

# Turbocharger Control-Oriented Modeling: Twin-Entry Turbine Issues and Possible Solutions

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## ABSTRACT

The paper presents possible solutions for developing fast and reliable turbocharger models, to be used mainly for control applications. This issue is of particular interest today for SI engines since, due to the search for consistent CO<sub>2</sub> reduction, extreme downsizing concepts require highly boosted air charge solutions to compensate for power and torque de-rating. For engines presenting at least four in-line cylinders, twin-entry turbines offer the ability of maximizing the overall energy conversion efficiency, and therefore such solutions are actually widely adopted.

This work presents a critical review of the most promising (and recent) modeling approaches for automotive turbochargers, highlighting the main open issues especially in the field of turbine models, and proposing possible improvements. The main original contribution is then on solving specific issues related to the twin-entry turbine, to develop a control-oriented model able to predict the machine behavior under all possible admission conditions.

The results of this study have been applied to a V8 high-performance GDI engine with twin-entry turbochargers. Experimental data are shown throughout the paper, to demonstrate the benefits of the proposed approach.

**CITATION:** Cavina, N., Borelli, A., Calogero, L., Cevolani, R. et al., "Turbocharger Control-Oriented Modeling: Twin-Entry Turbine Issues and Possible Solutions," *SAE Int. J. Engines* 8(5):2015, doi:10.4271/2015-24-2427.

## INTRODUCTION

The physics governing the turbocharger operation is particularly complex, since the flow is viscous, unsteady and three-dimensional, and a significant amount of heat is exchanged between turbine and compressor and with the surrounding environment [1]. For these reasons, a detailed and "white" box model (typically 3-D) of the turbocharger fluid dynamics is needed during the components design, development and optimization phases. Some authors have recently demonstrated that a 1-D model of the turbine still allows representing the turbine behavior under unsteady operating conditions quite accurately [2, 3]. Due to the complexity of this kind of approach, and to the need for expensive and time consuming experimental tests ([4]), a zero-dimensional, map-based (or "grey/black") representation of the turbocharger performance is still preferred for studying the interactions between the internal combustion engine and the turbocharger itself (so called turbo-matching), or for developing and testing engine control strategies [5, 6, 7, 8, 9, 10, 11]. This modeling approach considers the flow quasi-stationary, and the main

consequence is that the instantaneous mass flow rate and efficiency values (i.e., at the actual rotational speed and pressure ratio) are obtained from steady-state experimental data [1].

This work focuses on the analysis and control-oriented modeling of a turbocharging system for a high-performance V8 engine. For this reason, each turbocharger is equipped with a Waste-Gate (WG) by-pass system, and the turbines are of twin-entry type. During this project, different turbocharger models, produced by the same supplier, have been modeled and tested. The paper initially presents experimental data, control-oriented models, critical issues and possible solutions for single-entry, or "standard", turbines. The second part of the paper is instead entirely focused on twin-entry turbine modeling. The main original contribution is the development of a map-based, zero-dimensional twin-entry turbine control-oriented model, able to predict its behavior under all admission conditions.

Turbine and compressor maps are obtained by performing experiments on specifically designed test-benches. Such systems are usually equipped with temperature and pressure sensors both in intake and exhaust ports of the turbo-machines, and the compressor and turbine mass flows are also measured, as well as their instantaneous rotational speed. Such information allows calculating the efficiency of both machines, the Corrected Mass Flow (CMF) rate for the compressor, and the Reduced Mass Flow (RMF) rate for the turbine, respectively as a function of compressor Pressure Ratio (PR) and turbine Expansion Ratio (ER), for different external conditions [12, 13]. Such parameters are defined in the following equations and in the Acronyms/Abbreviations Section.

$$\text{CMF} = \frac{\dot{m}_C \sqrt{\frac{T_{01}}{T_{ref}}}}{\frac{p_{01}}{p_{ref}}} \quad (1)$$

$$\text{RMF} = \frac{\dot{m}_T \sqrt{T_{03}}}{p_{03}} \quad (2)$$

$$\text{PR} = \frac{p_{02}}{p_{01}} \quad (3)$$

$$\text{ER} = \frac{p_{03}}{p_{04}} \quad (4)$$

The experimental characterization of the compressor is quite straightforward, since the flow is “cold” and quasi-stationary. Compressor maps can be obtained with relatively simple test flow-benches, or even directly during engine operation. Conversely, the direct coupling of the turbine at the engine exhaust introduces several elements that make the experimental characterization of its performance particularly critical: fully unsteady (and three-dimensional) flow, very significant heat exchange, waste gate variable position and its effects on the turbine intake conditions. Such phenomena make it impossible to map the turbine performance directly during engine operation at the test bench, and require specifically designed and very complex experimental flow benches, which still do not allow to fully reproducing the turbine operation when coupled to the internal combustion engine, especially for what concerns unsteady phenomena. The most widely adopted solution is the so-called hot-gas test-bench, where the turbine is fed with a stationary hot gas flow produced by burning at a controlled temperature a stoichiometric mixture of air and methane. Intake turbine pressure may be controlled by varying the burner hot gas mass flow. In such type of test stands, the compressor acts as a brake with respect to the turbine, and therefore the operating conditions that can be tested at a given rotational speed are limited between compressor surge and choking lines, which respectively impose a lower and upper limit to the mechanical work absorbable by the compressor. To extend the range of measurable turbine expansion ratios, the compressor power consumption can be controlled in closed loop by varying the air density at the compressor inlet. A typical compressor closed-loop hot gas test stand is shown in Figure 1.

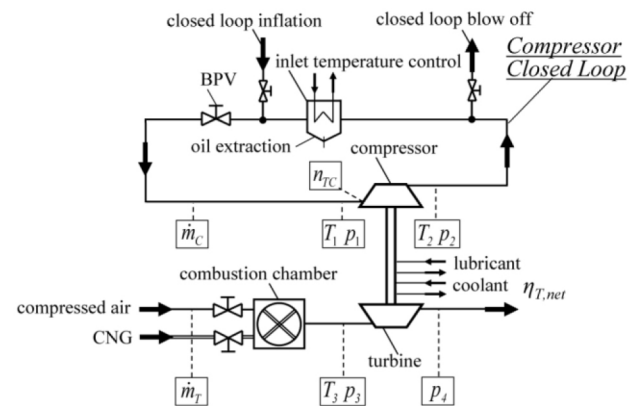


Figure 1. Hot gas test bench with compressor closed loop control. Source [13].

To further extend the turbine maps at the lowest ERs, run-away speed measurements are performed by replacing the compressor rotor with a bladeless one of the same mass. By neglecting the aerodynamics friction on the bladeless disc, it is possible to determine the minimum ER at which the turbine is still able to provide mechanical work, and therefore to identify the zero turbine efficiency condition for each rotational speed level (Figure 2).

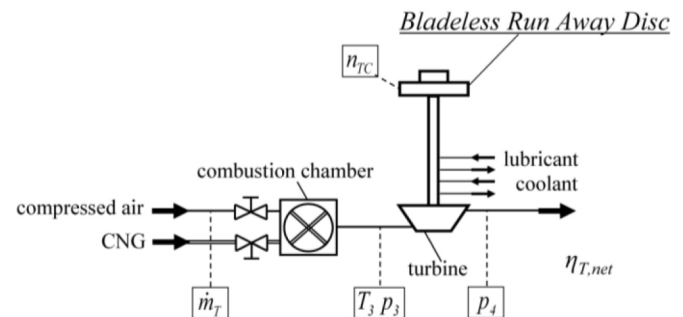


Figure 2. Hot gas test bench with bladeless compressor rotor for run-away speed measurements. Source [13].

The experimental determination of the compressor and turbine efficiency is nonetheless particularly challenging, since it is achieved by estimating the power delivered and absorbed respectively by the turbine and the compressor, and by measuring the inlet flows characteristics. The main issues that make such evaluation extremely complex are related to mechanical losses and heat exchange phenomena. The need to limit heat flows is reflected on a lower limit of the rotational speeds that can be investigated on a hot-gas test-bench, since the effects of heat being exchanged between turbine and compressor are more and more significant as the rotational speed is lowered. The main consequence of existing heat exchange is an over-estimation of the turbine efficiency (and an under-estimation of the compressor one), which masks the real aerodynamic effects contributions [1, 14]. Therefore, even when advanced hot gas test rigs are used for the experiments, the lower rotational speed range cannot be fully investigated while providing accurate evaluations. This implies the need to extrapolate both compressor and turbine maps towards low rotational speed regions, which are particularly interesting because they are frequently experienced during medium-low load engine operation, such as under urban driving conditions. As a possible alternative approach, recently some researchers have

proposed the use of a cold-gas test-bench [13, 14]. For what concerns friction losses determination, specific test benches may be developed using electric machines for measuring the mechanical efficiency. A simplified approach consists of including such losses in the turbine efficiency map [1]. Finally, it should be observed that flow-bench tests are typically performed with the waste-gate in closed position (sometimes the waste gate is mechanically sealed to prevent leakage), and this makes it very complex to model the turbine performance under all other operating conditions.

It is now worth recalling that a full experimental characterization of a given turbocharger is not always available, due to cost and time reasons. Typically, the selection of the turbocharger and the development of its control-oriented model are based on experimental data provided by the manufacturer, which are usually derived by hot-gas standard flow-bench measurements, and are therefore limited in terms of both expansion ratio and rotational speed ranges. Interpolation and extrapolation methods become therefore key factors for achieving high accuracy even under operating conditions that have not been directly measured. In fact, although the compressor maps may be directly used to simulate its behavior (essentially due to the quasi-static flow inlet conditions), the high pressure and temperature pulses at which the turbine is subjected determine an expansion ratio cyclically variable in a much wider range than the one covered by flow-bench data. To capture the real turbocharger behavior, extended maps can be built by following methodologies proposed by various authors in the last decades. The idea is to use the available experimental data to extend turbine and compressor performance characteristics at low rotational speeds and over a wider range of pressure ratios. The next sections of the paper deal separately with the compressor and turbine maps interpolation and extrapolation problem, by taking into account the most promising solutions that have been proposed in the literature, and introducing new contributions in particular for the turbine model. The second part of the paper is dedicated to twin-entry turbine modeling issues, and possible solutions.

## COMPRESSOR MAPS INTERPOLATION AND EXTRAPOLATION

As already mentioned, the extrapolation of compressor experimental maps is far less critical than the turbine one, since experimental measurements are able to describe the compressor performance over its entire operating range (i.e., between surge and choking conditions), for a given rotational speed. However, it should be observed that surge conditions also depend on the compressor surroundings, so that different test benches will generate different surge limits (and generally different to the ones experienced on the engine). Moreover, surge initiation definition is not unique, but it is based on different identification techniques (today mainly related to the frequency content of instantaneous pressures and flows).

The interpolation and extrapolation of experimental compressor data has been carried out following the well-known approach proposed by [15], and based on the non-dimensional parameters introduced by [16] (head parameter  $\Psi$  and non-dimensional flow rate  $\phi$ ):

$$\Psi = \frac{c_p T_{01} \left[ PR^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\frac{1}{2} U_c^2} \quad (5)$$

$$\phi = \frac{\dot{m}_c}{\rho_{01} \frac{\pi}{4} D_c^2 U_c} \quad (6)$$

The blade tip speed  $U_c$  is defined by the following Equation (7):

$$U_c = \frac{\pi}{60} D_c n_{TC} \quad (7)$$

In [15] it is demonstrated that the mathematical relationship between the two non-dimensional parameters may be expressed in the form of a homographic function, such as the one reported in Equation (8).

$$\Psi = \frac{K_1 + K_2 \phi}{K_3 - \phi} \quad (8)$$

Where the parameters  $K_i$  are defined by the following equation:

$$K_i = K_{i,1} + K_{i,2} M \quad (i = 1, 2, 3) \quad (9)$$

$$M = \frac{U_c}{\sqrt{\gamma R T_{01}}} \quad (10)$$

Figure 3 shows the results obtained by applying the methodology (5), (6), (7), (8), (9), (10) to interpolate and extrapolate experimental data that were available for one of the compressors investigated during the project.

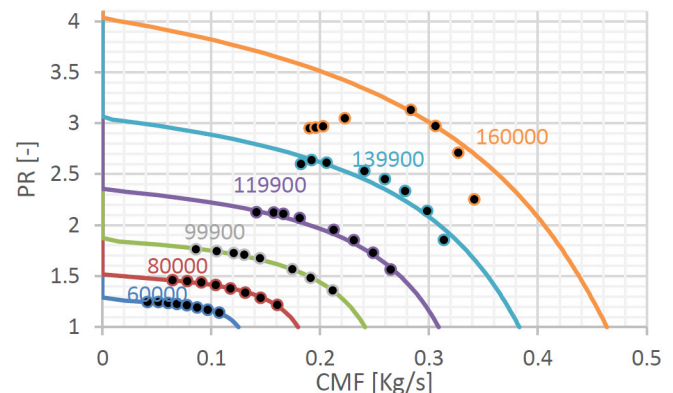


Figure 3. Interpolation and extrapolation over a wide PR range of a compressor map (PR vs CMF) using Jensen and Kristensen [15] method.

The coefficients  $K_{i,j}$  have been identified using Matlab-based custom genetic algorithms, converging over minimum Root Mean Squared Error (RMSE) between experimental and modeled data. A key issue is the algorithm initialization, and the genes of the starting population (the initial values of the coefficients  $K_i$ ) have been selected by forcing the corresponding homographic function to pass through three specifically selected experimental values, for each speed line (max PR, max CMF, and intermediate conditions).

In [15] it is also stated that the proposed methodology is not particularly accurate for the highest compressor rotational speeds, when  $M$  becomes greater than 1.1 and the flow characteristics change significantly. The accuracy loss is clearly visible in Figure 3. To overcome this limit, some authors have proposed solutions purely based on mathematical fitting techniques (such as spline functions or neural network algorithms [7, 10]), while others have attempted to maintain the parametric description introduced by [15, 16], evolving it according to what proposed by [11, 14]. In particular, in [11] it is suggested that each constant-speed group of experimental data should be treated separately, and that a set of  $K_i$  parameters should be identified for each constant-speed dataset. Then quadratic functions are used to fit the  $K_i$  coefficients with respect to turbocharger speed, thus allowing to extrapolate data also at turbocharger speeds that were not experimentally tested. By following this approach, a much more robust description of the compressor behavior has been achieved, as shown in Figure 4.

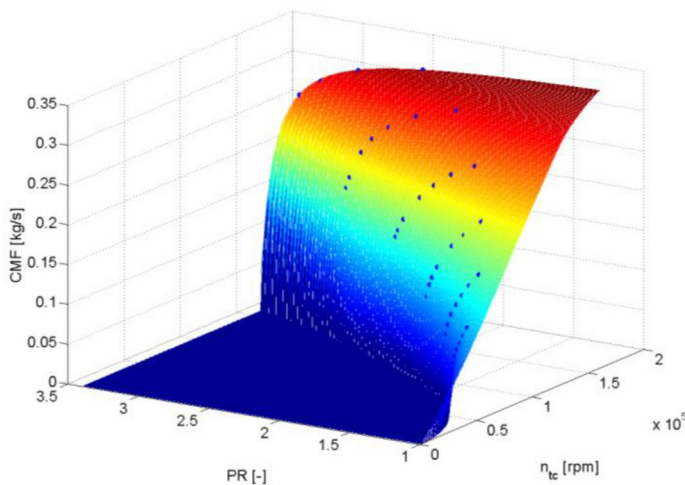
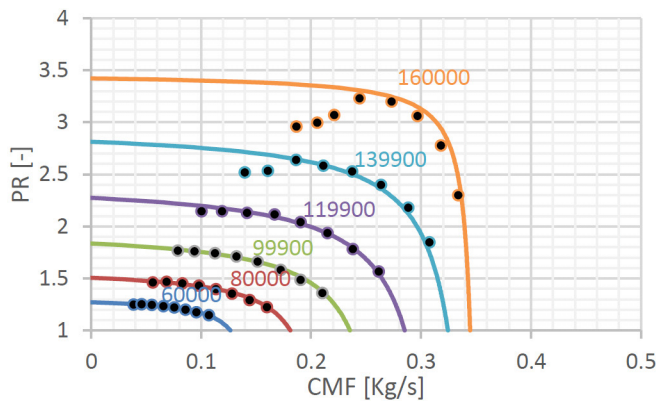


Figure 4. Interpolation and extrapolation over a wide PR range of a compressor map (PR vs CMF), according to [15] as modified by [11]. 2-D and 3-D view.

Figure 4 shows that the interpolation and extrapolation of experimental data is now much more robust. In particular, extrapolation at low PRs is now consistent with the compressor experimental choking behavior. Still, the extrapolation towards smaller mass flow rates is strictly monotone, while experimental data denote a PR reduction when approaching the surge line (mainly due to incidence losses [1]), especially at high speeds. Two further considerations are needed to completely clarify this issue: Since the extrapolated maps will be used in a filling-emptying 0-D model, providing mass flow rate in response to PR inputs, the curves need to be monotone and a more mathematically fit function would introduce uncertainty. Secondly, such operating conditions (highest speeds and PRs) are typically not reached during engine-turbocharger interaction.

It should finally be noted how the extrapolation at even higher rotational speeds (in this case higher than 175500 rpm) is not reliable, and this is due to the type of function used for data interpolation/extrapolation. A homographic function as the one defined by Equation (8) presents a discontinuity as  $K_3 \rightarrow \phi$ , and it provides reliable information only if the left branch of the function is involved.

For describing the compressor efficiency characteristics, the methodology proposed by [11, 14] has been adopted, which is based on the following definition of isentropic efficiency:

$$\eta_c = \frac{\Delta h_{is}}{\Delta h} \quad (11)$$

Where

$$\Delta h_{is} = c_{p1} T_{01} \cdot \left( PR^{\frac{\gamma-1}{\gamma}} - 1 \right) = \frac{1}{2} \Psi U_c^2 \quad (12)$$

After analyzing the behavior of several different compressors, some researchers isolated the existence of a linear relationship between the enthalpy difference and the Corrected Mass Flow (CMF) rate across the compressor, at constant speed [8, 11, 14]:

$$\Delta h = a + bCMF \quad (13)$$

For each rotational speed available in the experimental dataset, the two parameters  $a$  and  $b$  may be estimated theoretically or identified mathematically [8, 11]. In our approach, they have been identified using genetic algorithms. Finally, to interpolate/extrapolate the efficiency model at different rotational speeds, the parameters  $a$  and  $b$  may be modeled as a function of turbocharger speed through parabolic fittings passing through the axes origin.

One advantage of this approach is that the compressor characteristic maps (mass flow and efficiency) are now related one another through Equation (13), thus increasing simulation stability. At the same time, such relationship propagates the CMF interpolation/extrapolation errors (such as the monotonic trend when reducing PR at constant speed) onto the interpolated/extrapolated efficiency maps.