diesel Engine

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ABSTRACT

The present paper focuses on a system and an appropriate controller concept for an advanced air management system of a turbocharged passenger car diesel engine. The proposed air management system consists of a VTG turbocharger, two separate EGR loops, a non-cooled high-pressure EGR and a cooled low-pressure EGR loop. In the low pressure EGR loop, the exhaust gas leaving the particulate filter is mixed with fresh air just in front of the compressor inlet.

The first step consisted of developing a sensor and actuator concept. Prior to conducting engine tests on this system, GT power simulations were performed. Additionally, a MATLAB/Simulink model was created to design a model-based predictive controller. This model is mainly founded on physical equations, allowing for easy adaptation to various systems.

At the beginning of the engine test stage, stationary measurements were conducted to examine the influence of variations of the EGR rate, boost pressure, fresh air mass, etc. These tests were carried out in an open loop without an integrated controller for the air management system. The results were used to optimize the Simulink simulation model.

As to the controller concept, a model-based predictive approach is presented which uses a simplified simulation model of the complete air path for the prediction.

Furthermore, the system design on the dynamometer and the interaction of the individual steps are described. Simulation results of the new controller type and first results of the controller on the dynamometer are presented, as well.

Due to modifications of the state-of-the-art high-pressure system arising from the integration of a low-pressure EGR line, additional measures need to be taken to protect the components in the inlet air path from harmful impact resulting from corrosion and particulates.

INTRODUCTION

In LD diesel applications, VTG turbochargers have been commonly used for some years, now. Also, almost all modern LD diesel engines use high-pressure cooled EGR to reduce emissions. Application of the high-pressure EGR technology allows for conducting the exhaust gas in upstream direction from the turbine and leading it into the fresh air mass stream directly in front of the intake manifold. In many applications, the VTG and EGR valve are actuated pneumatically. However, as regards VTG and EGR valves, the trend is toward substituting pneumatic actuators by electric ones, as well. The advantage of this option lies in the improvement – i.e. faster and more accurate – of VTG vane and EGR valve position control. Moreover, controlling also becomes independent of the availability of vacuum pressure.
In 2003, BorgWarner initiated an internal cross-divisional project entitled 'Turbo/EGR Air Management System'. The BorgWarner Engine Group and the Divisions E/TS (as supplier of EGR valves and throttles) and Turbo Systems (as supplier of VTG turbochargers) are involved in this project. This project has for its purpose to investigate the system hardware as well as the actuator control system. In this paper, the main focus is placed on the project's control aspect.
The benefit of this cross-divisional project lies in the synergy effects using the know-how of both Divisions to improve system components, the system as such and finally to reduce engine emissions to comply with future emission standards, especially Euro 5 and US 07.

Fig. 1 depicts the previous, current and future NO_x and particulate emission standards, both for the US as well as the European market. NO_x and PM emissions stand for the most critical emissions of diesel engines. The US 07 standards will become operative in 2007, the Euro 5 specifications as from 2008. The final limits of the Euro 5 standards are still under discussion. Presently, diesel engines are marketed which comply with the Euro 4 standards, some of which even without using a diesel particulate filter (DPF). Because of the significant reduction of permissible PM emissions in the transition from Euro 4 to Euro 5 specifications, meeting the future emission standards without integrating cost-intensive after-treatment systems will stand for a major challenge.

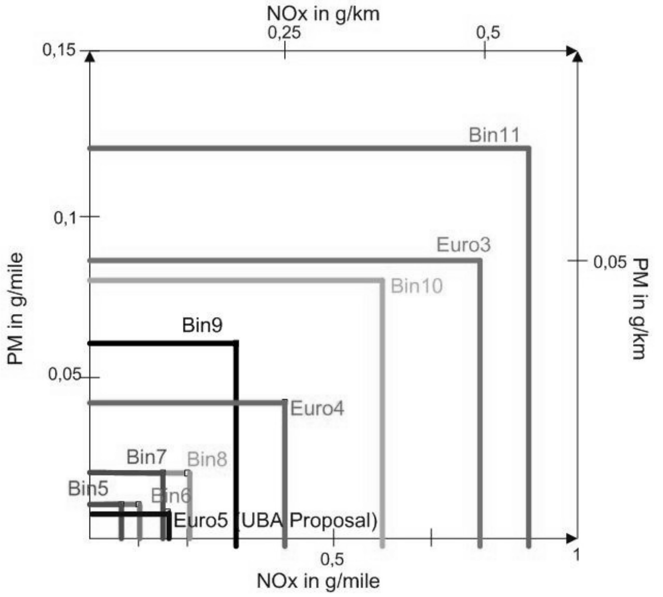

Fig. 1: Emission standards [1]

NO_x emissions increase dramatically, when the in-cylinder temperature and the air-fuel ratio (AFR) increase, whereas PM emissions increase at a rich AFR and lower temperatures. Generation of NO_x and PM is a local phenomenon, so the local AFR and temperature at each point of the cylinder are of relevance to the generation of emissions.

The known basic interrelationship between NO_x and PM emissions necessitates that any modification of the combustion toward lower NO_x emissions produces higher PM emissions and vice versa. Many emissions occur during transients, e.g. during acceleration phases causing peak PM emissions due to insufficient fresh mass air flow supply required for increased fuel injection. During acceleration, the fresh air mass flow does not follow immediately, because of the volumes between the air cleaner and the intake manifold. Furthermore, the turbocharger stands for the most critical part, as it needs to speed up first to generate a higher air mass flow and satisfy the increasing demand for boost pressure.

How does the EGR influence combustion?
Ideally, the EGR may be regarded as an inert gas. Yet, in a diesel engine generating a high AFR, the exhaust gas still contains some oxygen – however, compared to the system's fresh air concentration, its oxygen concentration is much lower. Hence, using the EGR to replace part of the fresh air mass with exhaust gas will cause a decreasing oxygen concentration inside the cylinder. This lag of oxygen leads to reduced NO_x production. The thermal capacity of exhaust gas is higher than the one of fresh air, as it contains CO_2 and water. This results in lower in-cylinder temperatures and, as a consequence, lower NO_x emissions. Cooling down the EGR additionally helps decrease the temperature. On the other hand, the boost pressure needs to be increased to reach the same AFR and avoid higher PM emissions.

To control boost pressure p_2 and fresh air mass flow m_{air}, it is important to reach the required p_2 and m_{air} values as accurately and – to reduce emissions during transients – as quickly as possible. Therefore, any improvement of the emission behavior during transients highlights the control concept thus becoming one of the most important factors in this context. Due to the fact diesel particulate filters (DPF) will form a standard component in future applications, the main focus should be placed on NO_x reduction. The objective is to decrease the scope of required NO_x after-treatment or, if possible, to avoid it completely.

SYSTEM CONFIGURATION

The cross-divisional project of BorgWarner does not just focus on control aspects. The main part of the project revolves around developing a new feasible system configuration, including relevant hardware components. For the engine test and simulation stage, a mass-produced passenger car DI Diesel engine was used. The air path configuration of this engine met conventional standards, i.e. it was equipped with a high-pressure cooled EGR and pneumatic actuators for the VTG and EGR. First, the engine was tested in its baseline configuration on the engine test bed. In parallel, a GT power analysis was initiated to evaluate various potential new hardware configurations. After that analysis, one of the hardware configurations was selected and set up on the dynamometer. For that new engine configuration, steady-state tests were performed on the dynamometer. Then, its transient behavior was examined by conducting load steps, first with recalibrated baseline control equipment, i.e. the current mass-produced control system from the ECU which was recalibrated to drive the new actuators. The impact of the new system configuration on emissions was analyzed on the basis of those data. Finally, it was planned to repeat those load steps by applying the newly developed control strategy.

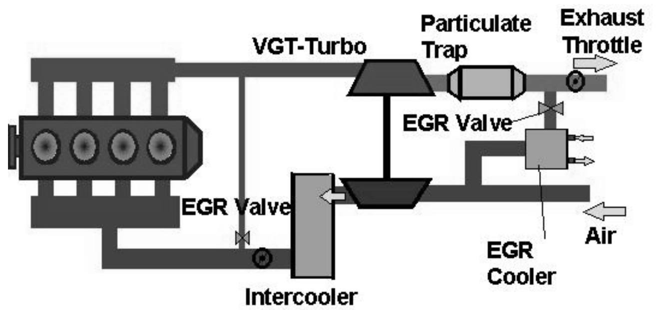

Fig. 2: New system configuration

The new system configuration as shown in Fig. 2 is an extended version of the conventional baseline system. It holds two EGR loops, i.e. a high-pressure loop which – except for the missing EGR cooler – is almost identical to the one integrated in the baseline configuration, and a second low-pressure loop. For this low-pressure loop, the exhaust gas is extracted behind the DPF and mixed with the fresh air mass flow just in front of the compressor. To increase the pressure drop via this low-pressure EGR line, an exhaust throttle is required. Under low load conditions in combination with a reduced exhaust mass flow, the backpressure generated by the muffler is not high enough.

To lower the volumes, the charge air cooler is replaced by a water-to-air cooler operated on the basis of an additional low-temperature water-cooling circuit. This option offers the possibility to integrate the cooler closer to the intake manifold instead of at the front end of the vehicle where the conventional air-to-air cooler is located. The low-pressure EGR loop holds a two-stage water-to-air cooler made by Behr. In the first stage, the exhaust gas is cooled by the engine's usual cooling water. To reach lower temperatures in front of the compressor, the exhaust gas is additionally cooled in a second step via the extra low-temperature water circuit.

The low-pressure EGR loop offers both a better intermixture of the EGR and the fresh air as well as a better cylinder-to-cylinder EGR distribution. This cooling approach (i.e. the EGR is cooled in the EGR cooler and later on also in the charge air cooler) has for its purpose to lower intake manifold temperatures which help decrease NO_x emissions. Due to the lower reaction time of the high-pressure EGR loop, this loop is only used for cold starts to warm up the engine and for transients. That is why this loop does not require an additional EGR cooler.

SENSOR AND ACTUATOR CONCEPT

ACTUATOR CONCEPT

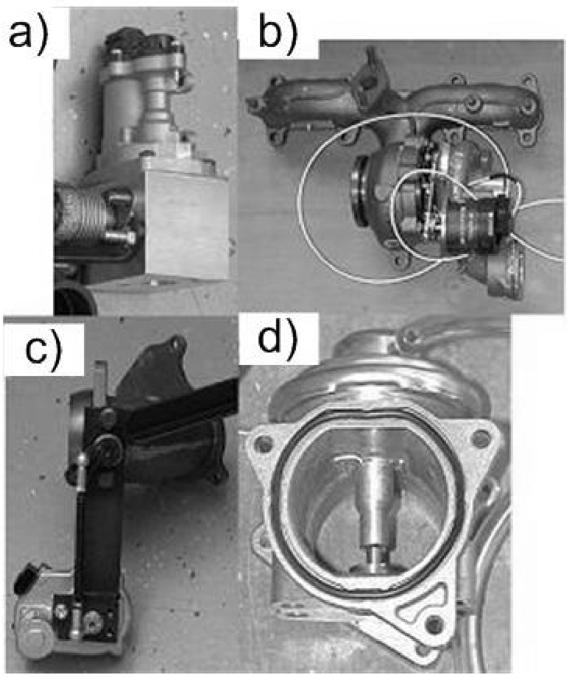

Fig. 3: Actuator components

The decision to use electric actuators for almost all components originates in their improved control behavior. The non-smart EGR valve (see Fig. 3a) is a valve made by BorgWarner's E/TS Division and equipped with a DC motor-operated actuator and a potentiometric position sensor. To control it on the engine test bed, external electronic equipment holding an integrated position controller is applied. This way, the EGR valve may be easily regulated via a control PWM signal. The only modification to the mass-produced VTG turbocharger of this engine affects the substitution of the pneumatic actuator by an electric one. This actuator type is a non-smart torque motor (see Fig. 3b) equipped with a Hall sensor as position sensor. Consequently, the position is controlled from the outside, i.e. inside the ECU or on the engine test stand, with the aid of the ASCET hardware-in-the-loop (HIL) system. A throttle by Jacobs and usually applied as engine brake for Dodge RAMs was used as exhaust throttle (see Fig. 3c). A smart actuator equipped with internal electronics for position control acts as an electric actuator replacing the mass-produced pneumatic actuator. This actuator holds a non-contacting position sensor which is controlled via a control PWM signal and sends back a PWM signal indicating the current position. For the high-pressure throttle and the HP EGR valve (see Fig. 3d), standard mass-produced components were used. The serial throttle is electrically driven; yet, the system approach aims at fully opening this throttle at all operation points, as there is no use in throttling the engine on the exhaust side and in the air inlet. The high-pressure EGR valve is pneumatically actuated. On the dynamometer, the pneumatic actuator was exchanged for an electric one to become independent of the availability of vacuum pressure.

SENSOR CONCEPT

An important part of a controller concept is to identify the sensors required in the air path. The task aimed at using the minimum set of sensors to avoid unnecessary costs. The result was the sensor concept shown in Fig. 4.

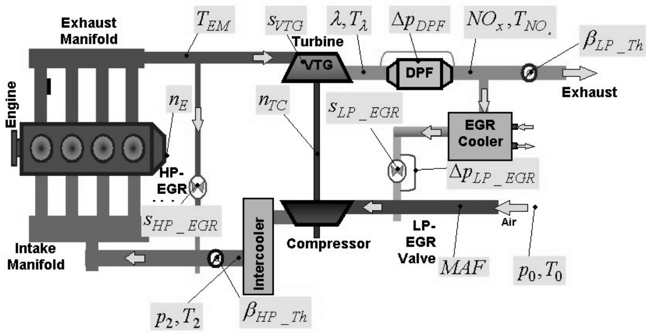

Fig. 4: Sensor concept for the dual-loop EGR system

Some of these sensors are already widely used for the baseline system, e.g. the MAF sensor, the sensor for p_0, T_0 (ambient conditions), p_2, T_2 (boost pressure and temperature) and n_E (engine speed). Actuator position data (s_{HP_EGR}, s_{VTG}, s_{LP_EGR}, β_{HP_Th}, β_{LP_Th}) are usually available when electric actuators are used. These signals are needed for position control and as information for a model helping calculate the missing states of the system. The sensors for T_{EM} (temperature in exhaust manifold), λ, T_λ (lambda and temperature in front of the DPF), Δp_{DPF} (pressure drop via the DPF) and NO_x and T_{NOx} (temperature behind DPF) are needed to determine the regeneration capacity of the DPF. The sensor for Δp_{LP_EGR} (pressure drop via the LP EGR valve) and the position of the LP EGR valve help estimate the mass flow through the valve. By measuring the turbocharger speed (n_{TC}), it is possible to prevent overspeed and calculate the mass flow, temperatures and pressures in front of or behind the turbocharger. Since many of these sensors are already needed for other functions, there are only two additional sensors (n_{TC} and Δp_{LP_EGR}) for this system.

The additional sensors on the dynamometer are only used to monitor the system's behavior and help detect potential errors of our controller model in the development stage. The sensors described in Fig. 4 are the only sensors which are used in series application.

INFORMATION FLOW FOR CONTROLLER DEVELOPMENT

The controller uses a model-based predictive approach to determine the ensuing necessary actuator positions. For the prediction, a model of the engine's air path system with real-time capability is applied. This model is a reduced version of a more detailed model which was developed first and which is also used for software-in-the-loop tests of the controller.

The engine model used for the simulation during the controller development is mainly a physical zero-dimensional model implemented in MATLAB/Simulink. This model was developed in parallel with the steady-state engine tests on the dynamometer. This parallel development process is shown in Fig. 6. Consequently, the results from a GT power model of this specific engine and the measurements of the baseline engine (e.g. as shown in Fig. 5) were used to parameterize the model, first. With the ongoing engine test bed measurements of the new air path system, these results were used to optimize the simulation model.

The Simulink engine model forms the basis for the controller development, since the controller contains a reduced version of the simulated model. The simulation model is also used to verify and pre-calibrate the controller. Hence, a software-in-the-loop (SIL) simulation was carried out (see Fig. 7). In this simulation, the engine simulation model replaces the engine.

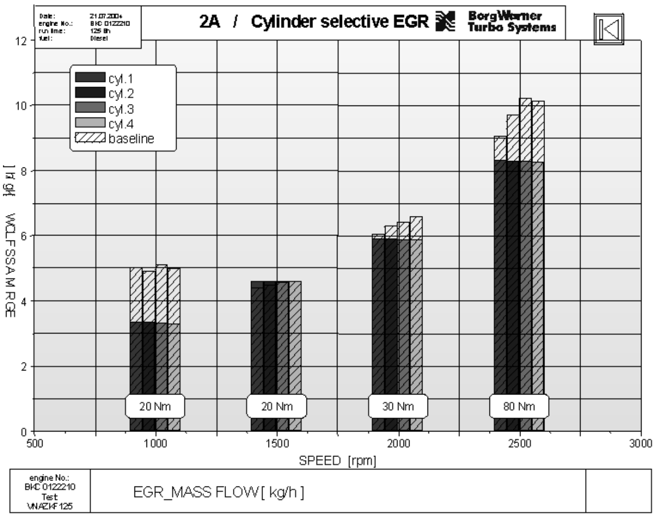

Fig. 5: Example of dynamometer measurement results for the new system (EGR distribution)

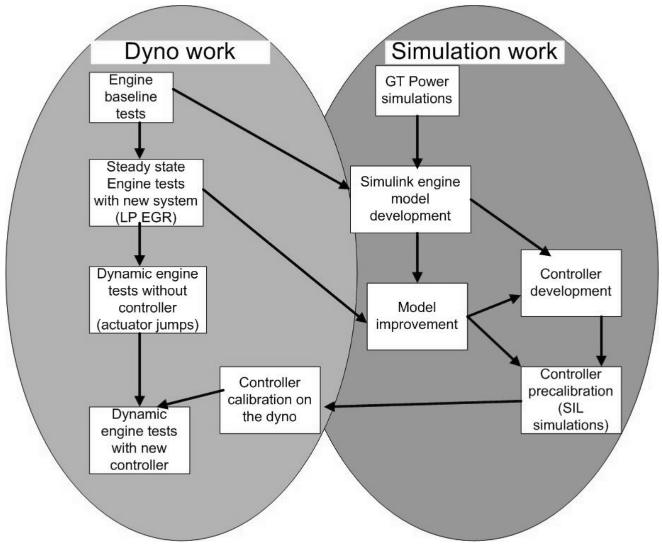

Fig. 6: Development process

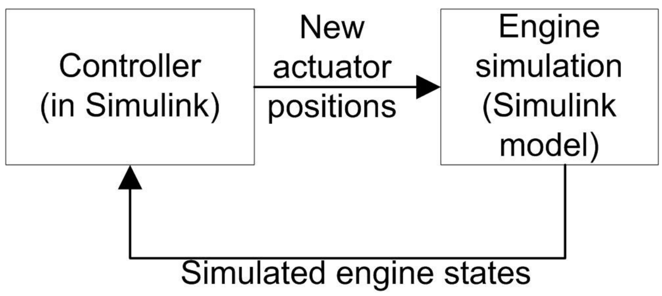

Fig. 7: SIL simulations to verify and pre-calibrate the controller

SYSTEM ASSEMBLY ON THE ENGINE TEST STAND

ON THE DYNAMOMETER, the new BorgWarner control algorithm controls the HP EGR valve, the HP throttle, the LP EGR valve, the exhaust throttle and the VTG turbocharger (see Fig. 9). Since there are no plans to develop an own ECU, the control algorithm needs to be integrated into a bypass on the standard ECU. The development ECU still controls most of the engine functions, e.g. injection timing. Via the ETK interface of the development ECU, sensor signals and other ECU variables may be read. An ASCET ES1000 real-time system is used for the bypass functionality. The ASCET system captures sensor signals via the ETK interface and via additional I/O ports (for additional sensors not used in the standard ECU). The Simulink controller model may be implemented into the ASCET system to carry out all previously simulated controller functions. The actuators are operated via the D/A or PWM output drivers of the ASCET ES1000 system.

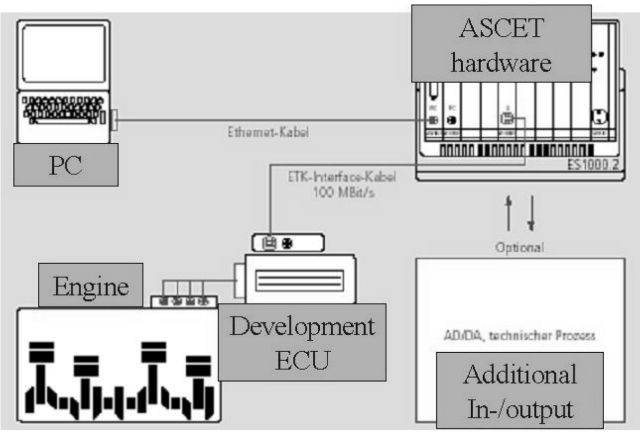

Fig. 8: Signal flow on the dynamometer

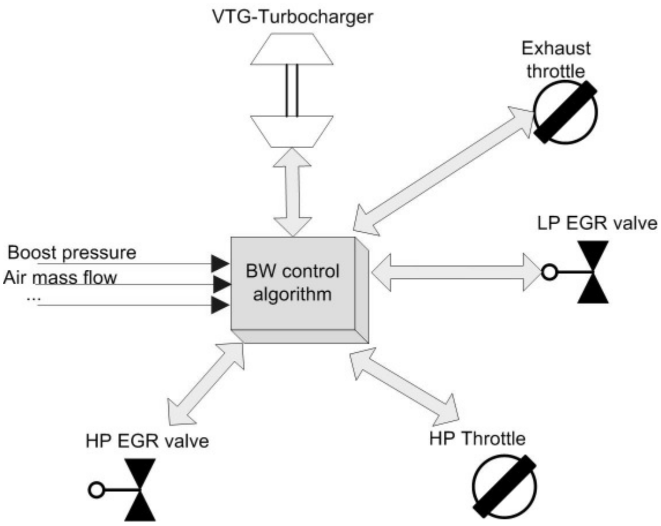

Fig. 9: Actuators controlled by the BW control algorithm

SIMULINK ENGINE MODEL

The engine model, named **Model_E** here, is developed in part on a physical basis (e.g. for manifolds and all other volumes) and in part on an empirical basis (e.g. for the turbocharger (TC) and cylinder). Fig. 10 shows all modeled physical components as well as the interaction variables between them.

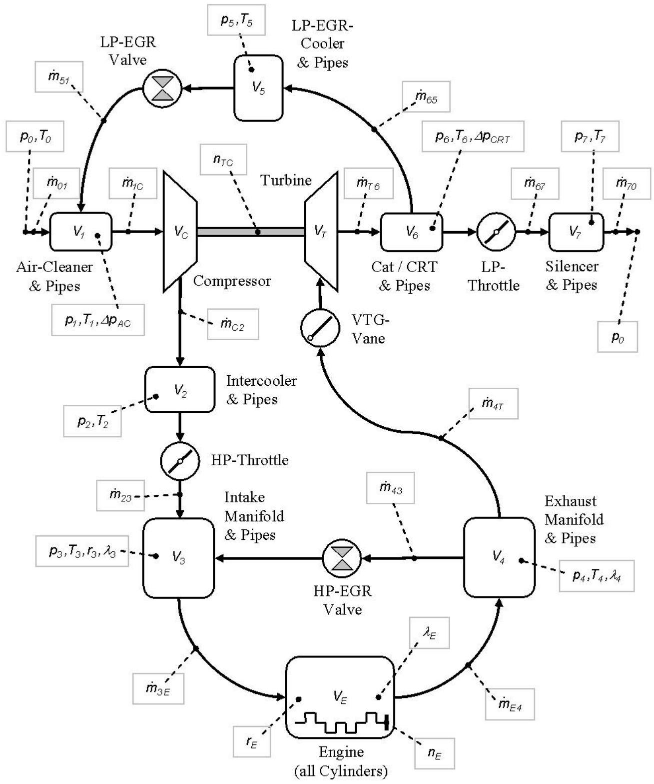

Fig. 10: Schematic representation of **Model_E**

VOLUME (PLENUM) SUB-MODEL

All volumes are described by means of the balance equations for the mass of the gas mixture (1), the EGR rate (2) and the energy (3), i.e. by means of the filling and emptying method. The air mass of the entire system treated as perfect gas is only concentrated in these volumes.

$$\frac{d}{dt}m = \sum_i m'_i - \sum_j m'_j \qquad (1)$$

$$\frac{d}{dt}r_k = \frac{1}{m}\cdot\sum_i\left[m'_i\cdot\left(r_{k_i} - r_k\right)\right] \quad \text{where} \quad r_k = \frac{m_k}{m} \qquad (2)$$

$$\frac{d}{dt}T = \frac{1}{m}\cdot\left[\frac{Q'}{c^\circ_V} + \kappa\cdot\left[\sum_i\left(m'_i\cdot T_i\right) - T\cdot\sum_j m'_j\right] - T\cdot\frac{d}{dt}m\right] \qquad (3)$$

With:

States in the actual volume

m	mass of the gas mixture
r_k	mass fraction of gas component k (e.g. EGR)
T	temperature

Exchanged quantities

m'_i	input mass flow rate
m'_j	output mass flow rate
Q'	heat flow rate

Other

t	time
c'_v	isochor thermal capacity of gas mixture
κ	isentropic exponent

The mass flow rates between these volumes are regarded as isothermal, since this provides a more realistic description of the system (closer than an isentropic system). For visualization purposes, the reader may assume the integration of a throttle or a valve between two pipe parts. This system may be considered adiabatic with a steady flow running through the throttle. Both assumptions need to be transferred to an isentropic system, as well. Accordingly, the mass and energy balance equations change.

$$\frac{d}{dt} m = m'_{in} - m'_{out}$$

$$\frac{d}{dt} U = Q' + m'_{in} \cdot h_{in} - m'_{out} \cdot h_{out}$$

where steady process implies

a) $\frac{d}{dt} m = 0$ thus $m'_{in} = m'_{out} = m' = const$

b) $\frac{d}{dt} U = 0$ for internal energy,

and adiabatic system $Q' = 0$

One results $h_{in} - h_{out} = 0$ for specific enthalpy

which for perfect gas means $T_{in} = T_{out}$ (isothermal).

TURBOCHARGER (TC) SUB-MODELS

The TC maps are processed to eliminate current interpolation difficulties.

The classic compressor maps (Fig. 11) represent the pressure ratio in relation to the reduced mass flow rate and the isentropic efficiency in relation to the reduced mass flow rate depending on various TC speeds. If the pressure ratio and the TC speed are known and the reduced mass flow is supposed to be specified, two or more solutions may be obtained at higher TC speeds (due to the flat curves).

To avoid such situations, the maps are appropriately (i.e. physically) processed by combining the information of both maps. This results in new maps, by way of which it is possible to find an iterative solution to the problem described above.

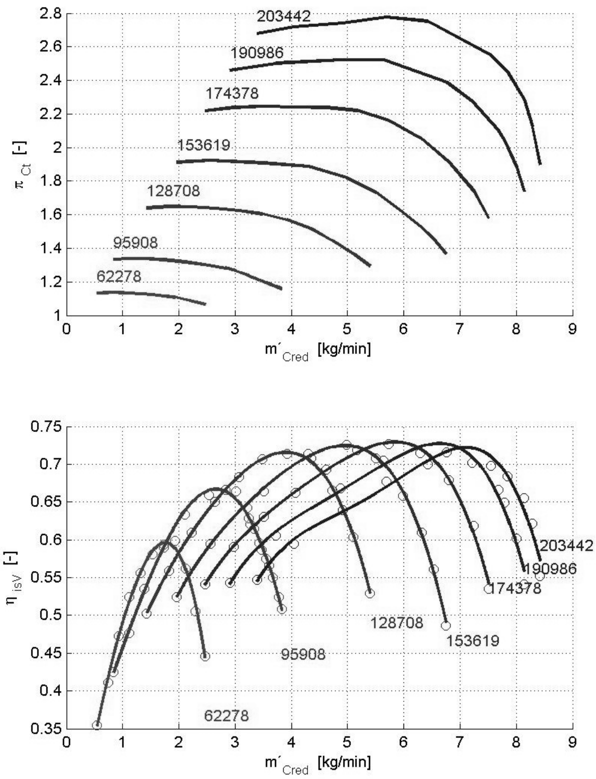

Fig. 11: Classic compressor maps

The situation on the turbine is nearly the same, but much more complicated because of the impact of the VTG. The turbine maps of Fig. 12 represent the reduced mass flow rate in relation to the pressure ratio and the product of isentropic and mechanical efficiency in relation to the reduced pressure ratio, both with the reduced TC tip speeds and the VTG position as parameter. Solely the bottom map appears to be rather complicated because of the fragmented curves.

The problem may be resolved in a fashion comparable to the compressor map. This way, not only may the difficult problem originating in the unsteady curves be resolved. Additionally, the TC speed is no longer a parameter of the maps.

CRANK TRAIN AND TC SHAFT SUB-MODELS

The dynamic of the crank train of the TC shaft is modeled by means of the following equation

$$\frac{d}{d\,t}\omega = \frac{1}{J} \cdot \left(\sum_i M_i \right) \qquad \text{with}$$

ω	angular speed
J	moment of inertia
M	torque

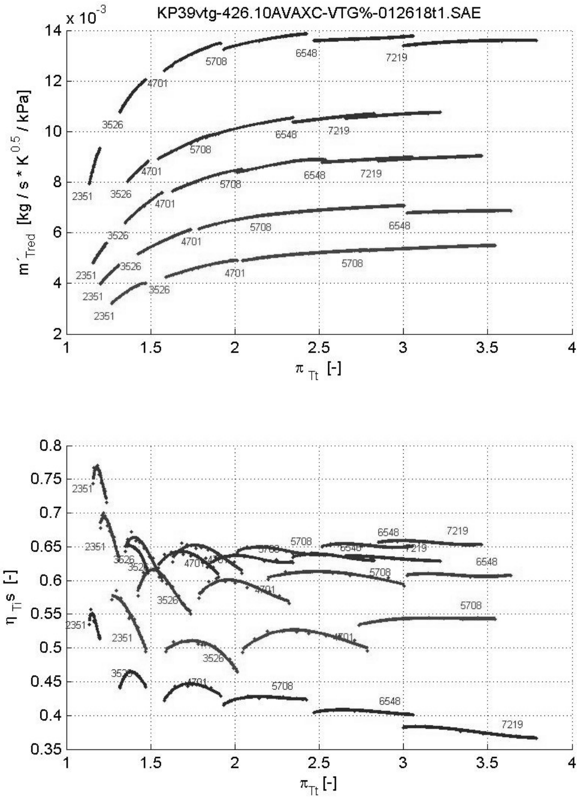

Fig. 12: Classic turbine maps

CYLINDER SUB-MODEL

The cylinder sub-model is partly physical (modeled as plenum) and partly empirical, i.e. it is based on some dynamometer measurements carried out solely on mass-produced engines (with mass-produced ECUs). Therefore, e.g. no data of EGR rate variations at the operation points (EOP) were available at that time.

Consequently, if the EGR rate shows a substantial deviation from the original measurements, this sub-model may be more or less inaccurate.

The cylinder sub-model is developed as a **mean values model**, i.e. all its states are not a function of the crank angle. The following states of this sub-model need to be determined: (a) the per-cycle gas mixture sucked into the cylinders, (b) its EGR rate, (c) the temperature of the exhaust gas leaving the cylinder and (d) the engine power.

The following considerations were made during the development of the model:

1. Since the entire model is projected for real-time operation, no detailed physical model may be used here.
2. The mass flow sucked into the cylinder depends more on density than on the pressure from the intake manifold.
3. The gas mixture of air and EGR from the intake manifold is homogenized.
4. The exhaust gas mass leaving the cylinder is equal to the sum of the sucked gas mixture mass and fuel mass (steady-state gas exchange process).
5. The exhaust gas temperature and the engine power depend on (a) the filled cylinder (volume) and therefore on the density of the intake manifold and engine speed, (b) the air-fuel-ratio (lambda), (c) the EGR rate into the cylinder and (d) the start of injection (SOI, not considered in this model, as measurements were not available).

Fig. 13 shows an example of one of these cylinder calibration functions (Index *3* denotes the intake manifold).

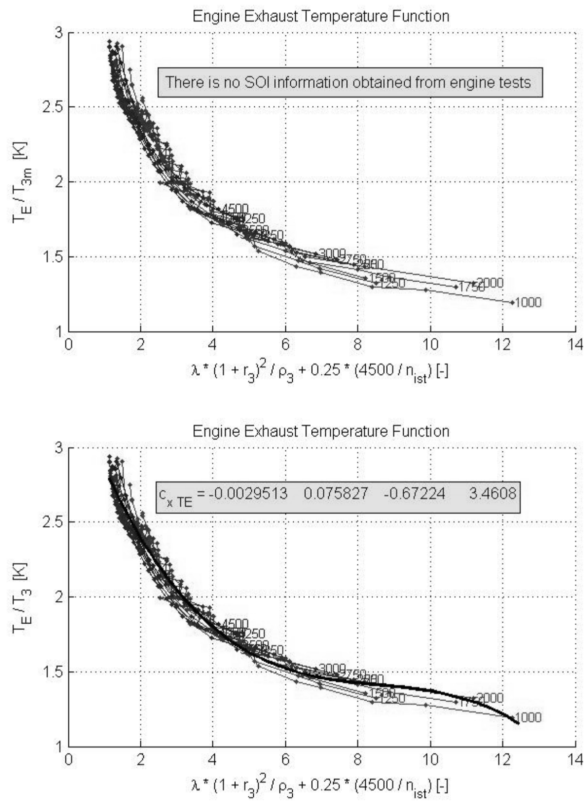

Fig. 13: Example of engine characteristics based on dynamometer measurements of a mass-produced engine

Fig. 14 shows the results of a simulation during a VTG actuator step. The engine was simulated at an operation point of 3000 rpm and 9 bar BMEP.

MODEL-BASED PREDICTIVE CONTROLLER

The model-based non-linear predictive controller (short MNPC or non-linear MPC) developed for this application contains an observer, a predictor and an optimization algorithm (Fig. 15). The observer model, Model_S, uses all available sensor signals to calculate the unmeasured states of the system. These states form the basis for the predictor, Model_G. The predictor estimates the future behavior of N different combinations of actuator positions. The optimization algorithm (J) judges all estimations by calculating a cost function for each simulated actuator position set. Both the observer (Model_S) and the predictor (Model_G) are model-based and both models are derived from the model presented above (**Model_E**).

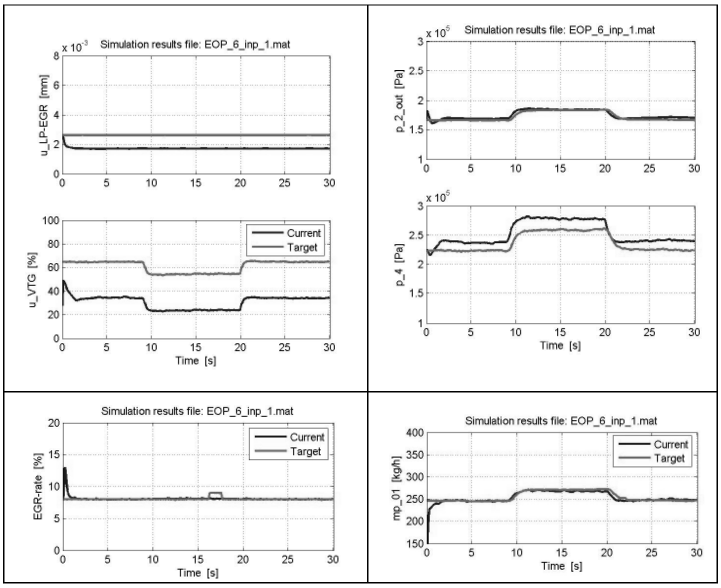

Fig. 14: Simulation results of the engine model during a VTG actuator step

OBSERVER MODEL

The observer model, named **Model_S**, calculates all states of the engine (taken as system) based on sensor signals (as system inputs) in real time.

This model is necessary to compute all other states of the engine (e.g. EGR rate in all volumes), except the ones provided directly as sensor signals. Hence, **Model_S** offers, in real time, a set of all required engine states to form the basis for prediction.

PREDICTOR MODEL

The predictor model, named **Model_G,** holds no sensor inputs, as it only predicts the future horizon (no sensor signals are available for the future). **Model_G** is stimulated with new actuator positions delivered by the optimization algorithm and simulates the engine's response in the future.

AIR CLEANER SUB-MODEL AS AN EXAMPLE OF MODEL REDUCTION

To explain the particularity of model reductions (simulation model (**Model_E)**, observer (**Model_S** and **Model_E**), and predictor (**Model_G**)**,** the transformations for the air cleaner sub-model (Volume V_1) are exemplified here.

Air Cleaner Sub-model of **Model_E**

The inputs of the air cleaner sub-model of **Model_E** include:

1. The ambient states,
2. the states of the connecting pipe between the LP EGR valve and the air cleaner (mass flow, temperature and EGR rate), and
3. the mass flow rate running through the compressor.

In this sub-model, the air mass flow rate and thus the three balance equations (1) to (3) (see VOLUME (PLENUM) SUB-MODEL) are calculated. The air cleaner sub-model outputs cover the states and the mass flow rate. The flow discharge coefficient is calibrated and deduced from the analysis of the dynamometer measurements.

Air Cleaner Sub-model of Observer **Model_S**

The air cleaner sub-model as part of **Model_S** (working on a real-time basis) does not need to calculate the air mass flow rate, as it is available in form of a sensor signal. Because of the low dynamic of the filling and emptying processes, the balance equations (1), (2), (3) may be neglected.

In consequence, only steady-state mixing processes for the EGR rate and EGR temperature may be implemented here. By comparing the computed and measured air mass flow rate, the flow discharge coefficient may be calibrated to use it in **Model_G**. The calculated gas mixture mass flow and the other states of this volume stand for initialization states of the predictor.

Air Cleaner Sub-model of Predictor **Model_G**

The air cleaner sub-model used in **Model_G** needs to compute the air mass flow rate because, as mentioned above, no sensor signals are available during predictions. Since this sub-model is expected to sensitively and accurately respond to different actuator settings, the balance equations (1), (2) and (3) may no longer be neglected here.

Before starting the predictions, all integrators need to be initialized externally and their states reset.

OPTIMIZATION ALGORITHM

After prediction, the simulation results of **Model_G** are used in the optimization algorithm (**J**) to establish the optimal new actuator positions required to achieve the target values. Because of the hardware and software restrictions inherent in an engine ECU (usual sampling time ~1 msec), no classic optimization algorithm may be implemented here.

In this case, the only feasible way consists of starting many predictions from the same basic states (provided by **Model_S**) for many actuator positions and deciding between them, i.e. deciding which of these actuator positions minimizes the cost function **J**.

At present, the optimization algorithm **J** is being developed to minimize the functional *J* of only two variables (*u1* and *u2*), e.g. LP EGR valve and VTG vane positions.

To find the global and not just a local optimum, the whole range of vector *u = (u1, u2)* needs to be considered. The *(u1, u2)* combinations cover an *N* points map. For each of these *N* combinations of *(u1, u2)* one prediction cycle consisting of the following steps is implemented:

1. The predictor **Model_G** is reset and initialized with the states from **Model_S**.
2. The prediction is started for the current combination *(u1, u2)* of the actuator positions.
3. Some integration steps of **Model_G** are carried out.
4. All results, i.e. the predicted states (*y_G*), actual current state deviations from target values *dy = (y_target – y_actual)* and actual actuator positions *u_actual* are saved.

Now, for all N predictions, the functional J is evaluated:

$$J_n = \sum_{i=1}^{3} \begin{bmatrix} c_{y1}\left|y1 - y1\_t\arg et\right|_{i,n} \cdots \\ c_{y2}\left|y2 - y2\_t\arg et\right|_{i,n} \cdots \\ c_u\left|u - u\_old\right|_{i,n} \cdots \end{bmatrix}$$

The minimum of the functional *J* corresponds to the optimal actuator position set *u_new*.

To improve the stability of the algorithm, i.e. to avoid oscillations, a weighted sharing method may be applied to smooth the behavior, or a windowing approach may be implemented to achieve the N actuator position sets.

SIL ENVIRONMENT FOR MNPC DEVELOPMENT

Fig. 15 shows the integration of the MNPC in parallel with a mass-produced ECU for engine control on the dynamometer. For implementation on the dynamometer, some ECU functions need to be bypassed.

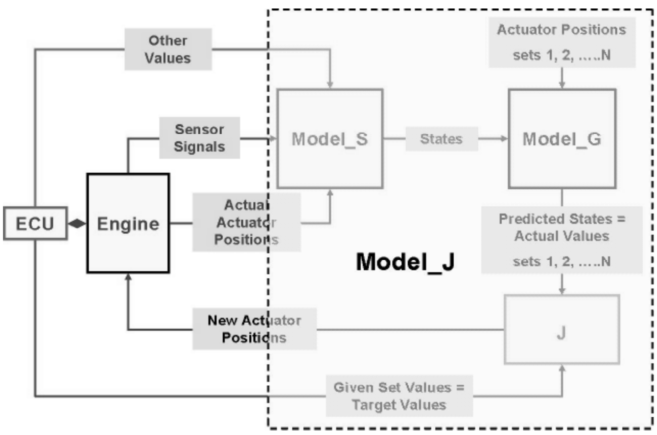

Fig. 15: Combining the MNPC with a mass-produced ECU

To develop and pre-calibrate this MNPC, a SIL environment is built up. In this SIL environment, the simulation model (**Model_E**) replaces the engine and the mass-produced ECU (as shown in Fig. 16).

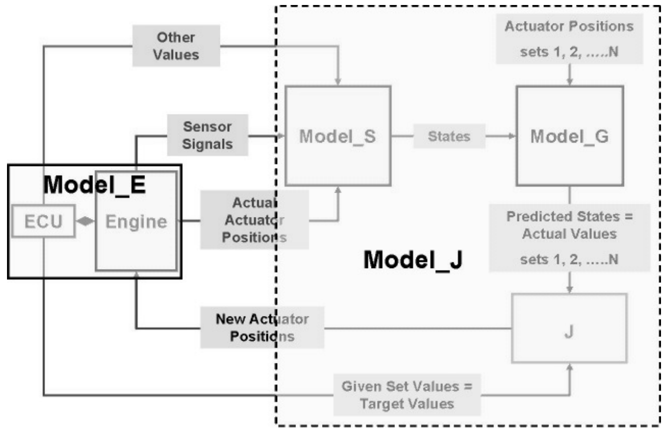

Fig. 16: SIL environment for MNPC development

To implement this SIL environment of **Model_J,** which works on a sampling time of 1 msec, into Simulink, a multi-rate application needs to be realized, since **Model_E** uses a much shorter sampling time (here 5 µsec) for numeric stability and accuracy reasons.

RESULTS OF SIL TESTS ON THE MNPC

Choosing the targets has a major influence on the performance of this MNPC algorithm. Since the desired values for the air mass flow rate, EGR rate and boost pressure are taken over from the mass-produced ECU, and thus optimized only for HP EGR, they will not fit exactly into the modified engine holding an added low pressure EGR line.

Compared to LP EGR, HP EGR leads to higher intake manifold temperatures. Hence, to reach the same air and EGR mass flow rates, HP EGR requires a higher boost pressure (due to thermal state equation).
Consequently, selecting the boost pressure from a mass-produced ECU as a target value will result in slightly excessive boost pressures.

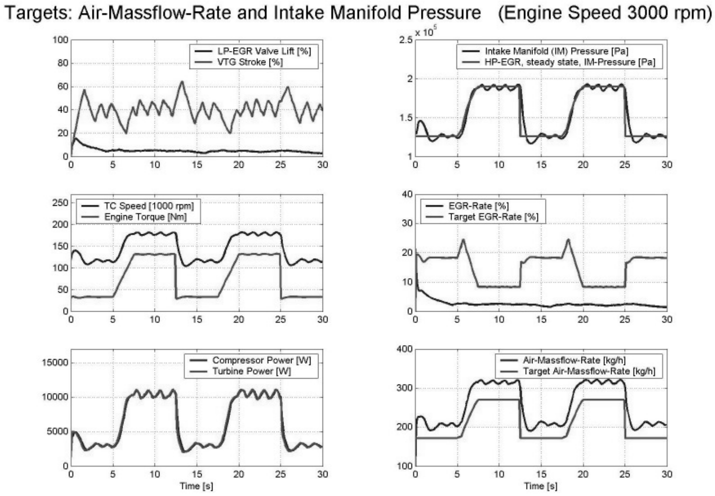

Fig. 17: SIL Test of MNPC; targets: air mass flow and boost pressure

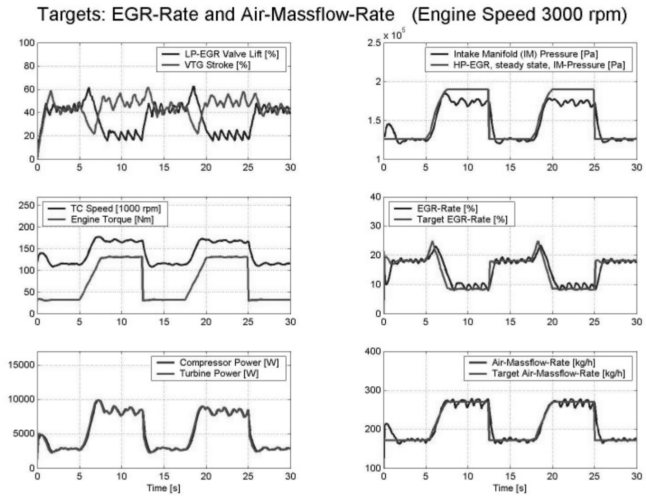

Fig. 18: SIL Test of MNPC; targets: EGR rate and air mass flow

As regards the cylinder-filling process, the air and EGR mass flow rates are more important than the boost pressure. The simulation results shown in Fig. 17 and Fig. 18 confirm this statement. In Fig. 17, the air mass flow and boost pressure are used as targets for the controller, and in Fig. 18, the EGR rate and air mass flow stand for the targets. These tests reveal that the targets should be chosen in consideration of the changed requirements of the modified hardware. This fact renders the development of an MNPC more difficult, as adequate target values are missing when this controller is tested together with the modified engine and its mass-produced ECU on the engine test bench.

HARDWARE MODIFICATIONS TO REALIZE A LOW-PRESSURE EGR LINE

Using EGR is a tried-and-tested method to minimize NO_x emissions. Finally, the efficiency of this method is also based on both a decrease of the in-cylinder peak temperature as well as on a decrease of the in-cylinder oxygen concentration. Both parameters are very important for NO_x generation. The potential of using EGR to reduce NO_x emissions increases in view of the possibility to perform higher EGR rates and lower EGR temperatures. For the diesel combustion principle, EGR rates up to 60% and more are conceivable. In reality, however, the pressure difference of the intake and exhaust manifold limits EGR rates.

At present, two methods are applied in connection with EGR. Internal engine EGR (valve overlap) hardly allows for cooling. Moreover, controlling the ratio is possible only to a limited extent. By contrast, the external EGR mixes the exhaust gas in front of the turbine of the turbocharger and leads it toward the intake manifold via an EGR valve. This system stands for the state-of-the-art external high-pressure EGR. Often, it is difficult to reach an equal cylinder-to-cylinder distribution of the EGR when applying this method. This affects NO_x generation to a great extent. With a low-pressure EGR, which means that the EGR is extracted behind the turbine and mixed with fresh air in front of the compressor, a better distribution and also some other advantages may be achieved. A combination of high- and low-pressure EGR offers the best potential, although the complexity of the system increases.

LP EGR necessitates the conveyance of exhaust gas through the TC compressor. Depending on the EGR rate, the inlet temperature of the compressor increases. In addition, the required higher pressure ratios lead to higher compressor outlet temperatures. Hence, the thermal stress on the components, in particular on the compressor wheel, increases. Some substances contained in the exhaust gas may cause corrosion on the air path components, as well. This negative impact increases, if the EGR is cooled below the dewpoint, so that liquid acids can attack the components. Furthermore, unburned hydrocarbons and other components of the exhaust gas tend to cover both the compressor wheel and the diffuser with resin-like deposits. This leads to a considerable reduction of the efficiency of the compressor. From today's point of view, the most critical impact on the compressor wheel arises from relatively large droplets hitting the compressor blades at an unfavorable angle. These droplets cause material fatigue within a short period of time. Consequently, parts of the blades' surface become brittle, thus starting destroying the wheel within quite a short time. Additionally, a layer of soot particles increases the stress on the compressor.

Obviously, the components cannot resist these extra loads without additional measures. Detailed analysis and subsequent adaptation of the whole EGR, cooling and turbocharging system forms the basis for a solution.

First results show that droplets cause the most harmful damage. Therefore, big droplets which cannot follow the gas flow need to be avoided under any circumstance. This implies that the cooler inside the EGR line has to be operated in a way ensuring that the dewpoint is not reached and thus preventing the occurrence of liquids within the EGR line during normal operation. Now, the only remaining area where a liquid phase may crop up (depending on the thermodynamic state) is in the mixing zone between the EGR and the fresh air. Optimizing the intermixture directly in front of the compressor is of utmost importance here. Due to the fact that condensation and especially the growth of droplets are time-related processes, the length behind the mixing area of the EGR and the fresh air in front of the compressor needs to be minimized. If the critical components are arranged favorably, the condensate generated during cold start occurs only on the pipe walls. Additionally optimized piping may force the condensate to flow into the exhaust pipe without contacting any critical components. The condensate then leaves the vehicle together with the exhaust gas. With these proposed modifications it is possible to ensure that the components do not suffer from significant damage during the cold start stage. Coating the critical components makes them resistant to wettening – at least temporarily. These measures should allow for reliable operation over the vehicle's complete service life. These coatings do not only improve corrosion behavior, they also change the wettability of the surface and therefore reduce the problem of growing resin-like coatings.

CONCLUSION

The combination of the state-of-the-art HP EGR and an LP EGR seems to be a system which is capable of complying with the requirements of new emission standards. The benefits of this system may only be realized in combination with a new controller concept. The proposed predictive model-based controller ensures a good transient and steady-state control behavior.

The development process guarantees a good comparison of SIL and HIL simulations due to the fact that both methods are used in parallel.

The objective of the ongoing development is to modify the hardware to enable a low-pressure EGR line used in a series application. Many components, which need to be made compatible with this new system configuration are already included in BorgWarner's portfolio. The compressor of the turbocharger stands for one of the most critical components. Improvement of the components is essential; yet an overall system solution is even more important. The package of the whole air management system needs to be designed accordingly.

Additionally, further optimizations of the controller concept need to be evaluated and proven in the test runs on the engine test bench. Only the combination of an optimized control strategy and a system approach as regards the required changes of the hardware seems to pave the way for the feasibility of such combined HP and LP EGR systems realizable in subsequent stock production. BorgWarner focuses on improving both the required hardware improvements and the advanced controller concepts.

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