14

Advanced Torque Control

C. Fritzsche and H.-P. Dünow
Hochschule Wismar – University of Technology, Business and Design
Germany

1. Introduction

Torque is one of the fundamental state variables in powertrain systems. The quality of motion control highly depends on accuracy and dynamics of torque generation. Modulation of the air massflow by opening or closing a throttle is the classical way to control the torque in gasoline combustion engines.

In addition to the throttle and the advance angle several other variables are available to control the torque of modern engines - either directly by control of mixture or indirectly by influence on energy efficiency. The coordination of the variables for torque control is one of the major tasks of the electronic control unit (ECU).

In addition to the generation of driving torque other objectives (emission threshold, fuel consumption) have to be taken into account. The large number of variables and pronounced nonlinearities or interconnections between sub-processes makes it more and more difficult to satisfy requirements in terms to quality of control by conventional map-based control approaches. For that reason the investigation of new approaches for engine control is an important research field (Bauer 2003).

In this approach substitute variables instead of the real physical control variables are used for engine torque control. The substitute variables are used as setpoints for subsidiary control systems. They can be seen as torque differences comparative to a torque maximum (depending on fresh air mass). One advantage of this approach is that we can describe the controlled subsystems by linear models. So we can use standard design methods to construct the torque controller. Of course we have to consider variable constraints. Due to the linearization the resulting control structure can be used in conventional controller hardware.

For the superordinate controller a Model Predictive Controller (MPC) based on state space models is used because with this control approach constraints, and also setpoint- and disturbance progressions respectively, are considered.

In chapter 2 we briefly explain the most important engine processes and the main torque control variables. The outcome of this explanations is the use of a linear multivariable model as a bases for the design of the superordinate torque controller. The MPC- algorithm and its implementation are specified in chapter 3. Chapter 4 contains several examples of use.

In this chapter it is also shown how the torque controller can change to a speed- or acceleration controller simply by manipulation of some weighting parameters.

Source: New Approaches in Automation and Robotics, Book edited by: Harald Aschemann, ISBN 978-3-902613-26-4, pp. 392, May 2008, I-Tech Education and Publishing, Vienna, Austria

2. Process and torque generation

Combustion and control variables: In figure 1 the four cycles of a gasoline engine are shown in a pressure/volume diagram. The mean engine torque results from the difference energy explained by the two closed areas in the figure.
The large area describes the relation of pressure and volume during compression and combustion. The charge cycle is represented by the smaller area.
For good efficiency the upper area should be as large as possible and the area below should be minimal (for a given gas quantity). The best possible efficiency in theory is represented by the constant-volume cycle (or Otto cycle) (Grohe 1990, Urlaub 1995, Basshuysen 2002).

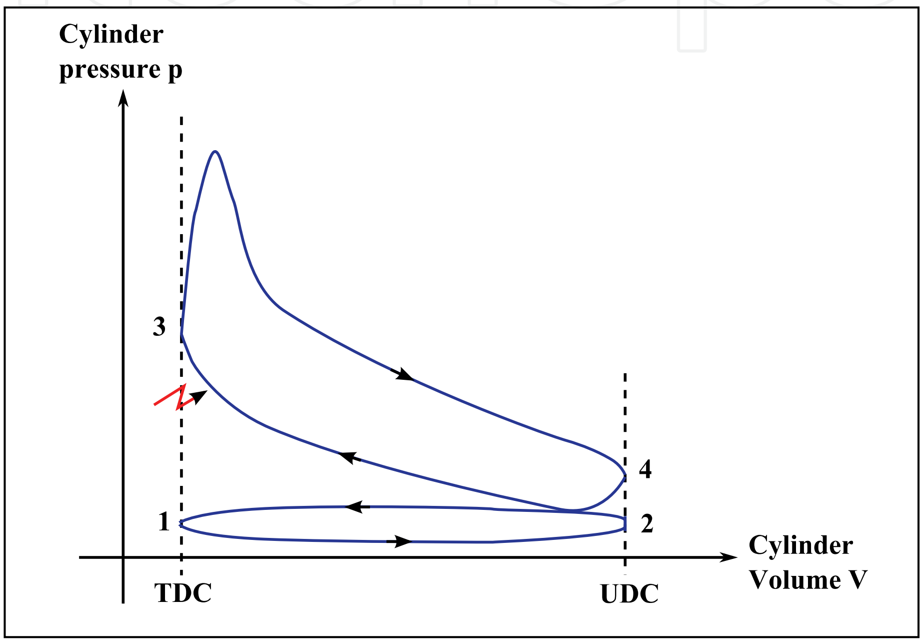


Fig. 1. p-V-Diagram (gasoline engine)

The maximum torque of the engine basically depends on the fresh air mass which is reaching the combustion chamber during the charge cycle. A desired air/fuel mixture can be adjusted by injecting an appropriate fuel quantity. The air/fuel-ratio is called lambda (λ). If the mass of fresh air corresponds to the mass of fuel (stoichiometric ratio) the lambda parameter is one (λ=1). By several reasons the combustion during the combustion cycle is practically never complete. So after the cycle there remain some fresh air and also unreacted particles of fuel. A more complete reaction of the air can be reached by increasing fuel mass (λ<1). This results in rising head supply and hence to an increase of the potential engine torque. The maximum of torque is reached by approximately λ=0.9. The torque decreases more and more if lambda increases. So one can realize fast torque changes for direct injection engines by modulation of the fuel.
The maximum of lambda is bounded by combustibility of the mixture (depends on operating conditions). For engine concepts with lean fuel-air ratio (λ>1) the fuel feed is the main variable for torque control. Because of the exhaust after treatment λ=1 is requested for most gasoline engines at present. In case of weighty demands however it is possible to change lambda temporary.

In addition to air and fuel the torque can be controlled by a number of other variables like the advance angle or the remaining exhaust gas which can be adjusted by the exhaust gas recirculation system.

For turbocharged engines the air flow depends in addition to the throttle position on turbocharger pressure which can be controlled by compressors or by exhaust gas turbines.

Pressure charging leads to an increase of the maximum air mass in the combustion chamber and hence to an increasing maximum of torque even if the displaced volume of the engine is small (downsizing concepts).

Because of the mass inertia of the air system it takes time until a desired boost pressure is achieved. The energy for the turbocharging process is taken from the flue gas stream. This induces a torque load which can result in nonminimumphase system behaviour.

In addition to the contemplated variables there are further variables imaginable which influence the torque (maximum valve opening, charge-motion valve, variable compression ratio, etc.).

Furthermore the total torque of the engine can be influenced by concerted load control. For example one can generate a negative driving torque by means of the load of the electric generator. The other way around one can generate a fast positive difference torque by abrupt unloading. This torque change can be much faster than via throttle control.

In hybrid vehicles we have a number of extra control variables and usable parameters for torque control.

More details on construction and functionality of internal combustion engines are described by Schäfer and Basshuysen 2002, Guzella and Onder 2004 and Pulkrabeck 2004.

For control engineering purposes the torque generation process of a gasoline engine is multivariable and nonlinear. It has a large number of plant inputs, a main variable to be controlled (the torque) and some other aims of control like the quality of combustion concerning emission and efficiency.

Considerable differences in the dynamic behaviour of the subsystems generate further problems particularly for the implementation of the control algorithms. For example we can change the torque very fast by modulation of the advance angle (compared to the throttle). Normally the value that causes the best efficiency is used for the advance angle. However to obtain the fastest torque reduction it can be helpful to degrade the efficiency and so the engine torque is well directed by the advance angle. Fast torque degradation is required e. g. in case of a gear switching operation. For the idle speed control mode fast torque interventions can be useful to enhance the stability and dynamics of the closed loop. Fast load changes also necessitate to generate fast torque changes. In many cases the dynamic of sub-processes depends on engine speed and load.

Whether the torque controller uses fast- or slow-acting variables depends on dynamic, efficiencies and emission requirements. Mostly the range of the manipulated variables is bounded.

In the normally used "best efficiency" mode one can only degrade the torque by the advanced angle. For special cases (e. g. idle speed mode) the controller chooses a concerted permanent offset to the optimal advance angle. So a "boost reserve" for fast torque enhancement occurs. Because of the efficiency this torque difference is as small as possible and we have a strong positive (and changeable) bound for the control variable. This should be considered in the control approach. The main target variables and control signals are pictured in figure 2.

We can summarize: The torque control of combustion engine is a complex multivariable problem. In addition to the torque we have to control other variables like lambda, the efficiency, EGR , etc. The sub-processes are time variant, nonlinear and coupled. The control variables are characterized by strong bounds.

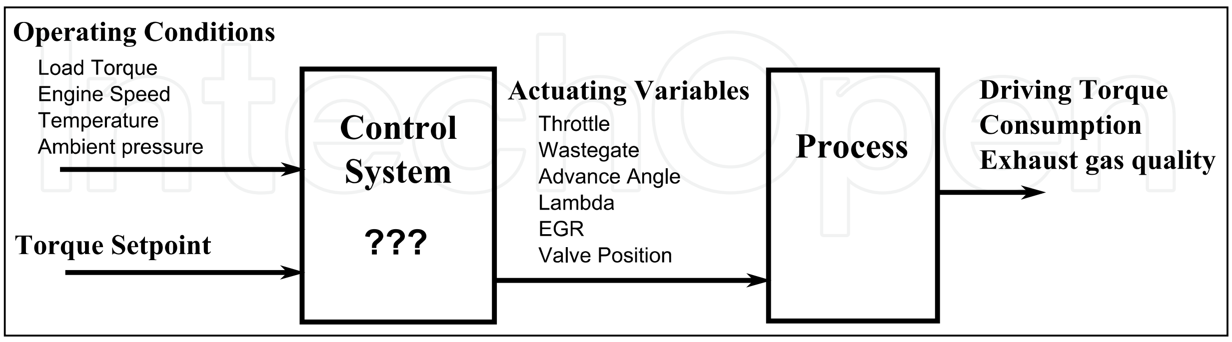


Fig. 2. Actuating and control variables

Torque control: Below we describe the most important variables for torque control and the effect chain.

a.) fresh air mass:

As aforementioned the engine torque mainly depends on the fresh air mass as the base for the maximum fuel mass. Classical the air mass is controlled by a throttle within the intake. In supercharged engines the pressure in front of the throttle can be controlled by a turbocharger or compressor. The result of supercharging is a higher maximum of fresh air mass in the combustion chamber. This leads to several benefits. In addition to the throttle or charging pressure the air mass flow can also be controlled by a variable stroke of the inlet valve.

It is not expedient to use all variables of the air system for torque control directly. For the torque generation the air mass within the cylinder is mainly important (not the procedure of the adjustion). Hence it is more clever to use the setpoint for the fresh air mass as the plant input for the torque controller. This can be realized by the mentioned control variables with a the aforementioned sub-control system. One advantage of the approach is, that the sub-control system at large includes measures for the linearization of the air system. Furthermore the handling of bounds is a lot easier in this way.

b.) exhaust gas recirculation (EGR):

By variable manipulating of the in- and outlet valves it is possible to arrange, that both valves are simultaneously open during the charge cycle (valve lap). So in addition to the fresh air some exhaust gas also remains the combustion chamber. This is advantageous for the combustion and emission. In case of valve lap the throttle valve must be more open for the same mass of fresh air. This is another benefit because the wastage caused by the throttle decreases. The EGR can also be realized by an extern return circuit. Here we only consider the case of internal EGR. The EGR induced a deviation from the desired set point of fresh air. This deviation usually will be corrected by a controller via the throttle. As aforementioned this correction is relatively slow. If the mechanism for the valve lap allows fast shifting we can realize fast torque adjustments because the change of air mass by the valve lap occurs immediately.

c.) advance angle:
An important parameter for the efficiency of the engine is the initiation of combustion. The point of time mainly depends on the advance angle. The ignition angle which leads to the best efficiency is called the optimized advance angle. It depends on engine speed and the amount of mixture. A delay in the advance angle leads to less efficiency and to the decrease of engine torque. The relation between advance angle and torque is nonlinear.
d.) lambda:
The amount of fuel determines (for a given fresh air mass) the torque significantly. As aforementioned bounded we can manipulate the torque by lambda. For direct injection concepts the torque changes immediately if lambda is modified. Torque control by means of lambda is limited for engine concepts with λ=1.
e.) other variables
The quality of mixture has also an effect on the efficiency and so on the torque. For the quality several construction parameters are important. Further the quality depends on adjustable variables like fuel pressure, point of injection or the position of a charge-motion valve. The setpoints for these parameters are primarily so chosen that we obtain best mixture efficiency.
In figure 3 the main variables and the effects on the process are outlined.

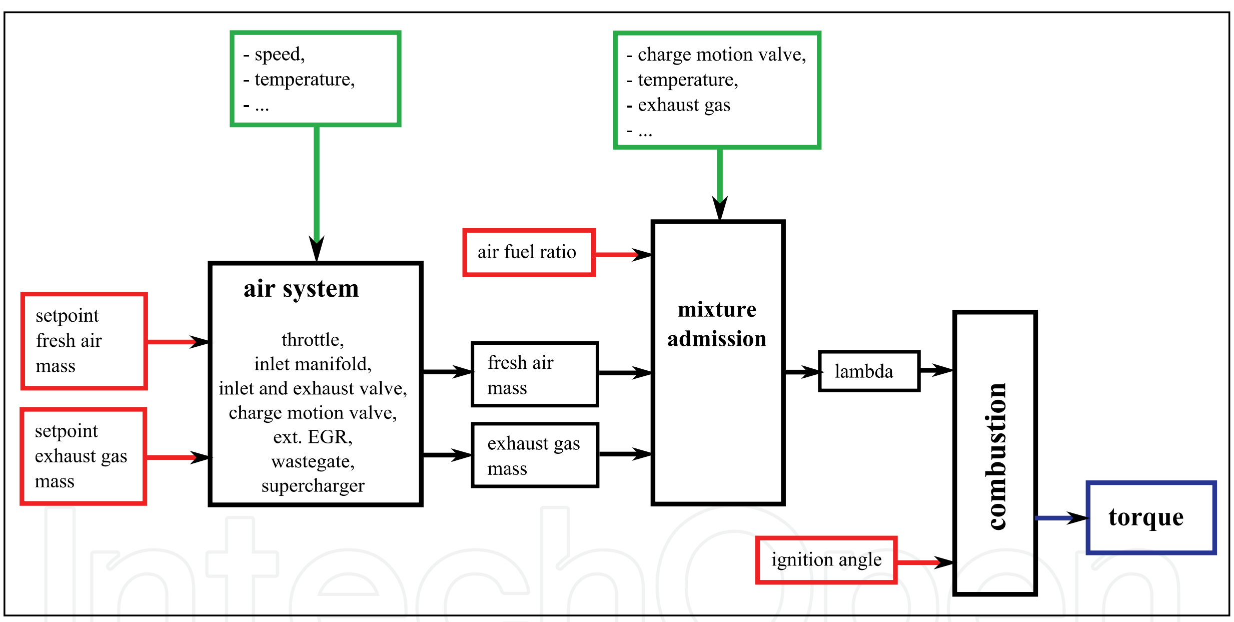


Fig. 3. Main variables to influence the energy conversion and the torque generation

From the explanation up to now we can deduce appropriate variables for torque control:
– fresh air mass in the combustion chamber (adjustable via charge pressure, throttle, valve aperture and valve lap),
– fuel, lambda (adjustable by injection valve),
– advance angle,
– internal exhaust gas recirculation (adjustable via valve lap)

All variables are bounded.
Subsidiary control systems: For simplification of the torque control problem it is expedient to divide the process in to several sub-control circuits. The following explanations are aimed

to gasoline engines with turbo charging, internal EGR, direct injection and homogeneous engine operation (λ=1).
The sub-control circuits are outlined in figure 4. The most complicated problem is the design of the subsystem for fresh air control. Because of the strong connection of fresh air and EGR it makes sense to control both simultaneously in one sub-system. The control system has to consider or compensate a number of effects and nonlinear dependences.
The main task is to control the fresh air mass and the desired amount of exhaust gas which reaches the combustion chamber during the charge cycle. The control system should reject disturbances and adjust new setpoint values well (fast, small overshoot). Another requirement is that it should be possible to describe the complete control circuit by a linear model.
For the control structure we propose a reference-model controller (figure 5). One of the aims of this control structure is to achieve a desired dynamic behaviour for the circuit. The mentioned requirement for simplification of the modelling is implicitly given in this way. Nonlinear effects can be compensated by appropriate inverted models.

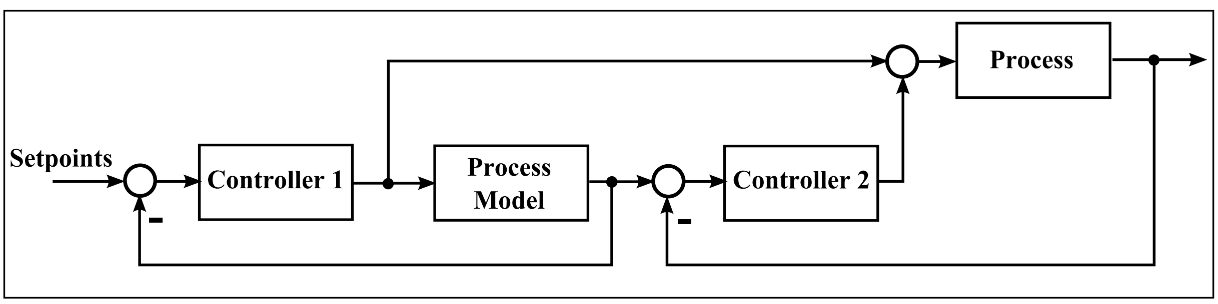


Fig. 4. Model Following Control (MFC)

In the explained approach the fresh air mass is used as the main control variable. From this air mass results the maximum possible torque with approximately λ = 0.9.
In case of modification of the exhaust gas portion we only consider the influence on the fresh air. The influence on combustion quality is disregarded. Increasing of exhaust gas portion leads to decreasing of fresh air and so to reduction of torque. This influence is compensated by a special controller so that the action of EGR on the engine torque is comparable to the characteristic of a high-pass filter. An advantage is obtained if the acting effect from EGR-setpoint to the fresh air is faster than via the throttle. In this case it is possible to realize fast (but transient) changes of the torque.
For torque manipulation via the advance angle it is not necessary to use a feedback system. The influence on the efficiency of the engine can be modelled by a characteristic curve. So for this control path a feedforward control action is adequate. We can find the same conclusion for lambda.
Second-level control system: By means of the described sub-control systems the actuating variables of the torque control system can be defined. The main variables are the setpoints of the fresh air mass, the exhaust gas, the advance angle and the setpoint for lambda. The relation between the torque and the mentioned variables is nonlinear. Following the torque controller also had to be nonlinear and the controller design could be difficult.
A more convenient way is to linearize the sub-control systems first by stationary functions at the in and outputs. In the described control approach the control variables are substituted by a number of partial torque setpoints (delta torques). The partial torques represent the

influence of the used control variables to the whole engine torque. The over-all behaviour of the system to be controlled is linear (widely).
Base control variable is the theoretical maximum of the torque for a given fresh air mass. The other variables can be seen as desired torque differences to a reference point (the maximum of torque). That means if the delta variables all set to zero, the maximum of torque for the given fresh air mass should be generated by the engine.
The difference variables are both: actuation variables for the torque and variables to be controlled. In this way it is possible to adjust stationary other setpoints for lambda, the residual exhaust gas or the advance angle.
We can find the bounds for the substituted variables from the physical bounds also by using the linearization functions. It is assumed that the nonlinear functions are monotone in the considered range.
From the explanations above the total control structure follows (figure 5). The plant for the torque controller consists of the sub-control-systems (fresh air, residual exhaust gas, and lambda and advance angle) and the stationary linearization systems. The whole system can be described by a linear multivariable model. Only the bounds of the input variables have to be taken into account. The plant outputs are the engine torque and the difference variables. Other plant outputs could be estimated values for exhaust gas emission or for fuel consumption (can be estimated by simplified models).
The setpoints for the torque are generated by the driver or by other ECU-functions (e.g. for changing gears). For the realization one can find different requirements.
For example we have to realize a requested torque as fast as possible. For other situations the time could be less important and one wish to realize a desired torque with a minimum of consumption. The control system should be able to handle all the different requirements.
The control variables are not available at all time. In general the range of the control variables is bounded. Sometimes they may not differ from the setpoint. The torque controller should consider this variability of bounds.
An appropriate control principle with the ability to solve this problem is the well known model predictive controller (MPC). This approach can handle variable bounds as well as known future setpoints and disturbances. Future setpoints are known for instance for changing gears. Switch-on of the air conditioning system is an example for known future disturbances. Matching to miscellaneous priorities is possible by appropriate weighting matrices in the control algorithm.
The model structures for the superordinate torque control system can be explained from figure 5.
For example we can arrange the model equation

$$\begin{bmatrix} y_1 \\ y_2 \\ y_3 \\ y_4 \end{bmatrix} = \begin{bmatrix} G_{11}(s) & G_{12}(s) & G_{13}(s) & G_{14}(s) \\ 0 & G_{22}(s) & 0 & 0 \\ 0 & 0 & G_{33}(s) & 0 \\ 0 & 0 & 0 & G_{44}(s) \end{bmatrix} \cdot \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \end{bmatrix} \tag{1}$$

if we use all of the aforementioned control variables (fresh air, residual exhaust gas, lambda and advance angle). The output signals of the model are the engine torque (y_1) and the three difference control variables which appear in fact ($y_2 \ldots y_4$).

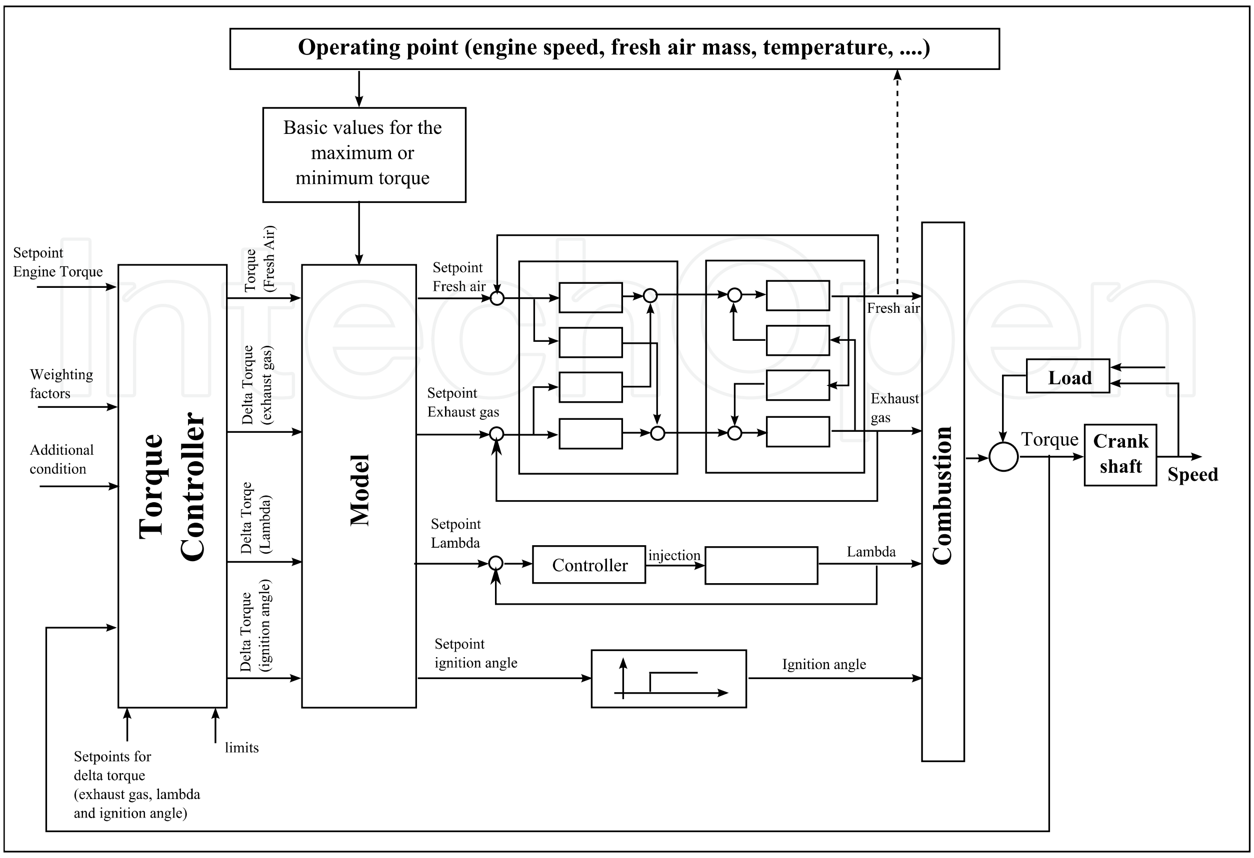


Fig. 5. Two layer control approach

The engine torque (y_1) depends on all actuating variables. $G_{11}(s)$ describes the effect of the fresh air path to the engine torque, $G_{12}(s)$ models the effect of the exhaust gas path on the engine torque, $G_{13}(s)$ describes the behaviour of the torque depending on lambda and $G_{14}(s)$ the effect of the ignition angle path. We may assume, that the output variables y_2 ... y_4 are only dependent on the corresponding setpoints because of the decoupling effects of the sub-control circuits. $G_{22}(s)$, $G_{33}(s)$ and $G_{44}(s)$ are simple low-order transfer functions.
If we have (in addition to the torque setpoint) other control requirements (e.g. consumption) it is easy to extend the model by appropriate transfer functions for the estimation of the control variable effects on the considered output variables. For exhaust gas emission and fuel consumption this results in a model as follows:

$$\begin{bmatrix} y_1 \\ y_2 \\ y_3 \\ y_4 \\ y_5 \\ y_6 \end{bmatrix} = \begin{bmatrix} G_{11}(s) & G_{12}(s) & G_{13}(s) & G_{14}(s) \\ 0 & G_{22}(s) & 0 & 0 \\ 0 & 0 & G_{33}(s) & 0 \\ 0 & 0 & 0 & G_{44}(s) \\ G_{51}(s) & G_{52}(s) & G_{53}(s) & G_{54}(s) \\ G_{61}(s) & G_{62}(s) & G_{63}(s) & G_{64}(s) \end{bmatrix} \cdot \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \end{bmatrix} \qquad (2)$$

The extension of the model is possible by using any other variable of the engine as plant output signals. Possibilities are, for instance, the engine speed or the acceleration.

In this way we can consider the torque and the speed simultaneously. The model has to be completed to:

$$\begin{bmatrix} \vdots \\ y_7 \end{bmatrix} = \begin{bmatrix} \vdots & \vdots & \vdots & \vdots \\ G_{71}(s) & G_{72}(s) & G_{73}(s) & G_{74}(s) \end{bmatrix} \cdot \begin{bmatrix} \vdots \\ u_4 \end{bmatrix} \quad . \tag{3}$$

y_7 describes the engine speed. The transfer functions of the lower line describe the effects of all inputs on the speed.
Certainly, it is not possible to adjust torque and speed independently. The structure allows however to switch between torque and speed control smoothly. This can be done by the weighting parameters of the MPC. In the same way it is feasible to extend the model by a line which describes the acceleration depending on the input signals. A disadvantage of the model extensions is the increasing complexity.

3. MPC-algorithm and implementation

3.1 Approach

As mentioned the model predictive control concept seems to be an appropriate approach for torque control. In the following we describe the basics of this control concept. The explanations are focused mainly on a state space solution.
MPC is widely adopted in industries, primarily in chemical engineering processes. This control principle is an established method to deal with large constrained multivariable control problems (Dittmar 2004, Dünow 2004, Grimble 2001, Maciejowsky 2002, Salgado et al. 2001). The main idea of MPC is to precalculate the control action by repeatedly solving of an optimization problem (see figure 6). The optimization aims to minimize a performance criterion during a time-horizon subject to selected control and plant signals. An appropriate and usual criterion is the quadratic cost function

$$J(k) = \sum_{i=0}^{H_p} (\hat{y}(k+i\,|\,k) - r(k+i))^T Q_i (\hat{y}(k+i\,|\,k) - r(k+i)) + \sum_{i=0}^{H_u-1} \Delta\hat{u}^T(k+i\,|\,k) R_i \Delta\hat{u}(k+i\,|\,k). \tag{4}$$

$\hat{y}(k+i\,|\,k)$ denotes the predicted output for the time $k+i$ calculated at the time instant k.
$\Delta\hat{u}$ denotes the difference control signal. Q and R are the weighting matrices for the input and the output variables of the process. H_p and H_u are prediction and control horizons. The use of difference control signals for the cost function leads to better control structuring and inhibits high-frequency signals. Unlike huge computing potential in other application fields of MP-Controllers the hardware capacity of engine control systems is limited in memory and computing power. For this reason we implemented a MPC-algorithm which is suitable for realtime application in engine control units. As minimum requirements the control algorithm should allow the application to multivariable systems with variable constraints in the control output. A main design requirement was the achievement of reasonable computing efforts.
For the MP-Controller we use a set of linear constant time-discrete state-space process models of the form

$$
\begin{aligned}
x(k+1) &= Ax(k) + Bu(k) \\
y(k) &= Cx(k)
\end{aligned}
\tag{5}
$$

(u-input signal, y-output signal, x-state of the system, A-system matrix, B-input- and C-output matrix).

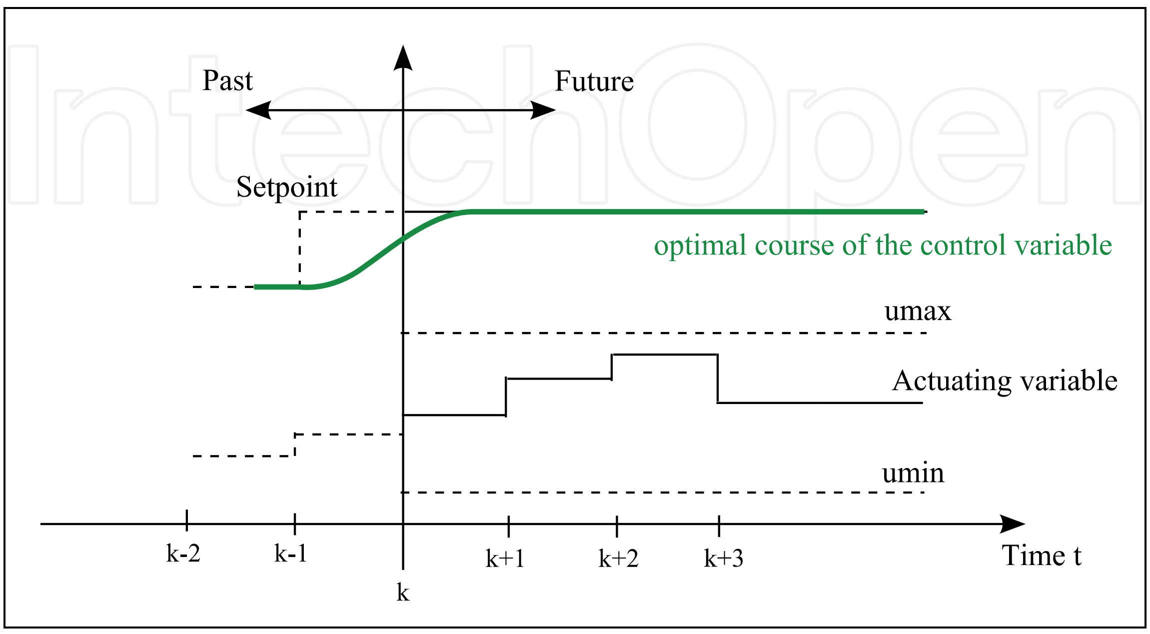


Fig. 6. Predictive Control: The basic idea

In case of control signal constraints we have to consider the conditions

$$
\begin{array}{ccccc}
u_{\min}(k) & \leq & u(k \mid k) & \leq & u_{\max}(k) \\
u_{\min}(k+1) & \leq & u(k+1 \mid k) & \leq & u_{\max}(k+1) \\
\vdots & & \vdots & & \vdots \\
u_{\max}(k+H_u-1) & \leq & u(k+H_u-1 \mid k) & \leq & u_{\max}(k+H_u-1)
\end{array}
\tag{6}
$$

From the model we obtain the prediction for the output of the system for a horizon of H_p steps at time k.

$$
\begin{bmatrix} \hat{y}(k+1 \mid k) \\ \vdots \\ \hat{y}(k+H_p \mid k) \end{bmatrix}
=
\begin{bmatrix} CA \\ \vdots \\ CA^{H_u} \\ CA^{H_u+1} \\ \vdots \\ CA^{H_p} \end{bmatrix} x(k)
+
\begin{bmatrix} CB \\ \vdots \\ C\sum_{i=0}^{H_u-1} A^i B \\ C\sum_{i=0}^{H_u} A^i B \\ \vdots \\ C\sum_{i=0}^{H_p-1} A^i B \end{bmatrix} u(k-1)
+
\begin{bmatrix}
CB & \cdots & 0 \\
C(AB+B) & \cdots & 0 \\
\vdots & \ddots & \vdots \\
C\sum_{i=0}^{H_u-1} A^i B & \cdots & CB \\
C\sum_{i=0}^{H_u} A^i B & \cdots & C(AB+B) \\
\vdots & \vdots & \vdots \\
C\sum_{i=0}^{H_p-1} A^i B & \cdots & C\sum_{i=0}^{H_p-H_u} A^i B
\end{bmatrix}
\begin{bmatrix} \Delta\hat{u}(k \mid k) \\ \Delta\hat{u}(k+1 \mid k) \\ \vdots \\ \Delta\hat{u}(k+H_u-1 \mid k) \end{bmatrix}
\tag{7}
$$

$$
= y(k) = \Psi x(k) + \Upsilon u(k-1) + \Theta \Delta U(k) \tag{8}
$$

With the setpoint signal $r(k+i \mid k)$, $i = 0 \ldots H_p$, the equation

$$\varepsilon(k) = \begin{bmatrix} r(k \mid k) \\ r(k+1 \mid k) \\ \vdots \\ r(k+H_p \mid k) \end{bmatrix} - \Psi x(k) - \Upsilon u(k-1) \tag{9}$$

and the weight matrices

$$Q = \begin{bmatrix} Q(0) & 0 & \cdots & 0 \\ 0 & Q(1) & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & Q(H_p) \end{bmatrix} \quad \textit{respectively} \quad R = \begin{bmatrix} R(0) & 0 & \cdots & 0 \\ 0 & R(1) & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & 0 & R(H_u-1) \end{bmatrix} \tag{10 and 11}$$

the cost functional (4) becomes to

$$J(k) = -\Delta U(k)^T G + \Delta U^T H \Delta U(k) \tag{12}$$

with

$$G = 2\Theta^T Q \varepsilon(k) \quad \textit{and} \quad H = \Theta^T Q \Theta + R \tag{13 and 14}$$

The constraint conditions from equation 6 we can formulate as a linear matrix inequation like

$$A \Delta U(k) \le U_{constraint} \tag{15}$$

with

$$A = \begin{bmatrix} 1 & 0 & 0 & \cdots & 0 \\ 1 & 1 & 0 & \cdots & 0 \\ \vdots & \vdots & \vdots & \ddots & \\ 1 & 1 & 1 & \cdots & 1 \\ -1 & 0 & 0 & \cdots & 0 \\ -1 & -1 & 0 & \cdots & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ -1 & -1 & -1 & \cdots & -1 \end{bmatrix} \quad \textit{and} \quad U_{constraint} = \begin{bmatrix} u_{\max}(k) - u(k-1) \\ u_{\max}(k+1) - u(k-1) \\ \vdots \\ u_{\max}(k+H_u-1) - u(k-1) \\ -u_{\min}(k) + u(k-1) \\ -u_{\min}(k+1) + u(k-1) \\ \vdots \\ -u_{\min}(k+H_u-1) + u(k-1) \end{bmatrix} \tag{16 and 17}$$

The main problem of the model predictive controller is to minimize the cost function 12 subject to the constraints expressed in inequation 15. To solve this standard convex optimization problem (which is summarized in figure 7) efficient numerical procedures (quadratic programming (QP)) are available. For the realtime application we developed a computing time optimized QP program which uses an active set algorithm (Dünow et al. 2005, Lekhadia et al. 2004, Lekhadia 2004a). We implemented this optimization algorithm for Mathworks xPC-Target and for an Infineon TriCore microcontroller board. This microcontroller is actually also used in ECU-Systems.

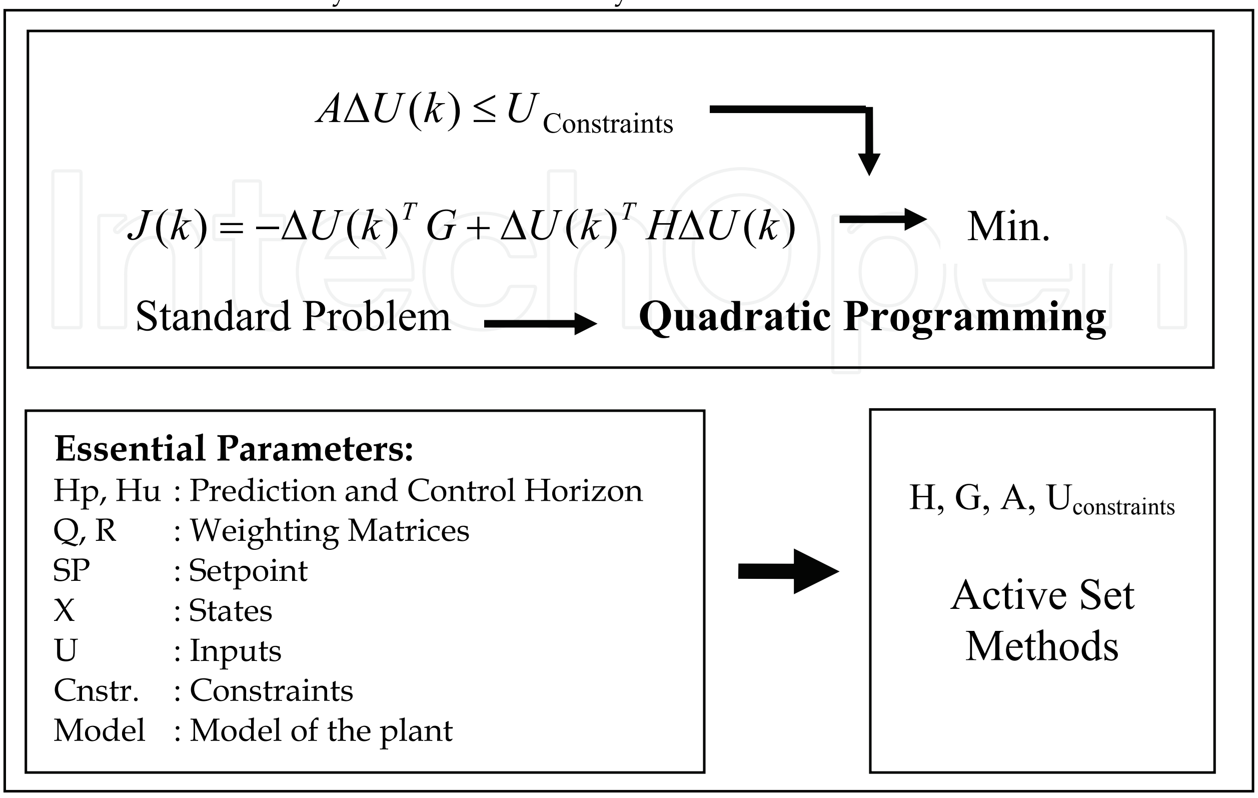


Fig. 7. MPC-algorithm (summarized)

3.2 Implementation

The model predictive controller was implemented by means of the packages LAPACK and BLAS for C-language (Dünow et al. 2005). These Libraries are efficient concerning the computing effort and widely used in research and industrial applications. The MPC-algorithm was implemented by a C-S-function for Matlab / Simulink to use it on a xPC Target system. Because of the limited capabilities of the ECU the implemented control algorithm should need less computing time and memory space. For the unconstraint case we can find a constant MPC solution. Here the bounds will be ignored. The optimal $\Delta U(k)$ for the unconstrained case we find by

$$(dJ / d\Delta U) = \nabla_{\Delta U(k)} J = -G + 2H\Delta U(k) = 0 \tag{18}$$

From equation 18 follows the optimal set of control variables:

$$\Delta U(k)_{opt} = \frac{1}{2} H^{-1} G \tag{19}$$

Condition 16 minimizes the cost function because of the fact that

$$\frac{\partial_2 J}{\partial \Delta U(k)^2} = 2H = 2 \cdot \left(\Theta^T Q \Theta + R\right) \tag{20}$$

is positive defined if $Q \geq 0$ and $R > 0$. In figure 8 the unconstrained case is summarized in a block diagram. The implementation of a controller which based on equation 19 is outlined in figure 9. Here the controller consists of a linear state space model, three constant gains and a simple wind up clipping structure (Dünow et al. 2005).

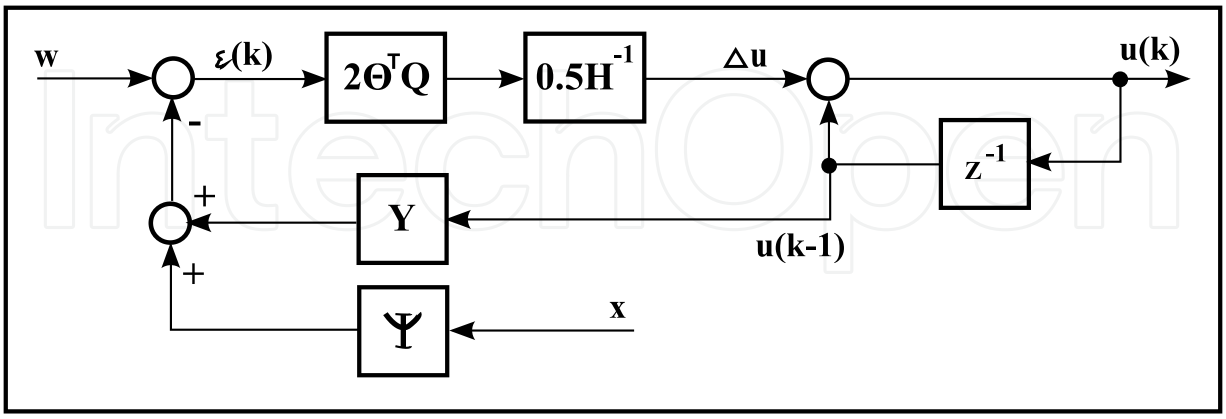


Fig. 8. Block diagram of the unconstrained case

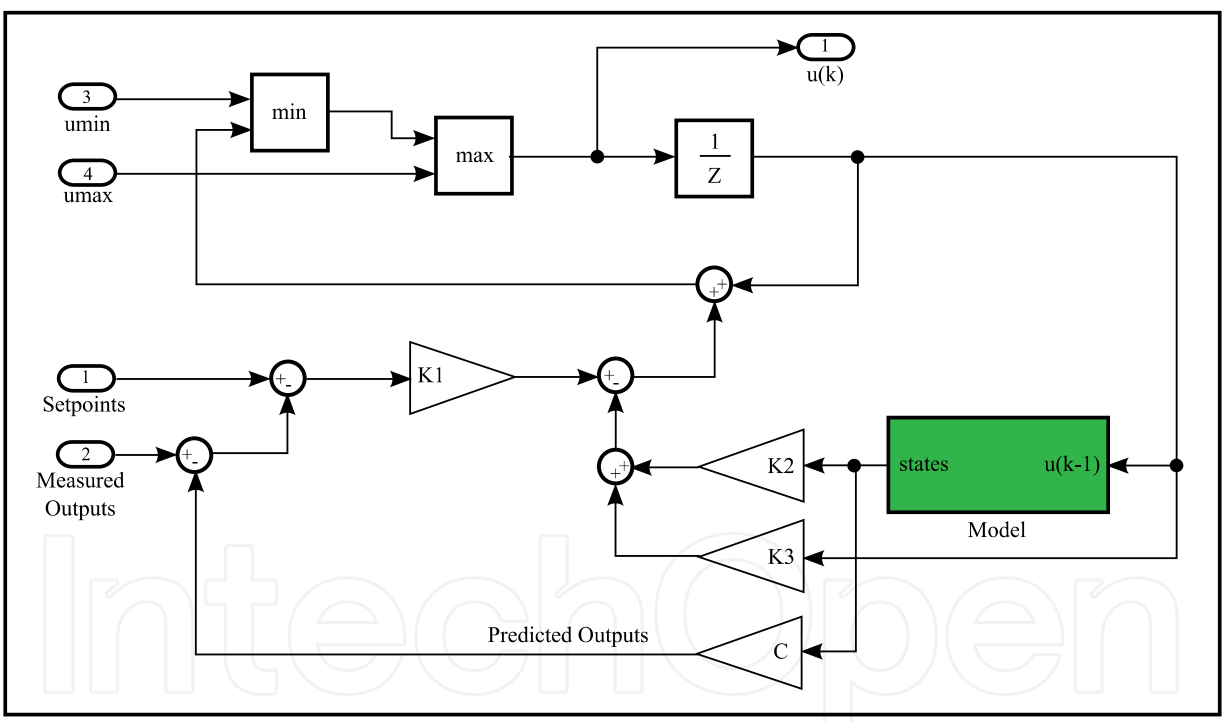


Fig. 9. Implementation of the constant Controller in Matlab/Simulink (Dünow et al. 2005)

For the constrained case we have to minimize the cost function (see equation 12) regarding to the controller constraints output marked in equation 15. With the help of active set methods we solved this standard quadratic programming optimization problem. In figure 10 the flow chart, which based on Fletscher 1997, Gill et al. 1991 and Maciejowski 2002, of the optimization algorithm is shown. The active set method involves two phases. At first a feasible point is calculated. The second phase involves the generation of an iterative sequence of feasible points that converge to the solution. The λ-check below in figure 10 is a test for Karush-Kuhn-Tucker condition for a global optimum (Fletscher 1987).

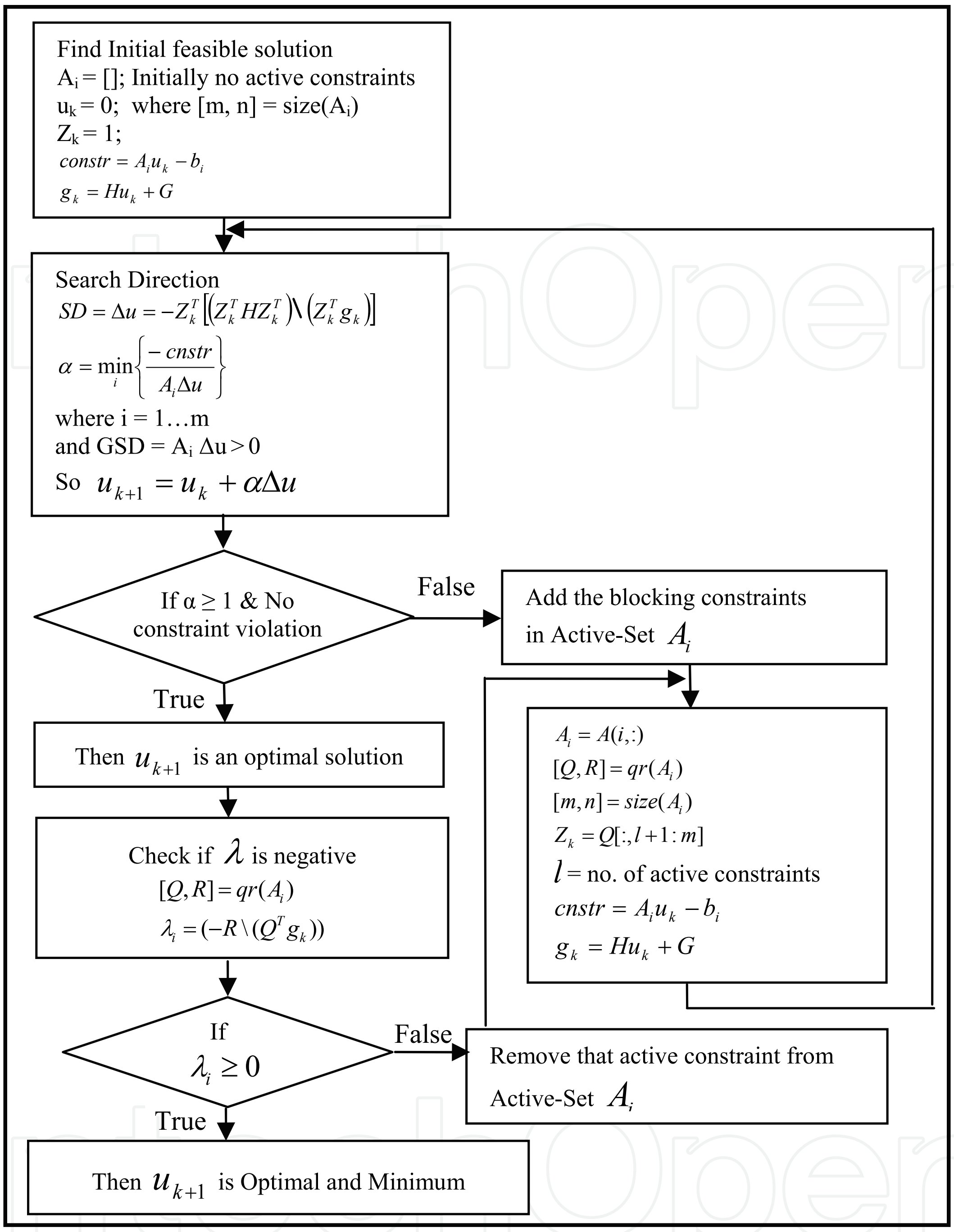


Fig. 10. Flow chart of the implemented active set algorithm (Dünow et al. 2005 and Lekhadia 2004a)

4. Practical applications and results

In this section we will investigate practical aspects of the described approach. As a test environment we used a complex nonlinear model of a four-cylinder direct injection engine. This model includes the physical elements of the engine as well as the necessary ECU-functions for torque control. The model is usable at Matlab/Simulink. The model is briefly depicted in figure 11.

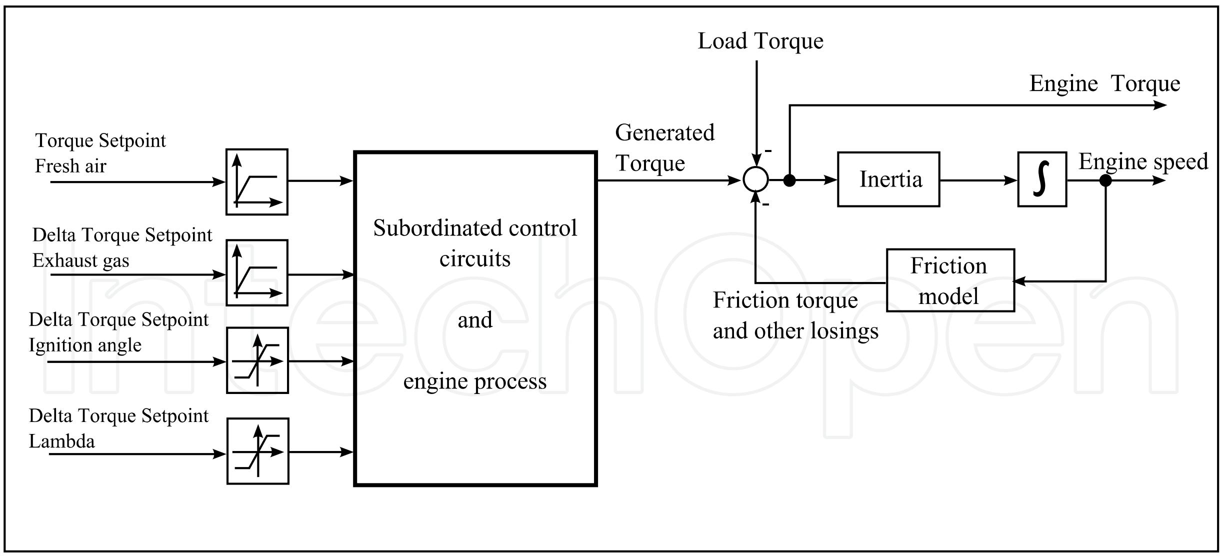
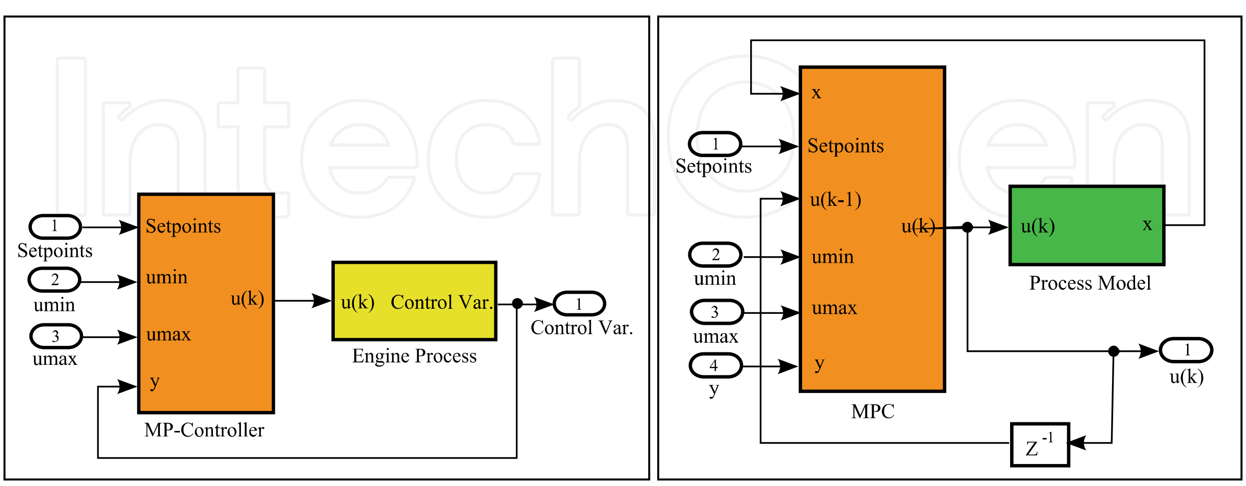


Fig. 11. Engine model (simplified)

The model was completed by a part for exhaust gas emission and one part for fuel consumption.

Because of the different dynamics of the control pathways the sampling time is relatively small. This results in a corresponding large prediction horizon of the MPC. Regarding the control horizon we btained the best compromise between performance and computational effort for $H_u = 5$.

Figure 12 foreshadows the structure of the simulation system. We used a state space method for the modelling and solution of the control problem. The MPC-Block in the left of the simulink-model includes the control solution explained in section 3. The process model can be seen as an alternative to a real engine. The background of the model was illustrated in section 1 and 2. The model in the controller block is conform to equation 2.



a) MPC and engine process   b) Optimization and internal model

Fig. 12. Simulation environment

In figures 13a to 13d an application case of the torque controller is shown. The main objective of the control process in this example is to follow the torque setpoint as accurately as possible. The upper figure (13a) shows the setpoint and the real values of the torque. Below (13a) the four control variables are pictured. At about 1.5 seconds we have a setpoint step. At 2.5 seconds a load step occurs. This load step is predictable (turn-on procedure of the air conditioner). In the example the setpoints for the delta torques of the ignition and exhaust gas path are zero. The setpoint for the lambda torque path is here about -8 Nm. This corresponds to λ=1. The torque value of 8 Nm can be seen as a "reserve" and the MPC can boost the total engine torque due to the fuel path immediately by this value. The upper bound for the delta torque via the fuel path is zero in the example.

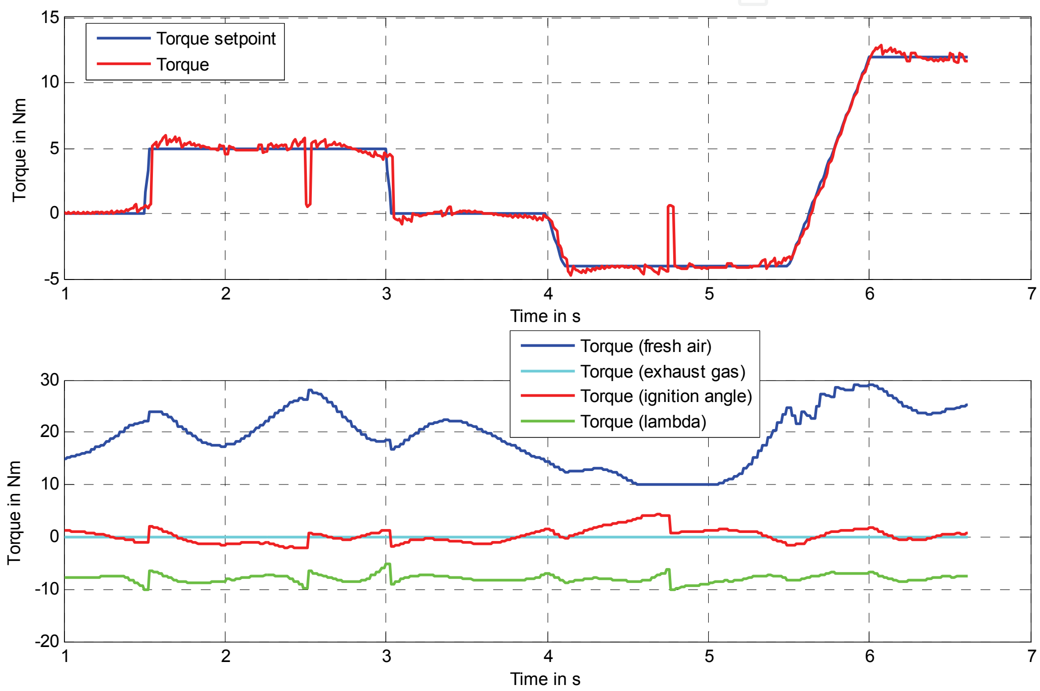


Fig. 13a. Results – max torque priority

The upper and lower bounds for the exhaust gas path in the example are equal (zero). That means the MPC must not use this variable for torque control until the constraints are changing. The value of the controller output always is equal to the setpoint (as required). Because of the fact that the controller outputs for the delta torques at the same time are variables to be controlled these outputs are used only temporarily by the MPC (also visible in figure 13b). The lower bounds for the fuel and the ignition path in the example are -15 Nm. The setpoint step for the total engine torque at 1.5 seconds was predictable. It is visible that the MPC automatically prepared this action. The maximum torque (represented by the air path) increases. The available delta variables (fuel and ignition path) are decreasing. So the total torque doesn't change up to 1.5 seconds. In this way the MPC builds up a torque reserve which can be used (temporarily) to advance the setpoint adjustment.

The compensation of a predictable load step (in the example about at 2.5 seconds) occurs in the same way. Figure 13b and 13d give a zoomed view to the load compensation. The controller is able to readjust the torque very fast due to the torque reserve.

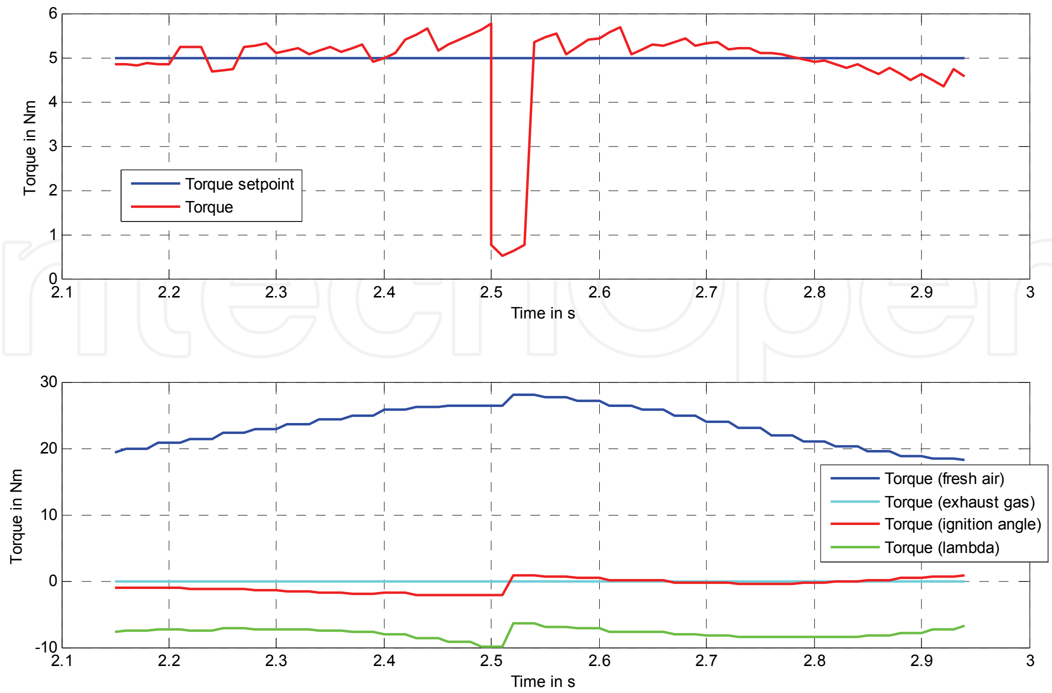


Fig. 13b. Results – max torque priority

In the example above the control request was focused mainly to minimize the engine torque deviation from the desired setpoint. This was achieved by appropriate weighting variables within the model predictive control algorithm.

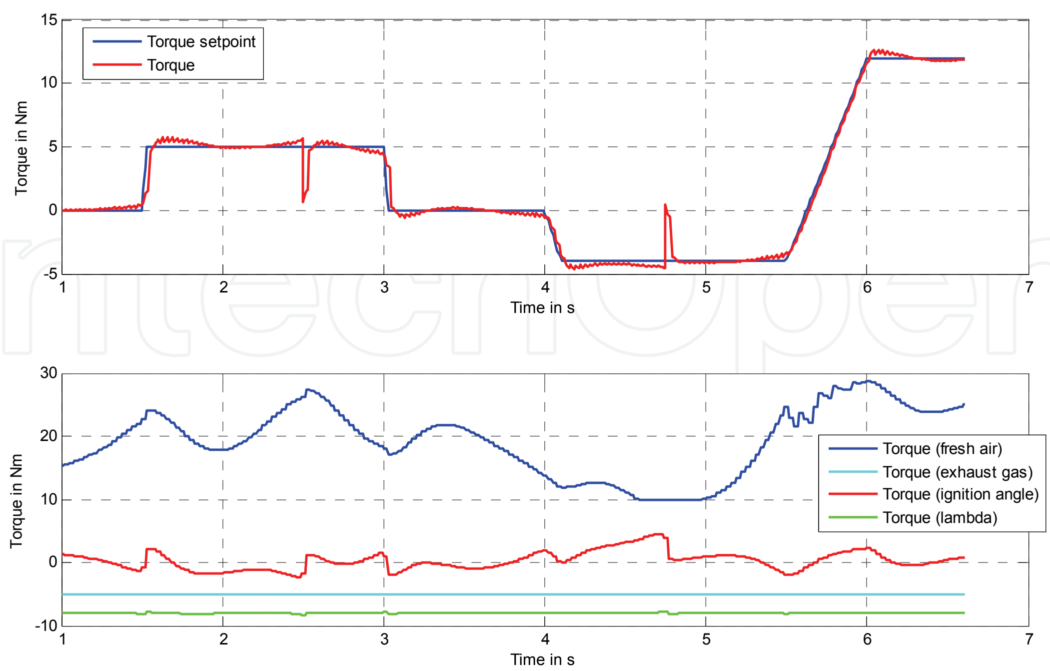


Fig. 13c. Results – min torque priority

By changing the weighting factors it is also possible to shift the focus more towards other attributes. For instance one can achieve a reduction of the waste gas emission by a setpoint for corresponding λ=1 and increasing the weighting factors of the lambda path.
The results are shown in figure 13c for the same setpoint and load steps as in the example above. It is visible that the torque adjustment is (compared to figure 13a) slower and the controller output for the fuel path is used only marginally.

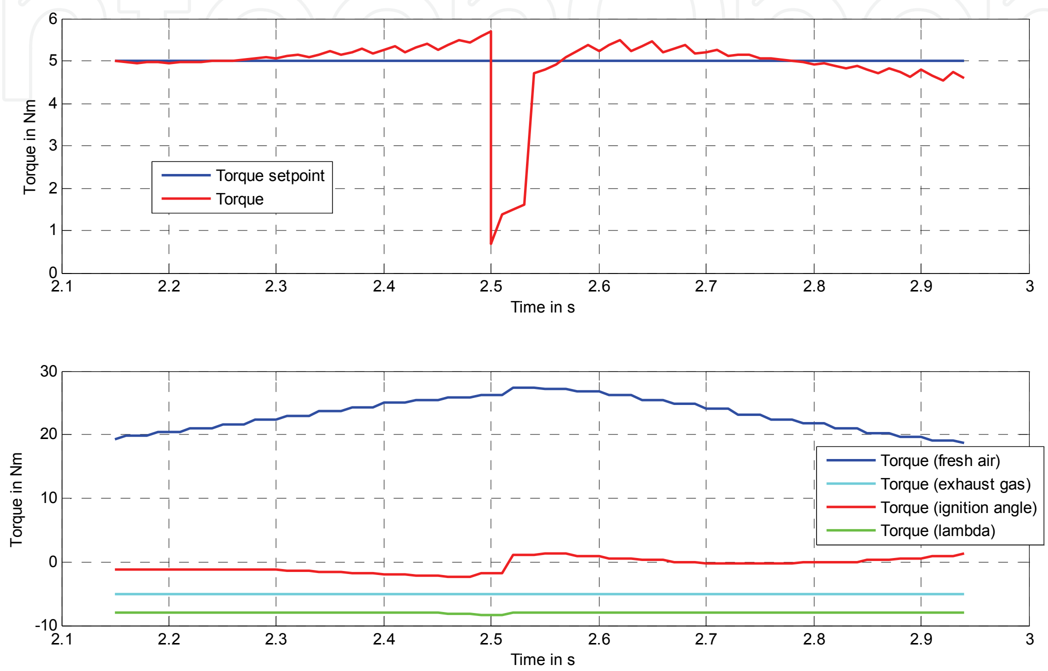


Fig. 13d. Results – min torque priority

As mentioned in section 2 the described control approach allows to switch smoothly between torque, speed or acceleration control. This can be simply achieved by the model extension (described in section 2, equation 3) and appropriate weighting parameters for the MPC. Only by shifting the weighting factors one can switch from speed control to torque control and back.
The plots in figure 14 demonstrate how this control mode works. In this example we used only the fresh air and the ignition path for torque control. Up to 3 seconds the controller should work as a speed controller. From then it should work as a torque controller. At approximately 6 seconds the weighting factors where switched back to speed control mode values. Equation 21 represents the linear model which is used internally by the MPC for this example.

$$\begin{pmatrix} \text{Engine Torque} \\ \text{Engine Speed} \\ \Delta Torque_{\text{Ignition Angle}} \end{pmatrix} = \begin{pmatrix} G_{11}(s) & G_{12}(s) \\ G_{21}(s) & G_{22}(s) \\ 0 & G_{32}(s) \end{pmatrix} \cdot \begin{pmatrix} Torque_{\text{Fresh Air Setpoint}} \\ \Delta Torque_{\text{Ignition Angle Setpoint}} \end{pmatrix} \tag{21}$$

Switching between speed and torque control can simply be realized by changing some weighting factors. The behaviour of the MPC changes automatically. This is a very convenient way to change for instance to the idle speed control mode and back to torque control.
In figure 14 first the speed control is active. The speed follows the desired characteristic whereas the torque is different from the corresponding setpoint. After changing the weighting factors the torque will be adjusted to the setpoint and the engine speed is only marginally considered by the controller.

Time period in s	**Active Control**	**Torque weighting factor**	**Speed weighting factor**
0 … 2.5	Speed control	0	1
2.5 …6	Torque control	1	0
6 … 8.5	Speed control	0	1

Table 1. Enabled control

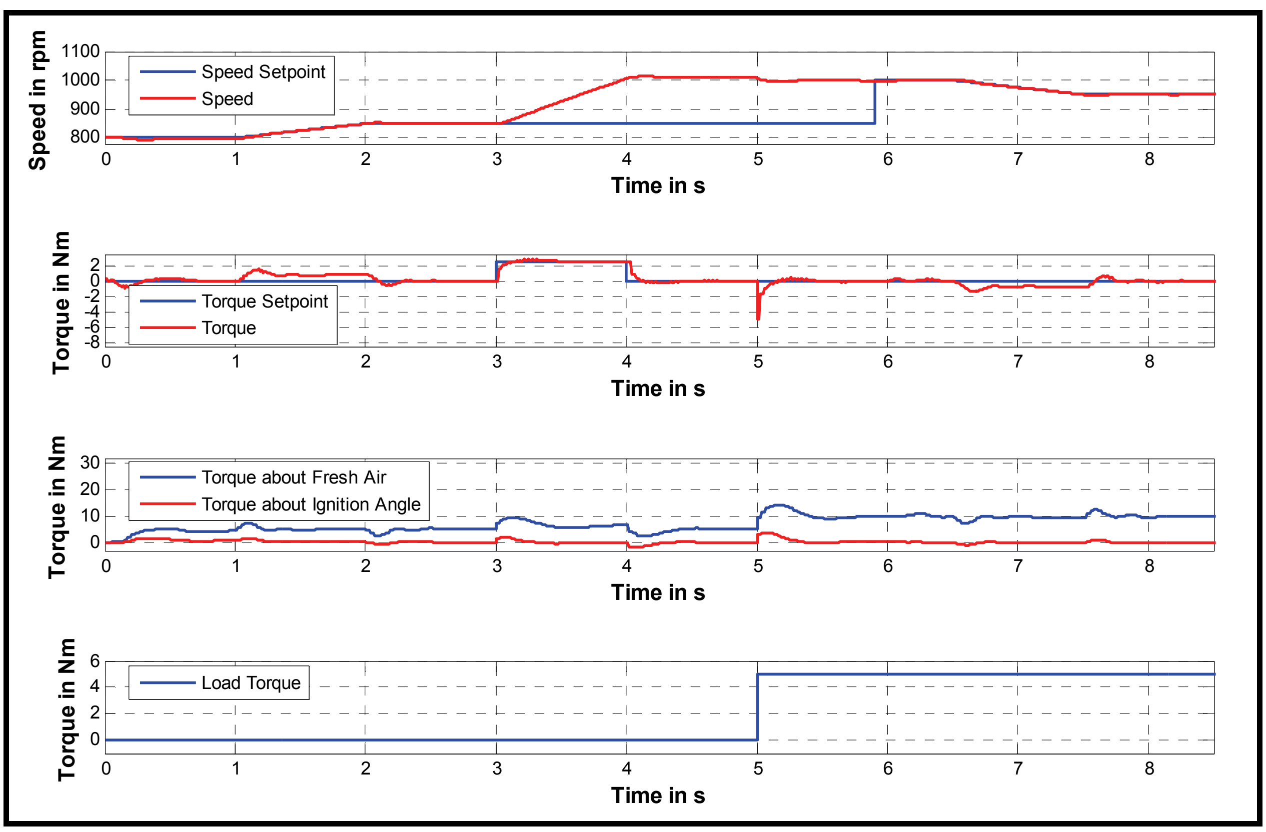


Fig. 14. Speed and Torque Control

At approximately 6 seconds the engine speed control mode was activated again. The speed should decrease here from the current value smoothly along a given trajectory. It is visible in figure 14 that the controller solves the differing control tasks properly. In the third plot (from the top) one can see also that the ignition path control variable is used transient by the controller. Stationary the setpoint is adjusted.

5. Conclusion

In the chapter we discussed a control approach for torque control of gasoline engines. Because of several actuating variables and control requirements the process to be controlled is multivariable. The actuating variables are usually bounded and the effects on the engine torque are nonlinear. Hence direct use of the actuator variables for torque control generally produces plenty of problems.

The two layer approach described in the chapter allows the application of standard control methods. The main idea of the control structure is to compensate or alleviate the nonlinearity behaviour by subordinate control circuits. All the physical actuating variables are substituted by setpoints of the subordinate systems. The torque controller so can be designed on base of linear models. Additionally only bounds of some control variables have to be considered.

An appropriate standard control concept for the superordinate torque controller is the model predictive control principle. For the implementation we used a state space approach. The optimization problem is solved by the active set algorithm. For lower computation effort a solution with constant parameters was introduced. But this solution doesn't consider constraints and loss performance can occur.

The three examples in section 4 show the capability of the control concept. The extension of the control structure is possible simply by completing the model and appropriate weighting parameters. In this way the controller should be able to handle more actuating variables or other requirements.

The control quality depends on the quality of the model. Although the subordinate control circuits contribute to the linearization of the process behaviour the dynamic parameters may be dependent on engine speed or load. For that case a set of linear models could be useful.

The control approach described in this chapter demonstrates that modern control approaches have considerable potential to improve the performance of embedded control systems. In addition to better performance also the variability of the systems and the ability to handle different control requirements could improve.

6. References

Auckenthaler, T.S. (2005). Modelling and Control of Three-Way Catalytic Converters, Institute of Technology, Zürich, Dissertation

Back, M. (2005). Prädiktive Antriebsregelung zum energieoptimalen Betrieb von Hybridfahrzeugen, Institut für Regeluns- und Steuerungssysteme Universität Karlsruhe, Dissertation

Bauer, H. (2003): Ottomotor-Management: Systeme und Komponenten, Friedr. Vieweg & Sohn Verlagsgesellschaft mbH

Brand, D. (2005). Control-Oriented Modeling of NO Emissions of SI Engines, Institute of Technology, Zürich, Dissertation

Basshuysen, R. van; Schäfer, F. (2002): Handbuch Verbrennungsmotor – Grundlagen, Komponenten, Systeme und Perspektiven, Vieweg und Sohn Verlagsgesellschaft mbH, Braunschweig/Wiesbaden

Dittmar, R.; Pfeiffer, B. (2004). Modellbasierte prädiktive Regelung, Oldenbourg Verlag München, Wien

Dünow, H.-P. (2004). Anwendung Modellprädiktiver Regelungen, Forschungsbericht, Hochschule Wismar, Fachhochschule für Technik, Wirtschaft und Gestaltung – Fachbereich Elektrotechnik und Informatik

Dünow, H.-P.; Lekhadia, K.N.; Köller, M.; Jeinsch, T. (2005). Model based predictive control of spark ignition engine, *Proceedings of MMAR*

Fletscher, R.R (1987). Practical Methods of Optimization, Wiley, 2nd edition

Fritzsche, C.; Dünow, H.-P.; Lampe, B.; Schultalbers, M. (2007). Torque Coordination of Spark Ignition Engines based on Predictive Control, *Proceedings of MMAR*, Szczecin, Poland

Gill, P.E.; Murray, W.; Wright, M.H. (1991). Numerical Linear Algebra and Optimization, Volume 1, Addison Wesley

Grimble, M.J. (2001). Industrial Control Systems Design, John Willey & Sons, LDT Chichester, New York, Weinheim, Brisbane, Singapore and Toronto

Grohe, Heinz (1990). Otto- und Dieselmotoren, Vogel Fachbuchverlag, Auflage 9, Würzburg, Germany

Guzella, L. and Onder, C.H. (2004). Introduction to Modeling and Control of Internal Combustion Engine Systems, Springer Verlag Berlin, Heidelberg, Germany

Johansen, T.A. (1994). Operating Regime based Process Modelling and Identification, PhD Thesis, Department of Engineering Cybernetics – The Norwegian Institute of Technology – University of Trondheim

Johansen, T.A.; Murray-Smith, R. (1997). Multiple Model Approaches to Modelling and Control, Taylor & Francis London

Kristoffersson, I. (2006). Model Predictive Control of a Turbocharged Engine, School of Electrical Engineering and Information Technology, Stockholm

Lekhadia, K.N.; Dünow, H.-P.; Jeinsch, T. (2004). Active Set Method Approach for Real Time Model Predictive Engine Control, In: Modellierung, Regelung und Diagnose von Verbrennungsmotorprozessen, 2. ASIM Workshop Wismar

Lekhadia, K.N. (2004a). Development of OP Algorithms for Real-Time Predictive Control Systems, Master Thesis, FH Darmstadt

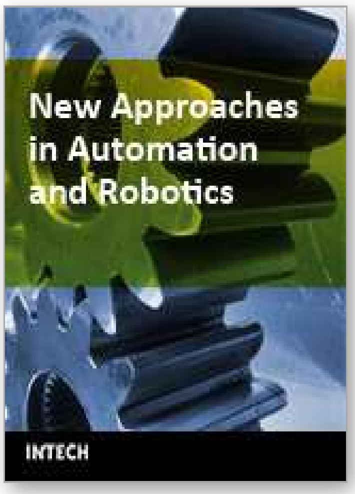
Maciejowski, J.M. (2002). Predictive Control with constraints, Prentice Hall

Pulkrabek, Willard (2004). Engineering Fundamentals of the Internal Combustion Engine, Prentice Hall

Rückert, J.; Richert, F.; Schlosser, A.; Abel, D.; Herrmann, O.; Pfeifer, A.; Pischinger, S. (2003). Konzepte zur Regelung von Ladedruck und AGR-Rate beim Nutzfahrzeug-Dieselmotor, GMA-Kongress – VDI-Berichte 1756, VDI-Verlag Düsseldorf

Salgado, M.E.; Godwin, G.C.; Greabe, S.F. (2001). Control System Design, Prentice Hall London

Urlaub, Alfred (1995). Verbrennungsmotoren – Grundlagen, Verfahrenstheorie, Konstruktion, Springer Verlag Berlin, Heidelberg, Germany



New Approaches in Automation and Robotics
Edited by Harald Aschemann

ISBN 978-3-902613-26-4
Hard cover, 392 pages
Publisher I-Tech Education and Publishing
Published online 01, May, 2008
Published in print edition May, 2008

The book New Approaches in Automation and Robotics offers in 22 chapters a collection of recent developments in automation, robotics as well as control theory. It is dedicated to researchers in science and industry, students, and practicing engineers, who wish to update and enhance their knowledge on modern methods and innovative applications. The authors and editor of this book wish to motivate people, especially under-graduate students, to get involved with the interesting field of robotics and mechatronics. We hope that the ideas and concepts presented in this book are useful for your own work and could contribute to problem solving in similar applications as well. It is clear, however, that the wide area of automation and robotics can only be highlighted at several spots but not completely covered by a single book.

InTech Europe
University Campus STeP Ri
Slavka Krautzeka 83/A
51000 Rijeka, Croatia
Phone: +385 (51) 770 447
Fax: +385 (51) 686 166
www.intechopen.com

InTech China
Unit 405, Office Block, Hotel Equatorial Shanghai
No.65, Yan An Road (West), Shanghai, 200040, China
中国上海市延安西路65号上海国际贵都大饭店办公楼405单元
Phone: +86-21-62489820
Fax: +86-21-62489821