

Inversion-Based Intake Manifold Pressure Control System for Modern Diesel Engines

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ABSTRACT

An improved model-based two-degree of freedom control system for the intake manifold pressure in passenger car diesel engines is described in this paper. The aim of this control system is to track the air charge setpoint rapidly and precisely. To achieve this, an inverse model of the intake manifold dynamics is included in the feedforward control path. The system parameters which are necessary to calculate the inverse model are setpoints from other control loops in the gas system. These generated setpoint values allow for decoupling of the individual control loops in the gas system as far as possible. The parallel linear feedback controller is designed to further improve the accuracy of the control system. The calculated feedforward control signal and the feedback control variable additively generate the effective opened area of the intake throttle valve. Due to its model-based structure, the improved control system proposed will require less calibration effort than current production-code controllers and that the calibration can be partially carried out using simulation. This is also the main motivation for the development of this controller. The proposed control algorithm can be easily implemented on the standard electronic control unit (ECU) of passenger car diesel engines. For testing and validation, the production-code controller function of the ECU has been replaced by the new control approach using a bypass with a rapid control prototyping. Experimental results are provided and show that the new control system achieves a performance that is similar to the production-code controller.

CITATION: Reiß, J., Bohn, C., Märzke, F., Meinecke, R. et al., "Inversion-Based Intake Manifold Pressure Control System for Modern Diesel Engines," *SAE Int. J. Engines* 7(3):2014, doi:10.4271/2014-01-1709.

INTRODUCTION

Due to rapidly changing torque requirements based on highly dynamic driving situations, a fast and precise control of the air charge in diesel engines is becoming increasingly important. This is especially important in view of the exhaust gas components DPF (Diesel Particulate Filter) and NSC (NOx Storage Catalyst). These require (at varying time intervals and under specific chemical conditions) the removal of the stored exhaust gas components. The engagement of the throttle valve allows for this by setting the air charge. The air charge is essentially associated with the amount of air stored in the intake manifold, which is a function of the intake manifold pressure and is governed by the equation of the state of ideal gases. Therefore, this intake manifold pressure is the setpoint for the charge control system. This goes back to the fact that the number of sensors is limited in the gas system. One of the sensors is usually the intake manifold pressure sensor (see [section 2](#)). This sensor directly measures the controlled variable.

Most control concepts used for this purpose in practice consist of PID controllers where the control parameters are calculated through operating-point-dependent look-up tables. This leads to a high calibration effort and requires extensive experience on the part of the calibration engineer. In [1], a control approach was suggested to control the intake manifold pressure without the need for extensive look-up tables. The proposed control concept also considers the nonlinearities of the controlled system. However, due to the lack of feedback, the control system cannot cope with disturbances and model inaccuracies. The introduction of a parallel linear PI controller, as proposed in [2, 3, 4], overcomes this drawback.

It is also possible to control the intake manifold pressure via system prediction of the intake manifold and the throttle dynamics. This approach compensates for the two delays of the response model [5]. However, the algorithm is designed for one operating point only, so it seems hardly applicable in practice.

Another concept to control the gas system of diesel engines is model predictive control (MPC) [6, 7, 8]. In this approach two or more actuators are considered together. This has the advantage that a decoupling of the control loops is not necessary. However, it is difficult to integrate additional functions. Also it is not easy to take individual exception situations into account in practice, because of the high complexity involved. Thus, the usage in production vehicles is limited. This is also one of the reasons why usually only simulation results are shown as proof of the functionality of the MPC algorithm. Therefore, the control system described in this paper is based on the more classical two degrees of freedom concepts of [3] and [4].

The remainder of this paper is organized as follows. In [section 2](#), a description of the intake manifold and the throttle valve is given. In [section 3](#), the two-degree of freedom control system is introduced. The measurement setup and measurement results are presented in [section 4](#). Finally, the paper concludes with a summary and an outlook on future work ([section 5](#)).

SYSTEM MODEL DESCRIPTION

[Figure 1](#) shows a schematic representation of the gas system of modern diesel engines. The high pressure EGR (Exhaust Gas Recirculation) valve remains closed in situations where the throttle controls the air charge. This is necessary in order not to interfere with the chemical reaction in the exhaust gas components involving the recirculated exhaust gas.

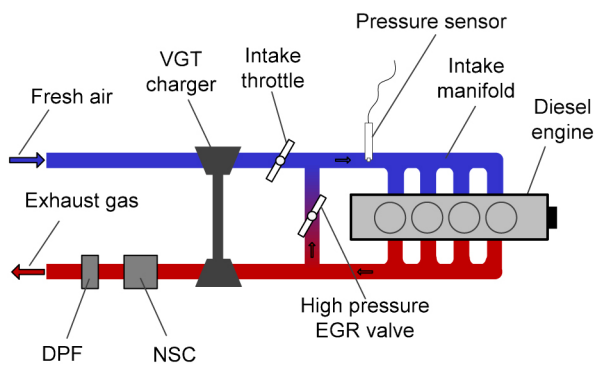


Figure 1. Schematic overview of the gas system of a modern diesel engine.

A simple physical model of the intake manifold is based on the state of ideal gases

$$p_{in} \cdot V_{in} = m_{in} \cdot R \cdot \vartheta_{in}. \quad (1)$$

The time derivative of this equation

$$\frac{dm_{in}}{dt} = \frac{d}{dt} \left(\frac{p_{in} \cdot V_{in}}{R \cdot \vartheta_{in}} \right) \quad (2)$$

leads (under the assumption that the variation of the intake manifold temperature ϑ_{in} can be neglected) to the approximation

$$\frac{dm_{in}}{dt} = \frac{V_{in}}{R \cdot \vartheta_{in}} \cdot \frac{dp_{in}}{dt}. \quad (3)$$

Here, R is the specific gas constant and $\dot{m}_{in}$, p_{in} and V_{in} denote the mass flow, the pressure and the volume of the intake manifold, respectively. The net mass flow

$$\frac{dm_{in}}{dt} = \dot{m}_{in} = \dot{m}_{thr} - \dot{m}_{cyl} \quad (4)$$

is the difference between the throttle mass flow $\dot{m}_{thr}$ and the cylinder mass flow $\dot{m}_{cyl}$. The throttle mass flow is calculated from the temperature upstream of the throttle ϑ_{us} , the effective open area of the throttle A_{eff} , the pressure upstream of the throttle p_b (the boost pressure) and the flow rate function ψ by the throttle equation of Saint Venant

$$\dot{m}_{thr} = A_{eff} \cdot \sqrt{\frac{2}{R \cdot \vartheta_{us}}} \cdot \psi \cdot p_b. \quad (5)$$

The flow rate function

$$\psi = \begin{cases} \sqrt{\frac{\kappa}{\kappa-1} \cdot \left[\left(\frac{p_{in}}{p_b} \right)^{\frac{2}{\kappa}} - \left(\frac{p_{in}}{p_b} \right)^{\frac{\kappa+1}{\kappa}} \right]}, & \frac{p_{in}}{p_b} > \frac{2}{\kappa+1} \\ \frac{2}{\kappa+1} \cdot \sqrt{\frac{\kappa}{\kappa+1}}, & \frac{p_{in}}{p_b} \leq \frac{2}{\kappa+1} \end{cases} \quad (6)$$

is dependent on p_b , p_{in} and the isentropic exponent κ .

The mass flow in the cylinder is given as

$$\dot{m}_{cyl} = \frac{\lambda_{ve} \cdot n \cdot V_d}{120} \cdot \frac{p_{in}}{R \cdot \vartheta_{in}}. \quad (7)$$

In this equation, λ_{ve} is the volumetric efficiency, n is the engine speed and V_d is the displaced volume.

The [equations \(3\)](#), [\(4\)](#), [\(5\)](#) and [\(7\)](#) lead to the differential equation of the intake manifold

$$\frac{dp_{in}}{dt} = \frac{R \cdot \vartheta_{in}}{V_{in}} \cdot \left(A_{eff} \cdot \sqrt{\frac{2}{R \cdot \vartheta_{us}}} \cdot \psi \cdot p_b - \frac{\lambda_{ve} \cdot n \cdot V_d}{120} \cdot \frac{p_{in}}{R \cdot \vartheta_{in}} \right) \quad (8)$$

The dynamic behavior of the throttle valve is approximated as a second order system [9]. The values of the time constants were experimentally obtained by least square estimation (minimizing the sum of the squared errors between real system output and simulation results). For the engine investigated in this paper, the resulting time constants were obtained as $T_1 = T_2 = 22$ msec. The measured position and the model output are shown in figure 2 for a step change in the setpoint.

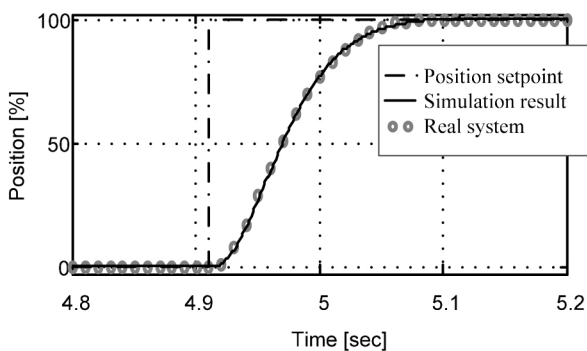


Figure 2. Measured and modeled throttle valve position.

The complete model consisting of the physical model of the intake manifold and the approximate second-order model of the throttle valve behavior was verified by simulation. The simulation parameters A_{eff} , n , p_b , ϑ_{us} and ϑ_{in} are based on a measurement performed with the same engine considered in section 4. Figure 3 shows the measured intake manifold pressure $p_{in,meas}$ and the modeled intake manifold pressure $p_{in,sim}$. The two curves correspond well and show a consistent behavior. This confirms the validity of the model.

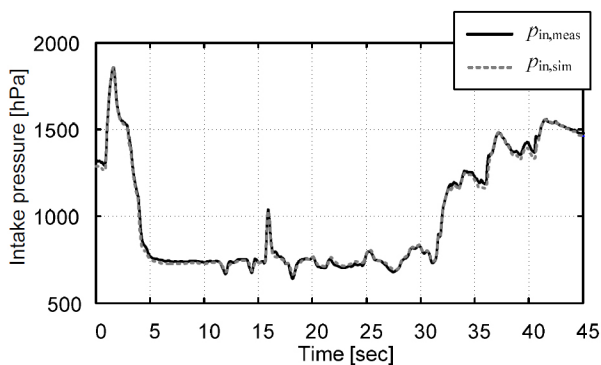


Figure 3. Simulated and measured system performance of the intake manifold pressure.

TWO-DEGREE OF FREEDOM CONTROL SYSTEM

To control the intake manifold pressure, a two-degree of freedom structure with the transfer function G_M/G_S in the feedforward path and the controller transfer function G_C in the feedback loop is used, see figure 4. The controlled system can be divided up into a linear part G_S and a nonlinear part, see next section. In this case the nonlinear operator H includes the system gain with the nonlinear flow rate function.

The feedforward control improves the dynamics of the system without affecting the stability of the control loop. The control structure has been extended by a transfer function G_M that describes the desired tracking behavior. The desired model is needed for the realization of the control loop. Furthermore, it lowers the intervention of the parallel linear PID controller. This PID controller is intended to ensure correct control of the intake manifold pressure in the event of model inaccuracies and disturbances.

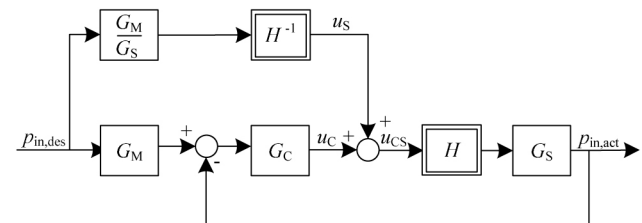


Figure 4. Two-degree of freedom control system.

The calculated feedforward control signal u_s and the feedback control variable u_c additively generate the setpoint for the effective open area of the throttle u_{CS} . However, the effective area is not equal to the geometrical area of the throttle valve. To enable conversions for this, a mass flow dependent look-up table must be calibrated. The desired throttle area is limited to its physically possible values and then becomes the setpoint value for the underlying throttle position controller of the throttle actuator. This conversion and limitation is shown in figure 5.

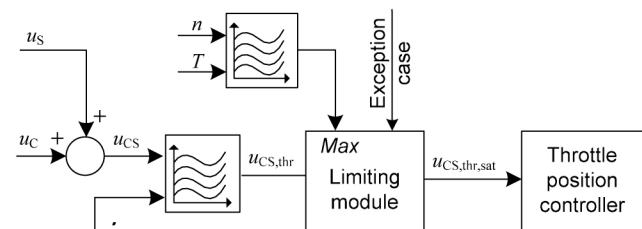


Figure 5. Advanced controller structure with conversion of controller output and operating point dependent limitation.

Feedforward Control

The aim of the inversion-based feedforward control is to map the controlled system inversely. The central element here is the throttle equation, Eq. (5), transposed to the area, that is