

Fast Simulation of Wave Action in Engine Air Path Systems Using Model Order Reduction

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ABSTRACT

Engine downsizing, boosting, direct injection and variable valve actuation, have become industry standards for reducing CO₂ emissions in current production vehicles. Because of the increasing complexity of the engine air path system and the high number of degrees of freedom for engine charge management, the design of air path control algorithms has become a difficult and time consuming process. One possibility to reduce the control development time is offered by Software-in-the-Loop (SIL) or Hardware-in-the-Loop (HIL) simulation methods. However, it is significantly challenging to identify engine air path system simulation models that offer the right balance between fidelity, mathematical complexity and computational burden for SIL or HIL implementation.

In this paper, a new modeling approach is presented to predict the performance of the exhaust system of a downsized boosted engine, including the ability to predict the influence of pressure wave propagation on the exhaust flows and turbine performance. A key feature of the proposed model is the use of a model order reduction technique that approximates the conservation laws in partial differential equation form as a reduced-order system of nonlinear ordinary differential equations. The result is a model that can be easily implemented into SIL or HIL simulation tools as a “virtual engine” and offers significantly lower computational requirements compared to 1D gas dynamic simulators.

To evaluate the accuracy of the reduced-order model, simulation results are compared against the industry standard tool used for engine performance simulation (GT-Power), for a range of different operating conditions. A set of metrics to quantify the accuracy and computation time of the model is proposed and used to perform a complete analysis.

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INTRODUCTION

The increasingly stringent government mandates on fuel economy and CO₂ emissions reduction have pushed the automotive industry to the widespread adoption of IC Engine downsizing, turbocharging, direct injection and flexible valve actuation technology as a standard solution for fuel economy improvement of passenger cars and light-duty trucks [1,2,3,4,5]. Extreme downsizing, which couples a dramatic reduction of the engine displacement and number of cylinders to advanced boosting technologies to achieve very high BMEP, has been recently proposed to further extend the benefits on fuel economy and drivability [6, 7]. Such evolution in the engine design has significantly increased the complexity of the air path system, particularly introducing a large number of degrees of freedom to control the cylinder charge, as well as its thermodynamic condition and composition. Advanced control algorithms for the next

generation of downsized boosted engines will require knowledge of the pressure at different locations within the air path system, and during specific events within the engine cycle. This information will enable control engineers to design more precise algorithms to control the cylinder charge and its composition, ultimately with benefits on engine performance, fuel consumption, and emissions. Furthermore, Software-in-the-Loop (SIL) and Hardware-in-the-Loop (HIL) simulation methods present the opportunity to significantly reduce the development time and costs associated with engine control system design, calibration and verification processes, and ultimately shorten the time to deploy advanced powertrain technologies into production. In particular, SIL simulation offers the opportunity to conduct the calibration process in a virtual environment before transitioning to an engine test bench. In this case, the compiled engine controller software is incorporated into a mathematical simulation tool that

contains a physical model of the complete engine system. To fully benefit from this development tool, the plant model used in the SIL simulation must be characterized by high fidelity physical consistency with the actual system. Specifically in the case of engine air path system control, the “virtual plant” should provide a physics-based, high-frequency prediction of the pressure at different locations within the intake and exhaust manifolds. In addition, the model describing the wave action effects must be implemented such that it can be coupled with the existing charge prediction models currently used by the Industry. On the other hand, such requirements go beyond the predictive ability of the control-oriented models that are currently in use for engine air path control design verification. Specifically, the ability to predict pressure at multiple locations within the air path and with a high time resolution requires the ability to predict the wave propagation dynamics in the manifolds, which is currently available only in high-fidelity engine performance simulation software, such as GT-Power [8]. Conversely control-oriented models are typically based on low-frequency approximated lumped-parameter models (also known as Mean-Value Engine Models), and require extensive calibration on engine dynamometers and in vehicle to achieve sufficient accuracy in predicting the cylinder charge and cycle-averaged variables [9,10,11,12].

Traditionally, gas dynamic models are based upon the mass, energy and momentum equations in one-dimensional form, which are solved by discretizing the engine air path system geometry and applying numerical methods [13]. Several 1D gas dynamic modeling tools are commercially available, principally targeted to engine design optimization and performance simulation [14, 15]. On the other hand, there are fundamental limitations of current 1D gas dynamic models that limit their application to SIL or HIL simulation, and ultimately to control algorithm prototyping and verification. Some of the existing commercial products, such as GT-Power, do allow for co-simulation of model and controller in different environments (for instance, Matlab/Simulink [16]), which enables for control verification. However, there is a significant tradeoff between simulation time and model fidelity.

This paper presents a novel approach for high-fidelity, fast simulation of a turbocharged multi-cylinder engine based upon a predictive gas dynamic model. The tool is based upon a model order reduction methodology developed by the Authors that approximates the governing equations (conservation laws in partial differential form) onto a set of low-order nonlinear ordinary differential equations. This approach leads to a Reduced-Order Model (ROM) that is readily implementable into SIL and HIL simulation tools. The model order reduction methodology was originally proposed to characterize nonlinear wave action phenomena in one-dimensional ducts [17], and subsequently applied to a single cylinder engine simulation with experimental verification [18, 19]. This paper presents the first-ever application of the aforementioned methodology to a downsized, turbocharged multi-cylinder engine. The focus of this work is the characterization of the exhaust system, with the objective of capturing the influence of the engine air path system actuators (waste-gate position and variable valve timing), and of the engine operating conditions on the pressure waves propagation through the exhaust system, and ultimately on the cylinder charge composition and turbine inlet conditions.

DESCRIPTION OF THE ENGINE

The engine considered in this work is a downsized boosted 4-cylinder engine with Variable Valve Timing and waste-gate valve. As mentioned above, this paper focuses on characterizing the performance of the exhaust system, whose layout is illustrated in Figure 1.

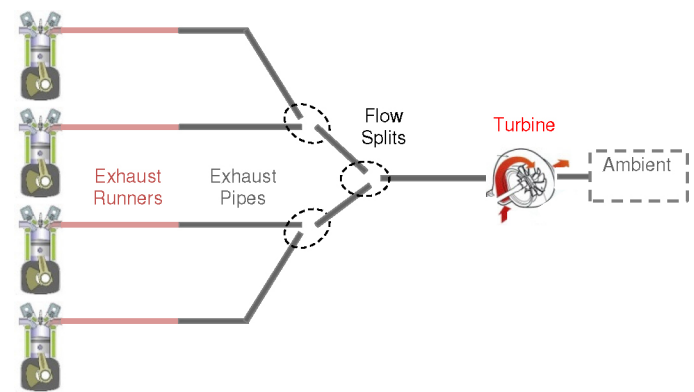


Figure 1. Layout of the 4-Cylinder Engine.

Characterizing the pressures and flow rates throughout the exhaust system in the crank angle domain poses several challenges compared to the intake side. For instance, the higher in-cylinder pressure values at exhaust valve opening (EVO) excite the exhaust system causing large fluctuations of the pressure and temperature, which affect the turbine inlet conditions in a highly nonlinear fashion. Furthermore, higher exhaust gas temperatures increase the local speed of sound, which is a known problem in gas dynamic simulation tools that may decrease the accuracy of the solution, or lead to computational issues. Finally, the presence of complex boundary conditions due to the turbine and waste-gate valve, introduces a challenge in correctly capturing the wave interactions and coupling at that location.

In this paper, model prediction of the thermo-fluid properties at different locations in the exhaust is compared against the industry standard tool used for engine air path system performance simulation (GT-Power), which is considered here as a benchmark for evaluating the accuracy and computational complexity of the proposed model.

The GT-Power model was previously validated against experimental data.

The exhaust system is characterized using both models, with reference to several engine steady-state conditions that cover the entire operating range. In addition to the crank-angle domain responses, integral metrics are also considered in the analysis, specifically to evaluate the ability of the model to quantify backflows through the exhaust valve, and blow-through across the valves. The ability to accurately and reliably predict such phenomena, critical for the charge exchange and combustion process, is important for SIL and HIL simulation tools used to verify cylinder charge control strategies.

While this paper focuses on comparing the proposed model against simulation results, a thorough experimental validation of the modeling approach was previously conducted in [19], showing the

ability to accurately predict the pressure wave propagation and overall performance of a single-cylinder engine air path system, including intake, cylinder, exhaust and muffler.

DESCRIPTION OF THE MODELING APPROACH

The major drawbacks of using 1D gas dynamic models for control algorithm calibration or verification is the significant computational effort, and the use of numerical (algebraic) solution methods that limit direct implementation into SIL and HIL environments. For this reason, a fast simulation tool for wave action phenomena was first proposed in [17]. The approach relies on a projection method to reduce the nonlinear hyperbolic PDEs that govern the wave action dynamics in engine manifolds into a set of nonlinear ODEs implementable in fast simulation tools.

This section summarizes the governing equations and outlines the model order reduction approach. Additionally, the derivation of the boundary conditions required for this case study is shown. A rigorous characterization of the boundary conditions is a critical step in the model development, as the ability of gas dynamic models to correctly predict the wave action dynamics depends largely on the way boundary conditions are described and coupled with the pipe elements. For a more detailed description of the derivation process of the reduced-order model, the reader is invited to refer to [17].

Pipe (One-Dimensional) Elements

Wave propagation phenomena in engine manifolds are generally predicted using the 1D Euler equations for compressible flows [13], which are generally expressed as a set of three coupled nonlinear hyperbolic PDEs in matrix form:

$$A \frac{\partial \mathbf{W}(x,t)}{\partial t} + \frac{\partial \mathbf{F}(\mathbf{W})}{\partial x} + \mathbf{B}(\mathbf{W}) = 0 \quad (1)$$

where:

$$\mathbf{W}(x,t) = \begin{bmatrix} \rho \\ \dot{m} \\ \rho e_0 \end{bmatrix}$$

$$\mathbf{F}(\mathbf{W}) = \begin{bmatrix} \dot{m} \\ (\gamma-1)A^2 \rho e_0 - \frac{\gamma-3}{2} \frac{\dot{m}^2}{\rho} \\ \gamma \frac{\dot{m}}{\rho} \rho e_0 + \frac{\gamma-1}{2A^2} \frac{\dot{m}^3}{\rho^2} \end{bmatrix}$$

$$\mathbf{B}(\mathbf{W}) = \begin{bmatrix} 0 \\ \frac{\dot{m}|\dot{m}|}{\rho} f \sqrt{\frac{\pi}{A}} \\ 2h_c \sqrt{\pi A} \left[T_w - \frac{\gamma-1}{R} \left(e_0 - \frac{\dot{m}^2}{2\rho^2 A^2} \right) \right] \end{bmatrix} \quad (2)$$

and where ρ is the density, $\dot{m}$ the mass flow rate, e_0 the stagnation internal energy per unit mass, A the cross-sectional area of the pipe, R the specific gas constant, γ the specific heat ratio, f is the friction coefficient, h_c the convective heat transfer coefficient of the pipe and T_w is the wall temperature.

The above equations express the conservation of mass, momentum and energy for an ideal gas flowing in a straight pipe with constant cross-sectional area. For engine gas dynamic simulations, the above equations are generally solved numerically, for instance using the finite difference method or the finite volume method.

The model order reduction method proposed in [17,18,19] starts from a discretized representation of the air path system similar to the finite volume method, where a staggered grid is applied to discretize the system geometry. The governing equations are then integrated over a control volume that corresponds to one element of the discretized pipe domain:

$$\int_{cell} \left[A \frac{\partial \mathbf{W}(x,t)}{\partial t} \right] dx + \mathbf{F}(\mathbf{W}) \Big|_{bounds} + \int_{cell} \mathbf{B}(\mathbf{W}) dx = 0 \quad (3)$$

At this point, a projection method is defined by approximating the system variables as the following separated form:

$$\mathbf{W}(x,t) \cong \mathbf{W}^x(x) \cdot \bar{\mathbf{W}}^t(t) \quad (4)$$

where $\mathbf{W}^x(x)$ is a time-invariant set of *spatial basis functions*, which represents the spatial distribution of the system variables $\mathbf{W}(x,t)$ over a pipe element. This allows for separating the spatial from the temporal dependency. A spatial average for the vector $\mathbf{W}(x,t)$ is then introduced:

$$\mathbf{W}^*(t) = \frac{A \int_{cell} \mathbf{W}^x(x) \bar{\mathbf{W}}^t(t) dx}{V} = \frac{\bar{\mathbf{W}}^x \bar{\mathbf{W}}^t(t)}{V} \quad (5)$$

By applying Eq. (3, 4, 5), the governing equations in matrix form expressed by Eq. (1-2) can be transformed into a system of coupled nonlinear ODEs in time domain, for each computation cell:

$$AL \frac{d\mathbf{W}^*(t)}{dt} + \mathbf{F}(\bar{\mathbf{W}}^t(t)) \Big|_{bounds} + \mathbf{S}(t) = 0 \quad (6)$$

where L is the length of the pipe element and $\mathbf{S}(t)$ represents the average source terms in the control volume.

To couple the spatial averages, denoted by $()^*$, and the fluxes at the boundaries, a set of piecewise polynomial basis functions is selected to provide an analytical expression for $\mathbf{W}^x(x)$ [17]. In particular, for zero order polynomial basis functions, the final form of the governing equations describing the wave propagation in the i^{th} control volume is obtained as follows [18, 19]:

$$\begin{aligned}
 AL \frac{d\rho_i^*(t)}{dt} &= \dot{m}_{i-1}^*(t) - \dot{m}_i^*(t) \\
 AL \frac{d\dot{m}_i^*(t)}{dt} &= A^2(\gamma-1) \left[\rho_i^*(t) \cdot e_{0,i}^*(t) - \rho_{i+1}^*(t) \cdot e_{0,i+1}^*(t) \right] + \\
 &\quad + \frac{3-\gamma}{2A} \left[\frac{(\dot{m}_{i-1}^*(t))^2}{\rho_i^*(t)} - \frac{(\dot{m}_i^*(t))^2}{\rho_{i+1}^*(t)} \right] - \rho_0 \bar{G} A \\
 AL \frac{d(\rho_i^*(t) \cdot e_{0,i}^*(t))}{dt} &= \gamma(\dot{m}_{i-1}^*(t) \cdot e_{0,i-1}^*(t) - \dot{m}_i^*(t) \cdot e_{0,i}^*(t)) + \\
 &\quad - \frac{1-\gamma}{2A^2} \left[\frac{\dot{m}_{i-1}^{*3}(t)}{\rho_{i-1}^{*2}(t)} - \frac{\dot{m}_i^{*3}(t)}{\rho_i^{*2}(t)} \right] + \rho \dot{q} A
 \end{aligned} \quad (7)$$

where $\bar{G}$ is the mean friction and $\dot{q}$ is the heat generation term in the control volume. The superscript (*) indicates that the variable is spatially averaged over the control volume.

Boundary Conditions (BCs)

For modeling the gas dynamic processes in engine air-path system, several boundary conditions are required to correctly characterize the system components and their interaction with pressure waves. In typical 1D numerical simulation tools, the boundary conditions are defined as quasi-steady elements where the physical size of the boundary can be neglected when compared to the total length of the pipe resulting in 0D elements. The same approach is used here to derive the equation for the boundary conditions necessary for the considered exhaust system. Specific boundary conditions have been developed to account for the inflow and outflow conditions, valves, junctions and flow split, and the turbine with waste-gate.

BC for Open Pipe Termination

The open pipe termination is used to connect the pipe to the surrounding environments. For the outflow condition, corresponding to the fluid exhausting into the ambient, an isobaric transformation is assumed to calculate the flow properties. With the notation illustrated in Figure 2, the isobaric conditions results into [13]:

$$T_2 + \frac{u_2^2}{2c_p} = T_3 + \frac{u_3^2}{2c_p}; \quad p_2 = p_01 \quad (9)$$

where u is the local flow velocity.

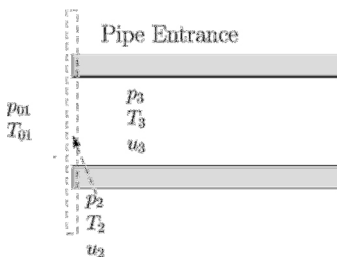


Figure 2. Model of an Open Pipe Termination.

For the inflow condition, hence when the fluid flows from the ambient into the pipe, the fluid properties at the boundary are calculated assuming an isentropic expansion process.

$$\begin{aligned}
 \dot{m}_2 &= \frac{Ap_{01}}{\sqrt{RT_{01}}} \sqrt{\frac{2}{\gamma-1}} \sqrt{\left(\frac{p_2}{p_{01}}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_{01}}\right)^{\frac{\gamma+1}{\gamma}}} \\
 \rho_2 &= \rho_{01} \left(\frac{p_2}{p_{01}}\right)^{1/\gamma}
 \end{aligned} \quad (8)$$

BCs for Valves and Restrictions

Similar to the case of the inflow BC, the mass flow rate through the engine exhaust valves is modeled by the isentropic orifice equation:

$$\dot{m}_v = \begin{cases} \frac{(C_d A) p_{ref}}{\sqrt{RT_{ref}}} \sqrt{\frac{2\gamma}{\gamma-1} \left[\left(\frac{p_d}{p_{ref}}\right)^{\frac{2}{\gamma}} - \left(\frac{p_d}{p_{ref}}\right)^{\frac{\gamma+1}{\gamma}} \right]} & \text{if } p_d \geq p_{cr} \\ \frac{(C_d A) p_{ref}}{\sqrt{RT_{ref}}} \sqrt{\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} & \text{if } p_d < p_{cr} \end{cases} \quad (10)$$

where p_{ref} and T_{ref} are the reference pressure and temperature upstream of the orifice, p_d is the downstream pressure, and p_{cr} is the critical pressure defined as:

$$p_{cr} = p_{ref} \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad (11)$$

In Eq. (10), $C_d A$ is a parameter known as *Effective Area*, and its value is typically determined empirically through engine flow bench data. In this case, the reduced-order model and the GT-Power model require the same level of information, as they are based on the same mathematical framework. Consequently, if a calibrated GT-Power model of the engine is available, the same values of the parameters characterizing each component can be imported into the reduced model.

Finally, the variation of fluid temperature across each valve is accounted by neglecting heat losses, hence applying the conservation of the stagnation enthalpy:

$$T_{ref} = T_d + \frac{u_d^2}{2c_p} \quad (12)$$

BCs for Junctions and Flow Splits

To extend the methodology presented in [14-15, 17] to a multicylinder engine, a set of boundary conditions must be derived to account for the presence of junctions and flow splits.

Leveraging the same process described above, flow splits are considered as quasi-0D elements, where the fluid properties are uniformly distributed within the control volume. The governing equations are then obtained by applying the mass and energy conservation and assuming that the junction volume is large enough