

Scalable Component-Based Modeling for Optimizing Engines with Supercharging, E-Boost and Turbocompound Concepts

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ABSTRACT

Downsizing and turbocharging is a proven technology for fuel consumption reduction in vehicles. To further improve the performance, electrified components in the turbocharger arrangements have been proposed, and investigations have shown acceleration improvements, emission reductions, and further fuel conversion efficiency benefits.

Simulation tools play an important role in the design process as the interplay between component selection, control strategy, system properties and constraints is very complex. Evaluations are performed with respect to BSFC map, fuel consumption in a drive cycle, acceleration performance, as well as many other aspects. A component-based engine and vehicle model is developed and evaluated to facilitate the process of assessing and optimizing the performance of e.g. engine, charging system, and electrical machine components.

Considerations of the execution time and model fidelity have resulted in a choice of models in the mean value engine model family. The turbocharging and electrical system models have all been evaluated using experimental data from engine dynamometer tests and turbocharger gas stand measurements and other dedicated component measurements.

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1. INTRODUCTION

There is a demand for more efficient and cleaner vehicles and this demand becomes a driver for the technical development. One path to improve the fuel consumption in vehicles is to downsize and turbocharge the engines, which results in engines that are more efficiently utilized in the driving cycles. Complementing this there are also other engine technologies that help improve the overall system efficiency and performance, e.g. variable valve actuation and direct injection. In line with this an interesting area is the heat that is lost in the thermodynamic cycle to the exhaust system, and how this can be recovered. In turbocharged engines parts of this energy is recovered for the boosting, but other complementing arrangements are gaining interest e.g. turbo compounding, thermoelectric devices, Rankine cycle heat engines on the exhaust.

To reach the best system efficiency there is a need to combine different technologies and select components in such a way that the system performance is optimized. Assessing and evaluating different concepts is a difficult task and tools that support such investigations provide valuable insights into the design and optimization.

This paper develops and evaluates a mean value engine model that is used in a tool for both assessing and optimizing the performance of e.g. engine, turbocharger, electrical machine components. Evaluation is performed with respect to e.g. its BSFC map, fuel consumption in a drive cycle, acceleration performance, as well as many other aspects. The simulation tool plays an important role in the design process as the interplay between component selection, control strategy, system properties and constraints is very complex.

When assessing the performance it is necessary to be able to exchange components and also influence and optimize the component designs (sizing). Therefore a modeling methodology is used that supports a component-based view of the system and thus easy replacement of components in the complete system model. Another important usage of the developed tool is to enable evaluation and optimization of component selection in combination with other hybrid solutions. Therefore emphasis is put on developing models that are scalable with respect to design parameters.

CONTRIBUTIONS AND OUTLINE

This paper contributes with a complete mean value engine model for turbocharged engines designed with a component

based thinking, enabling investigations of novel turbocharging concepts. The model serves as a starting point for investigating new concepts, but also enables extensions and refinements of components if it is needed in a particular investigation. As a case study, a novel concept of a turbocharger arrangement with an electrical machine is presented. The system can operate in different modes, normal turbocharging, e-boosting, and exhaust heat recovery through both the electrical machine and turbo compounding to the crank shaft.

Section 2 summarizes different engine and supercharging concepts with electrical machines and turbocompounding. The variety of concepts that one would like to analyze gives requirements on the model and these requirements are summarized in Section 3. Based on the requirements a set of component models for a turbocharged engine and a vehicle model are developed, these are described in Sections 4, 5, 6, 7.

A case study is performed in Section 8 and it is used to demonstrate the capabilities of the model and the developed tool. The most involved concept is studied and the importance of modeling the back pressure for a fair fuel consumption analysis is discussed. In addition the potential for turbocompounding is also illustrated. Finally the conclusions are given in Section 10.

EXPERIMENTAL DATA

Experimental data from several engines have been used to develop and validate the component models described in Section 4. The majority of the data has been collected on engine installations in the Vehicular Systems lab and the engines used were: a SAAB B235R 2.3 liter, a SAAB variable compression 1.6 liter, a SAAB L850 2.0 liter, a GM LNF 2.0 liter, and a Volvo experimental engine. To complement these, data from turbocharger gas stands have been used to build the turbocharger models and to validate them.

The turbocharger in the concept with VGT, gearbox, clutch and electric motor/generator has been tested in a gas stand. Tests have been performed both with the turbocharger as a stand alone device and with the motor/generator mounted and in use. An engine with complete turbocharger and motor/generator system, has been measured in an engine test cell.

2. BACKGROUND

This section summarizes investigations that have been made on the enhancement of turbocharging systems. The configurations and concepts that have been studied are illustrated in [Figure 1](#) and summarized in [Table 1](#).

Extracting or adding power to the turbocharger unit using a mechanical connection has been used for long time, and is well covered in the literature [1, 2, 3]. A fixed gearing ratio limits the potential benefit of a turbocompound system [1], where a continuous variable transmission can be shown to eliminate this restriction [4, 5]. Using a variable geometry

turbine, instead of a fixed geometry, is another way to increase the turbocompound flexibility and increase the potential gains [4]. Different ways of increasing the transient response of the normal turbocharger system is discussed in [2, 6]. Systems that combine mechanical supercharger and turbocharger capabilities or turbocompound systems have been implemented and used in production vehicles e.g. Scania HD trucks [7] or the currently manufactured VW engines combining a mechanical supercharger with a turbocharger.

A more flexible system is obtained if the mechanical link is replaced by an electric machine, which is studied in more recent research [8, 9]. The number of electrically assisted systems used in reality have however been few. The benefits from connecting an electric machine to either a compressor or turbine or to a turbocharger shaft are commonly shown in simulations.

The control of the extra degrees of freedom, that turbocharger electrification introduces, affects the potential performance gains. Optimal control is applied to investigate the transient performance potential in [10, 11]. A constraint during these optimizations is that the total electric energy consumed by the electric machine during the transient is negative. The electric power input to the turbocharger is positive during the first part of the transient, and re-generated during the later parts. The transient responses of three different electrically assisted turbocharger setups are investigated in [12]. The three systems are: (i) an electric motor connected to the turbocharger shaft, (ii) an electrically driven compressor before the inlet of the main turbocharger compressor, and (iii) an electric driven compressor and a turbine driving a electric generator. Compared to a baseline turbocharged engine, all three concepts show improved transient performance as well as reductions in fuel consumption. Other researchers have focused on control design during drive cycles. In [13] the sliding mode framework is used to develop a control structure for a 0D model of a DI engine with an electrically assisted turbocharger. A coordinated control of an electrically driven compressor connected to an energy buffer and a separate generator connected to a turbine is presented in [14]. The charging system model is connected to a DI engine model, and compared to a normal turbocharged DI engine. The electrically assisted charging system is shown to give both improved transient response, as well as lower fuel consumption over a city driving cycle.

Many researchers have presented gains in BSFC for driving cycle simulations. Investigations of various turbocompound systems for a limited cooled diesel engine using simulations is presented in [4]. Assuming very efficient turbomachinery, it is shown that up to 20% improvement in fuel consumption can be obtained. The use of either a VGT or a CVT are proposed as the best solutions, however expensive. In [15] a heavy duty truck with a 40-60kW electric motor is shown to give up to 10% fuel consumption reduction over a test cycle. A 5% reduction is however expected for normal

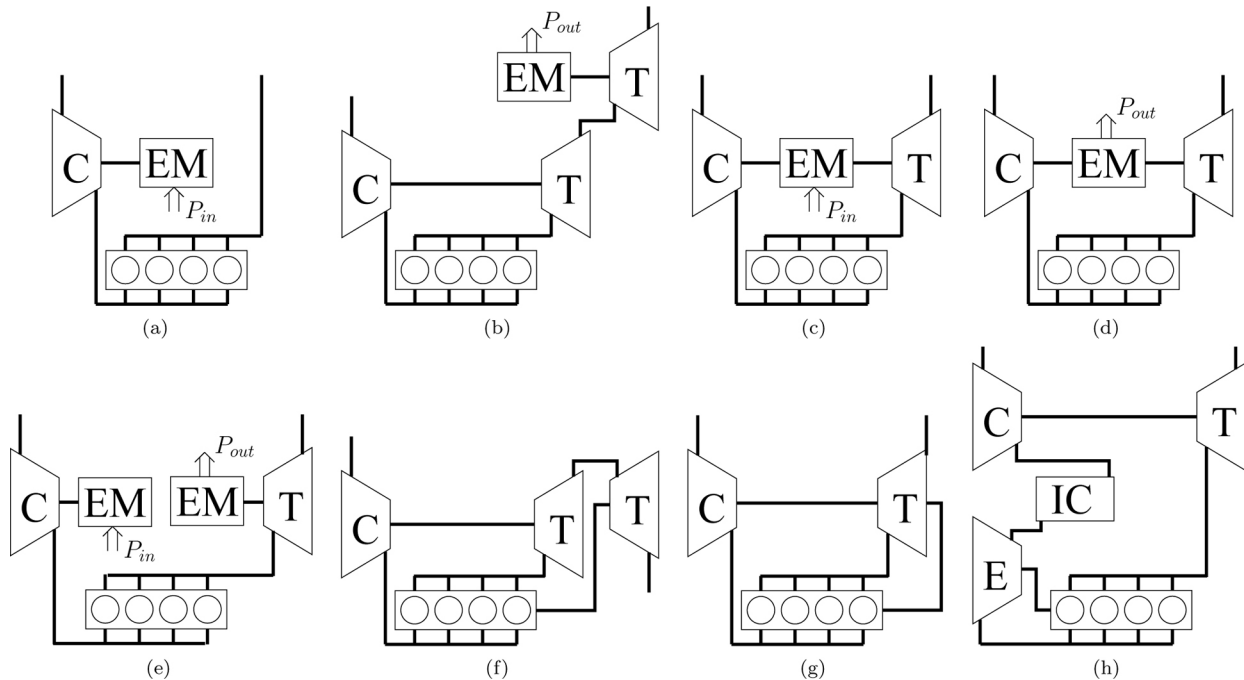


Figure 1. Different concepts for exhaust energy recovery and mechanical and electrical supercharging.

Table 1. An overview of the system configurations that are covered here and the different publications. As a case study for testing the models capability the leftmost configuration is used.

Reference	[23, 8, 9]	[10, 11, 22] [15, 18, 13]	[16]	[19]	[12]	[6]	[21]	[17]	[20]	[4, 5]	[14]	Case Study
Electric compressor, fig. 1a			X			X						
Turbo and electric turbocompound, fig. 1b					X		X					
Turbo with electric motor, fig. 1c	X	X		X	X	X						X
Turbo with electric generator, fig. 1d		X		X	X		X					X
Electric compressor and turbocompound, fig. 1e					X						X	
Mechanical compound turbine, fig. 1f				X					X	X		
Turbo with mechanical compound, fig. 1g								X		X		X
Turbo and mechanical expander, fig. 1h												

use. A 12% fuel economy improvement over an NEDC cycle is presented in [16], for a demonstrator vehicle having an electric centrifugal compressor. A novel approach is presented in [17], where a system having a positive displacement expander following the compressor on the intake side is presented. This system is said to potentially give intake manifold temperatures lower than ambient, enabling increased load without reaching knocking conditions. A 12 ton bus equipped with an electrically assisted turbocharger and using super-capacitors as intermediate energy storage is presented in [18]. The simulation study shows 0.6-10% improvement in fuel economy depending on driving cycle, and improvements in the dynamic response is also presented. In [19] a study is presented, comparing a standard turbocharged heavy duty diesel engine to: a mechanical turbocompound engine, an electrified turbocharger and a Rankine cycle system. The reductions in BSFC are reported to be 2-7.4% for the mechanical turbocompounding system (depending on turbine

efficiency and engine load), and 8-9% for both the electrical turbocharging and the Rankine cycle system. An investigation of the fuel economy improvements using a turbocompound system on an ethanol SI engine, running lean air to fuel ratios is presented in [20]. The brake engine efficiency is shown to be slightly improved when a turbocompound system is used, compared to the reference case with a normal turbocharger. The gain in efficiency amounts to at most 2%, with a decreasing gain with increasing air-to-fuel ratio. A radical downsizing (4.2 liter to 2.0 liter) 1D simulation study is presented in [5]. The vehicle performance is maintained at equivalent level using a turbocharger connected to the engine crankshaft through a CVT. Further, a duct from the compressor discharge to the turbine inlet is used to enable stable compressor operation for small engine mass flows at high pressures. The simulated fuel economy improvement is stated as 36% over the NEDC and EPA City drive cycles. In [21] a comparison of BSFC improvements for three different system is presented: an

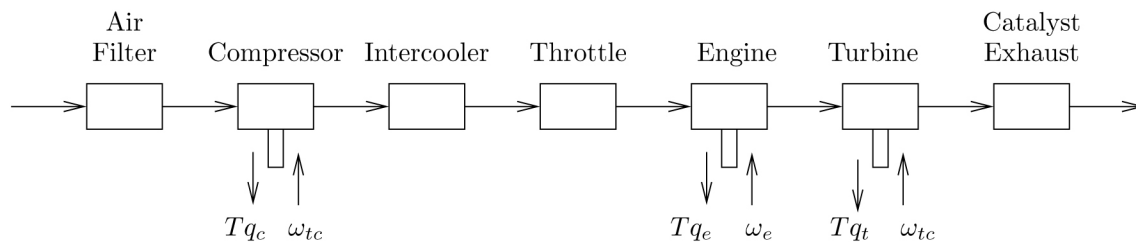


Figure 2. Components in the MVEM gas flow system. Control volumes are placed at the arrows in between the marked components. The arrows entering and leaving the system indicate information exchange with the mechanical subsystem and the direction of information transfer.

electrically assisted turbocharger, and two systems with a turbine connected to an electric generator. The two latter systems differ in that the first turbine is connected in parallel to the engine's normal turbocharger turbine, and the second is connected in series. The parallel configuration is shown to give the best fuel economy improvement: 4% for the US06 and 1.6% FTP75. For all tested cases the baseline engine structure is kept unchanged. The pumping losses due to the charging systems are further investigated.

The importance of different system properties e.g. turbocharger inertia, inertia to torque-ratio of the electric machine used and maximum power of the electric machine are studied in e.g. [23]. In [22] a study of the benefits from a direct drive motor-generator with the broad aims of optimizing the overall engine design for maximum thermal efficiency is presented. The importance of re-designing the base line engine parameters to get the largest benefit in fuel consumption is emphasized.

3. SUMMARY OF MODEL DEMANDS

A modular system of models is needed and it should be capable of handling all system configurations described in the previous section. The system can be used to evaluate different topologies and also provide support for component selection and optimization. There is a large deviation in the expected fuel economy improvements presented by different researchers in the previous section. The tool should thus further enable easy comparison of fuel consumption numbers from different system topologies and control system configurations.

A model or family of models that enables an engineer to evaluate properties of an engine with an flexible turbocharger arrangement in a vehicle setup. The demands are summarized in the following list:

- Enable design and evaluation of control strategies
- Analyze vehicle drive cycle fuel consumption
- Analyze single operating point fuel consumption
- Enable investigations of transient response
- Compare a traditional TC engine with e-boost or turbocompounding TC
- Capture interaction between engine and turbocharger and investigate different TC related matching properties:

- exhaust back pressure
- turbine inlet temperature
- potential for electrical power regeneration from turbocharger
- potential for turbo compounding

- Low computational complexity (fast iterations and ability to run an FTP-75 in a few minutes)

The last item is one of the most important properties as the model should enable rapid concept evaluation and testing together and control design studies.

The most common choice for modeling these systems is with 1D engine models with tools such as GT-Power, Wave or OpenWAM. However, here the choice of model family has been on, Mean Value Engine Models (MVEM), which gives a very favorable computational time, which is traded of for extrapolation capabilities. Since all models are simplifications and with the choice of MVEM, it becomes very important to ensure that important interactions are coupled between turbocharger, engine and other subsystems. To meet these demands and challenges the next section will compile a complete engine and vehicle model where the emphasis will be on the interaction between engine and turbocharger and in particular the exhaust back pressure that has a high impact on the fuel consumption.

4. MODELS, PARAMETERS AND SIMULATION

The total model of an engine, with or without electrification of the turbocharger and/or turbocompound, consists of several component subsystems and these are described in the following sections. The components described will be compiled to a complete engine model and coupled to the driveline and vehicle model in Section 7 and used in the case study in Section 8. The approach used here has become a standard approach in MVEM gas flow models (see e.g. [24, 25, 26, 27]) with a subdivision of the engine system in volumes in series, or parallel, with components that govern mass and energy flows. Here all components used in the model are given for the sake of completeness, although many of the engine component models like control volumes, flow restrictions, control valves, are well known in the field.

The completeness of a model is important since there are component details that have a direct impact on the global simulation results. The modeling is separated into three parts: MVEM Gas Flow Models (Section 5), Mechanical Models (Section 6), and Vehicle and Driver Models (Section 7). The essential addition to the standard MVEM gas flow models are the mechanical and electrical components (described in Section 6) that are added to the system and designed to be integrated with the turbo and engine models.

METHODOLOGY FOR MODEL PARAMETERIZATION

Complementing the models described below, it is also needed to handle all their parameters. These are required when running the simulations of the models. A tool has been constructed to handle the models and its parameters, as well as the simulation output. The idea has been that a user could take manufacturer data for all components, feed them into the simulation tool which then converts the data (like compressor maps) and parameterizes all model components using the least squares method, as well as prepares the complete model for simulation. Also handled by the tool is a set of driving scenarios, which a user can define, store and choose from, when setting up a simulation.

5. MVEM GAS FLOW MODELS

Component models for the gas flow in the mean value engine model, shown in [Figure 2](#), are collected below. Special emphasis in the modeling and description are on models that describe the interaction between engine and turbocharger arrangements.

CONTROL VOLUME

There are several pipes and volumes in an engine that can be modeled as control volumes. These volumes are for example found after components such as: the air filter, the compressor, the intercooler, the throttle (intake manifold) and the turbine. The exhaust manifold is modeled in a special way, as heat transfer is considered there.

A control volume is modeled as a dynamic element with two states taking mass and energy balances into account. The model is sometimes called adiabatic model, see [28] for a discussion. The mass balance gives the following time derivative for the mass in a control volume

$$\frac{dm_R(t)}{dt} = \dot{m}_{in} - \dot{m}_{out} \quad (1)$$

where m_R is the mass, and the mass flow in and out are given by $\dot{m}_{in}$ and $\dot{m}_{out}$ respectively. The fluid is assumed to be a perfect gas, with the temperature assumed to be the same in the whole volume. An immediate and complete mixing of the incoming gas, and no gas dynamics are further assumed. The temperature dynamics in the volume is derived from the

energy balance using the assumption of an ideal gas with constant specific heats. The internal energy of a volume is

$$U_R = m_R c_v T_R$$

where $c_v = \frac{R}{\gamma-1}$ is a gas constant and T_R is the temperature in the volume. The time derivative of U_R is

$$\begin{aligned} \frac{dU(t)}{dt} &= \frac{dm_R(t)}{dt} c_v T_R + \frac{dT_R(t)}{dt} c_v m_R \\ &= \dot{H}_{in} - \dot{H}_{out} + \dot{Q}_{in} \end{aligned}$$

where the enthalpy flows are described by, $\dot{H} = \dot{m} c_p T$. This yields the following expression for the time derivative of the temperature

$$\begin{aligned} \frac{dT_R(t)}{dt} &= \frac{1}{m_R} \left(\gamma \dot{m}_{in} T_{us} - \gamma \dot{m}_{out} T_{ds} \right. \\ &\quad \left. - (\dot{m}_{in} - \dot{m}_{out}) T_R + \frac{\dot{Q}_{in}}{c_v} \right) \quad (2) \end{aligned}$$

Finally, the pressure in the control volume is determined from the mass and temperature using the ideal gas law

$$p_R = \frac{m_R R T_R}{V_R} \quad (3)$$

Scalability of the control volume model is achieved by changing the component volume V_R directly.

The equations are also valid when flows in and out are reversed. For example a negative value of the inflow means an outflow of the control volume and this condition is accounted for in the equations above. In this model the states have been selected to be mass and temperature and they need initial values in a simulation. Temperature and pressure are easy to have knowledge about, while mass which is a state is not so easy to know. Consequently, the temperature is specified directly, while the initial value of the mass is calculated from the pressure and temperature using the ideal gas law $m_{init} = \frac{p_{init} V}{R T_{init}}$.

INCOMPRESSIBLE FLOW RESTRICTION

Flow restrictions are found in several places along the air flow path of the engine. Components that can be modeled as incompressible fluid restrictions are: air filter, intercooler and a lumped model of catalytic converter and exhaust system, see e.g. [25] for examples and model validations. The equations below are given for a flow from upstream to downstream, i.e. $p_{us} \geq p_{ds}$. The pressure drop for an incompressible flow over a component with wall friction,

change in area, or change in flow direction can be described by the following relation

$$\Delta p_r = p_{us} - p_{ds} = H_r \frac{T_{us} \dot{m}^2}{p_{us}}$$

where H_r is the flow restriction resistance. The mass flow can be obtained from this expression by

$$\dot{m} = \sqrt{\frac{p_{us}}{H_r T_{us}}} \sqrt{p_{us} - p_{ds}}$$

Problems can arise if this function is used directly in simulation. This is due to the fact that it does not fulfill the Lipschitz condition at $p_{us} = p_{ds}$, i.e. its derivative $\frac{d\dot{m}(p_{us})}{dp_{us}} \rightarrow -\infty$ when $p_{us} \rightarrow p_{ds}$, which causes problems for an ODE solver. Therefore the second root is exchanged for a linear component, with finite derivative, when the pressure difference is smaller than the limit p_{lin} which gives the following mass flow model

$$\dot{m} = \begin{cases} \sqrt{\frac{p_{us}(p_{us}-p_{ds})}{H_r T_{us}}} & p_{us} - p_{ds} \geq p_{lin} \\ \sqrt{\frac{p_{us}}{H_r T_{us}}} \frac{p_{us}-p_{ds}}{\sqrt{p_{lin}}} & \text{otherwise} \end{cases} \quad (4)$$

where $p_{lin} > 0$ is assumed in the implemented model. An isenthalpic restriction is assumed with no temperature change of the flowing fluid. Scalability for this model is achieved by varying the parameter $\frac{1}{\sqrt{H_r}} = K$ so that it changes proportionally to the cross section area of the flow.

If $p_{us} \leq p_{ds}$ backflow occurs. In this case the equations remain the same, but the pressure and temperature variables have to be exchanged, i.e. $p_{us} \leftrightarrow p_{ds}$ and $T_{us} \leftrightarrow T_{ds}$. Furthermore in this case the mass flow, given as output from the component, is negative.

COMPRESSIBLE FLOW RESTRICTION

The compressible fluid model is used when high velocities occur in the flow through the restriction, some examples are the throttle, the valves and the wastegate. For these components the area of the restriction usually changes as a result of control action. When considering compressible fluids there is also a distinction between choked and unchoked flow. The equations are summarized here, for a derivation see e.g. Appendix C in [3]. The mass flow through the restriction is

$$\dot{m} = \frac{C_D A_r p_{us}}{\sqrt{R T_{us}}} \Psi(\Pi) \quad (5)$$

where $\Pi = \frac{p_{ds}}{p_{us}} < 1$, C_D is the discharge coefficient, A_r the throat area and R is a gas constant. The product of C_D and A_r is lumped together to an effective area as

$$A_{eff} = C_D \cdot A_r$$

Scalability of this model is achieved by allowing the geometric area A_r to change.

The function $\Psi(\Pi)$ separates the cases of when the flow is choked or not

$$\Psi(\Pi) = \begin{cases} \sqrt{\gamma} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{2(\gamma-1)}}, & \Pi \leq \Pi_{crit} \\ \sqrt{\frac{2\gamma}{\gamma-1} \left[\Pi^{\frac{2}{\gamma}} - \Pi^{\frac{\gamma+1}{\gamma}} \right]}, & \Pi > \Pi_{crit} \end{cases}$$

The critical pressure ratio that determines when choking occurs is $\Pi_{crit} = (2/(\gamma+1))^{\frac{\gamma}{\gamma-1}}$. An isenthalpic restriction is assumed, with no temperature change over the restriction.

When the flow is in the forward direction, the temperature of the flow through the component is determined from the upstream conditions. For flows in the opposite direction the downstream temperature gives the temperature of the flowing fluid. The component also handles backflow, see the incompressible flow restriction above. This model does not fulfill the Lipschitz condition for $p_{us} = p_{ds}$ which can give problems in some cases.

THROTTLE PLATE AREA

The throttle plate is used to control the pressure in the intake manifold. It is modeled as a compressible flow restriction, using the block described in the previous section. The only addition here is a function for the effective area based on the throttle plate position. The throttle plate position is within the range $\alpha_{th} \in [0, 1]$ and the effective area is determined as follows

$$A_{eff,mod} = e^{(c_0 + c_1 \alpha_{th} + c_2 \alpha_{th}^2)}$$

The modeled effective area is limited so that the calculated exponential value is not allowed to grow beyond physical limitation of the throttle opening.

$$A_{eff} = \begin{cases} A_{eff,mod}, & A_{eff,mod} < A_{eff,max} \\ A_{eff,max}, & \text{otherwise} \end{cases}$$

where

$$A_{eff,max} = \frac{\pi D_{th}^2}{4}$$

where D_{th} is the maximum diameter of the throttle valve.

COMPRESSOR

The compressor type most frequently used in automotive applications is centrifugal, and the power to drive it usually comes from a turbine. The details of the selected implementation depends on two mapped or parameterized functions for the compressor performance, see e.g. [29] for