

Parameter Estimation for Physical Based Air Path Models of Turbocharged Diesel Engines – An Experience Based Guidance

Matthias Mrosek, Sebastian Zahn, Rolf Isermann

Institute of Automatic Control, TU Darmstadt, Germany

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ABSTRACT

Physical based air path models lead to a substructuring of the highly complex engine systems into several interacting submodels of low order. They offer detailed process information, support advanced control system design and allow to significantly reduce the calibration effort. Hence, physical approaches are predestinated to cope with the rise in system complexity and with the increasingly challenging demands concerning air system performance. Whereas the basic model equations are known a general methodology to obtain the model parameters is lacking. The purpose of this paper is to shed light on the identification procedure and to offer the automotive engineer helpful advice to gain well calibrated simulation models. Analysing the air path equations the determining factors on the parameter quality are investigated. Based on the results sensible modifications of the test bed setup and the measurement strategy are presented. In addition the need for careful data processing is discussed. The comparison of simulation results and measurement data from the engine test bed proves the potential of the developed methodology.

INTRODUCTION

Developing new combustion engines the car manufacturers are confronted with several concurrent goals: On the one hand they have to fulfil the future regulations concerning fuel consumption and pollutant emissions. On the other hand they are forced to respect the customer demands regarding driveability, performance and comfort. Further framework conditions are an increasing technology complexity, extended cost pressure and a limited time to market. A solution to this problem is the intensified usage of development methodologies. Especially model- and simulation-based methods for engine controller de-

sign and calibration offer the chance to seriously speed-up the development process and to shorten expensive test-bed times. The fuel consumption and the raw emissions of an engine depend on the gas composition within the combustion chambers. The gas composition, in turn, is strongly influenced by the dynamics of the air and EGR-path. For that reason the closed-loop control of the actuators in the air and EGR-path plays a decisive role. While former control-oriented models of the air path were black-box models (look-up tables, neural networks, etc.) now an emerging trend towards a physical modelling is observed. This even applies to ECU-internal models [1, 2]. In comparison to black-box models physical approaches give deeper insight into the process behavior. They need less measurement and parameterization effort and offer better extrapolation capabilities. Furthermore, due to their hierarchical and modular structure, they allow for an easy modification and extension of the engine configuration. Black-box models directly replicate the process output in dependance of a high number of inputs. In contrary, physical models are made up of several elements of low order whose interaction gives the process behaviour. Thus, the model accuracy results from the accuracy of the sub models [3]. This emphasises the importance of the model calibration.

From the vast pool of approaches for modeling the air path of combustion engines the lumped parameter approach (filling and emptying method) [3, 4] has become a standard for control purposes. Despite the fact that the basic physical equations are well known, the way to obtain the model parameters correctly is sometimes laborious and pitfalls are lurking in each working stage of model development. The objective of this paper is to give the automotive engineer a general methodology to obtain well-fitting parameters for his model. The contribution is structured as follows: After recapitulating the basic model equations the uncertainty contribution of each input variable on the

model parameters is analysed. Based on the results a revision of typical engine test bed setups is proposed. The next two sections highlight the choice of the measurement strategy and the appropriate processing of the measurement data. Subsequently possibilities to improve the model accuracy are mentioned. The practical assistance will be enriched with simulated and measured data from the engine test bed. The paper closes with a summary and outlook on further work.

AIR AND EXHAUST PATH MODEL

The modelling and identification procedure is introduced using the example of a modern passenger car diesel engine with common rail direct fuel injection (CRDI). Its configuration is shown in Fig. 1. The air management systems consists of a variable geometry turbocharger (VGT), a charge-air cooler and two exhaust gas recirculation loops (low-pressure and high-pressure EGR). Applying the filling and emptying method the air and exhaust path is described by an alternating series of storages (intake and exhaust manifold, ducts) and flow restrictions (valves, filters, coolers). The basic equations for modelling the storage components are the conservation laws for mass and energy

$$\frac{dm_{st}}{dt} = \sum_{i=1}^p \dot{m}_{in,i} - \sum_{j=1}^q \dot{m}_{out,j} \quad (1)$$

$$\frac{dU_{st}}{dt} = \sum_{i=1}^p h_{in,i} \dot{m}_{in,i} - \sum_{j=1}^q h_{out,j} \dot{m}_{out,j} + \dot{Q}_w \quad (2)$$

The calculation of the internal energy U_{st} as well as the specific enthalpies h_{in} and h_{out} is based on the Justi approach [5]. The wall heat flow $\dot{Q}_w$ is calculated by Newton's law of cooling

$$\dot{Q}_w = k_w A_w (T_a - T_{st}). \quad (3)$$

It is considered negligible for storages in the intake system.

Solving Eqs. (1) and (2) for the gas mass m_{st} and the gas temperature T_{st} the pressure in the storage is given by the ideal gas law

$$p_{st} = \frac{R m_{st} T_{st}}{V_{st}}. \quad (4)$$

In order to reflect the dynamic impact of the two EGR loops on the gas composition and to compute the gas properties an additional state variable is introduced. The air mass fraction $x_{st} = m_{st,air}/m_{st}$ is defined as ratio of the air mass and the total gas mass in the manifold. Its differential equation takes the form

$$\frac{dx_{st}}{dt} = \frac{1}{m_{st}} \left[\sum_{i=1}^p (x_i - x_{st}) \dot{m}_{in,i} \right] \quad (5)$$

where x_i represents the air mass fraction of the inlet flows.

The mass flow rate through the flow restrictions is obtained from the flow equation for incompressible fluids

$$\dot{m}_{res} = (C_D A_{ref}) \sqrt{\frac{2p_{in}}{RT_{in}}} \sqrt{p_{in} - p_{out}} \quad (6)$$

and compressible fluids respectively

$$\dot{m}_{res} = C_D A_{ref} \sqrt{\frac{p_{in}}{RT_{in}}} \sqrt{\frac{2\kappa}{\kappa-1} \left[\left(\Pi \right)^{\frac{2}{\kappa}} - \left(\Pi \right)^{\frac{\kappa+1}{\kappa}} \right]} \quad (7)$$

with $\Pi = \min \left[\max \left[\frac{p_{out}}{p_{in}}, \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \right], 1 \right]$

where C_D is the orifice discharge coefficient and A_{ref} a chosen reference flow area. The determining factor for the choice of the modelling approach is the gas velocity. Orifices with a fixed flow area (filters, coolers) are characterised by low velocities and a small pressure drop. They are modeled by Eq. (6). The effective flow area ($C_D A_{ref}$) is set constant. In flow restrictions with a variable cross sectional area (throttle valve, EGR valves) high flow velocities and large differential pressures occur. Here Eq. (7) is applied assuming ($C_D A_{ref}$) to be a function of the normalised actuator position s and the flow conditions.

Under the assumption of a nearly isenthalpic expansion process the downstream temperature equals the upstream one. In case of the charge-air and the EGR coolers the outlet temperature is approximated by a stationary model

$$T_{out} = T_{in} - \eta_c (T_{in} - T_{cl}) \quad (8)$$

where η_c is the cooler's effectiveness. The modelling can be extended to replicate the dynamic behaviour of the coolers. Details are given in a subsequent section.

The exhaust turbocharger model is divided into sub models for the compressor, the turbine and the shaft. To overcome the known drawbacks of the conventional map-based modelling a simplified one-dimensional approach (mean line method) is applied here. The model accounts for internal and external heat fluxes of the turbocharger and incorporates a physical based bearing loss model. The changes of enthalpy in the flow machines are modelled as a combination of an irreversible adiabatic compression/ expansion process and an isobaric heat transfer. The aerodynamic power is given by the Euler equation for turbomachinery. The heat input to the compressor as well as the heat losses of the turbine are provided by a simple heat transfer model. Contrary to common approaches the model is parameterised using measurement data from the combustion engine test bed instead of gas stand data. A detailed description of the turbocharger model is far beyond the scope of this contribution. The reader is referred to [6].

The estimation of the model parameters is mainly based on the evaluation of the upstream and downstream temperatures of the flow machines. The compressor and turbine aerodynamic power are obtained from cold test bed measurements (motored operation)

$$a = \Delta h_{adi} = c_p (T_{out,adi} - T_{in,adi}). \quad (9)$$

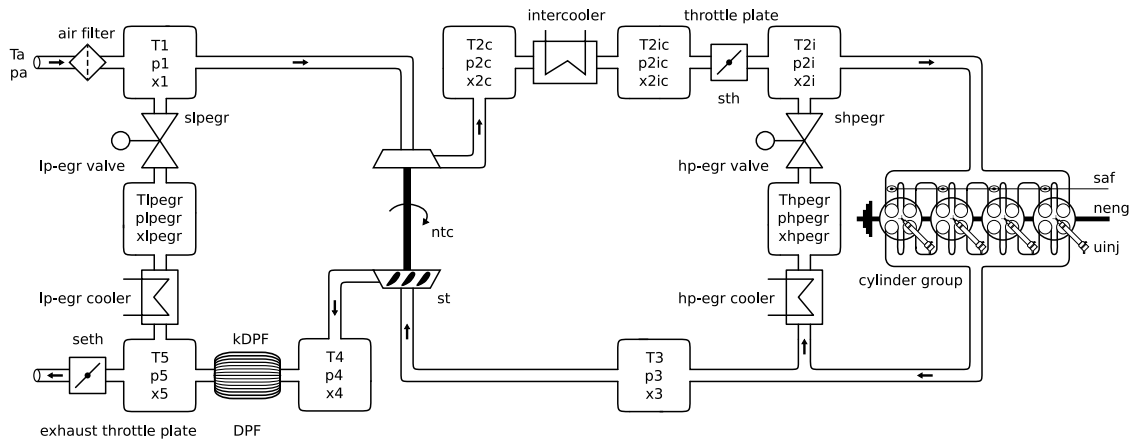


Figure 1: Scheme of the air and exhaust path model for a diesel engine with high pressure and low pressure EGR

The transferred heat q can be estimated from hot test bed measurements according to

$$a + q = \Delta h_{\text{dia}} = c_p (T_{\text{out,dia}} - T_{\text{in,dia}}). \quad (10)$$

The dynamic behaviour of the turbocharger is mainly caused by its mechanical inertia. It is modeled by

$$\dot{n}_{\text{tc}} = - \left(\frac{1}{2\pi} \right)^2 \frac{P_t + P_c + P_b}{n_{\text{tc}} J_{\text{tc}}} \quad (11)$$

where P_b represents the friction power of the bearings. The moment of inertia J_{tc} has to be identified from dynamic measurements.

PROPAGATION OF MEASUREMENT UNCERTAINTY

Measurements at the engine testbed provide the data to identify the unknown model parameters in the air path equations. These data sets are afflicted with measurement uncertainties mainly caused by sensor precision. Also neglected consideration of process dynamics in the test plan can falsify the raw data. The propagation of the errors in the measurement values in respect to the models' parameters is investigated in the following. Assuming errors of unknown sign the absolute error of a relationship $y = f(u_1, u_2, \dots, u_n)$ is defined as the sum of the absolute values of the partial derivatives regarding the inputs

$$\Delta y = \left| \frac{\partial y}{\partial u_1} \right| \Delta u_1 + \left| \frac{\partial y}{\partial u_2} \right| \Delta u_2 + \dots + \left| \frac{\partial y}{\partial u_n} \right| \Delta u_n. \quad (12)$$

Four representative engine operation points from the area covered by the New European Driving Cycle (NEDC) are chosen to investigate the sensitivity of the model parameters. Then the model inputs u_i are computed with a calibrated engine model. Tab. 1 shows the operation points OP1-OP4 at different engine speeds n_{eng} and injection quantities u_{inj} . To give further insight the measured air mass flow $\dot{m}_{\text{res}}$ is denoted.

Table 1: Representative engine operation points

	$n_{\text{eng}} [\text{min}^{-1}]$	$u_{\text{inj}} [\text{mm}^3/\text{cyl}]$	$\dot{m}_{\text{res}} [\text{kg}/\text{h}]$
OP1	1000	0	60
OP2	1660	10	97
OP3	2300	20	159
OP4	3000	30	232

All further investigations assume sensor precision taken from example testbed sensors. Pressure sensors have a tolerance of ± 25 mbar which is a precision class of $\pm 1\%$ at a metering range of $0 \dots 2.5$ bar. Type K thermocouples corresponding to DIN EN 60584-1 class 2 feature a measurement uncertainty of ± 2.5 K for measurements below 333°C . For measuring the air mass flow in the intake a Bosch HFM6-sensor is mounted. Its relative error is $\pm 2.5\%$ [7]. Simplifying the relative error to an absolute error of ± 2.5 kg/h completes the sensor configuration.

Solving the air path equations for the parameter to be identified and differentiating with respect to all measured quantities gives the sensitivity to the error in each measured variable. Regarding the flow equation for incompressible fluids (6) the propagation of uncertainty for the effective flow area $c_f = C_D A_{\text{ref}}$ follows applying (12) as

$$\begin{aligned} \Delta(c_f) = & \left| \frac{\sqrt{RT_{\text{in}}}}{\sqrt{2p_{\text{in}}\sqrt{p_{\text{in}} - p_{\text{out}}}}} \right| \Delta \dot{m}_{\text{res}} \\ & + \left| \frac{\sqrt{2RT_{\text{in}}\dot{m}_{\text{res}}}}{4\sqrt{p_{\text{in}}(p_{\text{in}} - p_{\text{out}})^3}} \right| \Delta p_{\text{out}} \\ & + \left| \frac{\sqrt{R\dot{m}_{\text{res}}}}{2\sqrt{2p_{\text{in}}T_{\text{in}}\sqrt{p_{\text{in}} - p_{\text{out}}}}} \right| \Delta T_{\text{in}} \\ & + \left| \frac{-\sqrt{2RT_{\text{in}}\dot{m}_{\text{res}}(2p_{\text{in}} - p_{\text{out}})}}{4\sqrt{(p_{\text{in}}^2 - p_{\text{in}}p_{\text{out}})^3}} \right| \Delta p_{\text{in}}. \quad (13) \end{aligned}$$

The calibrated model value of c_f is $6.8144 \cdot 10^{-4} \text{ m}^2$. Comparing the magnitude of c_{af} with the uncertainties caused by the single sensors is shown in Tab. 2 it is obvious that errors in the mass flow and the temperature have only small influence on the flow parameter c_f . The two pressure sensors p_{in} and p_{out} cause the main error on the flow parameter. Taking measurements at the operation points OP1 and OP2 results in an error ten times higher than the coefficient to parameterise. Only at high air mass flows (OP4) the error becomes smaller, caused by the rise in differential pressure.

Table 2: Uncertainty in the air filter parameter c_f

$ \Delta c_f $ [m ²]	$\Delta \dot{m}_{res} =$ $\pm 2.5 \text{ kg/h}$	$\Delta T_{in} =$ $\pm 2.5 \text{ K}$	$\Delta p_{in} =$ $\pm 25 \text{ mbar}$	$\Delta p_{out} =$ $\pm 25 \text{ mbar}$
$c_{af}(\text{OP1})$	$2.845e-5$	$2.766e-6$	$3.225e-3$	$3.216e-3$
$c_{af}(\text{OP2})$	$1.750e-5$	$2.766e-6$	$1.225e-3$	$1.217e-3$
$c_{af}(\text{OP3})$	$1.069e-5$	$2.766e-6$	$4.629e-4$	$4.543e-4$
$c_{af}(\text{OP4})$	$7.358e-6$	$2.766e-6$	$2.237e-4$	$2.151e-4$

Regarding to the flow equation for compressible fluids, Eq. (7), the propagation of error follows as

$$|\Delta C_{DAref}(s)| = \left| \frac{\sqrt{2RT_{in}}}{2p_{in} \sqrt{\frac{\kappa}{\kappa-1} \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa}{2}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa+1}{\kappa}} \right]}} \right| \Delta \dot{m}_{res} +$$

$$\left| \frac{R \dot{m}_{res}}{2p_{in} \sqrt{2RT_{in}} \sqrt{\frac{\kappa}{\kappa-1} \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa}{2}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa+1}{\kappa}} \right]}} \right| \Delta T_{in} +$$

$$\left| \frac{\dot{m}_{res} \kappa \sqrt{2RT_{in}} \left(\frac{\kappa p_{out} \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa-2}{2}}}{2p_{in}^2} - \frac{p_{out} (\kappa+1) \left(\frac{p_{out}}{p_{in}} \right)^{\frac{1}{\kappa}}}{\kappa p_{in}^2} \right)}{4p_{in} (\kappa-1) \sqrt{\left(\frac{\kappa}{\kappa-1} \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa}{2}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa+1}{\kappa}} \right] \right)^3}} \right| \Delta p_{in} +$$

$$\left| \frac{\frac{\dot{m}_{res} \sqrt{2RT_{in}}}{2p_{in}^2 \sqrt{\frac{\kappa}{\kappa-1} \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa}{2}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa+1}{\kappa}} \right]}}}{-\dot{m}_{res} \kappa \sqrt{2RT_{in}} \left(\frac{\kappa \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa-2}{2}}}{2p_{in}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{1}{\kappa}} \right)}{4p_{in} (\kappa-1) \sqrt{\left(\frac{\kappa}{\kappa-1} \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa}{2}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\kappa+1}{\kappa}} \right] \right)^3}} \right| \Delta p_{out}.$$

(14)

As an example the throttle valve is chosen. Simplifying the product $C_{D,th} A_{res,th}(s_{th})$ to $c_{th}(s_{th})$ gives the values

of $c_{th}(s_{th}=0) = 3.772 \cdot 10^{-4} \text{ m}^2$ for an opened throttle valve and $c_{th}(s_{th}=0.9) = 5.793 \cdot 10^{-5} \text{ m}^2$ for an almost closed one. Tab. 3 shows the sensitivity of the parameter c_{th} at the two operation points OP2 and OP3 for the two denoted positions of the throttle valve. When the throttle valve is opened the uncertainty of c_{th} is big and mainly depends on the precision of the pressure sensors. The sensitivity to measuring errors in the temperature can almost be neglected. Also the influence of the air mass sensor is small.

Table 3: Uncertainty in the throttle valve parameter c_{th}

$ \Delta c_{th} $ [m ²]	$\Delta \dot{m}_{res} =$ $\pm 2.5 \text{ kg/h}$	$\Delta T_{in} =$ $\pm 2.5 \text{ K}$	$\Delta p_{in} =$ $\pm 25 \text{ mbar}$	$\Delta p_{out} =$ $\pm 25 \text{ mbar}$
OP2 $s_{th}=0$	$5.120e-6$	$7.935e-7$	$1.070e-4$	$1.045e-4$
OP2 $s_{th}=0.9$	$1.447e-6$	$1.090e-7$	$8.465e-7$	$2.915e-7$
OP3 $s_{th}=0$	$3.119e-6$	$7.309e-7$	$4.737e-5$	$4.544e-5$
OP3 $s_{th}=0.9$	$1.369e-6$	$1.049e-7$	$6.137e-7$	$9.170e-8$

Fig. 2 shows the calibrated value of c_{th} and the summed uncertainty at the operation points OP2 and OP3. Table 3 reflects that the main source of uncertainty is the precision of the pressure sensors. The most impact is at small pressure differences over the flow restriction. When opening the throttle valve the differential pressure sinks. Therefore the band of uncertainty gets broader and complicates the estimation of a correct parameter c_{th} . This effect can also be recognised comparing the two operation points OP2 and OP3. At OP3 a higher air mass flow rate results in a higher differential pressure and therefore a smaller uncertainty. In general the error band size rises with decreasing mass flow and rising flow parameter c_{th} .

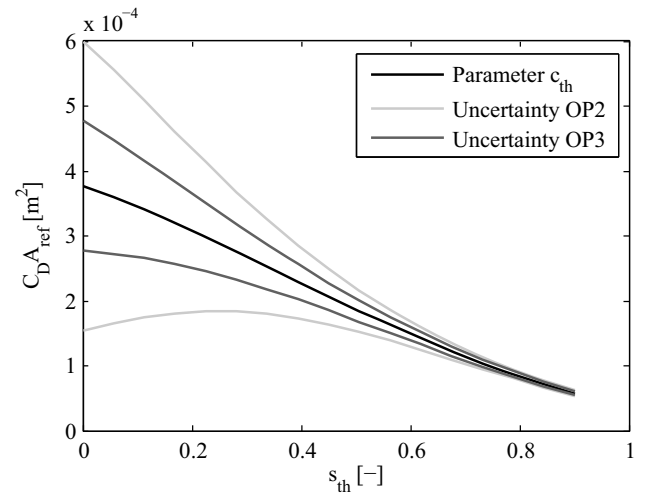


Figure 2: Throttle valve coefficient c_{th} and sensor uncertainty boundaries at engine operation points OP2 and OP3

For the cooler effectiveness, Eq. (8), the calculation of uncertainties results in

$$|\Delta\eta| = \left| \frac{T_{out} - T_{cl}}{(T_{in} - T_{cl})^2} \right| \Delta T_{in} + \left| \frac{-1}{T_{in} - T_{cl}} \right| \Delta T_{out} + \left| \frac{T_{in} - T_{out}}{(T_{in} - T_{cl})^2} \right| \Delta T_{cl} \quad (15)$$

In Table 4 the uncertainty of the intercooler's effectiveness is shown. The cooler exit temperature T_{out} has the biggest influence on the correct estimation of effectiveness. Especially at small air mass flows at operation points OP1 and OP2 the sensitivity to sensor tolerances is big. A temperature offset of only 2.5 K can result in a error of 100 % in effectiveness.

Table 4: Uncertainty in the charge cooler effectiveness η_{ic}

$ \Delta\eta_{ic} $ [-]	$\Delta T_{in} =$ ± 2.5 K	$\Delta T_{out} =$ ± 2.5 K	$\Delta T_{cl} =$ ± 2.5 K	$\eta_{ic} =$ [-]
OP1	0.258	0.498	0.234	0.48
OP2	0.171	0.266	0.095	0.36
OP3	0.043	0.058	0.015	0.26
OP4	0.018	0.024	0.006	0.23

Multiplying the enthalpy difference, Eq. (10), with the mass flow $\dot{m}_{c,t}$ through compressor and turbine results in their power. The sensitivity to sensor tolerances follows as

$$|\Delta P_{c,t,meas}| = |(T_{in} - T_{out}) c_{p,air}| \Delta \dot{m}_{c,t} + |\dot{m}_{c,t} c_{p,air}| \Delta T_{in} + |-\dot{m}_{c,t} c_{p,air}| \Delta T_{out} \quad (16)$$

The quality of turbine power and compressor power models mainly depend on the quality of the temperature measurement. Tables 5 and 6 show that the effect of temperature errors decreases with higher air mass flows at OP3 and OP4.

Table 5: Uncertainty in the compressor Power P_c

$ \Delta P_c $ [W]	$\Delta \dot{m}_c$ ± 2.5 kg/h	$\Delta T_{in} =$ ± 2.5 K	$\Delta T_{out} =$ ± 2.5 K	P_c [W]
OP1	3.506e+0	4.179e+1	4.179e+1	8.398e+1
OP2	6.565e+0	6.794e+1	6.794e+1	2.556e+2
OP3	3.018e+1	1.112e+2	1.112e+2	1.923e+3
OP4	7.297e+1	1.616e+2	1.616e+2	6.758e+3

Comparing the turbine power and the compressor power at OP1 shows the effect of transferred heat according to Eq. (10), since the absolute value of the turbine power is smaller than the compressor power.

Table 6: Uncertainty in the turbine Power P_t

$ \Delta P_t $ [W]	$\Delta \dot{m}_c$ ± 2.5 kg/h	$\Delta T_{in} =$ ± 2.5 K	$\Delta T_{out} =$ ± 2.5 K	P_t [W]
OP1	1.140e+0	4.179e+1	4.179e+1	-2.730e+1
OP2	1.131e+1	6.909e+1	6.909e+1	-4.479e+2
OP3	4.326e+1	1.144e+2	1.144e+2	-2.836e+3
OP4	9.023e+1	1.679e+2	1.679e+2	-8.680e+3

TESTBED SETUP

A revised testbed setup can dramatically improve the measurement quality. This section gives advice for an appropriate design of the measurement chain but doesn't claim for completeness. Fig. 3 shows an example testbed setup emphasising different sources of error. The measurement chain can be divided into the process and sensor layer, the signal layer and the data acquisition layer. Regarding the sensor layer the appropriate choice of the sensor measuring range in comparison to the range of the signal to be measured is well-known. For parametrising the air path equations mainly pressure, temperature and air mass flow sensors are used. The air mass flow sensor is not regarded here as it is an integral part of the engine instrumentation.

Thermocouples and PT100 resistance thermometer are the choice for precise temperature measurement. Both sensor principles have a response time to changes in the measurement fluid. The dynamic of the sensor can be described as a system of first order with a time constant mainly depending on the sensor diameter and the velocity or mass flow of the surrounding gas [8]. Mantle thermocouples are known for their small time constants (e.g. ≈ 1.8 s and ≈ 3 s for thermocouples of 0.5 mm and 1 mm diameter at a flow velocity of $v_{air} = 2 \text{ ms}^{-1}$) [9]. The main influences on the sensor error can be divided in three parts. Type T and K thermocouples of class A have precision of $\pm 1.5^\circ\text{C}$ or $\pm 0.004 T [^\circ\text{C}]$ and additional an error for the extension wires of $\pm 1.5^\circ\text{C}$. Finally an error of unknown quantity caused through measurement error in the reference temperature T_{cj} at the cold junction occurs. PT100 sensors are well known for their precision. Class A PT100 sensors have a relative error of $\pm(0, 15^\circ\text{C} + 0, 002 T [^\circ\text{C}])$ and a measuring range of $-200 \dots 850^\circ\text{C}$. Typical time constants for PT100 temperature sensors are 4...20 s [8]. Depending on the trade-off between fast dynamics and static precision either thermocouples or PT100 sensors fulfil the needs. In advance to Sect. EXPERIMENTAL DESIGN the authors recommend the measurement with PT100 sensors for the highly temperature depended parametrisation of the turbocharger turbine and compressor powers and the coolers' effectiveness.

Pressure sensors based on strain gauges or piezo resistive sensors are frequently used in testbed and automotive applications. The measurement error is very sensitive to changes in temperature. The main source of error