

A Scalable Modeling Approach for the Simulation and Design Optimization of Automotive Turbochargers

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ABSTRACT

Engine downsizing and super/turbocharging is currently the most followed trend in order to reduce CO₂ emissions and increase the powertrain efficiency. A key challenge for achieving the desired fuel economy benefits lies in optimizing the design and control of the engine boosting system, which requires the ability to rapidly sort different design options and technologies in simulation, evaluating their impact on engine performance and fuel consumption.

This paper presents a scalable modeling approach for the characterization of flow and efficiency maps for automotive turbochargers. Starting from the dimensional analysis theory for turbomachinery and a set of well-known control-oriented models for turbocharged engines simulation, a novel scalable model is proposed to predict the flow and efficiency maps of centrifugal compressors and radial inflow turbines as function of their key design parameters. The proposed approach is validated on a database of compressors and turbines for automotive boosting applications. Examples are given to illustrate how the characteristic curves can be scaled with key design parameters.

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INTRODUCTION

One of the main challenges in the automotive industry today is the trade-off between emissions reduction and fuel economy for both Diesel and SI engines. Research has taken several directions in tackling the problem and the downsizing and super/turbocharging of engines is currently the most followed trend in order to reduce CO₂ emissions and increase the powertrain efficiency. A key challenge for achieving the desired fuel economy benefits lies in optimizing the design and control of the boosting system, which requires the ability to rapidly sort different design options and technologies in simulation, and evaluating their impact on engine performance and fuel consumption [1,2,3].

Design optimization studies are currently performed in a virtual environment by relying on high-fidelity engine system simulators, such as GT-Power. Simulation of turbocharged engines requires however to import the characteristic maps of compressor and turbines, which are usually provided by the suppliers and are considerably limited in the number of available data points. Therefore, for engine system simulation it is often necessary to develop models that reasonably extrapolated the available data and

provide a representation of the turbocharger behavior at off-design conditions (such as at engine idle or low load points). The use of behavioral models to extrapolate the flow and efficiency maps of automotive turbochargers is a common practice and several approaches have been proposed [1,2,3,4,5,6].

On the other hand, developing virtual design studies, where the engine air path system design is optimized in simulation, requires the ability to model the behavior of compressors and turbines while varying their key design parameters related to the geometric features of the stator and rotor. Today, this can only be accomplished if a large database of maps is available, covering a wide range of geometries and, consequently, of flow rate and boost pressure conditions. Unfortunately, this is often not possible due to the limited information available from the suppliers. Furthermore, importing all the different maps into an engine simulator and extrapolating the data outside of the experimental region is extremely time consuming and difficult.

To this extent, this paper presents a model-based approach to predict the characteristic maps of automotive turbochargers when varying their most common design parameters. Specifically, this paper proposes a set of scalable models for compressors and turbines that

relate the flow and efficiency maps to their key design parameters. This approach makes it possible to model the behavior of a family of several turbocharger configurations varying their geometry, starting from only a few sample characteristic maps.

MODELING APPROACH

According to the dimensional analysis theory for turbomachinery, it is possible to introduce dimensionless variables to reduce the number of overall variables representing the performance of compressors and turbines [7,8,9]. For simplicity, the (dimensional) performance of a machine can be generally expressed as:

$$\begin{aligned} p_{out} &= f(\dot{m}, p_{in}, T_{in}, \gamma, N, \nu, D, l_1, l_2, \dots) \\ \eta &= g(\dot{m}, p_{in}, T_{in}, \gamma, N, \nu, D, l_1, l_2, \dots) \end{aligned} \quad (1)$$

where p is the pressure, $\dot{m}$ the mass flow rate, T the temperature, N the rotational speed, γ the specific heat ratio, ν the kinematic viscosity, and D, l_1, l_2 represent design parameters that influence the performance. According to the π -theorem, if there are n variables and m fundamental units, the equation relating the variables can be expressed in terms of $n-m$ dimensionless groups, leading to [8]:

$$\begin{aligned} \frac{p_{out}}{p_{in}} &= f\left(\frac{\dot{m}\sqrt{\gamma RT_{in}}}{p_{in} D^2}, \frac{ND}{\sqrt{\gamma RT_{in}}}, \gamma, \frac{ND^2}{\nu}, \frac{l_1}{D}, \frac{l_2}{D}, \dots\right) \\ \eta &= g\left(\frac{\dot{m}\sqrt{\gamma RT_{in}}}{p_{in} D^2}, \frac{ND}{\sqrt{\gamma RT_{in}}}, \gamma, \frac{ND^2}{\nu}, \frac{l_1}{D}, \frac{l_2}{D}, \dots\right) \end{aligned} \quad (2)$$

In order to design the characteristic curves, the above equations are usually simplified by neglecting the influence of the working fluid and Reynolds number on the characteristic curves, and by assuming that only one specific machine is modeled (hence, removing the dependence of the π -terms containing the design parameters). These approximations lead to the well-known dimensionless representation of the characteristic curves for compressors and turbines [8, 9].

In this study, Equation (2) is considered the foundation for studying the influence of design parameters on the characteristic maps, by applying the π -theorem to the flow and efficiency data obtained from the characteristic maps of different compressors and turbines.

The starting point for this study is a set of turbocharger maps that was made available for a family of compressors and turbines. The available maps cover different combinations of three design parameters, namely rotor diameter D , trim τ and A/R ratio Γ . These parameters are defined in Equation (3) and Figure 1.

$$\begin{aligned} \text{Trim: } \tau &= \left(\frac{d}{D}\right)^2 \cdot 100 \\ A/R: \Gamma &= \frac{A_{cs}}{R_{cl}} \end{aligned} \quad (3)$$

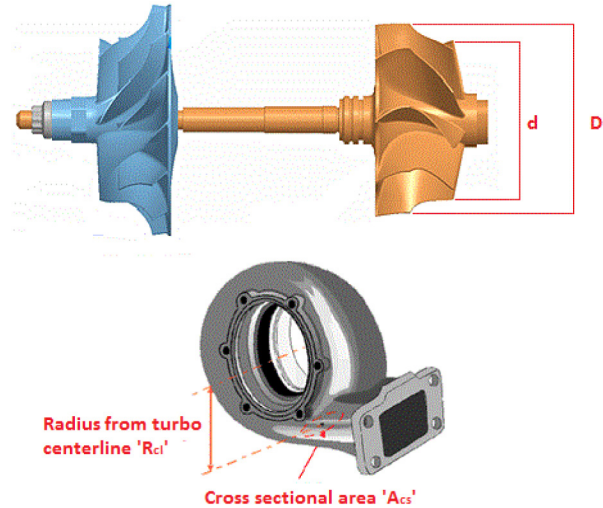


Figure 1. Main design parameters for turbochargers compressors and turbines.

The proposed approach for developing a scalable model starts from a set of semi-physical models predicting compressors and turbine maps. These models, illustrated in the following sections, have been proposed in literature and are generally well accepted for engine system simulation and control-oriented applications. Such models are first calibrated using data from different compressors and turbines. Then, the models are converted into dimensionless form, and a set of semi-empirical correlations is defined to link the model parameters to the π -terms dependent on the machine geometry. These correlations allow one to scale the flow and efficiency maps of compressors and turbines based upon dimensionless groups depending on the A/R ratio, trim and diameter.

Compressor Flow Rate Model

The scalable compressor flow rate model here proposed is based on the control-oriented model proposed by Jensen and Kristensen [10, 11], which will be referred to as the *JK model* in the following discussion.

According to this model, the compressor flow rate is predicted through a semi-empirical correlation expressed in terms of dimensionless variables, namely the head parameter Ψ , the flow parameter Φ and the blade Mach number Ma :

$$\begin{aligned} \Psi &= \frac{K_1 + K_2 \Phi}{K_3 - \Phi}; \\ K_i &= K_{i1} + K_{i2} Ma + K_{i3} Ma^2; \quad (i = 1, 2, 3) \end{aligned} \quad (4)$$

The dimensionless parameters are defined as:

$$\Psi = \frac{c_p T_{in} \left(\beta^{\frac{\gamma-1}{\gamma}} \right)}{\frac{u_2^2}{2}}; \quad \Phi = \frac{\dot{m} R T_{in}}{\frac{\pi}{4} p_{in} D^2 u_2}; \quad Ma = \frac{u_2}{\sqrt{\gamma R T_{in}}}$$

where D_2 is the impeller diameter and $u_2 = \pi DN/60$ is the impeller tip speed. The parameters K_{ij} , are generally identified on the compressor flow maps through a least-square curve fit [11].

The JK model is here used as a starting point to define a scalable model for the compressor flow maps. To this extent, the model has been identified on six different compressors, whose key data are listed in Table 1. The remaining compressor maps (listed in Table 2) will be later used for model validation. As an example, Figures 2, 3, 4 show how the JK model can be identified on the characteristic map of a single compressor, illustrating the agreement with the experimental data, and the functional relation of the three parameters with the Mach number.

Table 1. Summary of data of the compressors used for model calibration (diameter and A/R ratio are in mm).

Model	Diameter	Trim	A/R Ratio	Φ Range	Ψ Range
C1	56	50	0.56	0.040-0.139	0.462-0.927
C2	52	55	0.52	0.052-0.134	0.496-0.787
C3	52	50	0.52	0.047-0.120	0.506-0.779
C4	52	45	0.52	0.049-0.108	0.517-0.794
C5	60	55	0.60	0.045-0.131	0.443-0.846
C6	60	45	0.60	0.037-0.103	0.523-0.817

Table 2. Summary of data of the compressors used for model validation (diameter and A/R ratio are in mm).

Model	Diameter	Trim	A/R Ratio	Φ Range	Ψ Range
C7	60	50	0.60	0.041-0.117	0.493-0.829
C8	56	55	0.56	0.043-0.133	0.429-0.832
C9	56	45	0.56	0.034-0.106	0.477-0.837

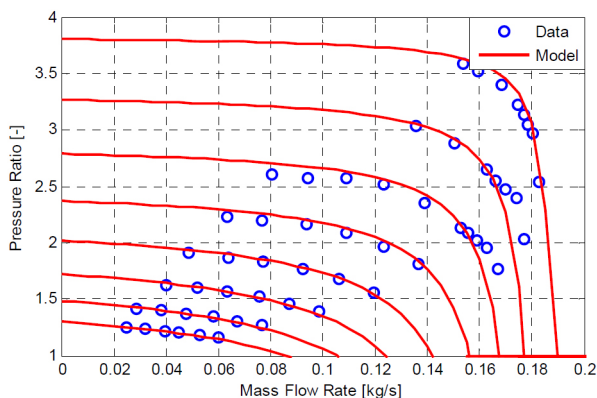


Figure 2. Results of JK model identified on the flow map of a single compressor (data from compressor C1).

As evident from Equations (4-5), the JK model depends only on the impeller diameter, through the definitions of the dimensionless variables. The influence of the other geometric parameters (A/R ratio and trim) is not immediately evident but can be elucidated by analyzing the structure of the above model and how it correlates with the operation of a centrifugal compressor at limit conditions.

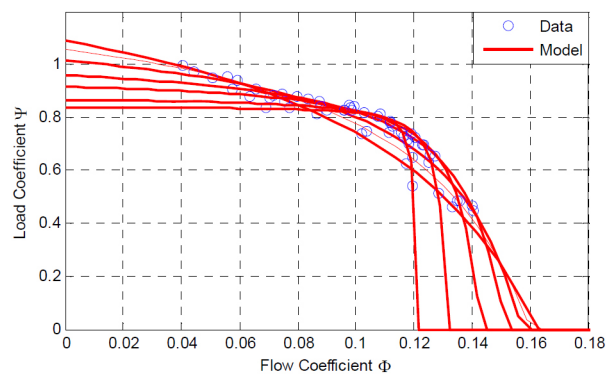


Figure 3. Results of JK model identified on the dimensionless map corresponding to compressor C1.

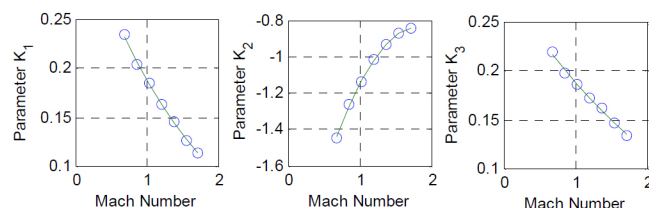


Figure 4. Parameters of JK model as functions of blade Mach number (data from compressor C1).

Recalling the similarity theory, two compressors that are geometrically and dynamically similar should have the same dimensionless characteristic curves, and should be represented by the same model and relative parameters. In other words, any difference in the parameters of the JK models describing two compressors can be only attributed to the influence of dimensionless groups formed by corresponding geometric parameters.

It is evident that this theory constitutes a significant approximation of the complex dependence between the design of a compressor and its characteristic maps. In fact, several factors must be neglected to proceed in this study, for instance the effects of friction losses, backflows, surge and stall, as well as design parameters typically not available (such as blade angles, impeller width and clearances). These simplifications are inevitable and will lead to limitations in the model accuracy, particularly if applied to compressors whose geometry significantly differ from those listed in Tables 1-2.

Figure 5 shows a normalized characteristic curve calculated at the same blade Mach number ($Ma=1.4$) for the six different compressors listed in Table 1. Each curve was obtained by applying the JK model calibrated to each compressor.

For illustrative purposes, the normalized characteristic curves have been extrapolated to the two axes. It is evident from Equation (4) that the JK model features a horizontal asymptote at $\Psi=-K_2$ and a vertical asymptote at $\Phi=K_3$. Neglects phenomena such as surge and stall, the functional form of Equation (4) suggests two limit conditions that can be imposed to obtain explicit expressions for the model parameters of the JK model.

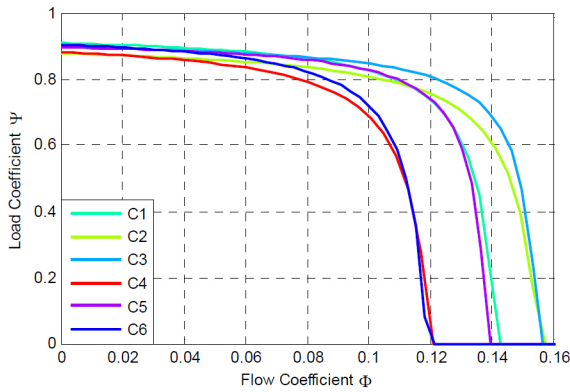


Figure 5. Extrapolated dimensionless characteristic curves for the six compressors in Table 1, at constant Mach number ($Ma=1.4$).

At zero flow condition ($\Phi=0$) all of the characteristic curves approximately converge to the same value. This limit behavior could be described as a condition where the work done by the blades is entirely spent to increase the kinetic energy of the volume of fluid contained in the compressor casing, and overcome friction losses. This condition can be represented mathematically through the radial equilibrium theory [8]:

$$c_p T_{in} \left(\beta^{\frac{\gamma-1}{\gamma}} - 1 \right) = u_2 c_2 - u_1 c_1 - f \frac{c_2^2}{2} \quad (4)$$

where c_1 and c_2 are the absolute velocity at the inlet and outlet of the impeller, u_1 and u_2 the corresponding blade speed values, and f is a friction coefficient, function of the mean flow velocity.

Under zero flow conditions, the velocity of the fluid in the casing is assumed equivalent to the blade speed u_2 . Further, the velocity at the inlet of the impeller can be neglected. Introducing the blade loading coefficient and blade Mach number, Equation (4) results into:

$$\Psi_0 \frac{u_2^2}{2} = u_2^2 - f(Ma) \frac{u_2^2}{2} \quad (5)$$

which leads to an expression for the load coefficient at zero flow:

$$\Psi_0 = 2 - f(Ma) \quad (6)$$

Note that the model obtained above is dimensionless, and does not depend on any design parameter.

Figure 6 compares the model described by Equation (6) to the limit condition Ψ_0 calculated from the JK models identified on the six compressors, for different Mach numbers. The parameter f can be

identified as a linear function of the blade Mach number based upon the data obtained from the JK model extrapolated at zero flow conditions, confirming a good agreement with the data.

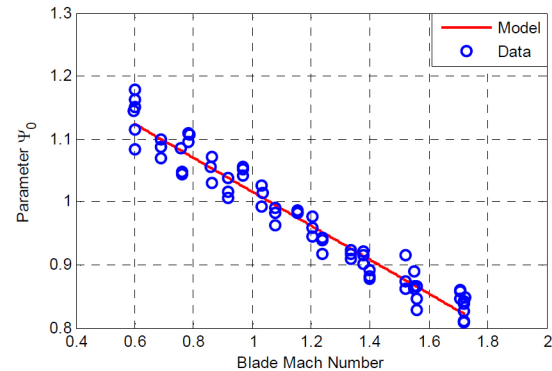


Figure 6. Identification of the load coefficient at zero flow condition.

The JK model presents also a vertical asymptote at $\Phi=K_3$. This property could be exploited to qualitatively describe the behavior of a centrifugal compressor approaching the choking at the inducer. For this limit condition, a mass and energy balance can be formulated to produce an analytical expression for the choking mass flow rate [8]:

$$\dot{m}_{choke} = \rho_1 c_1 A_t \left(\frac{2}{\gamma+1} + \frac{\gamma-1}{\gamma+1} \frac{u_1^2}{c_1^2} \right)^{\frac{\gamma+1}{2(\gamma-1)}} \quad (7)$$

where ρ_1 and c_1 are the density and absolute velocity at the inducer, and A_t the corresponding throat area. Converting Equation (7) to dimensionless form and introducing the trim τ as a design parameter for the compressor, the flow coefficient at the choking limit results:

$$\Phi_{choke} = k_1 \frac{T}{Ma} \cdot \left(\frac{2}{\gamma+1} + k_2 \cdot T \cdot Ma^2 \right)^{\frac{\gamma+1}{2(\gamma-1)}} \quad (8)$$

where the parameters k_1 and k_2 are calculated from the parameter K_3 (vertical asymptote) of the JK model calibrated against the data from the six compressors. This can be verified in Figure 7, where Φ_{choke} is calculated for different Mach numbers and represented against the data for different compressors (characterized by different design parameters). The accuracy of the model is higher for high Mach numbers, namely when the compressor characteristics curves become steeper at high flow rate conditions, validating the assumption that the parameter K_3 in the JK model approximates the choking limit Φ_{choke} .

Note that Equation (8) contains explicitly the trim, and is therefore suitable to describe the choking behavior of different compressors in geometric similarity. Since the trim τ influences Φ_{choke} in a nonlinear fashion, this approach for scaling the compressor flow maps is more accurate and physically consistent than simply applying linear scaling methods, as often done in common practice.