

Double Stage Turbocharger Control Strategies Development

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ABSTRACT

A major issue in the development of the future engines (diesel or gasoline) lies in the architecture and the control of the air intake system. In this context many setups are envisaged, and in particular the turbocharging systems are becoming more and more complex: variable geometry turbines, double stage turbochargers, variable geometry compressors... For these new architectures, the engine control strategies need to be modified in order to address the specific issues related either to the new system in itself or to its integration in the global engine.

Renault and IFP have studied together, from a control point of view, the integration of a double stage turbocharger in a diesel engine.

This paper presents the works undertaken during this study. The structure of the paper follows the different stages of the project. The functioning of the system was first analyzed in simulation, leading to the adaptation of existing turbocharger control strategies and the development of new specific ones. In a second stage, the algorithms were implemented on an engine and validated online. Simulation as well as real test results are exposed in the different sections, demonstrating a good correlation between online and offline tests and proving the validity of the approach based on the utilization of simulation tools.

INTRODUCTION

For the new generation of diesel engines, the capacity to provide fresh air to the system even with high EGR rates is crucial in order to achieve the stringent emissions constraints. This has led to the development of complex architectures of air intake systems. In this context, the turbocharger system provides two main benefits: on the one hand it enables the extension of the engine operating range with high EGR levels, and on the other hand it increases the quantity of air in the cylinder at high engine loads, and thus the power production. New technologies are investigated in order to improve the conventional fixed geometry turbochargers. After the

introduction of variable geometry turbines, now commonly used, newer concepts like variable geometry compressors and double stage turbochargers are under investigation [1], [2] and [3]. This latter option represents a good compromise between additional costs, technology maturity and functional advantages. However, its implementation in an engine is not straight forward. In particular, it requires a modification of the control strategies.

This paper describes a project made by IFP and Renault, consisting in the development of double stage turbocharger control strategies for a diesel engine. The approach adopted for this work (described in [4]) was based on simulation tools which allowed to analyze the functioning of the system in details before designing an adequate control structure. The model based strategies obtained after this first phase were tested offline before a validation phase performed on a test bench and on a vehicle. The following sections describe these different phases.

CONTEXT

ENGINE SETUP

The engine considered is a 2,0 L diesel engine fitted with high pressure EGR. This engine was adapted from a production one, in order to make it possible a higher torque production thanks to an excess of air provided by the supercharging system. Therefore, the combustion chamber was modified, with a decreased compression ratio. The air intake system was completely redesigned. Figure 1 shows the engine set up.

The specificity of this engine are the two turbochargers in series:

- a « low pressure » (LP) turbocharger that will be matched in order to set the maximum power.
- a « high pressure » (HP) turbocharger that will be used to achieve the requirements of torque at low speed and in order to decrease the turbo lag.

Both turbochargers have a fixed geometry and a waste gate valve. A by-pass valve is mounted on the HP compressor in order to avoid pressure drop through the HP compressor when operating only with the LP compressor.

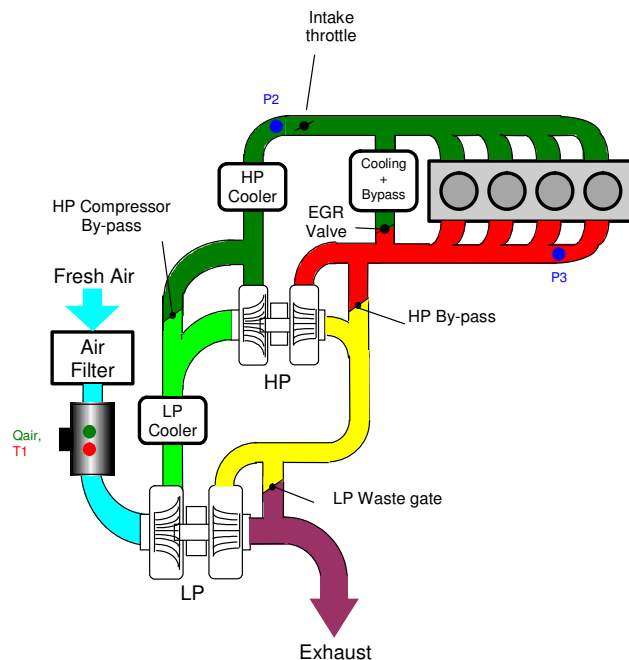


Figure 1 Two-stage turbocharger diesel engine set up

CONTROL GOALS

The main objective of the engine control strategies is to produce a required mechanical torque on the engine shaft. This torque is released during the combustion of the mixture of air and fuel in the cylinder. On a diesel engine, the combustion efficiency and the resulting pollutant emissions are linked with the mass and composition of gas aspirated by the cylinder. The engine gas mass flow depends on the intake manifold air density, while the intake gas composition depends on the balance between EGR flow and fresh air flow. Conventionally, the air path management structure considers two problems which are dealt with independently: the control of the fresh air mass flow via the actuation of the EGR valve, and the control of the intake manifold pressure via the turbocharger. In a single stage turbocharger system, the control strategy acts on the wastegate in order to follow a compressor downstream pressure set-point. Additionally, two constraints have to be respected for safety reasons: the turbocharger speed and the exhaust manifold pressure have to be maintained below maximum values. The control law must also be robust with respect to environmental conditions changes: the thermodynamic conditions at the boundaries of the system will affect its behaviour.

The Figure 2 represents the turbocharger control strategy that was used for a single stage turbocharger. This controller is composed of two cascaded controllers.

The first one (C1) computes a turbine pressure ratio set-point PR_{t_sp} in order to control the compressor outlet pressure measurement P_{spg_meas} to its set-point P_{spg_sp} . The second controller (C2) actuates the wastegate WG_{sp} in order to track this set-point. Feed forward terms depending on the engine operating conditions are added to both controllers. A similar structure is described for example in [5].

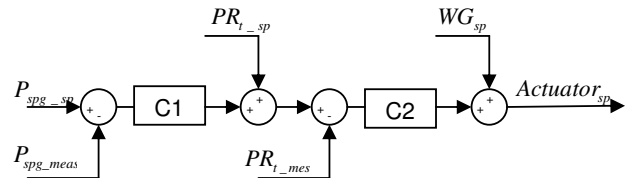


Figure 2 : single stage turbocharger control structure

The goal of this project was to adapt this structure to the specific issues of double stage turbocharging. A major constraint was to do so without adding any new sensor, particularly between the two compressors or the two turbines.

TURBOCHARGER MODELLING

The purpose of this section is to provide a representation of a turbocharger. Based on this model, the behavior of the system will be analyzed in the further sections. The equations given below can be found in many papers, for example [6], [7] and [8]. They will help for the understanding of the system, and provide a basis for the proposition of model based strategies.

Compressor map:

A typical compressor map is shown in Figure 3, which shows graphical representation of measurement data provided by the manufacturer. The performance variables PR_c and η_c are described using corrected quantities (see [7] for example). The dotted lines show constant corrected speed N_{c_corr} and the solid lines represent constant efficiency η_c .

This map defines the operating range of the compressor. Outside of its boundaries, three areas can be identified: the surge limit, the choke limit and the over speed limit. In order to guarantee the maximum life duration of the turbocharger, these operating areas must be avoided, in particular the over speed zone in which the system is irreversibly damaged.

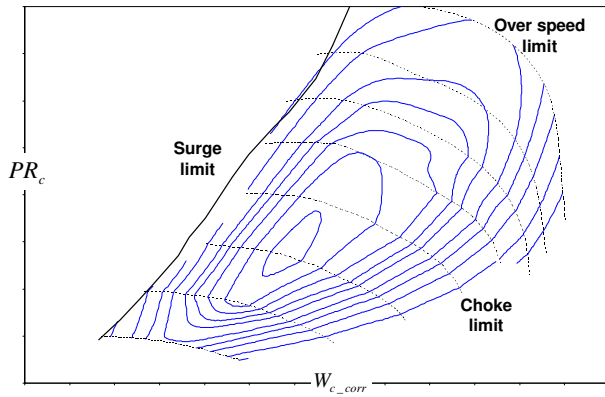


Figure 3: Compressor map showing the compressor efficiency map as a function of the pressure ratio PR_c and the corrected mass flow W_{c_corr}

Compressor equations:

The compressor pressure ratio and efficiency are given by steady static maps which depend on corrected compressor speed and mass flow. These maps characterize a given compressor, they are built from data provided by the manufacturer, using extrapolation formulas described in [9]. The dependencies can be described by equations (1) and (2).

$$PR_c = f_1 \left(W_c \cdot \frac{P_{ref}}{P_{uc}} \cdot \sqrt{\frac{T_{uc}}{T_{ref}}}, N_c \cdot \sqrt{\frac{T_{ref}}{T_{uc}}} \right) \quad (1)$$

$$\eta_c = f_2 \left(W_c \cdot \frac{P_{ref}}{P_{uc}} \cdot \sqrt{\frac{T_{uc}}{T_{ref}}}, N_c \cdot \sqrt{\frac{T_{ref}}{T_{uc}}} \right) \quad (2)$$

Turbine equations:

The turbine corrected mass flow and efficiency are given by steady state maps which depend on pressure ratio across the turbine and corrected turbine speed. As for the compressor, these maps characterize a given turbine, and are provided by the manufacturer. We describe the dependences with the equations (3), (4) and (5).

$$W_{t_corr} = f_3 \left(PR_t, N_t \cdot \sqrt{\frac{T_{ref}}{T_{ut}}} \right) \quad (3)$$

$$W_{t_corr} = W_t \cdot \sqrt{\frac{T_{ut}}{T_{ref}}} \cdot \frac{P_{ref}}{P_{ut}} \quad (4)$$

$$\eta_t = f_4 \left(PR_t, N_t \cdot \sqrt{\frac{T_{ref}}{T_{ut}}} \right) \quad (5)$$

Turbocharger equations :

The dynamic behavior of the turbocharger rotational speed is derived from the power balance on the turbocharger shaft.

$$P_t - P_c = J \cdot N_t \cdot \frac{dN_t}{dt} \quad (6)$$

Where J is the moment of inertia of the rotating parts.

The power developed by the turbine and the power required to drive the compressor can be expressed as follow:

$$P_t = W_t \cdot C_p \cdot \eta_t \cdot T_{ut} \cdot \left[1 - \left(\frac{1}{PR_t} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (7)$$

$$P_c = W_c \cdot C_p \cdot \frac{1}{\eta_c} \cdot T_{uc} \cdot \left[(PR_c)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (8)$$

Engine equations:

The air mass trapped at each engine stroke in the cylinder can be described from volumetric efficiency [10].

$$Mair_{cyl} = \frac{\eta_{vol} \cdot V_{cyl} \cdot P_{in}}{R \cdot T_{in}} \quad (9)$$

The mass air flow aspirated by the engine becomes :

$$W_e = \frac{30 \cdot \eta_{vol} \cdot V_{cyl} \cdot P_{in} \cdot N_e}{R \cdot T_{in}} \quad (10)$$

Making the assumption that the dynamics of the intake and exhaust manifold can be represented as a simple first order filter, the mass air flow across the turbine can be expressed as follow :

$$W_t = W_c \cdot e^{\frac{-t}{\tau}} + W_f \quad (11)$$

Where W_f is the injected fuel mass, W_c is the air mass flow across the compressor and τ the time constant of the first order filter to represent the transport time between the compressor and the turbine. This representation does not ensure a conservation of the mass. It is acceptable only in a context of strategy development.

Combination of equations:

As described above, the compressor map gives the compressor pressure ratio versus the corrected mass flow and the corrected turbocharger speed. The corrected mass flow across the compressor can be described from the equation (10). In the engine operating zones where the EGR valve is closed, and in particular in the high load region, it can be assumed that the compressor and engine mass flows are identical ($W_c = W_{eng}$). Figure 4 shows constant engine speed lines on the compressor map.

$$W_{c_corr} = \frac{\eta_{vol} \cdot V_{cyl} \cdot P_{ref}}{30 \cdot R \cdot T_{in}} \cdot PR_c \cdot N_e \cdot \sqrt{\frac{T_{uc}}{T_{ref}}} \quad (12)$$

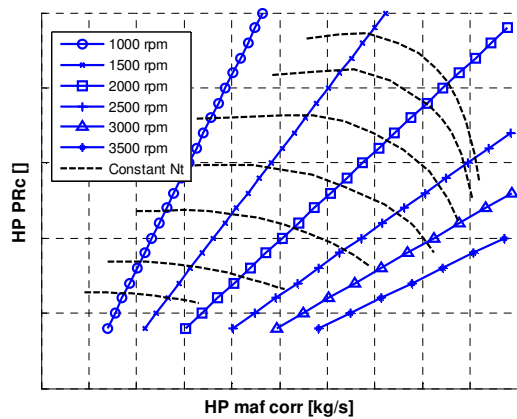


Figure 4: HP compressor map with constant engine speed measures and constant turbocharger speed.

The compressor pressure ratio can be expressed as a function which depends on the engine speed and the corrected turbocharger speed:

$$PR_c = f\left(\frac{\eta_{vol} \cdot V_{cyl} \cdot P_{ref}}{30 \cdot R \cdot T_{in}} \cdot N_e \cdot \sqrt{\frac{T_{uc}}{T_{ref}}}, N_c \cdot \sqrt{\frac{T_{ref}}{T_{uc}}}\right) \quad (13)$$

SYSTEM ANALYSIS

SIMULATION TESTS

As seen in [4], engine system modeling can have many uses (calibration, test bed set up support, architecture design...). In this study, the purpose of the simulator is to be used for the development of control strategies.

In this specific context, a high frequency 0D engine simulator has been developed on the AMESim platform thanks to a specific engine library called IFP-ENGINE. It represents the first order physical phenomena, as it will help to understand the impact on the system of the

addition of a turbocharger and to perform steady state and transient tests. A complete description of the engine simulator, its different sub-models and the comparison of the model and the bench results is presented in [11].

The main objective of the first tests performed on the engine simulator was to study the impact of the LP waste gate position on the global system, and in particular on the HP turbocharger dynamics.

The results presented in Figure 5 and Figure 6 show load transient tests with two different positions of the LP waste gate (15 % and 85 % of closing). The supercharging pressure is controlled by an action on the HP turbine by-pass. It can be noticed that a single stage controller can be applied on the HP stage, the LP stage being considered as a simple perturbation.

The first observation that can be made from these figures is that the HP turbocharger speed reaches high values in transient and stabilizes at lower values. The low time response of the LP turbocharger is compensated for by the faster HP stage. Because of this phenomenon, an over speed problem can arise if the HP system works near of its speed limits. An important fact is that the sensors available on the system do not give any information about the HP turbocharger speed. The HP compressor outlet pressure sensor provides a value that is the consequence of both compressions. They cannot be isolated.

Another important observation is that a closed position of the LP wastegate is always beneficial to the system. In transient, it makes the global system dynamics faster (intake pressure and air flow rate), whereas in steady state it decreases the difference between the exhaust pressure and the intake pressure, reducing engine pumping losses. The HP stage dynamics are not modified by the LP waste gate closing. The pressure ratio and the turbocharger speed dynamics are independent of the conditions at its boundaries.

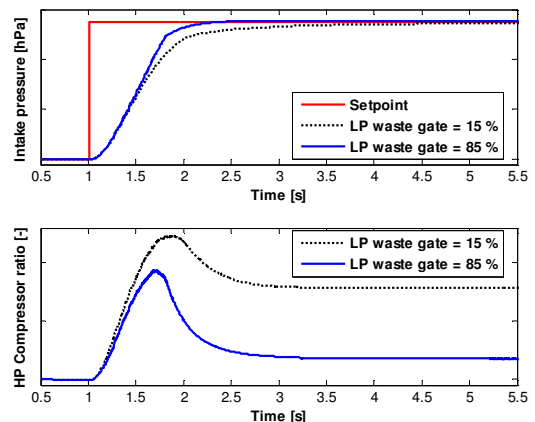


Figure 5: Effect of the variation of the LP waste gate opening on the intake pressure and the HP compressor ratio