

Control-Oriented Compressor Model with Adiabatic Efficiency Extrapolation

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ABSTRACT

Downsizing and turbocharging with single or multiple stages has been one of the main solutions to decrease fuel consumption and harmful exhaust emissions, while keeping a sufficient power output. An accurate and reliable control-oriented compressor model can be very helpful during the development phase, as well as for engine calibration, control design, diagnostic purposes or observer design. A complete compressor model consisting of mass flow and efficiency models is developed and motivated. The proposed model is not only able to represent accurately the normal region measured in a compressor map but also it is capable to extrapolate to low compressor speeds. Moreover, the efficiency extrapolation is studied by analyzing the known problem with heat transfer from the hot turbine side, which introduces errors in the measurements done in standard gas stands. Since the parameterization of the model is an important and necessary step in the modeling, a tailored parameterization approach is presented based on Total Least Squares. A standard compressor map is the only data required to parameterize the model. The parameterization is tested with a database of more than 230 compressor maps showing that it can deal well with different compressor sizes and characteristics. Also, general initialization values for the model parameters are provided using the complete database parameterization results. The results show that the model accuracy is good and in general achieves relative errors below one percent. A comparison of the model accuracy for compressor maps with and without heat transfer influence is carried out, showing a similar model accuracy for both cases but better when no heat transfer is present. Furthermore, it is shown that the model is capable to predict the efficiency characteristics at low speed of two compressor maps, measured with near adiabatic conditions.

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1. INTRODUCTION

The legislation pressure on the exhaust emission limits drive the automotive industry into researching more advanced technologies. Moreover, the increasing fuel prices, also push the need to develop more and more fuel efficient internal combustion engines (ICE). One of the most popular solutions to achieve these demands is downsizing and turbocharging the ICE. Many examples of boosting systems exist nowadays, e.g. from various turbocharger stages to electrically driven compressors. Introducing more and newer components into a ICE makes the complete system more complex to deal with. First, in terms of system design choices, later due to the complexity that arises from having to control the ICE as efficient as possible. In order to overcome this, having a simulation model to apply model-based control techniques can be very useful to benchmark different system architectures as well as to test different control strategies.

In model-based control, a model that is capable to capture the main dynamic characteristics of the system is required. At the same time, this model has to be computationally low demanding. Mean value engine models (MVEM) fulfill these two requirements, and have been successfully applied to many different types of combustion engines, see e.g. [1, 2, 3]. A control-oriented compressor model is

required as a part of the complete MVEM. This family of compressor models has also received significant attention in the automotive research literature, see among many others [4, 5, 6, 7, 8, 9].

In the research literature, the compressor efficiency modeling has received less attention than the mass flow modeling. One reason for that could be the known problem with heat transfer from the hot turbine gases in the gas stand, which introduces errors in the compressor efficiency measurements. This issue has a greater effect at low compressor speeds, as it is pointed out in [10, 11, 12, 13, 14] among many others. The fact that normal compressor maps are only measured down to 35% - 40% of the maximum rotational speed adds even more uncertainty to what is the true value of the compressor efficiency in this area. This in turn makes the validation of the efficiency extrapolation a difficult task.

The main objective of this investigation is to continue the work carried out in [15] for centrifugal compressors used in marine propulsion. The focus here is to apply the modeling approach together with an updated estimation procedure to automotive compressors. Hence, the general applicability of the model and parameterization method to any compressor size is demonstrated. At the same time an investigation of how the heat transfer affects the compressor efficiency is done. This motivates some changes in the

efficiency model compared to the one proposed in [15], in order to be able to describe as good as possible the adiabatic efficiency at low compressor speeds.

2. COMPRESSOR PERFORMANCE

The performance of a centrifugal compressor is characterized by its compressor map. The map contains measurements of several different operating points at different compressor speeds. Measurements with the same compressor speed are connected together in the compressor map forming a Speed Line (SpL). The usual map variables are pressure ratio $\Pi_c = \frac{p_{02}}{p_{01}}$, efficiency η_c , corrected speed $\bar{N}_c$ and corrected mass flow $\bar{W}_c$. The corrected variables are computed as in [16]

$$\bar{N}_c = N_c \frac{1}{\sqrt{T_{01}/T_{c,ref}}} \quad (1)$$

$$\bar{W}_c = W_c \frac{\sqrt{T_{01}/T_{c,ref}}}{p_{01}/p_{c,ref}} \quad (2)$$

where $T_{c,ref}$ and $p_{c,ref}$ are the reference values. A complete nomenclature list can be found at the appendix. The derivation of the correction formulas is done using dimensional analysis, a complete explanation can be found in [17, 18, 19]. The main reason to have corrected quantities in the compressor map is to cover different inlet conditions with a single compressor map. Thus, for the corrected quantities to fulfill its purpose, the same pair of values ($\bar{N}_c, \bar{W}_c$) achieved with different inlet conditions should give the same pressure ratio and efficiency values. However, deviations of pressure ratio for the same corrected variables but with different pressure inlet conditions are observed in [20]. A similar study but for the efficiency case would be very interesting, but it is out of the scope of this study.

Compressor Efficiency and Heat Transfer

In a combustion engine installation, compressor operation will not be adiabatic. Nevertheless, knowing the true adiabatic performance of the compressor is important, since the heat transfer characteristics of the gas stand are not the same as in the combustion engine. This mismatch will introduce errors as pointed out in [12]. In case an accurate compressor outlet temperature is desired, the model can be complemented with a heat transfer model of the turbocharger, see e.g. [12, 21].

Compressor efficiency measures of how much of the mechanical energy available in the compressor shaft is used to increase the pressure of the working fluid. There are many definitions of compressor efficiency, see e.g. [18]. Through this study the isentropic definition is going to be used. When measuring a compressor performance map, efficiency is usually calculated from the total pressures and temperatures at the inlet, (T_{01}, p_{01}), and at the outlet, (T_{02}, p_{02}). Total pressures are computed as the sum of the static value plus a term corresponding to the velocity of the fluid, see 17 for more information. Using the total to total isentropic definition, the efficiency is computed as

$$\eta_c = \frac{\Delta h_{is}}{\Delta h_{act}} = \frac{\frac{p_{02}}{p_{01}}^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{02}}{T_{01}} - 1} \quad (3)$$

where Δh_{is} represents the required power to compress the fluid in the ideal case that the compression is done isentropically. On the other hand, Δh_{act} , represents the actual power required to compress the fluid. More information about this efficiency definition and its derivation can be found in [18, 19].

From this expression it is easy to identify that errors in the measured pressure ratio will affect the efficiency, see [22]. However, the inlet and outlet temperatures are the most critical measurements for the efficiency accuracy. If the temperature measurement is influenced by external heat transfer from the hot turbine side, the measurement and the calculated efficiency will be inaccurate. Furthermore, temperature measurements are always difficult to carry out. Heat transfer effects on efficiency measurements have received much attention in literature, see e.g. [10, 12, 13, 14, 21, 23, 24, 25, 26]. In particular, as pointed out in [10] and [23], the effects of the heat transfer in the efficiency measurements have a greater influence at low compressor speeds and low flows. Coolant water and lubricating oil temperature and pressure will affect the measured efficiency as well, see [10, 14]. On the other hand, heat transfer practically does not affect the mass flow and pressure ratio measurements as it can be seen in [10].

A method to assess whether or not the measured efficiency is affected by heat transfer is to plot the Euler work input coefficient λ_{Euler} as function of the flow coefficient at the impeller outlet ϕ_2 . These two variables should follow a linear dependence, see [27], following a mathematical function of the form

$$\lambda_{Euler} = 1 - \frac{c_s}{U_2} + \phi_2 \tan \beta_2 \quad (4)$$

where c_s/U_2 represents the slip factor and $\tan \beta_2$ is negative for a backswept impeller. The impeller outlet flow coefficient is not straightforward to compute with the usual signals available in a compressor map, thus it is better to rewrite (4) for the inlet flow coefficient as it is done in [27]. This can be done with the help of the change of density of the working fluid during the impeller stage, which yields to the following equation

$$\lambda_{Euler} = 1 - \frac{c_s}{U_2} + \phi_1 \frac{D_2}{b_2 \pi} \frac{\tan \beta_2}{[1 + (\gamma - 1)\gamma_{imp}\lambda M_{U_2}^2]^{\frac{1}{n_{imp}-1}}} \quad (5)$$

where λ is the work input coefficient, γ_{imp} is the kinematic degree of reaction, M_{U_2} is the tip-speed Mach number and n_{imp} is the polytropic exponent. The disc friction losses on the impeller disc establish the relation between the work input coefficient λ , and the euler work input coefficient λ_{Euler} , which can be expressed as

$$\lambda = \left(1 + \frac{k_{fric}}{\phi_1}\right) \lambda_{Euler} \quad (6)$$

According to [23, 24], the different speed lines of the compressor should collapse into a single line in the $\lambda_{Euler} - \phi_2$ plane, unless there is heat transfer affecting the efficiency measurements. Note that in order to compute ϕ_2 , compressor geometry details are required to calculate the velocity triangle at the impeller outlet. Since this information is not usually available in the usual compressor maps, the iterative method presented in [27] can be used to obtain ϕ_2 for vaneless diffuser compressors. For vaned diffuser compressors, a similar method but with different assumptions could be derived. The only geometrical parameters required are the impeller outlet wheel diameter, D_2 , and the not always available diffuser width, b_2 . Based on empirical measurements of compressor impellers and data available in the database, a value between 7 - 10% of the diameter D_2 is a fair approximation of b_2 . With ϕ_2 available, the correction method based on constant heat transfer from the hot turbine side, described in [23], can be applied. It requires a single parameter for the complete map correction that needs to be properly calculated.

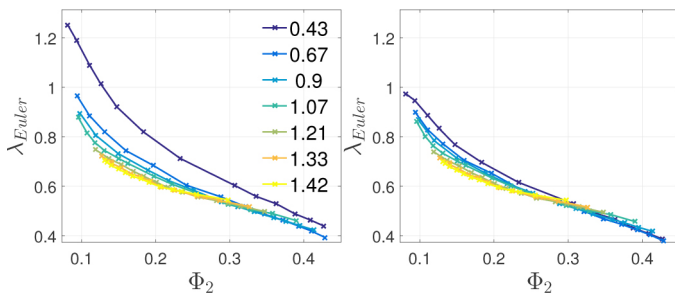


Figure 1. Euler work input coefficient vs outlet flow coefficient. Left; no heat transfer correction. Right; with the correction applied. The values in the legend correspond to different tip speed Mach numbers.

An example of a corrected Euler work input coefficient vs outlet flow coefficient can be seen in Figure 1. The Figure contains the same plot with and without applying the heat correction method. It clearly shows the lowest speed line having a much larger working input coefficient than the rest in the uncorrected measurements. When the Euler work input coefficient is corrected for adiabatic conditions, the efficiency lines are moved towards higher efficiency values.

It is important to mention that the obtained efficiency after the correction cannot be considered the exact adiabatic efficiency, since the correction method introduces uncertainties and assumptions. Moreover, the usual compressor maps do not contain much information about how the measurements are made. For example knowing if the turbine inlet temperature has been the same for all measured SpLs is important to be able to apply the correction method successfully.

3. EXPERIMENTAL DATA

A database of compressor maps is used to develop and validate the compressor model. The database contains 234 automotive compressor maps of different sizes from several manufacturers.

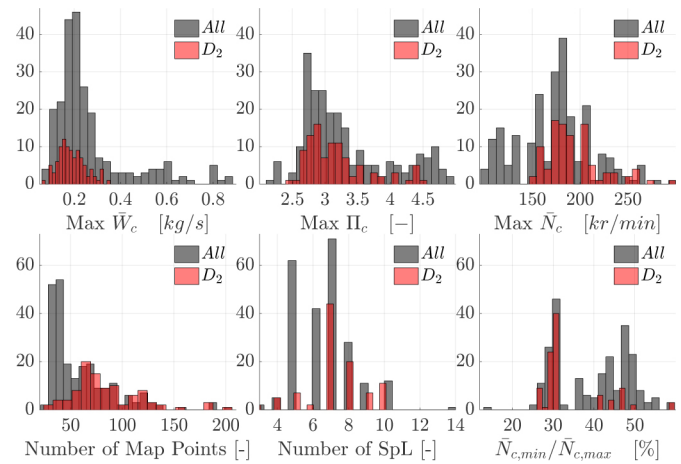


Figure 2. Histogram of the main characteristics of the compressor map database in gray. In red the main characteristics of the compressors with available outer impeller diameter, D_2 .

The main characteristics of the database are depicted in Figure 2. A subset of 98 compressor maps with available outer impeller diameter D_2 is selected from the database. This is done in order to be able to apply the heat correction method discussed in Section 2 and originally developed in [23]. An overview of the main characteristics of these maps is presented in Figure 2. It is interesting to note, as it can be seen in Figure 2, that there are very few maps with measured compressor speeds below 30% of the maximum compressor speed.

Two extended compressor maps, which are not included in the 234 database maps, are used to validate the extrapolation of the model at low compressor speeds. These maps, named TD04 and K04, are measured respectively down to 12% and 13% of the maximum measured speed. Furthermore, in order to minimize the heat transfer effects on the measured efficiency, the turbine inlet temperature in the gas stand is kept at 300K for the lower speed lines. Also the oil temperature and the coolant water temperature are kept as close as possible to the compressor outlet temperature. These conditions are considered to be really close to the true adiabatic case, see [10], thus these maps are called near adiabatic maps.

4. COMPRESSOR MODEL

The main purpose of the compressor modeling is to describe the compressor performance as a set of mathematical functions. The complete compressor model consists of a submodel to compute the compressor mass flow, and another submodel to compute the compressor efficiency. In a MVEM simulation, the surrounding pressures and the turbocharger speed are normally state variables. Hence, the compressor model is called with the following input-output relation

$$[W_c, \eta_c] = \text{CompressorModel}(\Pi_c, N_c) \quad (7)$$

with this setup, is important to note that efficiency and mass flow will be signals computed with the model, consequently its accuracy will be dependent on the modeling errors.

where $\bar{W}_{el}$ is an artificial mass flow, introduced here to allow the model to adapt better to the measured data. $\bar{W}_{el}$ is defined by the following ellipse function

$$\bar{W}_{el} = \bar{W}_{ZS} \left(1 - \left(1 - \frac{\bar{W}_c}{\bar{W}_{ZS}} \right)^{c_{15}} \right)^{1/c_{15}} \quad (13)$$

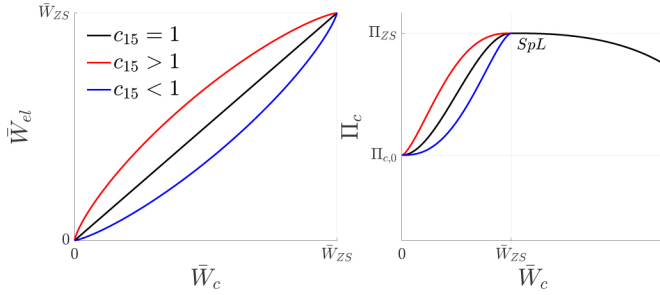


Figure 4. Sketch of the shape of the unstable arc depending on the parameter c_{15} . Left plot corresponds to the shape of equation (13). Right plot shows for the same c_{15} values, the shape change in equation (12).

where c_{15} is an estimation parameter that shapes the third-order polynomial in (12). When $c_{15} = 1$, the artificial mass flow is equal to the corrected compressor mass flow $\bar{W}_{el} = \bar{W}_c$, and thus (12) is not modified. If $c_{15} < 1$, the artificial mass flow lags behind the true compressor mass flow, and the SpL curve (12) moves to the right. On the other hand, when $c_{15} > 1$, the artificial mass flow advances the true compressor mass flow and thus the SpL curve (12) moves to the left. A sketch of the flexibility introduced in the model by (13) can be seen in Figure 4.

The pressure ratio at zero flow $\Pi_{c,0}$ is defined as

$$\Pi_{c,0} = \Pi_{ZS} - \Gamma_{\Pi_{cs}} (\Pi_{ZS} - 1) \quad (14)$$

where $\Gamma_{\Pi_{cs}}$ is a constant that can be adjusted for a given map if surge measurements are available. Since this is not normally the case in the standard compressor maps, the value is fixed here to $\Gamma_{\Pi_{cs}} = 1/2$, following the rules of thumb from [31]. Furthermore, the model can be extended for reverse flows using a turbine mass flow model, see [19], since in this region the compressor can be seen as a poorly designed turbine.

For mass flows greater than the choke flow ($\bar{W}_c > \bar{W}_{Ch}$) the model is assumed to be a vertical line. This is done to resemble the choking physical meaning. So for any pressure ratio smaller than the choking pressure ratio Π_{Ch} , the compressor mass flow is saturated at $\bar{W}_{Ch}$. In case this saturation is not suitable for the simulation model, it can be extended in this region with the equation proposed in [29], however this extension is not further investigated here.

Efficiency Model

The efficiency model presented here is an extension of the model presented in [15], but focused on vaneless automotive radial compressors. Moreover, the main goal in the efficiency modeling is to have a model capable of predicting the efficiency at low compressor

speeds for a given compressor map with adiabatic efficiency measurements. As pointed out in Section 2, heat transfer greatly influences the efficiency measurements accuracy, specially at low SpLs. As shown in [10], the maximum adiabatic efficiency per SpL is quite constant, so the drop in efficiency when speed decreases observed in most manufacturer maps is a direct consequence of the heat transfer. Thus, the model has to be able to keep a high maximum efficiency, per SpL, at low compressor speeds.

The novel idea in the development of the model is to relate the previous linear modeling in the $\Delta h'_{act} - W_c$ plane, see e.g. in [6, 32], to a more general model which incorporates the investigations from [27, 23] for the shape of the SpLs in the $\lambda - \phi_1$ plane. The derivation of the base functions for the efficiency model starts with equation (5). Using the definitions of the work input and flow coefficients from [18]

$$\lambda_{Euler} = \frac{\Delta h'_{act}}{U_2^2} \quad \phi_1 = \frac{W_c}{\rho_{01} D_2^2 U_2} \quad (15)$$

the equation (5) can be written in the $\Delta h'_{act} - W_c$ plane as

$$\Delta h'_{act} = \left(1 - \frac{c_s}{U_2} \right) U_2^2 + W_c \frac{U_2}{\rho_{01} D_2 b_2 \pi} \frac{\tan \beta_2}{[1 + (\gamma - 1) \gamma_{imp} \lambda M_{U_2}^2]^{\frac{1}{n_{imp} - 1}}} \quad (16)$$

where $\Delta h'_{act}$ is used instead of Δh_{act} to distinguish between the actual enthalpy computed with or without the disc friction losses defined by (6). In [15] the actual enthalpy was modeled as an affine function of mass flow. With the affine function parameters depending only on compressor speed with the following structure

$$\Delta h'_{act} = b(U_2) - a(U_2) \cdot W_c \quad (17)$$

identifying the terms b and a in (16) yields

$$b(U_2) = \left(1 - \frac{c_s}{U_2} \right) U_2^2 \quad (18a)$$

$$a(U_2) = \frac{U_2}{\rho_{01} D_2 b_2 \pi} \frac{\tan \beta_2}{[1 + (\gamma - 1) \gamma_{imp} \lambda M_{U_2}^2]^{\frac{1}{n_{imp} - 1}}} \quad (18b)$$

It is important to mention that the theoretical expression, (15), is not defined with the corrected quantities of mass flow and compressor speed introduced in (1) and (2). However, in the following definition and motivation of the base functions for the efficiency model, the corrected quantities are used. This is done because in the majority of the conventional compressor maps the only signals available are the corrected quantities. Furthermore, this is shown to give good modeling results later in Section 6 and it gives a convenient way to