

Design and Validation of a Control-Oriented Model of a Diesel Engine with Two-Stage Turbocharger

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ABSTRACT

Two-stage turbochargers are a recent solution to improve engine performance. The large flexibility of these systems, able to operate in different modes, can determine a reduction of the turbo-lag phenomenon and improve the engine tuning. However, the presence of two turbochargers that can be in part operated independently requires effort in terms of analysis and optimization to maximize the benefits of this technology. In addition, the design and calibration of the control system is particularly complex. The transitioning between single stage and two-stage operations poses further control issues.

In this scenario a model-based approach could be a convenient and effective solution to investigate optimization, calibration and control issues, provided the developed models retain high accuracy, limited calibration effort and the ability to run in real time. To this extent, this work documents the development and validation of a control-oriented model of a Diesel engine equipped with two-stage turbocharger. A bottom-up modeling approach, based on the principle of modularity, is adopted to characterize the turbocharger system, as well as the engine airpath, inclusive of a high-pressure EGR loop. The approach characterizes each component as a dynamical system, applying mass and energy conservation laws to derive a set of ordinary differential equations. The reliance on physical principles limits the number of calibration parameters, which can be identified from simple steady-state engine data at few operating points.

The validated model is able to fully characterizing the flows, pressures and temperatures throughout the engine systems. In addition, prediction of engine performance (i.e., torque, BMEP, volumetric efficiency, specific fuel consumption) is provided to support further analysis, optimization and control studies.

1. INTRODUCTION

Two-stage turbocharging is a design solution that is recently being adopted on automotive Diesel engines to increase the specific power output while maintaining low fuel consumption and pollutant emissions [1-3].

In two-stage turbocharging, a large low pressure (LP) turbocharger is arranged in series with a smaller high pressure (HP) turbocharger. If the compression of the inlet air is performed in two stages, higher boost pressures can be achieved, therefore improving the air charge of the cylinders. As a result, higher EGR rates can be achieved without penalizing fuel efficiency.

Each of the turbochargers is optimized either for low or high speed operation. The high pressure turbine is typically controlled through variable nozzle area, a bypass valve, or both. Other actuators may be present on the compressor side, for instance, to reduce the possibility of surge problems. The low inertia of the high pressure turbocharger ensures a faster response at low speeds, eliminating the 'turbo-lag' phenomena when compared to a conventional turbocharger with either variable or fixed geometry.

Although the use of two-stage turbochargers presents significant advantages for performance and transient response, several issues have to be accounted for. The connection of two compressors in series causes the extension of the surge and choking lines, which may limit the operating range of the system [4]. These limitations require great effort in the design and calibration of the control system. In addition, several control issues arise during the transitioning between single stage and twin-stage operations. Part of the complexity arises from the need to operate both turbochargers near to their optimum isentropic efficiencies while minimizing turbo-lag [5]. From a controls point of view, this is substantially more complex than in the case of single turbocharger.

As an example, the EGR-VGT control problem, characteristic of turbocharged Diesel engines [6, 7], becomes more acute in case of two-stage turbocharger systems, where the presence of multiple actuators (bypass valves, waste-gate valves, etc.) to regulate the

operating modes of the compressors and turbines considerably increases the complexity of the control.

For these reasons, a model-based approach becomes necessary to investigate the behavior and the control issues of two-stage turbochargers, while limiting the reliance on experimental tests. By developing suitable modeling tools, retaining accuracy and physical consistency, it is possible to characterize the system behavior in steady-state and transient conditions, with the ultimate goal of defining control objectives and algorithms [8, 9].

One of the most difficult challenges is to define a model structure that can characterize the system dynamics with sufficient fidelity, being at the same time flexible to serve a vast range of application, computationally fast and easy to calibrate on different engine configurations. In this scenario, mean-value models can be effective tools to accomplish the desired tasks, with a reasonable level of accuracy, as well as limited computation time and calibration effort [8-15].

As part of a research activity oriented to the model-based control of Diesel engine with two-stage turbochargers, this paper describes the design, calibration and validation of a high-fidelity dynamic model. The scope is to accurately characterize the system steady-state and dynamic behavior through real-time simulation, focusing on adequate prediction capability, flexibility, modularity and limited calibration and computation effort.

The modeling approach followed consists of dividing the engine system into fundamental components through an “input/state/output” representation [16-18]. Each component is then characterized applying the conservation laws, resulting into a low-order set of ordinary differential equations, with very limited number of calibration parameters. The modeling and calibration are done following a bottom-up approach, focusing first on each of the individual components and then assembling into partial subsystems. Validation of the complete model completes the design, evaluating the accuracy and computation time.

The paper is structured as follows. In section 2, a brief description is given for the engine system and the two-stage turbocharger. The following section describes in detail the assumptions and equations characterizing the system components. Finally, Sections 4 and 5 discuss the identification and validation procedure, and present simulation results.

2. DESCRIPTION OF THE SYSTEM

The system considered for model development is a GM Duramax medium-duty Diesel engine, equipped with a two-stage turbocharger, intercooler, intake air throttle and high pressure cooled EGR with coolant bypass. A schematic representation of the system is shown in Figure 1, while the essential data are in Table 1.

The turbocharger system includes a high pressure (HP) VGT in series with a low pressure (LP), fixed geometry turbocharger. The HP stage includes a controlled bypass valve that allows for commuting between single-stage and two-stage operations.

The fresh air is first compressed in the LP compressor and then fed to the HP compressor. In order to increase the air density, the air passes through an intercooler, and before entering the engine cylinders, it mixes with exhaust gases coming from the high-pressure EGR system.

Table 1: GM Duramax engine description.

Displacement:	6.599 l
Valvetrain:	OHV 4-V
Injection:	Direct; Bosch Common Rail
Torque:	820 Nm @ 1800 rpm
Power:	231 kW @ 3200 rpm

The two turbines convert the exhaust gas energy into shaft work, through a common shaft drives the connected compressor. On the HP stage, the flow through the turbine is controlled through the VGT and the bypass actuator. When the bypass is closed, the VGT and EGR positions determine the operating points of the two-stage turbocharger and control the back-pressure in the exhaust to drive the EGR flow.

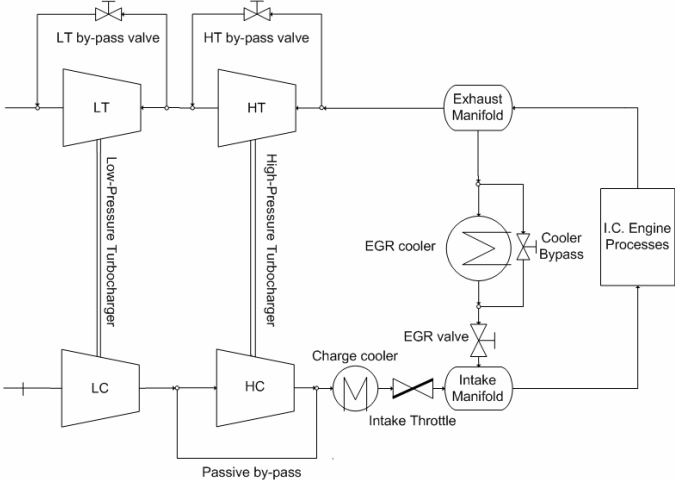


Figure 1: Diesel engine with two-stage turbocharger.

The nonlinear nature of the turbochargers, as well as the strong interaction between the EGR and VGT flow paths through the dynamics of the exhaust manifold and the common shafts between the compressor and the turbine create tremendous complexity in the control of the engine airpath. For this reason, a model of the engine system is proposed to investigate the problems and propose control schemes.

3. DESCRIPTION OF THE MODEL

The model developed focuses on the characterization of the engine air handling system, in order to study and control the two-stage turbocharger dynamics and its interaction with the EGR system and engine breathing dynamics. The engine in-cylinder processes are simplified, to provide estimates for the exhaust gas temperature and the shaft torque.

The mean-value modeling (MVM) approach is used to study the engine system dynamics [8-14, 19].

capturing only the low frequency spectrum of a system input/output behavior [18]. With reference to turbocharged Diesel engines, the bandwidth of interest is typically associated to the transients that result from variations of the engine load torque, which can be translated into correspondent variations in the intake and exhaust manifold pressures, temperatures and gas compositions. The high-frequency dynamics of the engine, for instance be associated to the fuel system dynamics, the engine cyclical behavior, or to the pressure wave propagation phenomena, are typically averaged over one or more engine cycles [8].

In order to facilitate the analysis of the system, the engine is divided into smaller subsystems, as outlined in Fig 2. The engine system model includes the intake and exhaust manifold systems dynamics models, as well as models characterizing the engine breathing dynamics, combustion and torque production. Finally, the EGR system model creates a coupling between the intake and exhaust, providing at the same time two control variables (related to the EGR valve opening and the EGR cooler bypass).

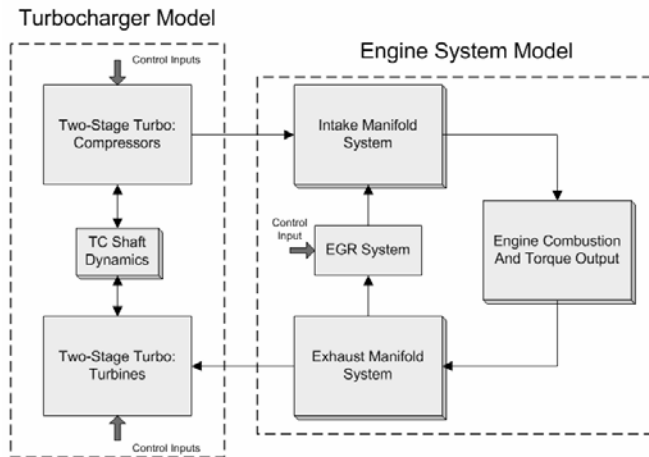


Fig 2: Functional scheme of the system model.

As shown in the following, the engine system is characterized by a five states dynamic model, where the states are the intake and exhaust manifolds pressures and temperatures, and the intake manifold composition (introduced to account for the EGR dilution). The combustion and torque production are modeled as purely algebraic functions (quasi-steady approach, [8, 9]). The control inputs to the model are the EGR valve position command and the EGR cooler bypass command.

The engine system model interfaces with the turbocharger model at the compressor side (mass flow rate and outlet temperature feeding the intake manifold model), and at the turbine side (the exhaust manifold states being the inputs of the high pressure turbine). This modular representation facilitates the possibility of interfacing the model with different turbocharger systems models, as well as implementing different technologies, such as low-pressure EGR loop.

The flow and energy outputs of compressors and turbines are typically modeled as quasi-steady elements

[8, 9, 13, 19]. In order to respect the cause and effect relationships [18], the two compressor stages must be separated by receivers in order to estimate the intermediate pressure (the same concept holds for the turbine stages). The receivers are treated as mass and energy accumulators, and their dynamics is characterized by the conservation laws [17, 18]. Similar methods are used to characterize the intake and exhaust manifolds, which characterize the engine system filling and emptying dynamics [8].

The following section describes the equation used to model the components of the engine system model and turbocharger model. In order to privilege the description of the methodology, the mathematical models are presented in a general form, *i.e.*, not associated to specific components.

3.1 Valves and flow restrictions

The flow restrictions of the system are modeled using a common framework, based on the steady-state, adiabatic reversible flow equations through an orifice [8, 17, 18, 20]. The model applied to the intake air throttle valve, the EGR valve and the bypass devices for the two-stage turbocharger system. For certain component, the possibility of reverse flow must be considered when developing the model equations.

Based on the equations for steady-state compressible flow, the mass flow rate through a restriction is given by the following form:

$$\dot{m}_{in-out} = C_d A \frac{p_{in}}{\sqrt{RT_{in}}} f_1 \left(\frac{p_{out}}{p_{in}} \right) \quad (1)$$

where

$$f_1 = \begin{cases} \sqrt{\frac{2\gamma}{\gamma-1} \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{2}{\gamma}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\gamma+1}{\gamma}} \right]} & \text{if } p_{out} > p_{crit} \\ \sqrt{\left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}}} & \text{if } p_{out} < p_{crit} \end{cases} \quad (2)$$

and C_d represents the discharge coefficient, A the actual

flow area and $p_{crit} = p_{in} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}}$. In case of reverse

flow (where $p_{out} < p_{in}$), the structure of Equations 1 and 2 the equations changes as in the following relations:

$$\dot{m}_{out-in} = C_d A \frac{p_{out}}{\sqrt{RT_{out}}} f_2 \left(\frac{p_{out}}{p_{in}} \right) \quad (3)$$

$$f_2 = \begin{cases} \sqrt{\frac{2\gamma}{\gamma-1} \left[\left(\frac{p_{in}}{p_{out}} \right)^{\frac{2}{\gamma}} - \left(\frac{p_{in}}{p_{out}} \right)^{\frac{\gamma+1}{\gamma}} \right]} & \text{if } p_{in} > p_{crit} \\ \sqrt{\left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}}} & \text{if } p_{in} < p_{crit} \end{cases} \quad (4)$$

$$\text{and } p_{crit} = p_{out} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}}.$$

Regardless of the direction of the flow, the model equations are physically based (deriving from mass and energy conservation in quasi-steady conditions), therefore requiring only the effective area $C_d A$ to be identified on experimental data for each specific component [18].

3.2 Receivers

Following a system dynamics approach, the receivers are considered as component where the storage of mass and energy is significant, hence characterized by filling and emptying dynamics. In this case, the mass and energy equations must be applied to the system in their general form, resulting into a set of nonlinear ordinary differential equations (ODE).

In a mean-value model, the typical components represented by receivers are the intake and exhaust manifolds. In the model shown in Figure 2, other components must be modeled as accumulators, to maintain consistency with the dynamics of the engine system and the cause and effect relationships. In particular, small accumulation of mass and energy are present between the two compressor stages and the two turbine stages. Depending on the thermal characterization required for each component, three possible modeling approaches are used, namely isothermal, adiabatic and diathermic [8, 17, 18, 21].

The isothermal approach is used to model the intermediate volumes of the two-stage turbocharger, whose thermal mass is significantly small. Therefore, the sole mass conservation equation is considered to derive a differential equation for the pressure [18]:

$$\frac{dp}{dt} = \frac{RT}{V} (\dot{m}_{in}(t) - \dot{m}_{out}(t)) \quad (5)$$

where V is the system volume. The adiabatic approach, applied to the intake manifold, considers two state variables, namely the temperature and pressure in the volume. The state equations are derived by the application of the mass and energy conservation [18]:

$$\begin{aligned} \frac{dp}{dt} &= \frac{\gamma R}{V} [\dot{m}_{in} T_{in} - \dot{m}_{out} T] \\ \frac{dT}{dt} &= \frac{\gamma RT}{pV} \left[\dot{m}_{in} T_{in} - \dot{m}_{out} T - (\dot{m}_{in} - \dot{m}_{out}) \frac{T}{\gamma} \right] \end{aligned} \quad (6)$$

As for the exhaust manifold, the mass and energy equations must be written in the most general form, to account for heat losses which may affect the temperature estimation. In this case, a heat loss term is introduced:

$$\begin{aligned} \frac{dp}{dt} &= \frac{\gamma R}{V} \left[\dot{m}_{in} T_{in} - \dot{m}_{out} T - \frac{\dot{Q}}{c_p} \right] \\ \frac{dT}{dt} &= \frac{\gamma RT}{pV} \left[\dot{m}_{in} T_{in} - \dot{m}_{out} T - (\dot{m}_{in} - \dot{m}_{out}) \frac{T}{\gamma} - \frac{\dot{Q}}{c_p} \right] \end{aligned} \quad (7)$$

where the heat transfer term can be expressed as $\dot{Q} = h A_w (T - T_{wall})$, being h is the convective heat transfer coefficient between the gas and the manifold walls, A_w the walls surface area, and T_{wall} the average temperature of the manifold wall.

3.3 Engine charge estimation

The engine charge estimation is a critical element for a MVM, as it is crucial for the model accuracy. A simple but fairly accurate approach is based on the speed-density equation, which relates the engine trapped charge to the intake manifold states [17,18]:

$$\dot{m}_{eng} = \lambda_v \frac{p_4 V_d}{RT_4} \frac{N_{eng}}{120} \quad (8)$$

where V_d is the engine displacement, λ_v the engine volumetric efficiency, p_4 and T_4 the pressure and temperature states for the intake manifold. The volumetric efficiency can be identified either through empirical methods ("black-box") or adopting physically-based approaches ("grey-box"). To retain accuracy and physical consistency, the method developed in [22] is applied, as yielding accurate results for turbocharged Diesel engines with a minimum number of calibration parameters [8-11]. This method relates the volumetric efficiency to the intake manifold pressure, in the form:

$$\lambda_v p_4 = S_i p_4 - Y_i \quad (9)$$

where S_i and Y_i are parameters depending on the engine speed, usually identified on experimental data.

3.4 Engine combustion and torque output

As mentioned earlier, the application of MVM techniques to the engine combustion and torque production assumes that the processes occurring in the engine cylinders are cycle-averaged. Therefore, the calculation of the in-cylinder processes is typically reduced to quasi-steady, algebraic relationships, mostly of empirical nature ("black-box") [17, 18].

The proposed model estimates the exhaust gas temperature and the engine torque through the brake mean effective pressure (BMEP). The exhaust temperature is estimated through the definition of a temperature increment ΔT , which is a regression function of the charge composition (fresh air, fuel, EGR), the engine speed and the injection parameters:

$$T_{exh} = T_4 + \Delta T(\dot{m}_{eng}, \dot{m}_{fuel}, \dot{m}_{EGR}, N_{eng}, DI_{comm}) \quad (10)$$

where the parameter DI_{comm} is related to the injection parameters (number of injections, duration, etc...).

Conversely, the engine torque model is derived following physical and empirical considerations, in order to achieve greater accuracy and consistency. A model purely based on regression would lead to poor results during the simulation of transient tests (such as driving cycles), where the operating conditions change frequently. As a requirement for closed-loop control of engine torque, the model must be able to accurately predict the torque output even in conditions where calibration data are not available.