

Fast Crank-Angle Based 0D Simulation of Combustion Engine Cold Tests including Manufacturing Faults and Production Spread

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ABSTRACT

During series production of modern combustion engines a major challenge is to ensure the correct operation of every engine part. A common method is to test engines in end-of-line (EOL) cold test stations, where the engines are not fired but tugged by an electric motor. In this work we present a physically based 0D model for dynamic simulation of combustion engines under EOL test conditions. Our goals are the analysis of manufacturing faults regarding their detectability and the enhancement of test procedures under varying environmental conditions. Physical experiments are prohibitive in production environments, and the simulative approach reduces them to a minimum. This model is the first known to the authors exploring advanced engine test methods under production conditions. The model supports a wide range of manufacturing faults (with adjustable magnitude) as well as error-free production spread in engine components. Modeled effects include engine gas dynamics and mechanics, variable valve lift and -timing and the special cold test conditions. The challenge is to simulate a large number (i.e. >1000) EOL tests to cover a realistic production scenario. Compared to maximum-precision development tools, our 0D model has the advantage of high computational speed and the ease with which a wide range of faults can be introduced. As a use-case, a modern 2.0 liter four cylinder four stroke SI engine was modeled using only data readily available from the development process. Compared to real EOL measurements and a 1D model from engine development, the 0D simulation reproduced all relevant dynamics.

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INTRODUCTION

Long-term business success in the automotive industry strongly depends on customer satisfaction [1]. High product quality and durability result in a small rate of breakdowns in the field and strengthen the image of the original equipment manufacturer (OEM). The consequences are low costs for recalls and continuous customer retention. OEMs rely on the zero-defects principle in production to generate high quality vehicles.

The complexity of combustion engines, which are still the main powertrain component of motorized vehicles, is rapidly increasing. Multiple sensors and actuators aim for high power at low emission levels over the entire operating range. Examples for actors are variable valve timing and -lifting as well as wastegates in turbocharged engines. With the quantity of components the possibilities of faults increase accordingly. The optimal interaction between all engine components already has to be proofed during the production and assembling process. The EOL test is used to check the performance at the end of the assembling procedure of modern

combustion engines. State of the art EOL tests are realized as cold test stations, where the combustion engine is not fired, but turned by an electric motor. In comparison to hot test cells the cold test possesses several ecological and economic advantages. Low exhaust gas and noise emissions establish a comfortable working situation and drop required investments for environment safety. Cold tests captivate with a higher rate of fault detection and test cycles of short duration and therefore enable a test coverage of 100% production volume [2]. During the cold test procedure the automated test bench is connected to the engine by adapters for electrical signals to control actuators and measure sensor signals. A clutch connects the crankshaft to a speed controlled electrical motor. While tugged without combustion in the operating cylinders, over 100 signals of thermodynamic, mechanical, electric and acoustic nature are dynamically measured in certain test stages with varying actuator configurations and rotational speed. Through comprehensive evaluation of the recorded signals the engine under test is classified as error-free or faulty. To be classified as error-free every engine

signal has to satisfy predefined characteristics within certain x-ranges. Figure 1 shows a condition that checks whether the signal-maximum in given x-boundaries lies within the given y-boundaries.

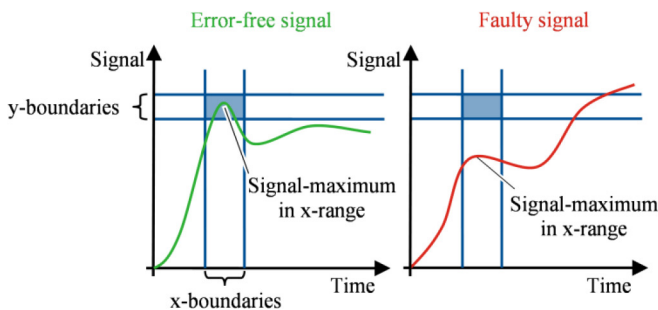


Figure 1. Example of a signal from an error-free and faulty engine and a predefined characteristic to check it

Covering the production spread the definition of the x/y-boundaries determines the balance between false alarms and detected faults. It is the cold test engineer's responsibility to find a reasonable compromise between undetected faulty engines (boundaries are too wide) and a high false alarm rate (boundaries are too tight). The determination of the boundaries fully relies on real measurement data from cold tests. This is because typical engine simulation models neither cover the special operating conditions the tugged engines are exposed to, nor the production spread of engine components. To improve the test procedure and boundaries, e.g. to check whether a certain fault can be detected reliably, physical experiments have to be conducted. However, physical experiments in full-scale production environments are highly prohibitive due to their cost. In this work we present a simulation model of combustion engines under EOL test conditions including production spread and engine production faults to resolve this.

In powertrain development, engine modeling has been an essential method in optimizing performance and building up control structures for several decades. Approaches supporting different depths and simulation speed have been derived [3]. Literature illustrates the widespread application of engine models in the development process, while references about modeling in the manufacturing context are very rare. [4, 4, 5] are dealing with cold test simulation and fault modeling to some extent, but only consider a Diesel engine in a elementary set-up stage composed of cylinders and valves covering valve train and sensor faults without an elaborate validation. In this paper a model describing the entire intake and exhaust system is explicitly derived in the cold test environment aiming for comprehensive fault studies. To reproduce a realistic production scenario, a large number of EOL tests (i.e. >1000) has to be simulated. Therefore a high simulation speed is desirable. To keep the simulation time as low as possible while still providing engine dynamics, the model has been built based on the 0D approach in crank-angle resolution. Recently 0D models gained back interest caused by the opportunity on solving them in real time systems. A lot of models regarding engine control have been developed using this technique [7, 8, 9, 9]. The model presented in this paper is the first known to the authors, which is used to explore advanced engine test methods under production conditions observing a completely assembled engine. Our model simulates mechanical and

thermodynamic cold test signals and represents all relevant physical effects in the intake and exhaust gas system caused by the individual cylinders. In comparison, a mean value approach can't provide the requested system dynamics [11, 12, 13].

The rest of this paper is structured as follows. In section "Special EOL Conditions and Engine Configuration" we give a detailed description of the operating conditions during cold testing. The model itself is derived in section "Generalized 0D Model Equations for Simulation in Crank Angle Resolution". As a use case we showcase the model in section "Implementation in Matlab/Simulink" for a modern 2.0 liter four cylinder four stroke SI engine from Audi ([14, 15] describe the 1.8 l version). Simulink® is used to implement the model equations, and simulation control is executed by a user-friendly structured Matlab® program. Finally, a validation of simulated cold test signals is performed with real data of EOL test benches as well as with a 1D GT-Suite® model. Both error-free and faulty engines with adjustable production spread are covered.

SPECIAL EOL CONDITIONS AND ENGINE CONFIGURATION

Building up a model simulating cold test signals, the special engine conditions in the EOL test stations have to be considered. The following description refers to the Audi four cylinder engine production line located in Győr (Hungary). In engine production advanced monitoring systems are applied, but consistent prevention of manufacturing faults is still not possible. In an extensive EOL test missed faults such as deviations in manufacturing tolerances (e.g. in shape cutting processes), missing or wrongly assembled parts (e.g. missing roller levers or valve springs), faults in actuator excitation (e.g. offset in throttle excitation) or sensor measurement (e.g. offset in crankshaft encoder signal) must be invariably revealed.

Before starting the cold test routine, the test bench automatically connects to the engine using an electronic adapter for actuator excitation and sensor signal logging, a multifunctional coupling disk supplying necessary operating media (e.g. oil) and a mechanical clutch linking the electric motor to the crankshaft. While oil lubrication is guaranteed by the test bench, no engine cooling fluid is used during the test process. Flow restrictions in terms of orifices at the inlet (inlet orifice diameter $d_{i,o} = 1 \text{ mm}$) and the outlet (outlet orifice diameter $d_{o,o} = 2.5 \text{ mm}$) of the air path constitute another specialty in engine cold testing (cp. Figure 2). The pressure drop caused by air flow through the orifices is used to make charge cycle effects in the intake and exhaust manifold measurable. For exact orifice attachment and positioning one relies on an industrial robot. The nomenclature cold test already implicates the unfired engine operation caused by missing fuel injection and ignition. Passing different test stages at variable actuator configuration and crankshaft speed, physical engine quantities are measured. Figure 2 shows the engine constitution and the sensor locations recording mechanical (i.e. engine torque) and thermodynamic signals (i.e. intake and intake manifold pressure, intake manifold temperature and exhaust gas pressure) dynamically over time or crank angle. Low rotational speed along with high restriction through the orifices in most of the test stages precludes the rotation of the exhaust turbocharger turbine. For

this reason, the turbocharger dynamics are not included in the presented model. In [9] a thermodynamic and fluid-dynamic turbocharger model approach is shown.

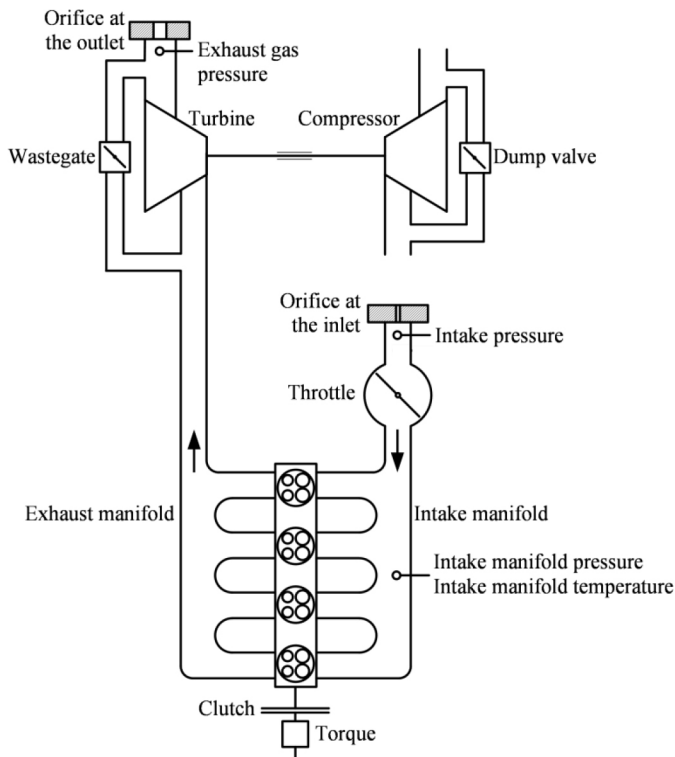


Figure 2. Schematic representation of the air path during cold test and the positions of the thermodynamic and mechanical sensors

Simulating mechanical and thermodynamic cold test sensor signals requires consequent inclusion of the mentioned EOL conditions.

Furthermore, engines in real production environments have an individual production spread, resulting from manufacturing tolerances of all engine parts. The model supports this effect and in addition a wide range of manufacturing faults. In the next section generalized 0D elementary models for engine components, which are used to build up the engine simulation tool in the subsequent step, are derived based on previous works and accounting for the EOL characteristics.

GENERALIZED 0D MODEL EQUATIONS FOR SIMULATION IN CRANK-ANGLE RESOLUTION

Like in control oriented modeling the application elaborated in this paper also calls for high simulation speed to represent a real production environment in short time. Since simulating multidimensional models is time-consuming, the zero dimensional approach with lumped parameters is used [9, 16]. The basic assumption for zero dimensional models is that thermodynamic states in a fixed volume are seen as location independent. Unlike mean value models for real time applications no simplifications, which renounce the proper reproduction of physical effects to the benefit of time saving, are allowed. All signals are persistently simulated over a

crank-angle scale, which leads to consideration of all engine gas dynamic effects for individual cylinders, but without the requirement of real-time capability.

Nowadays in 0D engine modeling two techniques have been established, called “Filling and Emptying” (F&E) and “Quasi-Steady-Flow” (QSF) method [17]. The gas path is considered as an alternating sequence of gas storages (e.g. manifold) and flow resistances (e.g. throttle). The F&E approach is based on physical laws in terms of mass and energy conservation as well as the ideal gas law in differential form, which result in ordinary differential equations (ODEs) [9, 11, 18]. Gas storages, which can be seen as control volumes, are the typical usage of the F&E method. None-volume parts, i.e. flow resistances, are described by QSF models using algebraic equations [8, 9, 17, 18, 19]. Combining the distinct models for storages and flow resistances yields the process model of the entire engine gas dynamic [9]. The dynamic engine torque generation includes the gas and the mass torque. Through the kinetic of the crank drive the gas and mass torques are calculated and subsequently summarized [9, 20].

To sufficiently predict the influences of manufacturing faults and spread on simulated state variables all specialties regarding EOL tests mentioned in section above have to be included in the models. In the following, elementary models for generalized engine components (i.e. storage, cylinder, resistance) using the lumped parameter approach based on literature are derived and presented. A short outline of the gas and mass torque calculation method is also given.

Storages

An engine consists of several gas storages such as the intake manifold, ducts or the exhaust gas manifold. In a narrow sense, the engine cylinders are a kind of storage, but due to volume alteration in time they are treated in a separate section. In this regard storages are characterized by a constant control volume with mass and energy permeable boundaries in order that the equations for thermodynamic open systems hold. One or more inlet and outlet mass flows transport internal energy over the control surface and cause a change of storage state variables (i.e. pressure p_{st} , temperature T_{st} and mass m_{st}) (cp. Figure 3). Using the F&E approach one obtains the differential equations of the three state variables.

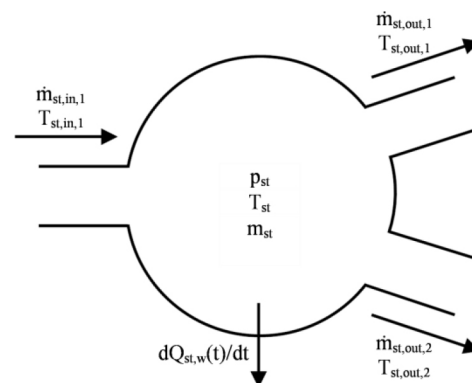


Figure 3. Gas storage with one inlet and two outlet mass flows taking heat transfer through the storage wall into account

The mass balance of a gas storage with an arbitrary number of inlet and outlet flows provides the first ODE [9]:

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

Based on the first law of thermodynamics the energy balance can be formed correspondingly [9]:

$$\begin{aligned} \frac{dU_{st}}{dt} = & \sum_{i=1}^p h_{st,in,i} \cdot \dot{m}_{st,in,i} - \\ & \sum_{j=1}^q h_{st,out,j} \cdot \dot{m}_{st,out,j} - \frac{dQ_{st,w}}{dt} \end{aligned} \quad (2)$$

where U_{st} is the internal energy of the stored air, $h_{st,in}$ and $h_{st,out}$ are the specific enthalpies of the mass flows and $dQ_{st,w}/dt$ is the heat flow through the storage walls. Neglecting the heat transfer is reasonable due to the missing engine cooling and the unfired operation in the cold test application. Together with the assumption of a permanent validity of the ideal gas law and constant specific heat capacities c_p and c_v [9], the differential equation for the air temperature is (The exact derivation can be found in the [appendix](#)) [7]:

$$\begin{aligned} \frac{dT_{st}}{dt} = & \frac{1}{m_{st} \cdot c_v} \left(\sum_{i=1}^p c_p \cdot T_{st,in,i} \cdot \dot{m}_{st,in,i} - \right. \\ & \left. \sum_{j=1}^q c_p \cdot T_{st,out,j} \cdot \dot{m}_{st,out,j} - c_v \cdot T_{st} \cdot \frac{dm_{st}}{dt} \right) \end{aligned} \quad (3)$$

Solving the ideal gas law in the differential form for the pressure changing and inserting [eq. \(3\)](#) results in the last intended gas storage ODE (The exact derivation can be found in the [appendix](#)) [7]:

$$\frac{dp_{st}}{dt} = \frac{R \cdot \kappa}{V_{st}} \left(\sum_{i=1}^p T_{st,in,i} \cdot \dot{m}_{st,in,i} - \sum_{j=1}^q T_{st,out,j} \cdot \dot{m}_{st,out,j} \right) \quad (4)$$

with the ratio of specific heat capacities κ and the specific gas constant of the air R . The mass flows and their temperatures are provided by the adjacent flow resistances using the QSF method. Integrating [eqs. \(1\), \(3\) and \(4\)](#) delivers the state variables of the gas storage as functions of time.

Cylinders

This section derives three equations for the description of the cylinder behavior in four stroke engines during EOL testing on the basis of the single-zone approach as presented in [20]. In contrast to engine development models, a combustion model is not necessary for the given cold test application. Since a cylinder is nothing else than a gas

storage whose volume has a time dependency, the F&E approach can be used again. The control volume of the cylinder and its system boundary shows [Figure 4](#). The three state variables are p_{cyl} , T_{cyl} and m_{cyl} . Depending on the mass flow in and out of the cylinder through the valves $\dot{m}_{iv}$ and $\dot{m}_{ov}$, the volume change dV and the heat transfer through the cylinder walls $Q_{cyl,w}$ the stored energy is changing.

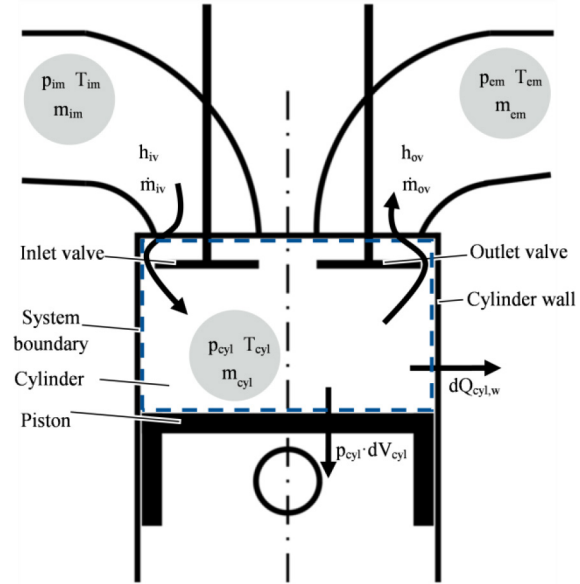


Figure 4. Single-zone model of the cylinder (according to [22])

The main difference in the model derivation in comparison to the storage model from the previous section is the volume change, which has to be taken into account. Mass and energy conservation together with the ideal gas law in differential form yield the three state equations, while heat transfer through the cylinder walls is not evaluated (for the exact derivation of the temperature and pressure ODE it is referred to the [appendix](#)):

- Cylinder mass for a cylinder with r inlet and s outlet valves:

$$\frac{dm_{cyl}}{dt} = \sum_{k=1}^r \dot{m}_{iv,k} - \sum_{l=1}^s \dot{m}_{ov,l} \quad (5)$$

- Cylinder temperature allowing for volume changes dV_{cyl} :

$$\begin{aligned} \frac{dT_{cyl}}{dt} = & \frac{1}{m_{cyl} \cdot c_v} \left(\sum_{k=1}^r h_{iv,k} \cdot \dot{m}_{iv,k} - \sum_{l=1}^s h_{ov,l} \cdot \dot{m}_{ov,l} - \right. \\ & \left. p_{cyl} \cdot \frac{dV_{cyl}}{dt} - \frac{dm_{cyl}}{dt} \cdot c_v \cdot T_{cyl} \right) \end{aligned} \quad (6)$$

- Cylinder pressure:

$$\frac{dp_{cyl}}{dt} = \frac{1}{V_{cyl}} \left(R \cdot \frac{dm_{cyl}}{dt} \cdot T_{cyl} + R \cdot m_{cyl} \cdot \right.$$

$$\left. \frac{dT_{cyl}}{dt} - p_{cyl} \frac{dV_{cyl}}{dt} \right)$$

- with the specific enthalpies of the inlet and outlet mass flow h_{iv} and h_{ev} estimated by the following relationship [9]:

$$h = c_p \cdot T \quad (8)$$

The cylinder volume as a function of time follows from engine kinetics [20, 23]:

$$\frac{dV_{cyl}}{dt} = \omega \cdot A_{piston} \cdot r_{cs} \cdot \left(\sin(t \cdot \omega + \varphi_0) + \frac{\lambda}{2} \cdot \sin(2 \cdot (t \cdot \omega + \varphi_0)) \right) \quad (9)$$

where:

- ω is the angular velocity of the crankshaft.
- A_{piston} is the pistons cross-sectional area.
- r_{cs} is the eccentricity of the crankshaft.
- λ is the connecting rod ratio.

Eqs. (6), (7), (8) and (9) are applied to all working cylinders taking the particular firing distance angles φ_0 into consideration.

Flow Resistances

Flow resistances are model as non-volume elements using the QSF method to estimate the mass flows and their specific temperatures for the adjacent storage or cylinder models (cp. section “Storages” and “Cylinders”) and cause pressure drops between consecutive volumes. Figure 5 shows the conjunction of two storages by one flow resistance and the associated nomenclature of the state variables.

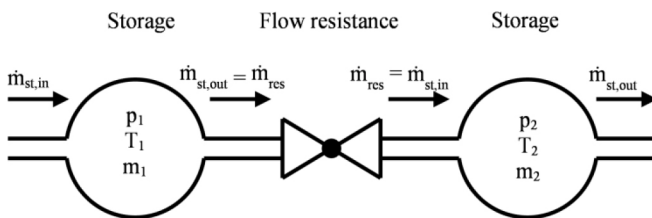


Figure 5. Combination of two storages and a flow restriction

Based on the algebraic flow functions for compressible fluids the air mass $\dot{m}_{res}$ streaming through a flow restriction in subcritical condition is calculated as a function of the prevailing pressure ratio [8, 9, 19, 24]:

$$\dot{m}_{res} = c_d \cdot A_{ref} \cdot \sqrt{\frac{2}{R \cdot T_1}} \cdot p_2 \cdot \psi\left(\frac{p_2}{p_1}\right) \quad (10)$$

with

$$\psi\left(\frac{p_2}{p_1}\right) = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\kappa}} \cdot \sqrt{\frac{\kappa}{\kappa-1} \cdot \left(1 - \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}}\right)} \quad (11)$$

In eqs. (10) and (11) ψ is the outflow function depending on the pressure ratio. The discharge coefficient c_d including friction losses and contraction effects [9] and the reference flow area A_{ref} follow from experimental measurements. According to Figure 5 T_1 and p_1 are the upstream stagnation temperature and pressure. Assuming negligible pressure recovery, which is reasonable for sharp edged restrictions [19], p_2 is equal to the pressure of the succeeding air reservoir. When the velocity of the air mass in the contraction reaches sound velocity, the outflow function ψ attains a constant value, which depends on the isentropic exponent exclusively, and a simplified equation for the mass flow holds [19]:

$$\dot{m}_{res} = c_d \cdot A_{ref} \cdot \sqrt{\frac{2}{R \cdot T_1}} \cdot p_2 \cdot \sqrt{\frac{\kappa}{2}} \cdot \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{2(\kappa-1)}} \quad (12)$$

The crucial quantity is the pressure ratio. For choked flow, i.e. the pressure ratio is less than or equal the critical value [19]

$$\left(\frac{p_2}{p_1}\right)_{crit} \leq \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}, \quad (13)$$

eq. (12) is appropriate for the mass flow calculation. Especially at flow resistances with small flow diameters high pressure losses result in high flow velocities near or at sonic speed [7]. In accordance with [9] for small pressure differences a isothermal change of state is assumed, which leads for the mass flow temperature T_{res} to:

$$T_{res} = T_1 \quad (14)$$

A separate approach for heat transfer in the resistance is not used, since heat transfer does not pertain to the primary function of the flow resistances (e.g. orifices, valves and leakages) in the considered engine [22]. The model equations are used for every occurring flow restriction in the system.

Torque

Many analytical formulations estimating engine torque exist in open literature [9, 20]. The total engine torque M_{eng} , i.e. the drive torque of the electric engine M_{el} , is composed of several individual parts [9]:

- gas force torque M_g
- oscillating masses torque M_m
- friction torque M_f
- auxiliary drive torque M_{aux}
- valve train torque M_v
- mass moment of inertia M_i