

Modeling and Analysis of a Turbocharged Diesel Engine with Variable Geometry Compressor System

2011-24-0123

Published
09/11/2011

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doi:[10.4271/2011-24-0123](https://doi.org/10.4271/2011-24-0123)

ABSTRACT

In order to increase the efficiency of automotive turbochargers at low speed without compromising the performance at maximum boost conditions, variable geometry compressor (VGC) systems, based on either variable inlet guide vanes or variable geometry diffusers, have been recently considered as a future design option for automotive turbochargers.

This work presents a modeling, analysis and optimization study for a Diesel engine equipped with a variable geometry compressor that help understand the potentials of such technology and develop control algorithms for the VGC systems.

A cycle-averaged engine system model, validated on experimental data, is used to predict the most important variables characterizing the intake and exhaust systems (i.e., mass flow rates, pressures, temperatures) and engine performance (i.e., torque, BMEP, volumetric efficiency), in steady-state and transient conditions. The model is used to explore the engine system sensitivity to the VGC control input; in particular, its effect on the engine performance and

compressor stability is evaluated. A preliminary optimization for steady-state (open-loop) control of the VGC system is then proposed.

1. INTRODUCTION

In the last few years, the application of turbocharger devices to internal combustion engines has considerably increased due to their proven potential to increase engine efficiency [1-2]. Variable geometry turbines have been largely adopted to optimize the exhaust energy recovery over a large operating range. Two-stage turbocharger systems have also been considered to improve engine low-end torque and efficiency, and are currently in production for automotive and off-road applications.

In order to further increase the system efficiency in the low engine speed range without compromising the peak performance of the turbocharger, variable geometry compressor (VGC) systems have been recently considered. Centrifugal compressors can be equipped with either a variable inlet guide vanes device or with a variable geometry diffuser. Both devices allow one to control the passage area (either at the inlet or at the outlet of the impeller) by means of

rotating vanes. Since the characteristic curves can be “scaled” in relation with the vanes position, the surge line can be properly repositioned, allowing one to maximize the boost pressure at low engine speed without generating stability issues.

In this work, a model-based approach has been considered for two main reasons. First of all, by developing modeling tools based on the physics of the system, it is possible to obtain an accurate characterization of the engine in steady-state and transient conditions, which helps to understand the VGC behavior and facilitate the control design [4, 5, 6, 7, 8, 9, 10, 11]. Second of all, a model-based approach could help reduce development and testing time and costs [12].

This paper presents a modeling and simulation study of a Diesel engine system equipped with a turbocharger consisting of a Variable Geometry Turbine (VGT) and a compressor with Variable Inlet Guide Vanes (VIGV).

The modeling approach followed consists in dividing the engine system into fundamental components through an “input/state/output” representation. Each component is then characterized applying 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 the components into partial subsystems.

The design is completed with the validation of the complete model which evaluates the accuracy and computation time.

In this study, the model is used as a “virtual engine” [3] to characterize the response of the turbocharger system to variations in the actuators commands. A sensitivity study conducted in steady-state engine operating conditions allows one to highlight the effects of the VGT and VIGV on the main engine performance variables.

The engine system response to transients in the VIGV command is also evaluated.

Finally, based on the results of the steady-state and transient tests, an optimization study of the VIGV control problem is conducted, outlining the basic controller structure and the objectives for the feed-forward compensation. The VIGV feed-forward maps so derived, were then integrated with the existing EGR and VGT control to form an open-loop control for engine system simulations in driving cycle conditions.

2. MODELING OF THE ENGINE SYSTEM

A GM Duramax medium-duty Diesel engine was considered for this study. The engine, whose air path is schematically

shown in Figure 1, is equipped with intercooler, intake air throttle and high pressure cooled EGR loop. The figure also shows the locations of the main sensors as installed on the production engine, namely the mass air flow meter (MAF), boost pressure (MAP), intake manifold temperature and engine speed. The principal actuators considered in this study are the EGR valve position and the turbocharger control devices, respectively VGT and VGC.

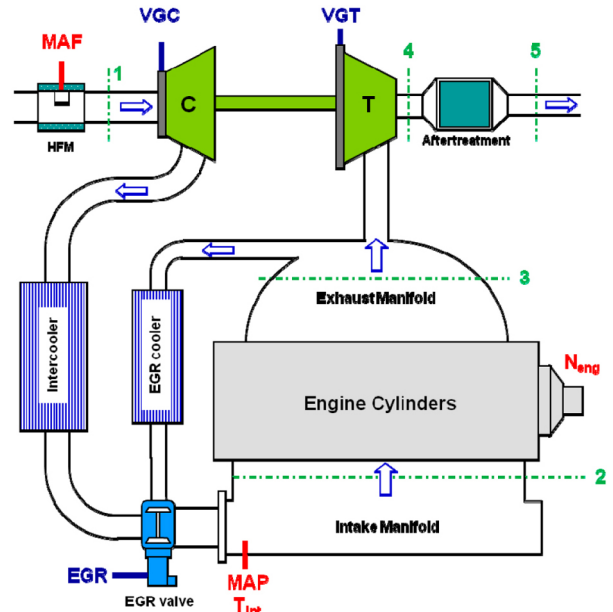


Figure 1. Simplified Air Path Scheme of Turbocharged Diesel Engine with VGC System.

Table 1. Description of Main Engine Characteristics.

Displacement:	V8, 6.599 l
Valvetrain:	OHV 4-V
Injection:	Direct; Bosch Common Rail
Torque:	820 Nm @ 1800 rpm
Power:	231 kW @ 3200 rpm

The turbocharger system considered for this model includes a turbine with variable geometry nozzle and a compressor with variable inlet guide vanes.

For the engine system considered, a cycle-averaged model (or mean-value model, [5, 6, 7, 8, 9, 10, 11, 12, 13, 14]) was developed to characterize the engine system dynamics that are relevant for performance evaluation under transient conditions and driving cycles. Cycle-averaged models are designed to capture the low frequency dynamic response, which is typically associated to the transients that result from variations of the engine load torque and speed. The high-frequency phenomena, for instance due to the pressure wave propagation in the air path system, are typically filtered out, and their effects are partially accounted for in model calibration [6, 7, 8, 14].

The following sections describe the equation used to model the components of the engine, with reference to [Figure 1](#) for the notation adopted.

FLOW RESTRICTIONS

Valves and flow restrictions (such as the aftertreatment system) are modeled based on the steady-state orifice flow equation, which predicts the mass flow rate as a function of the pressure ratio [5, 6, 7, 8, 14, 15, 16]. In the case of the EGR valve, such equation has the form:

$$\dot{m}_{EGR} = C_d A \frac{p_3}{\sqrt{RT_3}} f\left(\frac{p_2}{p_3}\right) \quad (1)$$

$$f = \begin{cases} \sqrt{\frac{2\gamma}{\gamma-1} \left[\left(\frac{p_2}{p_3}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_3}\right)^{\frac{\gamma+1}{\gamma}} \right]} & \text{if } p_2 > p_{crit} \\ \sqrt{\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}} & \text{if } p_2 < p_{crit} \end{cases} \quad (2)$$

where $p_{crit} = p_3 \left(\frac{2}{\gamma-1}\right)^{\frac{\gamma}{\gamma-1}}$ and the product $C_d A = A_{eq}$ represents the valve flow equivalent area, which is typically calibrated on steady-state experimental data for each specific component [16].

RECEIVERS

The receivers are considered as component where the storage of mass and energy is significant, hence characterized by filling and emptying dynamics.

Due to the presence of an intercooler and EGR cooler, the temperature variations within the intake manifold are typically limited, allowing for an adiabatic assumption to be introduced when formulating the state equations (mass and energy balance) that describe the dynamics of the system [7, 12, 14, 15]:

$$\begin{aligned} \frac{dp_2}{dt} &= \frac{\gamma R}{V_{IM}} [\dot{m}_c T_c + \dot{m}_{EGR} T_{EGR} - \dot{m}_{eng} T_2] \\ \frac{dT_2}{dt} &= \frac{\gamma R T_2}{p_2 V_{EM}} \left[\dot{m}_c T_c + \dot{m}_{EGR} T_{EGR} + \right. \\ &\quad \left. - \dot{m}_{eng} T_2 - (\dot{m}_c + \dot{m}_{EGR} - \dot{m}_{eng}) \frac{T_2}{\gamma} \right] \end{aligned} \quad (3)$$

As for the exhaust manifold, the mass and energy equations must be written in the most general form, to account for

variable inlet gas temperature and heat losses which may affect the temperature estimation and, ultimately, the prediction of the turbine flow and power:

$$\begin{aligned} \frac{dp_3}{dt} &= \frac{\gamma R}{V_{EM}} \left[\dot{m}_{eng} T_{eng} - \dot{m}_{EGR} T_3 - \dot{m}_{turb} T_3 - \frac{\dot{Q}}{c_p} \right] \\ \frac{dT_3}{dt} &= \frac{\gamma R T_3}{p_3 V_{EM}} \left[\dot{m}_{eng} T_{eng} - \dot{m}_{EGR} T_3 - \dot{m}_{turb} T_3 + \right. \\ &\quad \left. - (\dot{m}_{in} - \dot{m}_{out}) \frac{T_3}{\gamma} - \frac{\dot{Q}}{c_p} \right] \end{aligned} \quad (4)$$

where the heat transfer term can be expressed as $\dot{Q} = h A_w (T_3 - T_{wall})$, being h is the convective heat transfer coefficient, A_w the walls surface area, and T_{wall} the average temperature of the exhaust manifold wall.

ENGINE CHARGE MODEL

The engine charge model is quite important for achieving accurate prediction of engine torque and exhaust gas temperature. A simple but fairly robust approach is based on the speed-density equation [16, 18]:

$$\dot{m}_{eng} = \lambda_v \frac{p_2 V_d}{RT_2} \frac{N_{eng}}{120} \quad (5)$$

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") [5, 6, 7, 8, 9, 10, 11, 12, 13, 14]. To retain accuracy and physical consistency, a methodology that relates the volumetric efficiency to the intake manifold pressure is applied, in the form:

$$\lambda_v p_2 = S_i p_2 - Y_i \quad (6)$$

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

ENGINE EXHAUST AND TORQUE MODEL

As mentioned earlier, a cycle-averaged engine model reduces the in-cylinder processes to quasi-static, algebraic relationships, mostly of empirical nature.

The exhaust temperature is estimated through the definition of a temperature increment ΔT , which is approximated through a nonlinear regression model in relation with the

charge composition (fresh air, fuel, EGR), engine speed and the injection parameters:

$$T_{eng} = T_2 + \Delta T (\dot{m}_{eng}, \dot{m}_{fuel}, \dot{m}_{EGR}, N_{eng}, DI_{comm}) \quad (7)$$

where the parameter DI_{comm} represents the fuel injection parameters (number of injections, duration, timing).

For the engine torque prediction model, an approach based on the estimation of the indicated efficiency is adopted:

$$\eta_{ind} = \frac{IMEP \cdot V_d \cdot N_{eng}}{LHV_f \cdot 120 \cdot \dot{m}_{fuel}} \quad (8)$$

where LHV_f is the fuel lower heating value. The indicated mean effective pressure (IMEP) can be related to the engine BMEP through a simple model for the friction mean effective pressure (FMEP), based on the Willans line approach [5,16].

The indicated efficiency is then predicted by a phenomenological model that was built starting from existent results [5,10] and the analysis of experimental data for different Diesel engines, as a function of the engine speed, air fuel ratio and the differential pressure:

$$\eta_{ind} = \eta_0 (k_1 + k_2 N_{eng} - k_3 N_{eng}^2) \cdot (1 - k_4 AFR^{k_5}) \cdot (1 - k_6 \Delta p_d) \quad (9)$$

where the last term accounts for the backpressure that occurs at high power or high EGR conditions. The parameters of the indicated efficiency model were obtained through a least-square fitting of steady-state data and are summarized in Table 2.

Table 2. Summary of Torque Model Parameters.

Parameter	Values
η_0	0.5762
k_1	0.7148
k_2	0.4464
k_3	0.2600
k_4	1.2610
k_5	0.7830
k_6	0.0055

MIXTURE COMPOSITION

The fraction of exhaust gas trapped in the engine cylinder is an important variable to track for emission control. For this reason, a simple EGR mixing dynamics model was included to the intake manifold filling dynamics.

The fraction of combustion gases in the intake manifold F_I , defined as the ratio between the mass of residual gases m_{exl}

and the overall mass m_{IM} that occupies the intake manifold volume, can be determined by applying the conservation of the component mass:

$$\dot{F}_1 = \frac{\dot{m}_{EGR} - F_1(\dot{m}_{air} + \dot{m}_{EGR})}{m_{IM}} \quad (10)$$

where m_{IM} is calculated from the ideal gas law and Equation 3, and $\dot{m}_{EGR}$ is the mass flow rate through the EGR valve and given by Equations (1-2).

For simplicity, the above formulation assumes that the recirculated gas is composed exclusively by inert combustion products. However, the approach could be easily generalized to account for the presence of oxygen in the exhaust gas stream, by assuming F_I as a function of the instantaneous AFR and the composition of the fuel.

It is worth observing that the residual fraction is different from the EGR ratio, which is conventionally defined as the ratio between the mass flow rate from the EGR valve and the sum of the mass flow rates of air from the compressor (measured through the MAF sensor) and from the EGR valve:

$$EGR = \frac{\dot{m}_{EGR}}{\dot{m}_c + \dot{m}_{EGR}} \quad (11)$$

The above expression does not account for the intake manifold filling dynamics, thereby leading to significant errors in the estimation of the actual exhaust gas fraction in the intake during transient conditions.

TURBOCHARGER

The turbocharger model predicts the mass flow rate, efficiency and power for the compressor and turbine, as well as the shaft speed [3, 4, 5,10-11,19, 20, 21]. A common practice of cycle-averaged engine model is to assume the compressor and turbine as quasi-static components (similar to flow restrictions) and lump the turbocharger dynamics into the shaft speed:

$$\begin{aligned} \dot{m}_c &= f_1 \left(\frac{p_2}{p_1}, \omega_{TC}, x_{VGC} \right), & P_c &= f_2 \left(\frac{p_2}{p_1}, \omega_{TC}, x_{VGC} \right) \\ \dot{m}_t &= g_1 \left(\frac{p_3}{p_4}, \omega_{TC}, x_{VGT} \right), & P_t &= g_2 \left(\frac{p_3}{p_4}, \omega_{TC}, x_{VGT} \right) \\ \frac{d\omega_{TC}}{dt} &= \frac{1}{J\omega} (P_c - P_t) \end{aligned} \quad (12)$$

where ω_{TC} represents the turbocharger shaft speed, and P_c and P_t the compressor and turbine power, respectively. Note