

Validation of a 1D Compressor Model for Performance Prediction

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ABSTRACT

In the present paper, a recently developed centrifugal compressor model is briefly summarized. It provides a refined geometrical schematization of the device, especially of the impeller, starting from a reduced set of linear and angular dimensions. A geometrical module reproduces the 3D geometry of the impeller and furnishes the data employed to solve the 1D flow equations inside the rotating and stationary ducts constituting the complete device. The 1D compressor model allows to predict the performance maps (pressure ratio and efficiency) with good accuracy, once the tuning of a number of parameters is realized to characterize various flow losses and heat exchange.

To overcome the limitations related to the model tuning, unknown parameters are selected with reference to 5 different devices employing an optimization procedure (modeFRONTIER™). A unique set of tuning constants is identified by the optimizer, which is able to provide, in most of cases, a good agreement with the experimental maps of each compressor.

To verify the reliability of the previously selected parameters, the tuned model is employed to compute the performance map of an additional compressor of different geometry, not included in the tuning procedure. The predicted map shows a good agreement with the experimental one.

The proposed model can hence be very helpful in providing with good accuracy a first estimate of the performance maps of a centrifugal compressor, only basing on its geometry, and in absence of manufacturer data.

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INTRODUCTION

As widely known, the turbocharging is commonly employed for automotive Diesel engine applications with the aim to improve both the power/torque performance and to reduce the pollutant emissions [1,2]. More recently, this technology is widely used for Spark-Ignition (SI) engines, too. In the latter case, the turbocharging technique focuses on the research of strategies for fuel consumption reduction [3,4]

In particular, a reduced displacement (downsizing) and a turbocharging system are simultaneously employed to guarantee a fixed prescribed power output. In such a way, during a typical urban driving cycle, the downsized engine operates at higher loads with a positive impact on the specific fuel consumption. In addition, an increased low-speed torque, a better vehicle drivability and fun-to-drive are obtained.

The development of a new turbocharged engine involves the computation of the engine and turbocharger matching conditions. This is usually realized through the employment of 1D codes [5,6] which require the knowledge of compressor and turbine characteristic maps. Very often the manufacturers are reluctant to provide the turbocharger maps; the latter, moreover, usually cover a limited portion of the turbomachinery operating domain, in particular in case of the turbine.

Due to the above considerations, since some years, a 1D compressor model has been continuously developed at Dept. of Industrial Engineering of University of Naples, with the aim of reproducing the steady performance maps of the device. The computed map, moreover, covers the whole compressor operating domain, including unstable and reverse flow regimes [7,8]. Such analytical tool is hence very helpful to the engine designers, since it provides a feedback to the

turbocharger manufacturers and drives the iterative procedure aiming to select the best device to be selected for a specified engine.

The 1D procedure requires the direct measurement on the physical device of a reduced set of geometrical data. The compressor is schematized as a series of four pipes: the inlet duct, the impeller, the diffuser and the volute. The impeller itself is considered as a number of rotating channels in parallel. In particular, the geometry of the blade-to-blade duct is specified through the assignment of a limited number of geometrical data, which define the duct orientation in the space along its curvilinear abscissa. With the aim of describing the main phenomena that characterize the actual behavior of the device, some simple loss correlations have been introduced in the model [9], each of them multiplied by a tuning constant. Of course, the presence of the above unknown parameters introduces a lack of generality of the model. Tuning constants must be in fact precisely identified with reference to experimental data.

To overcome the above limitation, in this paper an optimization procedure is described to identify a generally applicable set of tuning constants. In particular, 5 different compressors are analyzed and a unique set of tuning constants is identified by the optimizer (modeFRONTIER™) which is able to provide, in most of cases, a good agreement with the experimental maps of each compressor.

In the following paragraphs, the geometrical module for the compressor schematization and the flow equations are recalled. The optimization tool is firstly used to identify a dedicated set of tuning constants for each compressor. The same method is then applied to perform a “multi-compressor” tuning procedure, being able to provide a satisfactory agreement with experimental data for all the investigated devices. Once calibrated, the procedure reliability is finally verified through the prediction of the performance maps of a further compressor, only knowing its geometrical data.

GEOMETRICAL SCHEMATIZATION

A centrifugal compressor is schematized as follows (fig. 1):

- A short inlet pipe (from section 0 to 1) of constant diameter, D_0 , and L_0 length; the inlet pipe end section is characterized by a sudden area contraction, due to the impeller eye obstruction.
- The impeller (from section 1 to 2), consisting of a Z rotating pipes, Z being the number of blades. The impeller geometrical schematization will be discussed in detail in the following. The procedure provides as an output the cross section area, the hydraulic diameter, the local blade/rotation angles and the radius evolution along the curvilinear abscissa, to be used in the 1D flow equations solution.
- The vaneless diffuser (from section 2 to 3); it is considered as a diverging pipe of constant width along the radial direction.

- The volute (from section 3 to 4); it is schematized as a linearly variable diameter duct, collecting the flow coming from the diffuser and ending with a cone. The volute equivalent diameters at inlet and outlet are derived by the diffuser outlet area and the cone inlet diameter, D_{ex-vol} , respectively. An equivalent length is considered as half the volute centerline length.

Table 1 reports the geometrical data required by the procedure to completely characterize the device geometry. The meaning of the geometrical parameters is depicted in figs. 2, 3, 4. The data refer to 5 different small and medium-size turbocharger compressors (*A*, *B*, *C*, *D* and *E*).

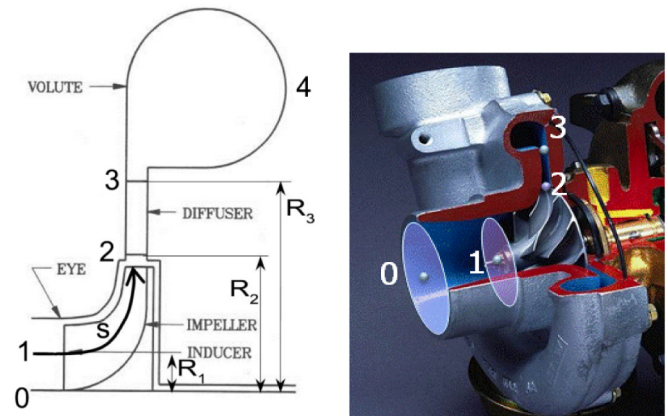


Figure 1. Main compressor stations

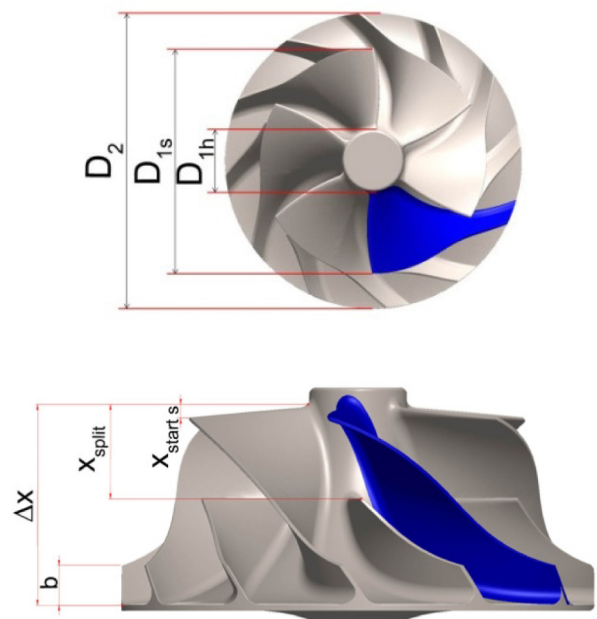


Figure 2. Main linear dimensions of the impeller

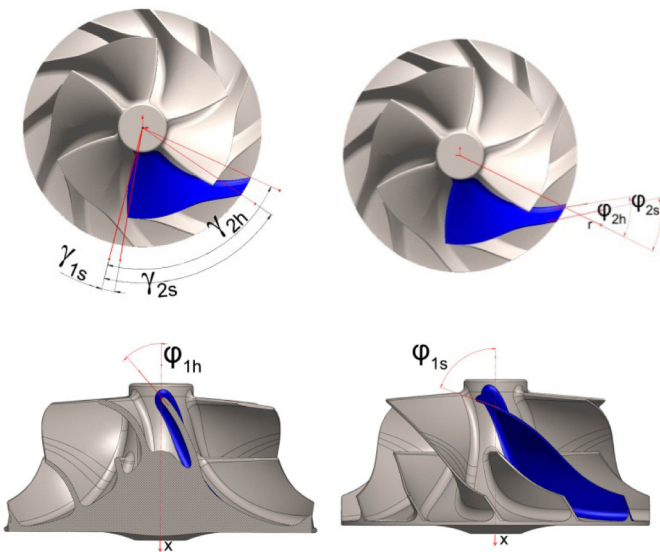


Figure 3. Main angular dimensions of the impeller

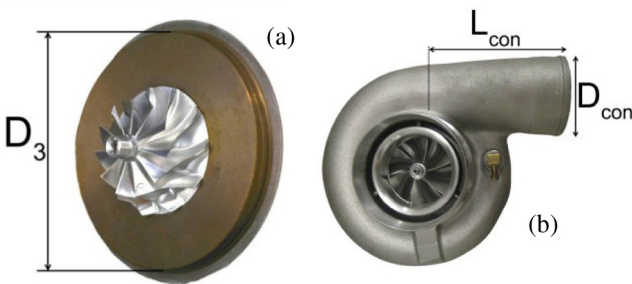


Figure 4. Characteristic dimensions of the volute (a) and diffuser (b)

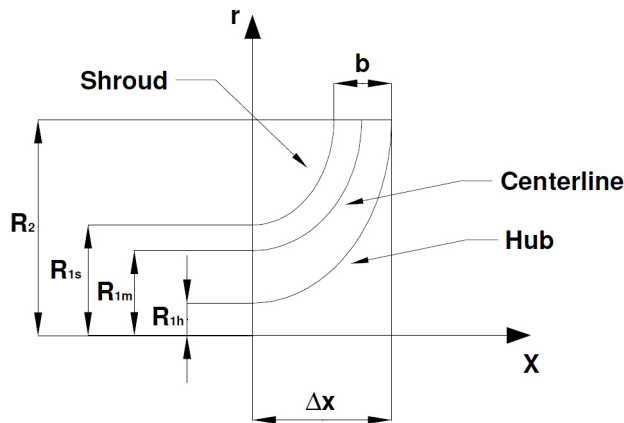


Figure 5. Data required for drawing the shroud, hub and mean elliptical meridian profiles

Table 1. Data required by the geometrical module

	A	B	C	D	E
INLET					
D_0	44.00	34.50	33.30	47.19	44.35
L_0	25.10	70.00	56.00	28.80	76.95
IMPELLER					
Z	12	8	12	8	12
Δx	24.03	13.20	13.80	16.20	14.90
b	4.69	3.00	3.30	3.50	3.50
$X_{start s}$	1.53	0.00	0.00	0.00	1.81
$X_{split h}$	11.05	4.50	5.00	5.60	4.28
$X_{split s}$	11.35	4.50	5.00	4.30	4.28
D_{1h}	12.04	10.00	9.80	13.10	10.02
D_{1s}	44.17	29.00	28.90	36.00	33.24
D_{2h}	58.00	40.00	38.80	51.00	45.25
D_{2s}	58.00	40.00	38.80	51.00	45.25
ϕ_{1h}	23.85	23.00	41.00	30.00	35.00
ϕ_{1s}	73.50	54.00	70.00	53.00	57.50
ϕ_{2h}	40.00	43.00	24.00	43.00	40.00
ϕ_{2s}	37.50	45.00	47.00	40.00	37.30
γ_{1h}	0.00	0.00	0.00	0.00	0.00
γ_{1s}	-6.50	5.00	2.00	0.00	6.44
γ_{2h}	77.06	70.00	67.00	67.00	59.35
γ_{2s}	70.75	64.00	59.75	58.00	50.55
DIFFUSER					
D_3	107.40	64.00	84.70	79.70	88.40
VOLUTE					
$D_{ex vol}$	30.85	26.80	29.00	30.00	32.40
D_{con}	55.00	26.80	29.00	55.00	32.40
L_{con}	102.00	62.00	44.30	59.00	76.90

Special attention is devoted to accurately reconstruct the complex 3D geometry of the impeller. To this aim, two approaches can be followed according to the desired accuracy degree. The simpler one is based on the assumption that the meridian profiles (hub, shroud and centerline) are elliptical and that the blade rotation angle γ is a cubic function of the curvilinear abscissa, m (single spline). In such a way, the meridian profiles are fully defined by the radii R_{1h} , R_{1s} , R_2 , and by the lengths b and Δx (fig. 5). The centerline inlet radius is assumed equal to the geometrical average between hub and shroud radii, eq. (1). This assumption ensures that the centerline divides the blade to blade duct in two flow pipes delivering the same mass flow rate. Since rotation angle γ and blade angle ϕ are related by eq. (2), as shown in fig. 6, the function $\gamma(m)$ is completely defined by the blade and rotation angles at inlet and outlet.

$$R_{1m} = \sqrt{\frac{R_{1h}^2 + R_{1s}^2}{2}}$$

$$\frac{d\gamma}{dm} = \frac{tg\varphi}{r} \quad (2)$$

Basing on the previous assumptions, the 3D blade geometry is reproduced simply rotating each point of the meridian profiles by the local γ angle.

A more refined geometrical schematization can be also used, providing a more accurate impeller reconstruction, at the expense of an increased amount of geometrical input data. First of all, the meridian hub and shroud profiles are described by 2-control points Bézier curves; the control points' positions hence represent 4 additional geometrical parameters required by the procedure (fig. 7). Fig. 8 highlights the improvement achieved in the meridian profile reconstruction employing the more sophisticated approach with respect the simple elliptical curve; geometrical data derived by an optical reverse of the physical device are used as a reference to compare the results of the different profile reconstruction logics.

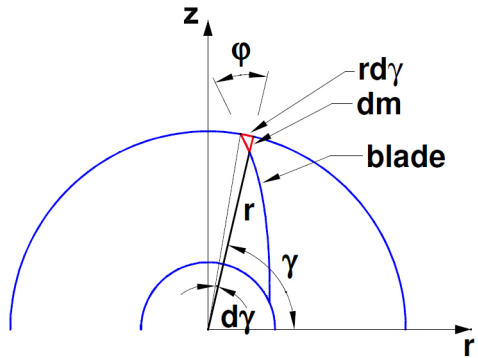


Figure 6. Rotation and blade angles of the impeller.

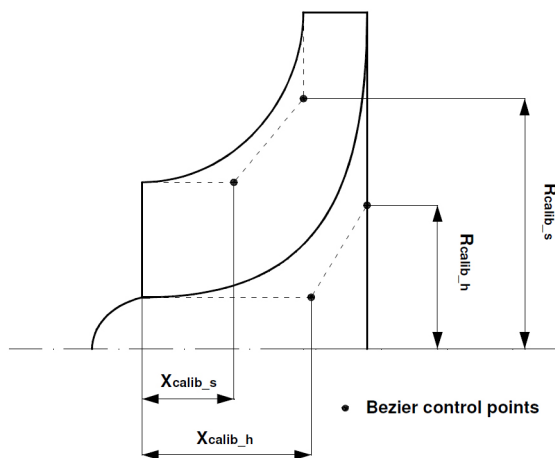


Figure 7. Control points for drawing Bézier meridian profiles

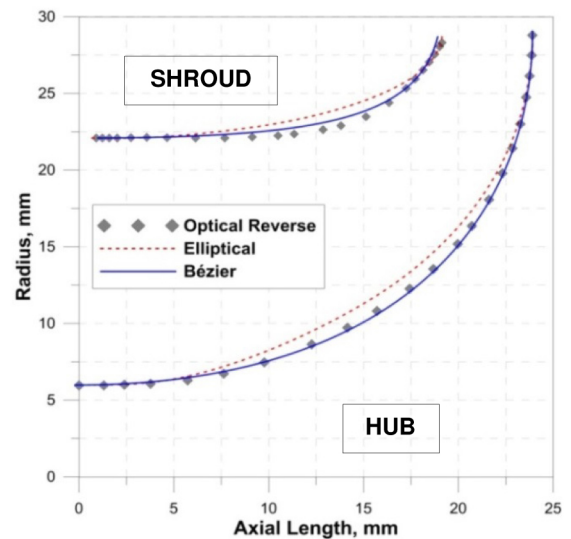


Figure 8. Meridional profiles, comparison between optical reverse data, elliptical and Bézier curves

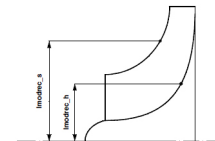


Figure 9. Control parameters for the rotation blade angle description using the "double spline" approach

Table 2. Data for the refined impeller geometrical reconstruction

	A	B	C	D	E
$X_{calib\ h}$	17.25	9.48	10.25	/	2.00
$X_{calib\ s}$	10.00	5.49	8.75	/	9.60
$R_{calib\ h}$	12.00	6.59	5.50	/	20.00
$R_{calib\ s}$	6.00	3.30	2.00	/	2.50
$Imodrec_h$	8.16	/	9.14	/	9.70
$Imodrec_s$	22.62	/	15.20	/	17.34
φ_{xh}	7.20	/	20.78	/	5.00
φ_{xs}	47.00	/	43.25	/	37.00
γ_{xh}	40.20	/	45.10	/	42.00
γ_{xs}	44.52	/	32.74	/	34.50

A further improvement in the geometrical schematization can be obtained using a double spline formulation for the $\gamma(m)$ function. Also in this case, an higher flexibility in the blade drawing is provided by assigning the radial position of the joint point between the two splines (*Imodrec* in [fig. 9](#)) and the related blade and rotation angles, ϕ_x and γ_x . [Fig. 10](#) shows that the refined approach allows to fit with very high accuracy the data derived by the actual impeller geometry. As already mentioned, however, the described sophisticated logics for a refined geometrical reproduction also require additional - and difficult to realize - measurements on the physical device. The latter are listed in [table 2](#).

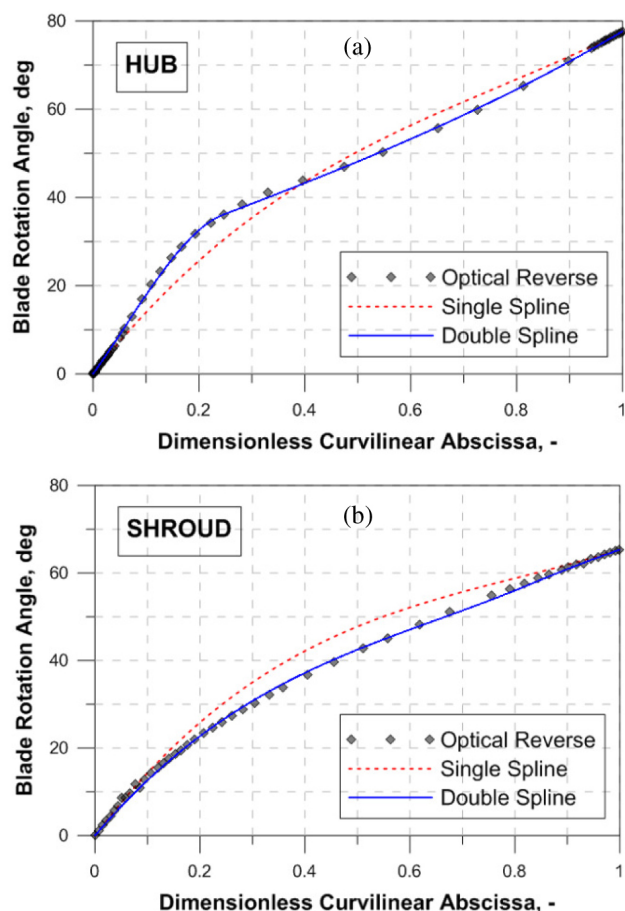


Figure 10. Comparison between optical reverse data, simple spline and double spline reconstruction for hub (a) and shroud (b) profiles

Basing on the above procedure, an excellent reconstruction of the 3D blade profile can be accomplished as shown in [fig. 11](#). The latter depicts the comparison (overlapping) between the actual impeller geometry (grey) and the reproduced one (blue).

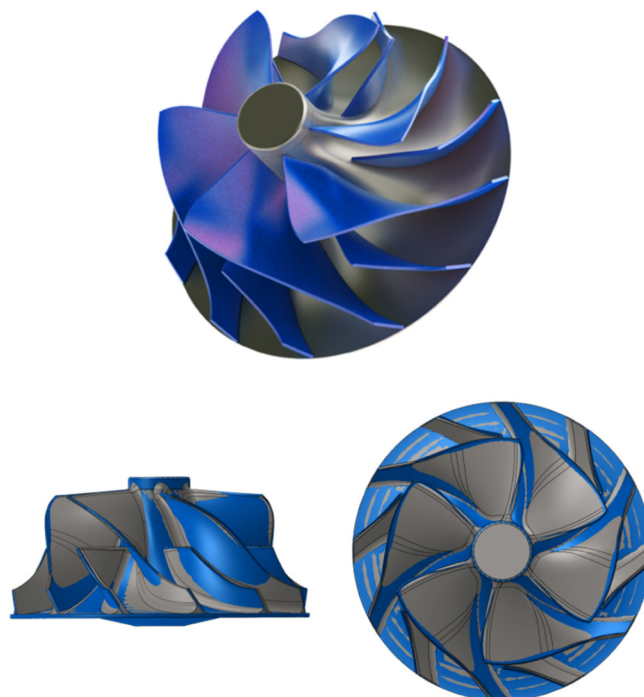


Figure 11. Impeller reconstruction (blue) and optical "reverse engineering" CAD (grey)

1D FLOW MODEL

Fixed Duct

The flow motion in the stationary ducts is described by the following steady 1D equations:

$$[\mathbf{F}(\mathbf{C})]_x = \mathbf{S}_C \quad (3)$$

$$\mathbf{C} = \begin{Bmatrix} \rho \\ \rho c \\ \rho E_C \end{Bmatrix} \quad \mathbf{F}(\mathbf{C}) = \begin{Bmatrix} \rho c \\ \rho c^2 + p \\ \rho c H_C \end{Bmatrix} \quad (4)$$

$$\mathbf{S}_C = - \begin{Bmatrix} \rho c \alpha \\ \rho c^2 \left(\alpha + 2 \frac{f}{d} \frac{c}{|c|} \right) \\ \rho c H_c \alpha - 4 \frac{q}{d} \end{Bmatrix} \quad (5)$$

The terms ρ , c , p , $H_C = c_p T_0 = c_p T + c^2/2$ in the system (3), (4), (5) represent the density, the absolute velocity, the static pressure, the total enthalpy per unit mass, respectively. The source term, $\mathbf{S}_C$, accounts for the friction coefficient, f , the rate of heat exchange, q , and the duct area variation, $\alpha = 1/\Omega \, d\Omega/dx$. The latter is provided at each curvilinear abscissa s by the previously described geometrical module.