CZECH TECHNICAL
UNIVERSITY
IN PRAGUE

FACULTY
OF MECHANICAL
ENGINEERING

DOCTORAL
THESIS
STATEMENT

Czech Technical University in Prague

Faculty of Mechanical Engineering

Department of Automotive, Combustion Engine and Railway Engineering

Doctoral Thesis Statement

Physical Modelling of Combustion Engine Process and Gas Exchange for Real-Time Applications

by

Jan Fortl, M.Sc.

Doctoral Study Programme: *Mechanical Engineering*

Field of Study: *Machines and Equipment for Transportation*

Supervisor: *Prof. -Ing. Jan Macek, DrSc*

Expert Supervisor: *doc. Ing. Oldřich Vítek, Ph.D.*

Teze disertace k získání akademického titulu "doktor", ve zkratce "Ph.D."

2022 Prague

Název anglicky: **Fyzikální modelování procesu spalovacího motoru a výplachu válce pro aplikace v reálném čase**

Disertační práce byla vypracována v kombinované formě doktorského studia na Ústavu automobilů, spalovacích motorů a kolejových vozidel, Fakulty strojní ČVUT v Praze.

Disertant: Jan Fořtl, M.Sc.
Ústav 12120, Fakulta strojní ČVUT v Praze
Technická 4, Praha 6

Školitel: Prof. Ing. Jan Macek, DrSc
Ústav 12120, Fakulta strojní ČVUT v Praze
Technická 4, Praha 6

Školitel-specialista: doc. Ing. Oldřich Vítek, Ph.D.
Ústav 12120, Fakulta strojní ČVUT v Praze
Technická 4, Praha 6

Oponenti:

Teze byly rozeslány dne: ....

Obhajoba disertace se koná dne .... v .... hod.

v zasedací místnosti č. 17 (v přízemí) Fakulty strojní ČVUT v Praze, Technická 4, Praha 6, před komisí pro obhajobu disertační práce ve studijním oboru Dopravní stroje a zařízení.
S disertací je možno se seznámit na oddělení vědy a výzkumu Fakulty strojní ČVUT v Praze, Technická 4, Praha 6.

doc. Ing. Oldřich Vítek, Ph.D.
předseda oborové rady oboru Dopravní stroje a zařízení
Fakulta strojní ČVUT v Praze

Abstract

The large majority of gas exchange models used for engine torque control within the Engine Control Unit (ECU) is data oriented. This means that they use look-up tables, or other more sophisticated methods (e.g., neuronal networks), to access calibration data during the engine operation. They require high calibration effort (e.g., in case of look-up tables, complexity is growing exponentially with the number engine actuators) and are usually limited to mean value information on engine stroke events. The use of physical based models is typically not possible due to low CPU performance of costly optimized production ECU's.

This work investigates the possibility to calculate a crank angle resolved, physical based, 1D and 0D thermodynamic engine simulation directly on a serial ECU (240MHz) in real-time. Transient flow in intake and exhaust manifolds, including pressure wave propagation, is described by conservation laws for mass, momentum and energy. Defined set of differential equations is solved by Runge-Kutta integration methods with a fixed time integration step.

A commercial 4-cylinder, turbocharged, spark-ignited engine is used for stationary as well as transient experiments. A detailed 1D model is defined, that satisfies accuracy requirements (e.g., deviation of in-cylinder air mass <5%) in a wide range of operating conditions. Different levels of simplifications between 1D and 0D are assessed in terms of the trade-off accuracy and real-time capability.

Anotace (CZ)

Velká většina modelů výplachu válce používaných pro řízení točivého momentu motoru v rámci řídicí jednotky (ECU) je datově orientovaná. To znamená, že pro přístup ke kalibračním datům během provozu motoru používají vyhledávací tabulky nebo jiné sofistikovanější metody (např. neuronové sítě). Tyto modely vyžadují velké úsilí při kalibraci (např. v případě vyhledávacích tabulek roste složitost exponenciálně s počtem stupňů volnosti motoru) a jsou obvykle omezeny na informace o středních hodnotách. Použití fyzikálních modelů obvykle není možné kvůli nízkému výkonu CPU, používaných v ECU z důvodů nízkých produkčních nákladů.

Tato práce zkoumá možnost výpočtu fyzikálních, termodynamických 1D a 0D simulací přímo na sériovém ECU (240 MHz) v reálném čase. Přechodné proudění v sacím a výfukovém potrubí, včetně šíření tlakové vlny, je popsáno zákony zachování hmoty, hybnosti a energie. Definovaná soustava obyčejných diferenciálních rovnic je řešena integračními metodami Runge-Kutta s fixním časem integračního kroku.

Komerční čtyřválcový, přeplňovaný, zážehový motor je pužit pro stacionární i přechodné experimenty. Je definován podrobný 1D model, který splňuje požadavky na přesnost (např. odchylka hmotnosti vzduchu ve válci < 5 %) v širokém rozsahu provozních podmínek. Různé úrovně zjednodušení mezi 1D a 0D jsou posuzeny z hlediska přesnosti a schopnosti operovat v reálném čase.

Contents

Introduction

For spark ignited engines, torque control is realized in the Engine Control Unit (ECU) by managing the cylinder charge exchange, while keeping the air-fuel ratio stoichiometric in order to minimize exhaust emissions [1] [2]. For this purpose, the ECU needs a real-time capable model, giving an accurate prediction based on the current sensor information. The knowledge of the in-cylinder state for each single combustion event allows a more efficient and emission optimized process control of the engine to meet the future CO_2 legislations.

In principle, there are two modelling strategies: 1. data driven models and 2. physical models [3]. Data driven models start from the result, typically approximated measured or simulated data. On the other hand, physical models rely on fundamental physical laws and natural constants.

The main obstacle for using physical models for direct engine control is the low CPU performance of state-of-art production ECU’s. The level of physical description for the particular application is in conflict with the runtime performance.

1 Literature Research

Coarse History – Publications

The history of automotive real-time tools starts in the 1980s with interpolation map-based models, improved during the 1990s to data based and semi-physical real-time tools [4] [5]. Around 2000, there started the use of neuronal networks and simple physical based models (semi-physical or grey-box models [4]). Neuronal network models have become widely popular in the last couple of years (e.g., 2006 LOLIMOT [6] [7], 2008 LMN [8]), especially due to their robustness and applicability of similar structures to completely different problems solved. Nevertheless, they remain still being data driven models, and therefore, require high calibration effort.

Real-time capable, physical based models solving differential equations are mainly used for ECU calibrations or other Hardware-in-the-Loop applications (Rösler 2013 [9], Ludwig 2011 [4], Friedrich 2008 [10]), but usually cannot be used directly for the engine control due to high computational effort.

Commercial RT Applications

There are several commercial black-box models available for the ICE engine simulations providing real-time capable solutions for SiL and HiL applications:

- Wave RT (https://ricardo.com/)
- GT-Power RT (https://www.gtisoft.com/)
- Simcenter AMESim (https://www.plm.automation.siemens.com)
- DYNA THEMOS (www.tesis.de)

Ready-to-use libraries are provided for different levels of physical complexity. For instance, results provided by 3D CFD simulations are used for dimensionless parametrization of complex 1D engine and gas exchange models, that are further simplified to fast-running 0D models and finally used for a HiL simulation or calibration of RT control-oriented models [6] [11].

In 2015, Ricardo presented real-time capable 1D engine application [12]. The rapid prototyping hardware platform rCube2 (based Infineon 2 core processors with 150 MHz internal clock frequency) was used for HiL simulation to calibrate a diesel engine ECU and improve emissions [13].

One of the most widely used engine simulation tools is the GT-Power, providing bride solutions from the possibility of co-simulation with 3D over different complexities in 1D an 0D. One example of RT application was presented in by Fiat Chrysler in SIA paper 2014 [14]. A detailed 1D model, previously used for engine performance an emission optimization (thus widely validated) was simplified to a fast-running 0D model. The fast-running 0D model was used for real-time HiL simulations with the objective to calibrate an ECU of a diesel engine. As a hardware platform, dSpace Autobox with a four cores processor 2.8GHz was used.

Last example to be mentioned is the German company TESIS that offers software package DYNA THEMOS, developed in cooperation with Technical University of Berlin (2007 Friedrich [10], 2013 Roesler [9]). The modular approach of THEMOS engine components is real-time capable on dSPACE HiL platforms. A CAN interface is used for communication and calibration of ECU’s.

One disadvantage of the commercial engine tools to be mentioned are the high licence costs. Another typical issue is the fact that as a black-box solution, only already available libraries can be used with restricted possibility for user defined library extensions for the particular application.

Conclusion / Current State-of-the-Art

The large majority of physical based real-time engine applications is used to calibrate data-oriented control models (look-up tables, neuronal networks etc.) used then for the purpose of the engine control during operation [6] [15] [9] [10] [4] [16] [14] [17]. With some exceptions like Ricardo Wave software [12], which is a commercial black-box model, there are no available codes with a potential to be real-time capable on an engine production ECU that has usually a restricted processor performance due to manufacturing costs. Most real-time applications using thermodynamic engine models require specialized HiL hardware like dSpace with high processor performance (e.g., CPU > 1GHz). There is also no available work that would transparently show the structure of computational effort (e.g., CPU load) of individual model components or modules.

2 Objective

The objective of this work is to

- create a physical model, based on differential equations, yielding detailed, crank angle resolved information on engine in-cylinder gas mixture and charge exchange including performance of a turbocharger, suitable for predictive model-based control of a turbocharged ICE. The model has to be real-time capable on a state-of-the-art production ECU. Required model calibration data should be less demanding than standard data-based models.

As an accuracy criterion,

- deviation of in-cylinder fresh air charge should be lower than 5% compared to steady-state and transient engine measurements

As a target hardware,

- a state-of-art production ECU with 240MHz processor clock should be used.

To keep the general validity of proposed models

- only global calibration parameters, valid as a single constant for the entire engine operating range should be used, based on natural constants and physically interpretable parameters derived from geometry.

Secondary objective:

- Because of the fact, that the transient 1D pressure propagation phenomena is the most time-consuming part of the numerical solution, but crucial aspect of the engines filling behaviour, it should be considered first. Theoretical real-time performance on target hardware should be assessed and shown in a transparent manner. The model should then be simplified to allow the real-time capability, but keeping the accuracy and inherited features from the detailed model at a reasonable level.

3 Theory

3.1 Numerical Solver

Thermodynamic engine components are defined individually as sets of ‘Ordinary Differential Equations’ (ODEs). From the mathematical point view, the system is formulated as an initial value problem with a given state $q = q(t) = (q_1, q_2, \dots, q_n)$:

Problem:

$$\dot{q} = f_{ODE}(t, q) \tag{3.1}$$

Solution:

$$q(t) = \int_{0}^{tEnd} \dot{q}\, dt \tag{3.2}$$

The initial value problem is solved with an explicit 2nd order Runge-Kutta integration method:

Predictor step (Euler)

$$q_{pre}^{k+1} = q^k + \Delta t \cdot \dot{q}_{pre}^{k} \tag{3.3}$$

Corrector step (Heun)

$$q^{k+1} = q^k + \frac{\Delta t}{2} \cdot \left(\dot{q}_{pre}^{k} + \dot{q}_{Corr}^{k+1}\right) \tag{3.4}$$

Causal modelling approach [18] is used to link the inputs of one component to the outputs of another via uniformly defined component ports.

3.2 Engine Process Simulation

The purpose of the exhaust and inlet processes is to remove the burned gasses and admit the fresh charge of reciprocating cycles. Fig. 3 illustrates the pulsating intake and exhaust gas flow in a conventional spark-ignited engine.

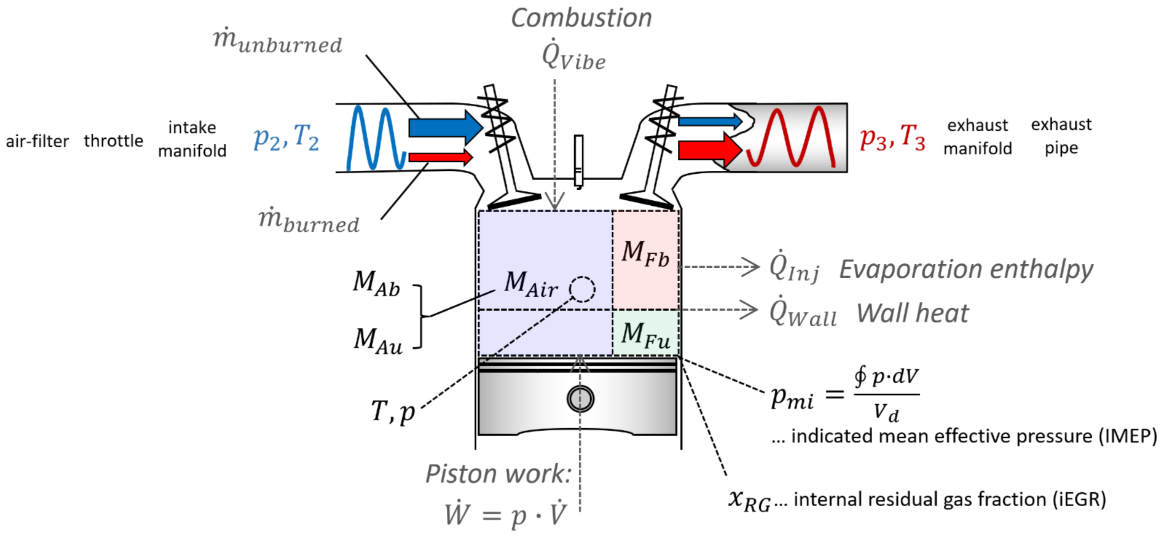

Fig. 1 Principle of gas exchange process, definitions of air-fuel composition and energy flow of the two-zone combustion model

The gas mixture is assumed to be composed by three specie components. The air, burned fuel and unburned fuel:

$$M = M_{Air} + M_{Fb} + M_{Fu} \tag{3.5}$$

The overall engine operation parameters of greatest interest, determined from a thermodynamic analysis of the engine operating cycle, are among others the indicated mean effective pressure (IMEP) [19] and the internal residual gas fraction (iEGR) [19] [1].

3.2.1 Thermodynamic Volumes

The 0D volumes are assumed as the open thermodynamic system (useful theory is in [19], [1], [3], [10], [4] or [9]). The change of mass states is modelled with the mass conservation law:

$$\frac{d}{dt}\begin{pmatrix} M_{Air} \\ M_{Fb} \\ M_{Fu} \end{pmatrix} = \begin{pmatrix} \Sigma\, \dot{m}_{Air,i} \\ \Sigma\, \dot{m}_{Fb,i} \\ \Sigma\, \dot{m}_{Fu,i} \end{pmatrix} \tag{3.6}$$

The energy conservation law (without thermodynamic work)

$$\frac{dE}{dt} = \sum \dot{m}_i \cdot h_i + Q \tag{3.7}$$

gets the form

$$\begin{aligned} \frac{d}{dt}\Big((M_{Air} + M_{Fb}) \cdot e_{gas} + M_{Fu} \cdot e_{fuel}\Big) \\ = \sum\big(\dot{m}_{Air,i} + \dot{m}_{Fb,i}\big) \cdot h_{gas,i} + \sum \dot{m}_{Fu,i} \cdot h_{fuel,i} \\ - Q_{Wall} \end{aligned} \tag{3.8}$$

, with specific internal energy e_{gas} end enthalpy h_{gas} calculated as gas properties dependent on the temperature and the richness factor [20].

3.2.2 Engine Cylinders

A 0D two-zone model [1] is used for the thermodynamic cylinder. Geometrical displacement volume is defined as function of crank angle. The conservation laws for masses are:

$$\frac{d}{dt}\begin{pmatrix} M_{Air} \\ M_{Fb} \\ M_{Fu} \end{pmatrix} = \begin{pmatrix} \dot{m}_{Air,in} - \dot{m}_{Air,ex} \\ \dot{m}_{Fb,in} - \dot{m}_{Fb,ex} + \dot{m}_{Vibe} \\ \dot{m}_{Fu,in} - \dot{m}_{Fu,ex} + \dot{m}_{inj} - \dot{m}_{Vibe} \end{pmatrix} \tag{3.9}$$

The mass transfer $\dot{m}_{Vibe}$ is calculated by using the Vibe combustion model [21] [1].

The most complex conservation law is the energy equation:

$$\begin{aligned} \frac{d}{dt}\Big((M_{Air} + M_{Fb}) \cdot e_{gas} + M_{Fu} \cdot e_{fuel}\Big) \\ = \big(\dot{m}_{Air,in} + \dot{m}_{Fb,in}\big) \cdot h_{gas,in} + \dot{m}_{Fu,in} \cdot h_{fuel,in} \\ + \big(\dot{m}_{Air,ex} + \dot{m}_{Fb,ex}\big) \cdot h_{gas,ex} + \dot{m}_{Fu,ex} \cdot h_{fuel,ex} \\ - p \cdot \dot{V} - \alpha \cdot A_{Wall} \cdot (T - T_{Wall}) - \dot{Q}_{Vibe} \end{aligned} \tag{3.10}$$

The Woschni model (1970 [22], implementation from Merker [21]) is used to calculate the heat transfer coefficient $\alpha\left[\frac{W}{m^2K}\right]$.

3.2.3 Orifices

Engine flow devices are modelled assuming subsonic or sonic flow through a flow restriction [1] [21] [19] [3], using the well-known Saint-Venant formula

$$\dot{m}_{SV} = A_{Eff} \cdot \sqrt{2 \cdot \rho_u \cdot p_u} \cdot \psi(x) \quad , \qquad \textit{with } x = \frac{p_d}{p_u}$$
$$\psi(x) = \sqrt{\frac{\kappa}{\kappa-1} \cdot \left(x^{\frac{2}{\kappa}} - x^{\frac{\kappa+1}{\kappa}}\right)} \qquad \textbf{(3.11)}$$

, where x refers to the ratio of the static pressure downstream of the restriction to the upstream stagnation pressure.

Optionally, the mass flow can be smoothed with a first order filter, giving an additional equation to the solved system

$$\frac{d}{dt}(\dot{m}) = \frac{\dot{m}_{SV}(x) - \dot{m}}{T_{delay}} \qquad \textbf{(3.12)}$$

The filter may be useful to stabilize dynamics. The time delay constant T_{delay} is also used as a calibration parameter to adjust transport delay behavior to the mass flow.

3.3 Turbocharger

Turbochargers in engine simulations are often represented by characteristic maps based on experimental data [23] [24] [6] [11] [47].

Compressor

The compressor is modelled as a flow device within the simulation and has therefore the same boundary conditions like the orifice. Additionally, mechanical boundary conditions based on the torque (power) equilibrium are used.

Due to the non-monotonic compressor flow map, a direct interpolation of the mass flow as a function of the pressure ratio $\dot{m}_c = f(\Pi_c)$ is ambiguous. Solution proposed by Friedrich [10], also published in Mecca 2022 [47], is being used. The idea is that a fluid mass between upstream and downstream boundaries must be accelerated by a pressure difference acting on cross section area on both sides of a control volume.

$$(\rho_1 \cdot A \cdot L) \cdot \frac{d}{dt}(u_c) = A \cdot p_1 - A \cdot p_2 - A \cdot (p_{2t} - p_{1t}) \qquad \textbf{(3.13)}$$

The momentum conservation (3.13) leads to an additional differential equation to be solved in the system

$$\frac{d}{dt}(\dot{m}_c) = \frac{A}{L} \cdot \left[\underbrace{\Pi_c\left(\dot{m}_{N,c},\ n_{N,c}\right)}_{\textit{map-interpolation}} \cdot p_{1t} - p_2\right] \qquad \textbf{(3.14)}$$

Used interpolation of the pressure ratio in (3.14) is already straight-forward and gives an unambiguous solution.

The resulting compressor output torque is calculated as

$$T_{qC} = \dot{m}_c \cdot \underbrace{\Delta h_C\left(\dot{m}_c,\ n_{N,c}\right)}_{\textit{map-interpolation}} \cdot \frac{1}{\omega_{TC}} \qquad \textbf{(3.15)}$$

, dependent on the given boundary conditions.

Turbine

The turbine is modelled as a flow restriction between two gas states and has therefore same boundary conditions like the orifice. A mechanical boundary condition at the turbine shaft (input speed, output torque) is defined analogously to the compressor.

The reduced turbine mass flow is obtained by an interpolation of the turbine characteristics with the given boundary pressure ratio Π_t and the reduced turbine speed $n_{R,t}$. The real turbine mass flow follows from the interpolated reduced mass flow by the given SAE9222 definitions.

$$\dot{m}_t = \underbrace{\dot{m}_{R,t}\left(\Pi_t,\ n_{N,t}\right)}_{\textit{map-interpolation}} \cdot \frac{p_{3t} \cdot 10^{-5}}{\sqrt{T_3}} \tag{3.16}$$

The real specific enthalpy difference is calculated from the ideal isentropic enthalpy difference and the interpolated turbine efficiency.

$$\Delta h_T = \underbrace{c_{p,exh} \cdot T_3 \cdot \left[1 - \left(\frac{p_4}{p_{3t}}\right)^{\left(\frac{\kappa_{exh}-1}{\kappa_{exh}}\right)}\right]}_{\Delta h_{sT}} \cdot \underbrace{\eta_T\left(\Pi_t,\ n_{N,t}\right)}_{\textit{map-interpolation}} \tag{3.17}$$

The resulting turbine output torque

$$T_{qT} = \dot{m}_t \cdot \Delta h_T \cdot \frac{1}{\omega_{TC}} \tag{3.18}$$

is directly depend on the given boundary conditions.

Torque Equilibrium (Inertial Mass)

The turbocharger model is supposed to operate under transient engine operating conditions. The map-based model is therefore extended with mechanical inertia mass [24] [6] [47]. The dynamic torque equilibrium at the turbocharger shaft has the following form:

$$T_{qT} - T_{qC} - T_{qI} = 0 \quad \text{, with} \quad T_{qI} = I \cdot \ddot{\varphi}_{TC} \tag{3.19}$$

This leads to two differential equations for the accelerated turbocharger shaft mass with a given inertia:

$$\frac{d}{dt}\begin{pmatrix}\varphi_{TC}\\ \omega_{TC}\end{pmatrix} = \begin{pmatrix}\omega_{TC}\\ \left[T_{qT} - T_{qC} - \omega_{TC} \cdot d\right]/I\end{pmatrix} \tag{3.20}$$

3.4 Pipe Systems

Complex Transient 1D Flow in Pipes

Transient fluid dynamics phenomena in engine duct systems represents the most demanding part of the numerical solution. The governing equations [21] with general validity for unsteady, non-homentropic flow including fluid friction are:

continuity equation – mass conservation

$$\frac{\partial \rho}{\partial t} + \rho \frac{\partial u}{\partial x} + u \frac{\partial \rho}{\partial x} + \frac{\rho u}{A}\frac{dA}{dx} = 0 \tag{3.21}$$

momentum equation

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + \frac{1}{\rho}\frac{\partial p}{\partial x} + f_r = 0 \tag{3.22}$$

energy equation (assumption of ideal gas)

$$\frac{\partial p}{\partial t} + u \frac{\partial p}{\partial x} - a^2 \left(\frac{\partial \rho}{\partial t} + u \frac{\partial \rho}{\partial x}\right) - (\kappa - 1)(\dot{q}_H + u f_r)\rho = 0 \tag{3.23}$$

From a mathematical point of view, these are 1D hyperbolic conservation laws, usually written in a more compact matrix form [25] [26]:

$$q_t + f(q_x) = s \tag{3.24}$$

, where q is a state vector with density, velocity and energy states.

Partial Differential Equations (PDEs) are discretized in space by using 1st order upwind scheme on in terms of the Finite-Volume-Method, giving a set of Ordinary Differential Equations (ODEs) to be solved with the "Numerical Solver".

Simplification to 1D Linear Acoustics

Simplifications according to the acoustic theory [21] are assumed:

Continuity equation (Acoustics)

$$\frac{\partial \rho}{\partial t} + \rho_0 \frac{\partial u}{\partial x} = 0 \tag{3.25}$$

Momentum equation (Acoustics)

$$\rho_0 \frac{\partial u}{\partial t} + \frac{1}{\rho} \frac{\partial p}{\partial x} = 0 \tag{3.26}$$

Energy equation (Acoustics) – def. speed of sound

$$a_0 = \sqrt{\kappa \cdot R \cdot T_0} \tag{3.27}$$

The linearized equations are suitable for systems, where only small pressure pulsations occur. Thermal effects are neglected.

Due to the linearization, the flux terms can be expressed as a product of multiplication with a constant matrix A (so-called Roe's matrix):

$$q_t + A \cdot q_x = s \tag{3.28}$$

The system is solved with a so-called Riemann solver, also known as the Godunov type method, that shows stability benefits when compared with standard discretization.

Numerical Testing of Pipe Components

Newly developed “complex” and “simplified” pipe components are validated with a so-called “Pipe Shock Test”, which is a discontinuous initial value problem with a known exact solution.

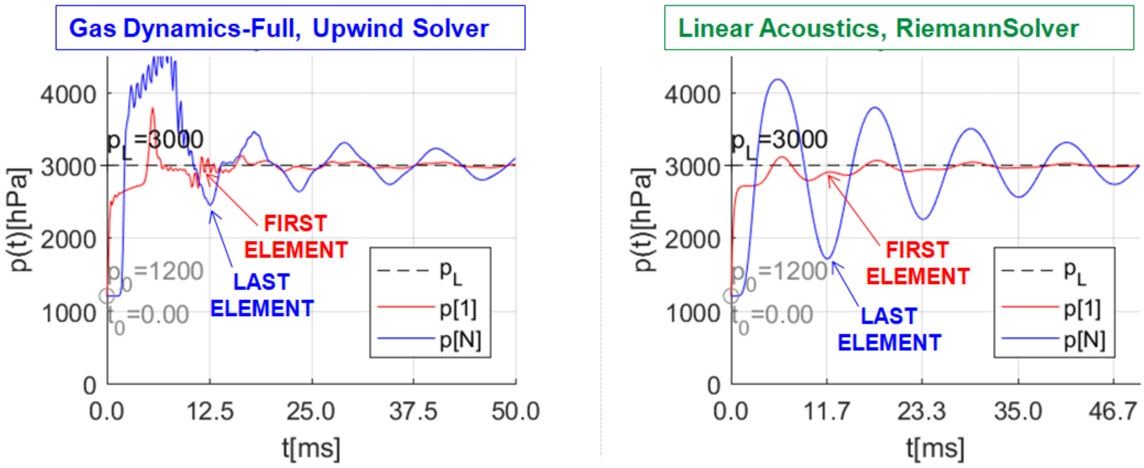

Fig. 2 Time dependent pressure pulsations of first and last pipe elements in ‘Pipe Shock Test’ by using a time step Δt=165µs

The Riemann solver used in the “simplified” pipe provides better stability than the previously used upwind scheme. Therefore, higher critical CFL number can be used. The real-time factor RT=0.72 on 240MHz processor of “complex” method, when applied on 10 discretization elements including variable gas properties, could be reduced to resulting real-time factor RT=0.06 of “simplified” pipe component with constant fluid properties. The overall reduction of real-time factor is 12 times.

4 The Engine Development Platform - Experiment

The investigations were performed on a turbocharged, 1.8 litre four-cylinder gasoline engine. The experimental engine was installed on a test bench with an asynchronous machine was tested under steady-state as well as transient conditions.

Fig. 3 shows the sensor positions during the experiment.

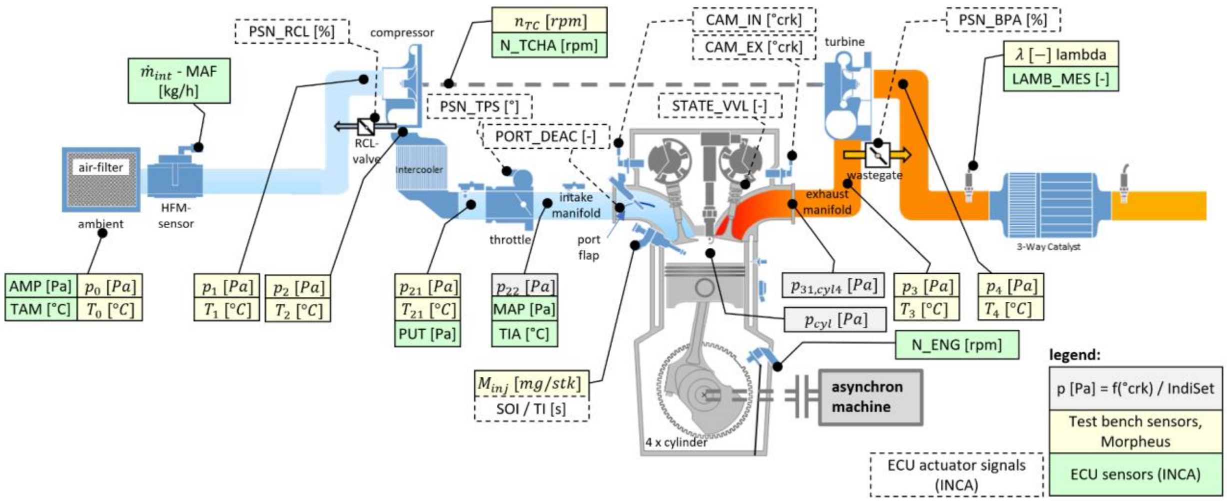

Fig. 3 Sensor positions on engine test bench during experiment (image source: Vitesco Technologies)

Recorded actuator positions define the later model inputs:

- Driver pedal value [%] – just for info, not used directly as input
- Throttle valve actuator position (opening angle [°])
- Boost pressure actuator position ([%] – wastegate opening)
- Intake/Exhaust cam phaser actuator position [°crk]
- Injection event start time and duration [s]

The later model validation is based on the measured in-cylinder air mass in specified engine operating conditions. Intake (p2) and exhaust (p3) pressures are measured with piezo-resistive absolute pressure sensors (type Kistler 4050, Kulite EWCT-312) with a 1° crank angle resolution. Thermocouples on both intake and exhaust side are used to measure gas temperature.

5 SI-Engine Process and Gas Exchange Model

The engine process is described by thermodynamic 1D and 0D simulation, based on newly developed C++, C and MATLAB code libraries.

Model Initialization and Convergence Criteria

The model is seen as being converged (adapted to manifold pressure sensor $p_{22,set} = MAP_{SP}$, when the deviation between measured manifold pressure and simulated manifold pressure is smaller than 0.5%.

Model Accuracy

5.1 Model M1: “detailed 1D model”

Model M1: “detailed 1D model” represents a baseline for the next investigations. The objective is to find a best possible trade-off between model accuracy and real-time capability.

Applied principles of the causal modelling technique, described in the section “Numerical solver” enable a high level of modularity, with a potential to be modified for different engine types and configurations.

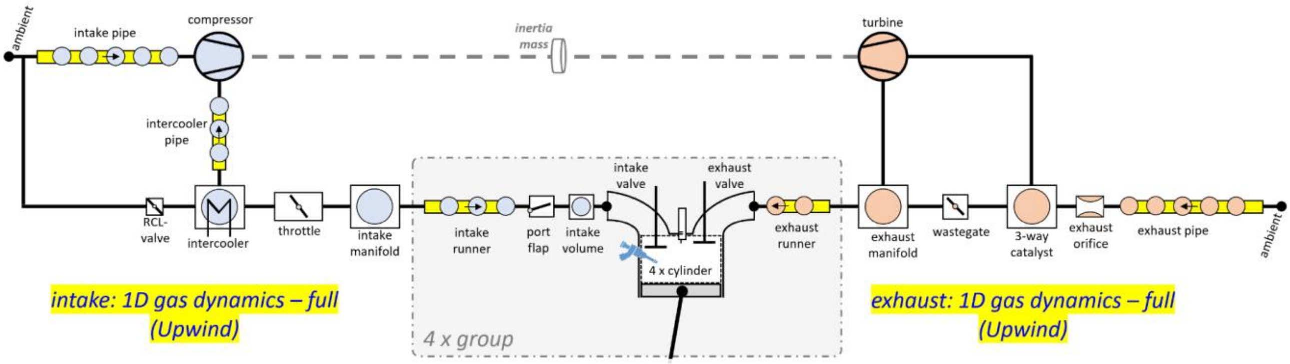

Fig. 4 Layout of detailed 1D model (M1: 234 ODEs, Δt=30µs, RT=41, RMSE=5.3%err)

The model provides good filling accuracy (RMSE=5.3%err, Fig. 8) with some exceptions at scavenging area and at low loads.

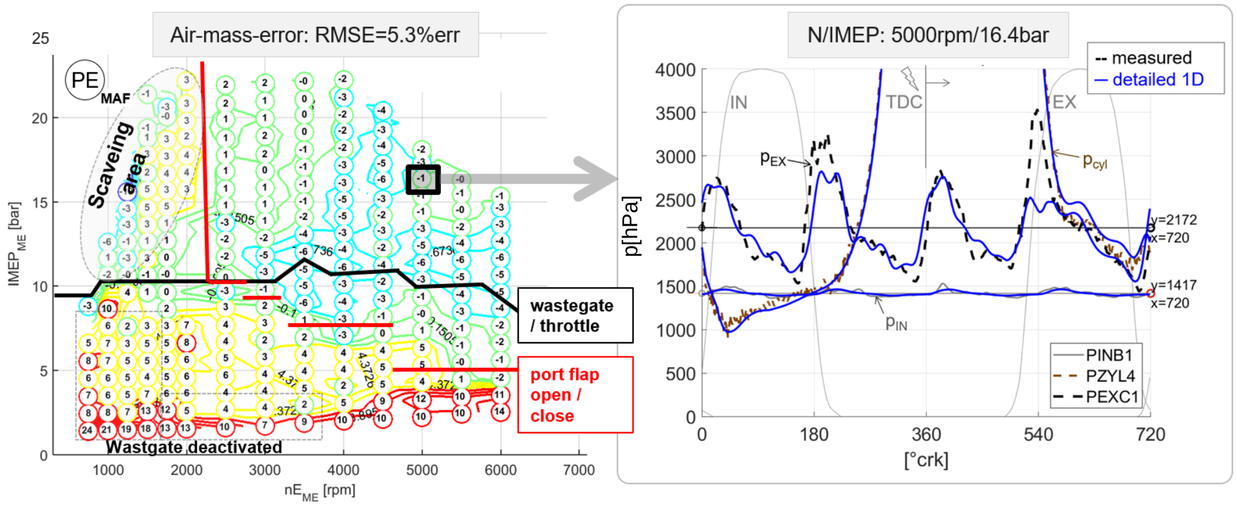

Fig. 5 Left: air mass accuracy of detailed 1D model compared to steady state measurements (M1: 234 ODEs, Δt=30µs, RT=41, RMSE=5.3%err)

Right: intake, cylinder and exhaust pressures compared to experiment

Fig. 8 right demonstrates the model capability to predict the high-frequency phenomena. The crank angle resolved intake, cylinder and exhaust pressures show a very good correlation compared to measurements.

However, the transient 1D flow in pipes shows to be the most time consulting part of the numeric solution. To satisfy the Courant-Lewy-

Friedrichs stability condition [21] [26], a relatively small integration time step $\Delta t = 30\mu s$ is required, yielding a high computational effort. This results in a real-time factor RT=41.

5.2 Model M2: “reduced 1D model”

Different levels of simplification, as described in section “Pipe Systems”, are applied on intake and exhaust side. The objective is to save computational time, while keeping the capability to resolve 1D pressure wave propagation.

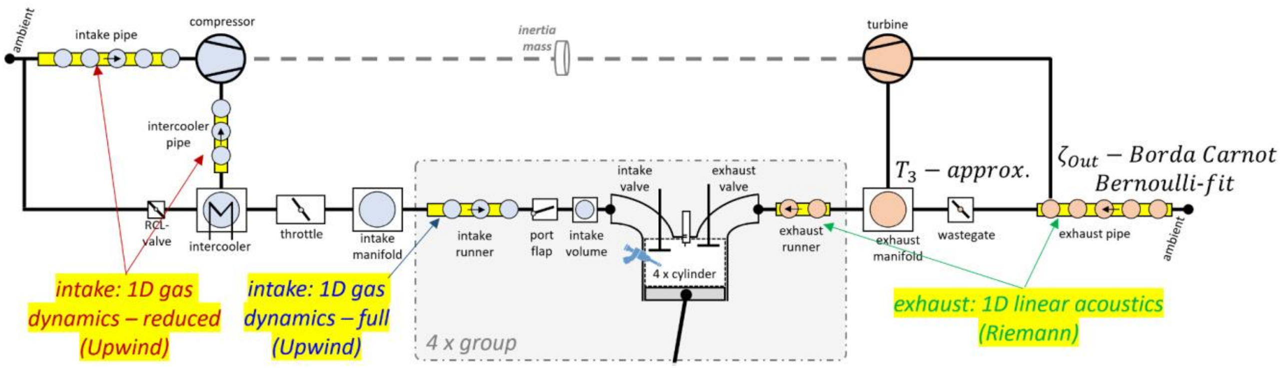

Fig. 6 Layout of the reduced 1D model (M2: 190 ODEs, ∆t=40µs, RT=20, RMSE=5.2%err)

The presented modifications on the exhaust side are provided by a recalibration, based on regressions obtained from steady-state measurements.

Due to the neglection of energy conservation on exhaust side, an assumption for model temperature must be made. The presented method provides reasonable results at steady-state conditions, but transient temperature behaviour cannot be captured.

The overall deviation is quite similar to the previously presented detailed 1D model (RMSE=5.2%err, Fig. 7). Due to presented recalibrations, even some minor improvements are observed.

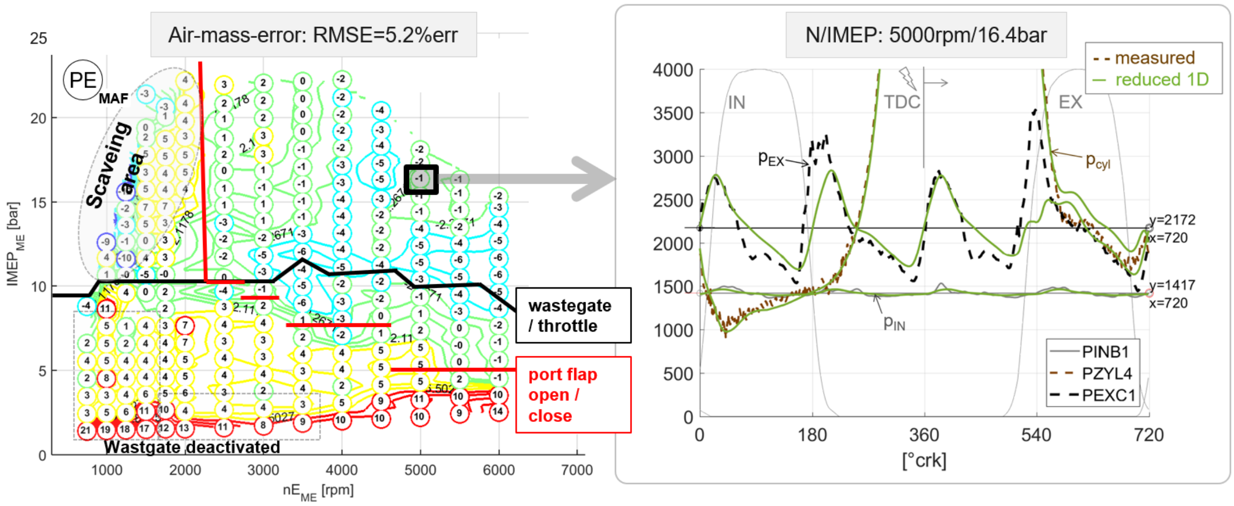

Fig. 7 Left: air-mass accuracy of reduced 1D model (M2: 190 ODEs, ∆t=40µs, RT=20, RMSE=5.2%err)

Right: intake, cylinder and exhaust pressures compared to experiment

The right side of the Fig. 7 shows that the model keeps the capability to capture pressure pulsations with a good correlation to measurements, but higher frequencies disappeared when compared to the detailed model (M1).

Due to the simplifications to linear acoustics and due to the explicit numerical stability of the Riemann’s solver, the real time factor could be reduced by 50% compared to previous model, but is with RT=20 still high.

5.3 Model M3: “fast-running 0D model”

The model is further simplified by removing of the pipe components out from the model layout (see Fig. 11). Remaining components interact based on the principles of the classical 0D filling-emptying method [21].

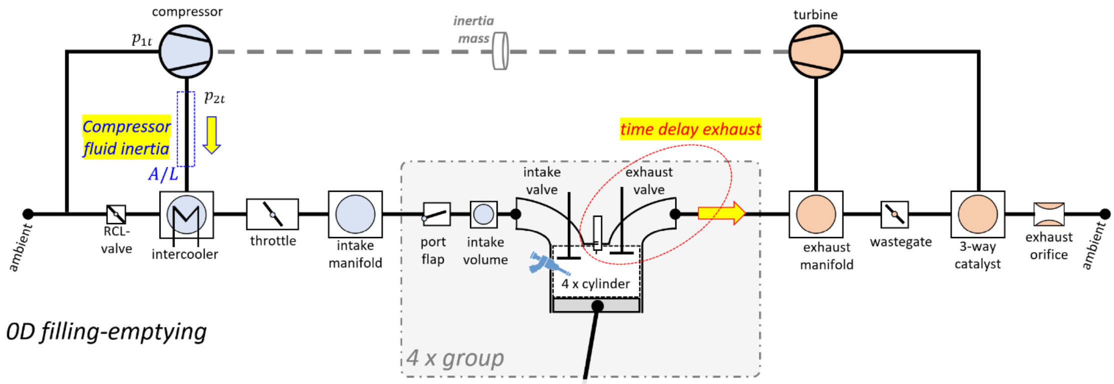

Fig. 8 Layout of fast-running 0D model (M3: 69 ODEs, Δt=300μs, RT=1.7, RMSE=7.4%err)

Nevertheless, the assumption of momentum conservation used within the compressor component (equation (3.14)) provides a certain potential to calibrate the time delay on intake side, and, thus approximate the missing wave propagation. Similarly to the intake side, a component feature of the exhaust valve, acting as the 1st order time delay of a signal (equation (3.12)) is used to calibrate previously neglected time delays due to wave propagation.

Despite of strong simplifications, the overall error RMES=7.4%err (Fig. 9) is still reasonable.

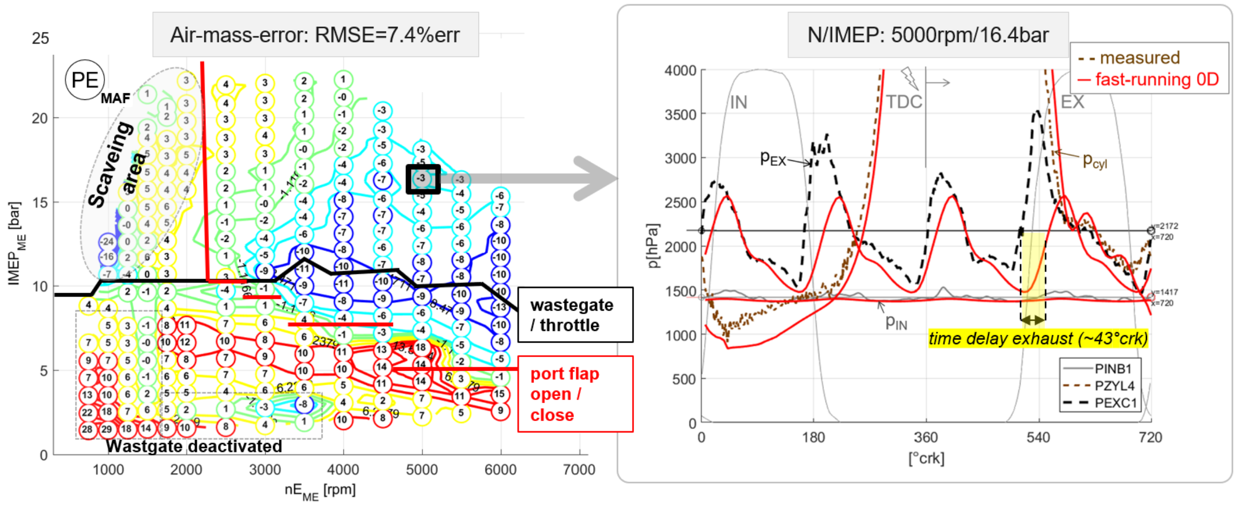

Fig. 9 Left: air-mass accuracy of fast-running 0D model compared to steady state measurements (M3: 69 ODEs, ∆t=300µs, RT=1.7, RMSE=7.4%err)

Right: intake, cylinder and exhaust pressures compared to experiment

Fig. 9 shows on the right side the resulting pressure pulsations of the fast-running 0D model compared to the measurement. The time delayed increase of exhaust pressure p_3 as a result of exhaust valve calibration can be observed. The calibration focus of the used time delay was to provide best possible solution during valve overlap, while compromising of the pressure pulsations seen at the turbine. Therefore, the model still has a relatively good capability to describe scavenging effects, including the prediction of the internal recirculation gas ratio (iEGR).

The neglecting of momentum conservation enables to use a significantly higher integration time step $\Delta t = 300\mu s$. This leads in connection with the reduced number of states to a real-time factor RT=1.7 on the target hardware (ECU with 240MHz).

5.4 Model M4: "reduced fast-running 0D model"

To come closer to the real-time capability, further model reduction is needed. The small intake volume before the cylinder in connection with the port-flap orifice represent the limiting factor from the point of view of the numerical stability. They are removed from the model layout (see Fig. 13).

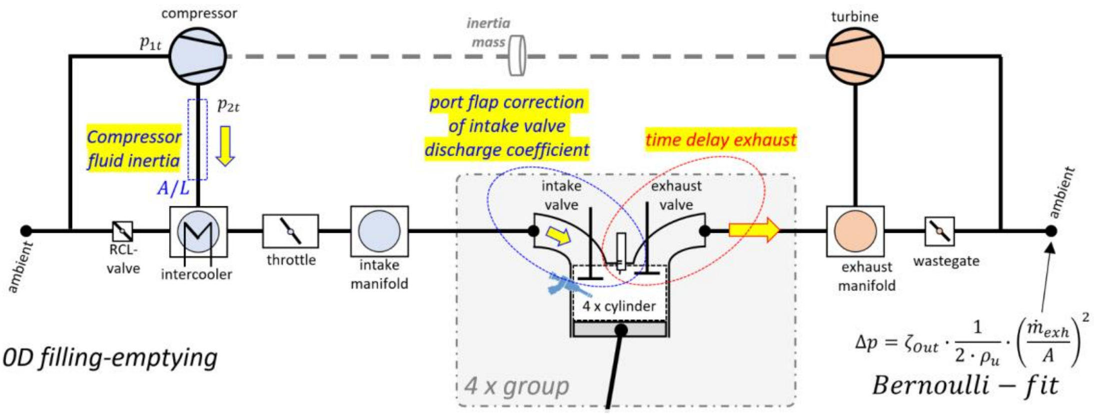

Fig. 10 Layout of reduced fast-running 0D model (M4: 44 ODEs, Δt=300µs, RT=1.1, RMSE=7.4%err)

This leads to a reduced capability of the prediction of cylinder gas composition (iEGR) and related scavenging effects. On the other hand, the neglected flow resistance of the two-stage port flap component can be preserved in the model by a relatively simple assumption. The effective flow area of the removed port flap component is accounted to the inlet valve characteristics. The equation for two resistances in series

$$A^*_{Eff,IN}(\varphi) = \sqrt{\frac{A^2_{Eff,PortFlap} \cdot A^2_{Eff,IN}(\varphi)}{A^2_{Eff,PortFlap} + A^2_{Eff,IN}(\varphi)}} \qquad \textbf{(5.29)}$$

is used to calculate an equivalent effective area [23].

In contrary to the previously shown reduced 1D model, the temperature T_3 is still part of the numeric solution of the energy conservation law in the exhaust manifold volume (no regression is used).

The overall air-mass accuracy of the reduced 0D fast-running model remains with RMSE=7.4%err (Fig. 14) comparable with the previous more complex model. The resulting real-time factor for target hardware is RT=1.1, while using of the Heun's 2nd order integration time method (see also section "Numerical Solver").

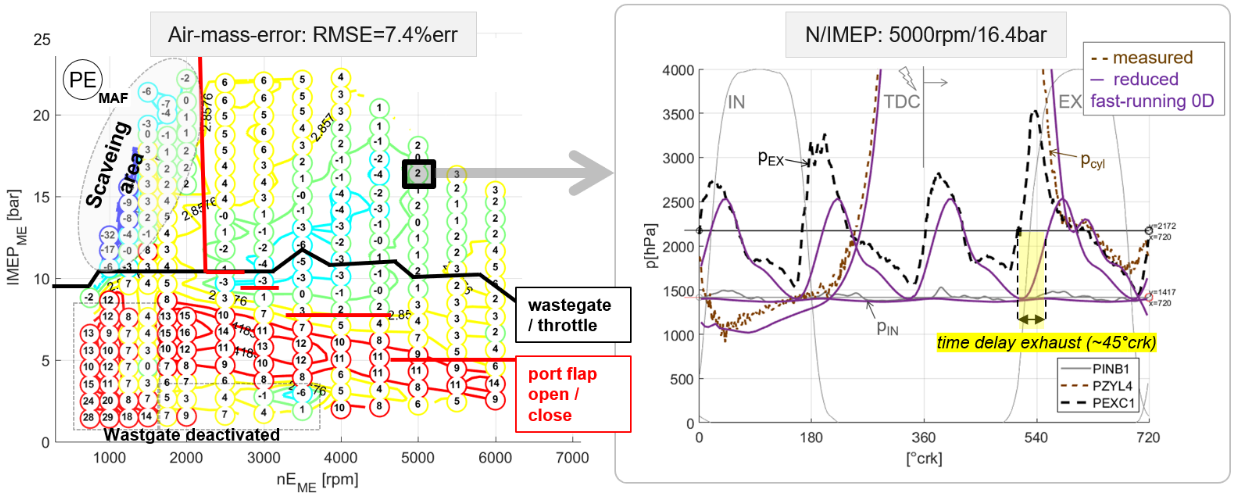

Fig. 11 Left: air mass accuracy of simplified fast-running 0D model compared to steady state measurements (M4: 44 ODEs, ∆t=300µs, RT=1.1, RMSE=7.4%err)

Right: intake, cylinder and exhaust pressures compared to experiment

The real-time capability of above defined models M1, M2, M3, M4 was not yet reached. Therefore, some other measures are investigated to reach RT<1 on the target hardware.

The first proposal to reach the real-time capability with RT=0.6 refers to further reduction of the model complexity. The reduced fast-running 0D model is further simplified by using of a symmetry condition for the first engine cylinder. Only the one of the 4 cylinders is being modelled with differential equations and therefore having an internal state. The last 3 cylinders including intake and exhaust valves are modelled as a time-delayed signal of the first cylinder. Resulting 1-cylinder model (with 3 fictive time-delayed cylinders) with 26 ODEs was successfully implemented and validated in transient engine operating conditions. The results were also published in Mecca [47]. However, the re-evaluation in a wide range of steady-state engine operating conditions and related validation work is out of the scope of current work.

Another engine models with similar level of detail were tested, giving stable results with the use of the Euler's 1st order integration method [47] and thus giving an RT=0.6. This seems to be promising to reach the desired RT<1, but it is also out of the scope of the current work. Additional validations of accuracy and stability would be required.

5.5 Estimation of Real-Time Capability

The target hardware for the model implementation is a multicore ECU used in the serial production. Each of the 3 cores has a processor with the clock frequency of 240 MHz. One core is reserved for the air-path model calculation.

The real-time capability on engine management system is defined as a ratio of turnaround time for model calculation and the ECU sample period, in which the model is supposed to run. The estimation real-time factor can be obtained even before going to real-time hardware, using the offline estimation formula:

$$RT = \frac{n_{R-K}}{\Delta t} \cdot \left(N_{add/sub} \cdot T_{add/sub} + N_{mul} \cdot T_{mul} + N_{div} \cdot T_{div} + N_{cpx} \cdot T_{cpx}\right) \quad \textbf{(5.30)}$$

, where n_{R-K} is the number of Runge-Kutta integration steps and Δt is the integration time step. The evaluation takes into account the necessary computational operations additions / subtractions, multiplications, divisions and complex operations involved in the code.

Fig. 12 shows the comparison of the resulting real-time factors obtained from equation (5.30) for each of the models that were previously validated in a wide range of operation conditions.

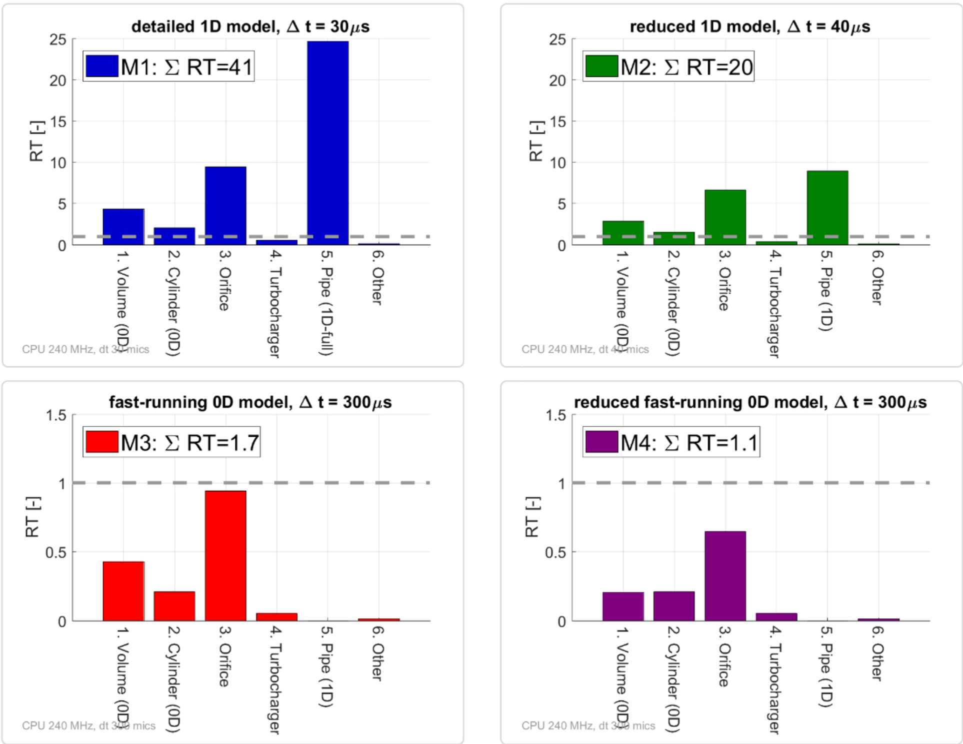

Fig. 12 Comparison of the offline estimated real-time factors on ECU with 240MHz processor for defined models, solved by the Heun's integration method (2nd order Runge-Kutta)

6 Conclusions

Modular, physical-based model and simulation environment was implemented based on the principles of causal modelling approach (see also "Numerical Solver"). The model provides crank angle resolved information on engine in-cylinder gas mixture and charge exchange including performance of a turbocharger (see also "Objective"). Fig. 16 illustrates the accuracy results and the real-time performance on the target hardware dependent on the complexity. The model complexity is expressed as the number of solved differential equations (ODEs).

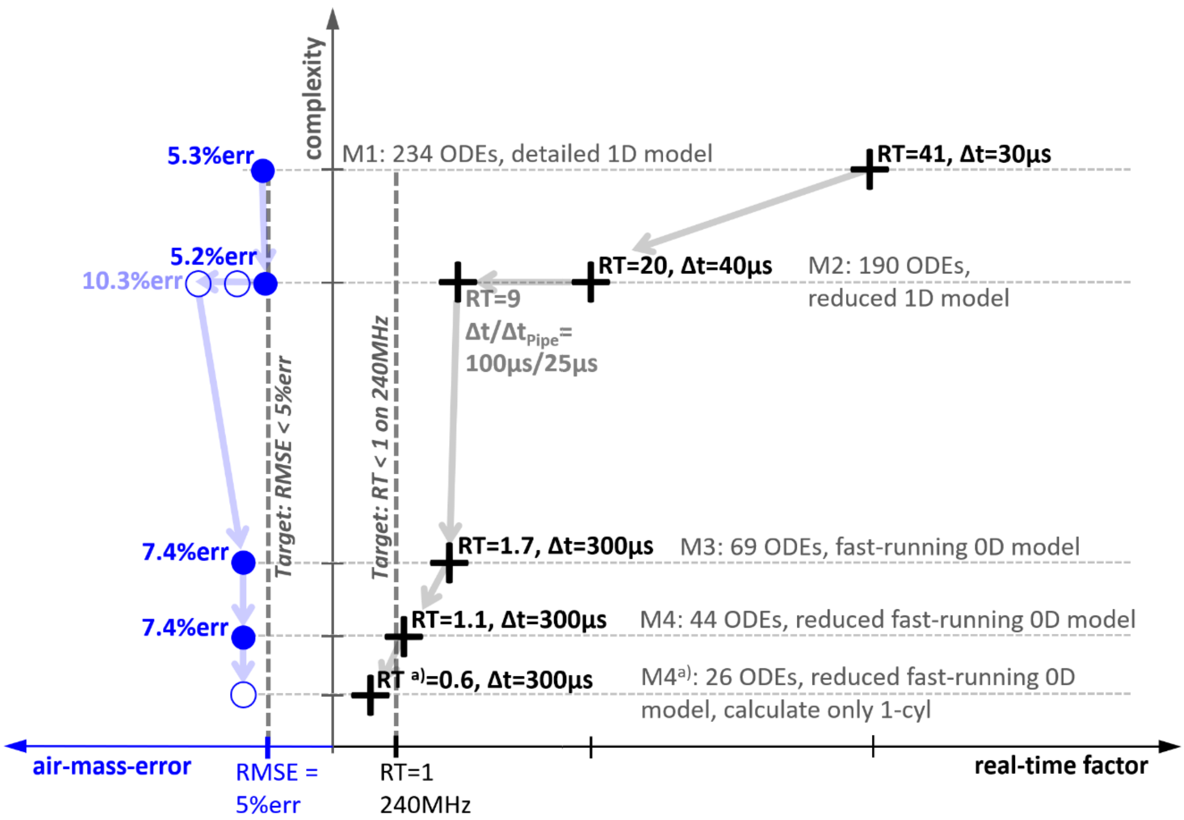

Fig. 13 Air-mass-error and real-time factors on production ECU (240MHz) in dependence on model complexity

The detailed 1D model provides overall reasonable accuracy on different time scales. The model requires, however, a small simulation time step of Δt=30µs to satisfy the CFL stability condition and has therefore high real-time factor RT=41.

To keep the model ability of resolving pressure wave propagation while reducing the CPU time, the 1D flow in pipe components was strongly simplified. Thanks to the higher stability of the Reimann based solver, the integration time step can be increased to Δt=40µs. This leads together with the reduced number of solved equations to resulting real-time factor RT=20 of reduced 1D model. The model accuracy RMSE=5.2%err shows even a minor benefit compared to detailed 1D model. However, proposed simplifications restrict the model validity to steady-state conditions.

The neglecting of the pipe components leads to an increase of the overall air-mass-error of the fast-running 0D model to 7.4%err. Because of the fact that one of the development objectives was the reduction of the calibration effort for model configurations, no locally valid calibrations are used. On the costs of the model accuracy, the integration time step can be significantly increased to Δt=300µs. This leads in combination with the reduced number of solved differential equations to a real-time factor RT=1.7.

Further simplifications to 44 solved ODEs of the defined reduced fast-running 0D model provide a real-time factor RT=1.1, close to real-time capability on the target production ECU with 240MHz clock frequency. The influence of engine port flap on the flow resistance before cylinder inlet was accounted to the intake valve effective area without decreasing of model accuracy. The deviations go to the bill of the large volumes used for discretization of (in reality complex) intake and exhaust manifolds.

Finally, one solution to reach the real-time capability on target hardware was found. The reduction of solved system to 26 ODEs by calculating only the first cylinder while assuming a symmetry condition for the last three cylinders allows a real-time factor RT=0.6. The model shows same accuracy like the previously defined model with 44 ODEs, when tested at selected individual stationary operating points. The model was successfully validated under transient operating conditions, using identical inputs from the engine actuators like the used engine's OEM ECU with serial calibration.

The proposed real-time capable model shows a good agreement with the measurements obtained from at individual stationary points and during the transitions (fresh in-cylinder air mass and turbocharger speed). Beside this, the model provides high quality information on each individual engine cycle such as internal recirculation ratio, indicated mean effective pressure and other relevant engine performance indicators without further increase of the model complexity for each individual output value (in contrary to classical data driven models). The comparison of modelled intake and exhaust pressures with high-sampled indication system shows that the model captures correctly the dynamic effects on different time scales from low frequency engine transitions to the crank angle resolved pressure pulsations. This represents a benefit, for example in comparison to classical mean value models.

The presented 1D models represent a good baseline for calibrating the derived fast-running 0D models. On the intake side, the first order time delay defined in compressor model can be used to calibrate previously neglected transport effects. Due to the used physically motivated assumption, based on the momentum conservation, proposed calibration parameters are physically interpretable (see equation (3.14)). Similarly to the intake side, a first order lag behaviour of the exhaust valve is used to calibrate previously neglected transport delay effects (see equation (3.12)).

7 Bibliography used in this Thesis

[1] **Merker, G. P., Schwarz, Ch., Stiesch, G., Otto, F.** *Grundlagen Verbrennungsmotoren; Simulation der Gemischbildung, Verbrennung, Schadstoffbildung und Aufladung.* Wiesbaden : Vieweg + Teubner, 2009. pp. 189-354, ISBN 978-3-8348-0740-3.

[2] **Beckmann, R.** *Beitrag zur exakten Füllungssteuerung am aufgeladenen Ottomotor.* Dissertation, Deutschland : Logos Verlag Berlin, Universität Rostock, 2015. pp. 1-6 and pp.96, ISBN 978-3-8325-3930-6.

[3] **Coraddu, A., Kalikatzarakis, M., Theotokatos, G., Geertsma, R. and Oneto, L., Book: Kumar A., A., Kumar, D., Sharma, N., Sonawane, N.,.** *Chapter 6. Physical, data-driven and hybrid approaches to model engine exhaust gas temperatures in operational conditions, Engine Modeling and Simulation.* Chapter 6: Delft-Netherlands, Book: Kanpur-India : Springer Nature Singapore, 2021. pp. 145-187, ISBN 978-981-16-8617-7.

[4] **Ludwig, O.** *Eine Möglichkeit zur echtzeitfähihen, physikalisch-basierten Motorprozessanalyse auf der Grundlage zeitlich fusionierter Messdaten.* Dissertation, Hamburg : Logos Verlag Berlin, Helmut-Schmidt-Universität / Universität der Bundeswehr Hamburg, 2011. pp. 55-174, ISBN 978-3-8325-2792-1.

[5] **Isermann, R.** *Engine Modeling and Control, Modeling and Electronic Management of Internal Combustion Engines.* Germany : Springer-Verlag Berlin Heidelberg, TU Darmstadt, 2014. pp. 5-25, ISBN 978-3-642-39933-6.

[6] **Macek, J., Polasek, M., Sika, Z., Valasek, M., Florian, M., Vitek, O.** *Transient Engine Model as a Tool for Predictive Control.* Detroit : SAE Technical Paper 2006-01-0659, Czech Technical University in Prague, 2006. pp. 1-17, doi:10.4271/2006-01-0659.

[7] **Kainz, J., Beer, J., Bänfer, O., Nelles, O.** *Einsatz von lokalen Modell-Netzen in einer Motorsteuerung zur Modellierung von Ventiltriebsvariabilitäten.* Essen : Congress "Haus der Technik" Variable Ventilsteuerung, March 3rd-4th, 2009. pp. 3-7.

[8] **Kainz, J., Beer, J., Banfer, O., Nelles, O.** *Local Model Networks, "The Prospective Method for Modeling in Electronic Control Units", ATZ Elektron Worldw 3.* Frankfurt am Main : November 2008ATZelektronik worldwide 3(6), 2008. pp. 36-39, doi:10.1007/BF03242200.

[9] **Roesler, C.** *Echtzeitfähiges physikalisches Motorprozessmodell - Potenziale für die Steuerung eines Pkw-Ottomotors.* Dissertation Berlin : Logos Verlag, Technical University of Berlin, 2013. pp. 71-96 and pp. 163-165, ISBN 978-3-8325-3359-5.

[10] **Friedrich, I.** *Motorprozess-Simulation in Echtzeit - Grundlagen und Anwendungsmöglichkeiten.* Dissertation, Berlin : Sharker Verlag Aachen, TU-Berlin, 2008. pp. 1-73 and pp. 92, ISBN 987-3-8322-7019-3.

[11] **Macek, J., Vitek, O.** *Determination and Representation of Turbocharger Thermodynamic Efficiencies.* USA : SAE Technical Paper 2016-01-1042, Czech Technical University in Prague, 2016. pp. 1-12, doi:10.4271/2016-01-1042.

[12] **Kouba, A., Navratil and J., Hnilicka, B. in Book: Liebl J., Beidl C. (eds).** *Engine Control using a Real-Time 1D Engine Model, VPC – Simulation und Test 2015, .* Wiesbaden, Germany : Springer Vieweg, Springer Nature, 2018. pp. 295-309, ISBN 978-3-658-20735-9.

[13] **Navratil, J., Hnilicka, B., Rawnsley, S.** *WAVE-RT Real Time 1-D Model Running on Engine Controller.* North American User Conference: Tools & Technologies to Meet CAFE Standards 17th April 2013 : Ricardo, 2013. pp. 1-26, www.ricardo.com/wave-rt.

[14] **Ruggiero, A., Montalto, I., Poletto P., Pautasso, E., Mustafaj, K., Servetto E.** *Development and assessment of a Fully-physical 0D Fast Running Model of an E6 passenger car Diesel engine for ECU testing on a Hardware-in-the-loop system.* Torino, Italy : International Congress SIA Powertrain - Rouen, May 21st-22th, SIA Technical Paper, 2014. pp. 1-10, Code R-2014-02-09.

[15] **Touimi, Z. B., Gaid, M. B., Bohbot, J., Dutoya, A.** *From Physical Modeling to Real-Time Simulation: Feed back on the use of Modelica in the engine control development toolchain.* Dresden & Rueil-Malmaison : Proceedings 8th Modelica Conference, March 20th-22th, pp. 763-771, 2011.

[16] **Jakovlev, S.** *Echtzeitfähige, kurbelwinkelaufgelöste Simulation des Motorstarts.* Dissertation, Braunschweig : Logos Verlag Berlin, Technische Universität Carolo-Wilhelmina zu Braunschweig, Wolkswagen AG, 2014. pp. 1 and pp. 47, ISBN 978-3-8325-3825-5.

[17] **Papadimitriou, I., Warner, M., Silvestri, J., Lennblad, J., Tabar, S.** *Neural Network Based Fast-Running Engine Models for Control-Oriented Applications.* Detroit, Michigan : SAE Technical Paper 2005-01-0072, 2005. pp. 1-9, doi:10.4271/2005-01-0072.

[18] **Sommer, R., Halfmann, T., Broz, J.** *Automated behavioral modeling and analytical model-order reduction by application of symbolic circuit analysis for multi-physical systems: Simulation Modelling Practice and Theory.* Ilmenau, Germany : IMMS Institut für Mikroelektronik- und Mechatronik-Systeme gemeinnützige GmbH, Frauenhofer Institute, 2008. pp. 1024–1039, doi:10.1016/j.simpat.2008.04.012.

[19] **Heywood, J. B.** *Internal combustion engines fundamentals.* New York : McGraw-Hill, 1989. pp. 204-304 and pp. 748-764, ISBN 0-07-08637-X.

[20] **Grill, M.** *Objektorientierte Prozessrechnung von Verbrennungsmotoren.* Dissertation, Deutschland : OPUS - Publication Server of the University of Stuttgart, Institut für Verbrennungsmotoren und Kraftfahrwesen der Universität Stuttgart, 2006. pp. 193-230, doi:10.18419/opus-4076.

[21] **Pischinger, R., Klell, M., Sams, T.** *Thermodynamik der Verbrennungskraftmaschine.* Wien : Springer Verlag Wien, 2009. pp. 1-68 and pp. 309-331, ISBN 978-3211-99276-0.

[22] **Woschni, G.** *A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine.* Maschinenfabrik Augsburg - Nürnberg AG : SAE Technical Paper 670931, 1967. pp. 1, doi:10.4271/670931.

[23] **Zinner, K., Puscher, H.** *Aufladung von Verbrennungsmotoren - Grundlagen, Berechnungen, Ausführingen.* Berlin, Heilderberg : Springer-Verlag, 4. Auflage, 2021. pp. 26-169, ISBN 978-3-642-28989-7.

[24] **Berndt, R.** *Einfluss eines diabaten Turboladermodells auf die Gesamtprozesssimulation abgasturboaufgeladener PKW-Dieselmotoren.* Berlin : Dissertation, Fakultät V - Verkehrs- und Maschinensysteme, Technische Universitat Berlin, 2009. pp. 24-37 and pp. 53-58, doi:10.14279/depositonce-2313.

[25] **Wesseling, P.** *Principles of computational fluid dynamics.* Heidelberg : Springer, 2001. pp. 339-445, ISBN 978-3-642-05145-6.

[26] **LeVeque, R. J.** *Finite volume methods for hyperbolic problems.* United Kingdom : The University of Cambridge, 2004. pp. 1-85 and pp. 291-309, ISBN 0-511-04219-1.

1 Bibliography, other References

Following references were additionally used within the disertation thesis:

[27] **Stockar, S., Canova, M., Guezennec, Y., Torre, A. D., Montenegro, G., Onorati, A.** *Modeling Wave Action Effects in Internal Combustion Engine Air Path Systems: Comparison of Numerical and System Dynamics Approaches.* s.l. : International Journal of Engine Research 14(4): pp. 391-408, 2013. doi:10.1177/1468087412455747.

[28] **Mahle GmbH.** *Kapitel 6 Variabilitäten im Ventiltrieb, ATZ/MTZ Fachbuch, Ventiltrieb – Systeme und Komponenten.* Stuttgart : Springer Vieweg, 2013. pp. 145-199, ISBN 978-3-8348-2490-5.

[29] **Basshuysen, R., Schäfer, F.** *Handbuch Verbrennungsmotor - Grundlagen, Komponenten, Systeme, Perspektiven.* Hamm, Deutschland : Springer Vieweg, 7. Auflage, 2002-2015. pp. 676, ISBN 978-3-658-04677-4.

[30] **Wurzenberger, J. C., Heinzle, R. , Schuemie, A. , Katrasnik, T.** *Crank angle resolved real-time engine simulation - integrated simulation tool chain from office to testbed.* USA : SAE Technical Paper 2009-01-0589, 2009. pp. 112, doi:10.4271/2009-01-0589.

[31] **Kouba, A., Navratil, J., Hnilicka, B., and Niven, P.** *Sensorless Control Strategy Enabled by a Sophisticated Tool Chain.* USA : SAE Technical Paper 2015-01-2847, 2015. pp. 1-6, doi:10.4271/2015-01-2847.

[32] **Cosadia, I., Silvestri, J., Papadimitriou, I., Maroteaux, D., Obernesser, P.** *Traversing the V-Cycle with a Single Simulation - Application to the Renault 1.5 dCi Passenger Car Diesel Engine.* USA : SAE Technical Paper 2013-01-1120, 2013. pp. 1-10, doi:10.4271/2013-01-1120.

[33] **Sellnau, M., Sinnamon, J., Oberdier, L., Dase, C., Quillen, M. V. K., Silvestri, J., Papadimitriou, I.** *Development of a Practical Tool for Residual Gas Estimation in IC Engines.* USA : SAE Technical Paper 2009-01-0695, 2009. pp. 1-10, doi:10.4271/2009-01-0695.

[34] **Ciesla, C., Keribar, R., Morel, T.** *Engine/Powertrain/Vehicle Modeling Tool Applicable to All Stages of the Design Process, also in: Electronic Engine Control Technologies-PT-73, Electronic Engine Controls 2000: Modeling, Neural Networks.*

USA : SAE Technical Paper 2000-01-0934, Gamma Technologies US, 2000. pp. 151-163, doi:10.4271/2000-01-0934.

[35] **Reine, A., Thomas, V., Manz, A.** *Particle Reduced, Efficient Gasoline Engines, Report on new prediction capabilities over regulatory and real driving emissions.* Germany : EUROPEAN COMMISSION, 2020. pp. 1-18, GA # 723954.

[36] **Cipollone, R., Sciarretta, A.** *A New Modeling for the Air and Gas Dynamics in ICE Manifold Oriented to Air-Fuel Ratio Control.* 1999. pp. 1, ASME Paper 99-ICE-170.

[37] **Hendricks, E., Chevalier, A., Jensen, M., Sorenson, S., Trumpy, D. and Asik, J.** *Modelling of the Intake Manifold Filling Dynamics.* Denmark : SAE Technical Paper 960037, Technical University of Denmark, 1996. pp. 1, doi:10.4271/960037.

[38] **Cipollone, R., Sciarretta, A.** *The Quasi-Propagatory Model: A New Approach for Describing Transient Phenomena in Engine Manifolds.* L'Aquila : SAE Technical Paper 2001-01-0579, University of L'Aquila, 2001. pp. 55-63, doi:10.4271/2001-01-0579.

[39] **Cavina, N., Migliore, F., Carmignani, L. and Di Palma, S.** *Development of a Control-Oriented Engine Model Including Wave Action Effects.* Italy : SAE Technical Paper 2009-24-0107, University of Bologna, Piaggio & C. S.p.A, 2009. pp. 1-12, doi:10.4271/2009-24-0107.

[40] **Cipollone, R., Martella, L., Scarpone, L. and Valente, R.** *A New Modeling to Predict the Fluid Dynamic Transient Phenomena in Ice Ducts.* L'Aquila : SAE Technical Paper 2008-01-2389, University of L'Aquila, 2008. pp. 1-8, doi:10.4271/2008-01-2389.

[41] **Hiroschi, A., MathWorks, Inc (1994-2022).** *1-D data interpolation (table lookup): User's Guide (r2021b).* s.l. : Retrieved February 14th, 2022 from. pp. 1, https://de.mathworks.com/help/matlab/ref/interp1.html.

[42] **Scharf, J. S.** *Extended Turbocharger Mapping and Engine Simulation.* Germany : Dissertation, Lehrstuhl für Verbrennungskraftmaschinen und Institut für Thermodynamik, RWTH Aachen, 2010. pp. 29, Interne Identifikationsnummern: RWTH-CONV-125328, Datensatz-ID: 63937.

[43] **Schreiber, G.** *Untersuchung von Verbesserungspotentialen hinsichtlich Verbrauch und Drehmoment bei Ottomotoren mit Hilfe 1-dimensionaler Simulationsrechnung.* Deutschland : Dissertation, Technische Universität

Kaiserslautern, 2006. pp. 122-134, Intene Diss. Nr. D 386, urn:nbn:de:hbz:386-kluedo-19526.

[44] **Borean, F., Message, S., Morgan, Ch., Neaves, B., Slaney, T.** *2020-2025 Challenges for Combustion Control in Automotive World: Hybridization, Emissions and Key Enablers.* Jaguar Land Rover, Coventry, UK : Symposium for Combustion Control, June 15th-16th, pp. 145-148, Aachen, 2016.

[45] **Follen, K. M.** *A system dynamics modeling metodology for compressible fluid flow systems with applicatins to internal combustion engines.* USA : Dissertation, The Ohio State University, 2010, pp. 5-40.

[46] **Gustavsson, E.** *Modelling of Wave Propagation in Combustion Engines.* Lund : Master thesis, Lunds Universitet, 2014. pp. 25-93, ISSN 0282-1990.

2 Authors Publications

[47] **Fortl, J., Macek, J.** *Turbocharger Modelling for Fast-Running Transient 0D ICE Simulations.* Prague : MECCA Technical Paper, submitted for review on February 22nd, 2022. pp. 1-18.

[48] **Fortl, J., Beer, J., Keller, J, Macek, J. and Borchsenius, F.** *Physical Model of SI-Engine Process and Gas Exchange for Real-Time Implementation in Engine Management System.* Prague : MECCA Technical Paper, 2018. pp. 11-17, 10.1515/mecdc-2018-0006.

[49] **Fortl, J.** *Presentaion of "Physical Model of SI-Engine Process and Gas Exchange for Real-Time Implementation in Engine Management System" in context of Real-Time Applications.* RWTH Aachen University, Germany : Symposium for Combustion Control, June 28th-29th, 2017. public speach with presentation, pp. 1-22.