COMPARISON OF THE PREDICTION PERFORMANCES OF DIFFERENT MODELS OF RADIAL TURBINE UNDER STEADY AND UNSTEADY FLOW CONDITIONS

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Abstract: *One of the techniques to reduce fuel consumption of automotive engines is downsizing. This technique leads to a great increase of the boost level, which corresponds to increase in the power recovered by the turbine. However, the turbine is driven by exhaust gases from the engine which are, by nature, highly compressible and unsteady. This has made critical, the understanding and prediction of the influence of unsteady phenomena on the turbocharger turbine's behavior. This paper deals with the comparison between experimental and numerical results of pressure waves influence on the turbine's behaviour. Regarding the part dedicated to 1D numerical calculation, two different turbine models were used: a literature one and a laboratory one. Finally, a new model suitable to modelise variable and fixed geometry turbines, in zero or one-dimensional fluid dynamics codes has been developed. It was conclude that this model is able to reproduce with good accuracy the fluid-dynamic behaviour of the turbine operating under unsteady flow conditions by using the maps supplied by the manufacturer as the only input data.*

Keywords: Turbochargers, radial turbine, unsteady flow, 1D modelling, compressible flow.

INTRODUCTION

Turbocharging can both increase the power of an engine as well as decrease the fuel consumption of a vehicle. This prediction capability is fundamental for choosing the right turbocharger early in the development phase, for the development of control strategies and for the selection of the required devices. Compressor and turbine characterization is usually made by turbocharger manufacturers under steady flow conditions. However, the exhaust gases of an IC engine are quite different from the steady flow test conditions. The unsteady exhaust flow nature and its effect on the turbine performance has been the subject of investigation for the past few decades (1-3), whereas the turbine is usually tested in steady flow conditions. Actually several different turbine models have been published in the literature using the turbine characteristic curves (4). The main difference between these models is in the way the fluid-dynamic behaviour is idealized by simple elements, thus simplifying the calculation procedures and reducing the involved computing times.

MODELISATION

This model is a "free" filling-emptying model, because unlike the model of Serrano *et al.*(5), it does not limit the effects of unsteady mass flow upstream of the turbine nozzle. The mass flow can go in and out from the volume representing the volute without any entropy change but with an adaptable pressure loss. Many researchers have observed and measured the unsteady phenomena (6), (7-8). As indicated by Copeland *et al.*(9), some fundamental discoveries have been made in this domain, two of the most important certainly are:

- The air volume (volute) before the nozzles (stator + rotor) acts as a reservoir, which can accumulate and empty the mass of fluid over a pulsation cycle. In addition to the processes of filling-emptying, there is an additional influence of the dynamic waves on the length of

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passage. The balance of these two effects determines the dynamic behaviour of the flow entering the rotor and thus the power it produces.

- For the typical frequency ranges encountered in an internal combustion engine, the rise in pressure is too fast to allow the complete mass accumulation. This hysteresis created between pressure and mass flow produces an operating loop which can encircle the quasi-steady operating line.

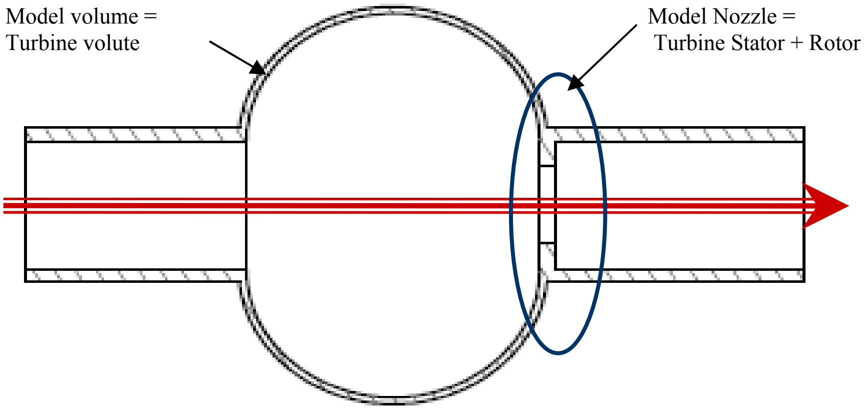

Figure 1. Turbine model scheme.

Figure2, the turbine model consists of a storage volume (equal to the volute capacity) coupled to a single nozzle (0D). Figure 3, as for the model of Serrano *et al.*(5), this model is coupled to pipe wave action models (1D) which solves the gas dynamics equations of the one-dimensional elements through the use of a modified Harten-Lax-Leer numerical calculation scheme Toro *et al.*(10). Pipes are respectively placed upstream (intake) and downstream (exhaust) of the turbine.

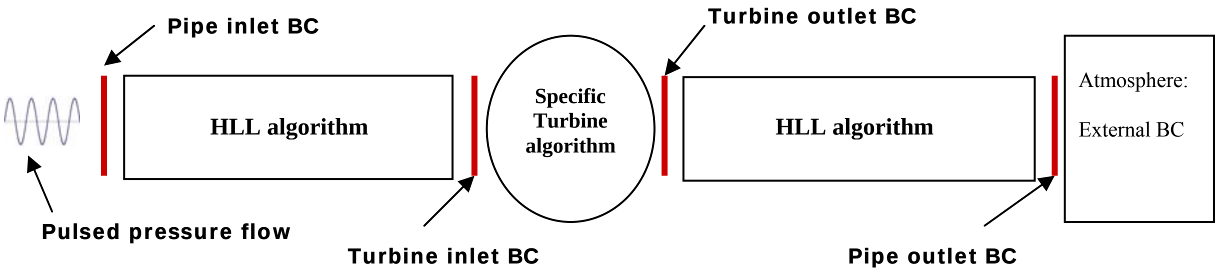

Figure 2. Simulation Code architecture.

The study of the phenomena occurring in each tube requires knowledge of their boundary conditions of them. This implies the use of open-end, nozzle boundary conditions and connection conditions for turbine boundary. The pressure excitation is made via the inlet boundary condition of the tube A. the pressure wave propagates along the inlet pipe, then across the turbine until the end of the outlet pipe.
The resolution of the system of equations for each boundary condition can be done by two different methods, the method of characteristics (Riemann variables) and by the method of fictive points/domains (meshed method) used by Chalet *et al.*(11,12). In this work, the latter is used, the fictive points method for the encoding of these turbine models, because it is a conservative method.

The aim of this work is to extrapolate the fields of turbine from the points measured on the turbocharger test bench. Indeed, measurements are generally carried out partially, that is to say they present only a few operating points by iso-speed data. Thus, based only on the test bench measurements, it would be impossible to estimate consistent mass flow rate values and operating performances on a single iso-velocity with a variation of the expansion ratio across the turbine. The

0D / 1D models accuracy depend strongly of the interpolation and extrapolation process and on a data map that doesn't take into account real engine conditions. So, for the cases where pure mathematical interpolation and extrapolation methods are used directly from the supplier data maps, the validity of the model results can be questioned. This is why, an extrapolation method inspired in large part, from the methods of Martin *et al.*(13) and Jensen *et al.*(14) is employed. It incorporates physical laws, with some extra-correlation tools as the corrected pressure ratio. Several attempts have been made to this framework; only the most effective extrapolation method is presented in this paper. To take into account the displacement of the critical expansion ratio in the turbine map, caused by the changes of flow relative velocity in the turbine, a parameter called "corrected pressure ratio" depending of the turbine rotation speed is introduced. This corrected expansion ratio is calculated via a simple linear equation.

Step 1: assess the critical expansion ratio for three (or more) iso-velocities. For a Barre St Venant equation (of a flow through a nozzle) the sonic blockage at the nozzle throat is product from an expansion ratio equal to:

$$\tau_{crit} = \left(\frac{P_e^*}{P_s}\right)_{crit} = \left(\frac{\gamma+1}{2}\right)^{\frac{\gamma}{\gamma-1}} \tag{1}$$

Step 2: Due to the moving parts inside the turbine which generate flow disturbances, the critical expansion ratio usually measured increases with the increase of the rotational rotor's velocity. For this reason, the corrected expansion ratio is introduced to take into account this phenomenon, it is expressed as a linear equation function of the real expansion ratio (equal to that measured on test bench) :

$$\tau_{crg}(\tau_p) = A_p(Nturb).\tau_p + B_p(Nturb) \tag{2}$$

The parameters $A_p(Nturb)$ and $B_p(Nturb)$ are defined from a polynomial regression (order 2 for my case) made in the evolution field of the parameters A and B defining the line of the corrected expansion ratio for each iso-velocity.

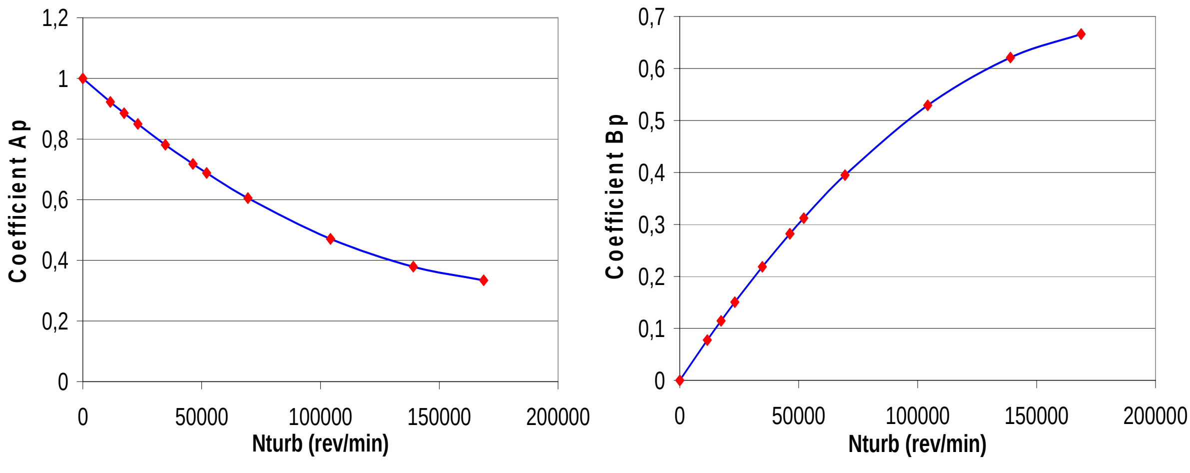

Figure 3. Evolution of the parameters Ap et Bp, in function of the rotor velocity.

Step 3: Now that the expansion ratio is fixed for all the defined iso-velocities (between 0 and 200 000tr/min), the effective area of the nozzle must be defined. Indeed, it is possible to characterize the reduced mass flow evolution via the nozzle equations of Barre St Venant (3), (4), as follows:

$$\dot{m}_{red} = S_{eff}\cdot\sqrt{\frac{2\gamma}{\gamma-1}\cdot\left[\left(\frac{1}{\tau_{crg}}\right)^{\frac{2}{\gamma}}\cdot\left(\frac{1}{\tau_{crg}}\right)^{\frac{\gamma}{\gamma-1}}\right]} \quad \text{when} \quad \tau_{crg} < \tau_{crit} \tag{3}$$

$$\dot{m}_{red} = S_{eff} \cdot \sqrt{\gamma \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma+1}{\gamma-1}}} \quad \text{when } \tau_{crg} > \tau_{crit} \tag{4}$$

Step 4: To adjust the values of the calculated effective section, two parameters are used, the corrected expansion ratio and rotor velocity of the turbine. The values of effective sections are calculated to verify that the reduced mass flow rate from the measurements corresponds to the reduced mass flows rate calculated using the nozzle equations of Barre St Venant (3), (4). From these values, a second degree trend is built which allows a good compromise between simplicity and accuracy (5).

$$S_{eff} = A_1.N_{turb}^2 + A_2.N_{turb} + A_3 \tag{5}$$

Now, all parameters needed for the extrapolation of the flow field of the turbine are defined.

Concerning the extrapolation of the turbine efficiency map; much of the method of Martin *et al.*(13) was used. Indeed, it is simple, based on the turbine physical characteristics, and shows that the enthalpy change in the turbine operates in a linear manner with is the corrected flow rate. So, the specific enthalpy change is expressed as:

$$\Delta h = A_e(Nturb).\dot{m}_{red} + B_e(Nturb) \tag{6}$$

The coefficients A_e and B_e are determined by the mean of a least square method based on the supplier's date map. Of course if the rotor is stationary, no power can be captured; so, in this case we will have $\Delta h = 0$. Furthermore, it is possible to make the same observation in term of mass flow, if the mass flow is null, the collected power is necessarily null. Knowing all this, it is possible to determine the origin points of the coefficient A_e and B_e, such as $A_e(0) = 0$ and $B_e(0) = 0$. Thus, the evolution of the parameters A_e and B_e can be plotted versus the rotational speed of the turbine rotor.

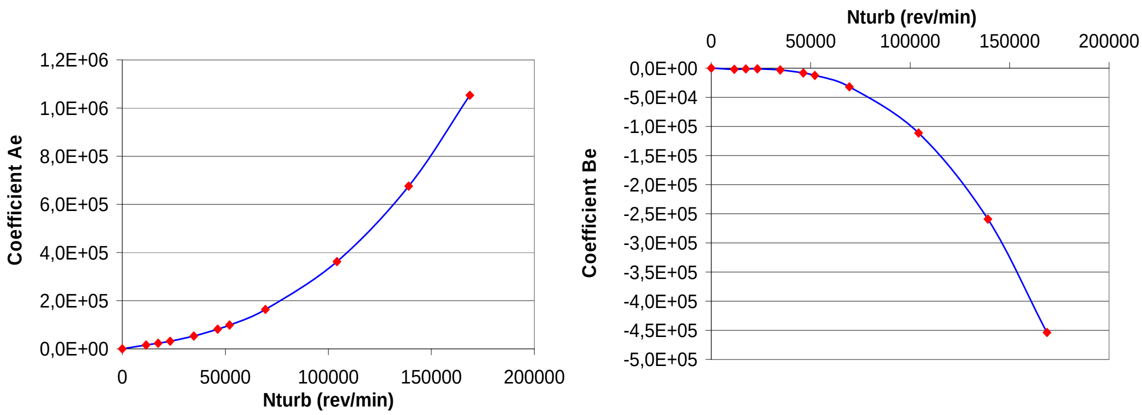

Figure 4. Evolution of the parameters Ae et Be, in function of the rotor velocity.

Considering a quasi-steady behaviour of the turbine at each time step of the calculation; all the thermodynamic variables of the fluid remain constant during this same time interval. So, it is possible to calculate the total-to-static turbine efficiency as follow:

$$\eta_{Ts} = \frac{\Delta h}{\Delta h_{is}} = \frac{h_e^* - h_s^*}{\left(h_e^* - h_s\right)_{Is}} \tag{7}$$

Where:

$h_e^* - h_s^*$: Represent the effective turbine energy consumption produced by the difference between the total enthalpy at the turbine inlet and the total enthalpy at the turbine outlet.

$\left(h_e^* - h_s\right)_{Is}$: Represent the ideal turbine energy consumption produced by the difference between the total enthalpy at the turbine inlet and the ideal static enthalpy at the outlet.

Hence, the performance values from the extrapolation can be easily determined by using the classic formula:

$$\eta_{Ts} = \frac{\Delta h}{\Delta h_{is}} = \frac{a(Nturb).\dot{m}_{red} + b(Nturb)}{\left(1 - \left(\frac{1}{\tau_P}\right)^{\frac{\gamma-1}{\gamma}}\right).Cp.T_e^*} \tag{8}$$

The results in figure 5 show a very good correlation between the extrapolated values and the experimental data. About the turbine efficiency extrapolation method, it can be noted that for the low rotational speeds of the turbine rotor, the quality and number of measurements become critical. Indeed, the operation range is very reduced at these low rotational speeds; and a large number of curves can be extrapolated from them (because of concentration of points in the zone). Thus, a substantial number of points on a wide range of expansion rate must be taken, in order to avert the possibility of having turbine efficiency predictions greater than 1.

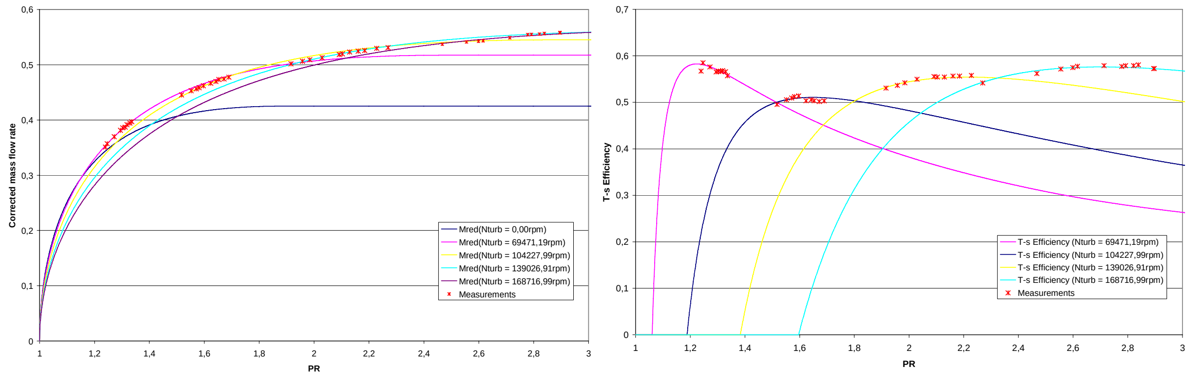

Figure 5. Results of the turbine data map extrapolations.

MODELS RESULTS

Steady prediction comparison

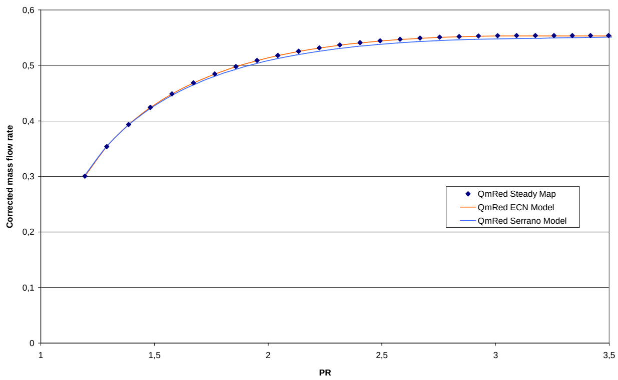

Figure 6: Reduced mass flow vs Pressure ratio plot: steady comparison of the models.

The calibration of the initial model is carried out by running under steady operation. When a wide range of expansion ratios has been covered, the computational reduced mass flow rate and the expansion ratio plot, shown in Figure 6, is constructed. The validated characteristics correspond to the experimental tests values under steady flow conditions with the turbine shaft running at 120 000 rev/min; note that only one turbine shaft speed has been presented here but other variables were also tested. Both models predictions are quite similar. But, due to the direct interpolation of the turbine extrapolated data map, the laboratory model has a slightly improved accuracy (at the middle of the pressure ratio range) compared to the model of Serrano *et al.*(5). Concerning the total-to-static turbine efficiency, both models offer comparable performances. As for the reduced mass flow parameter, the predicted and the measured data have a high degree of correlation.

Unsteady prediction comparison

The tests were performed on a turbocharger test bench; the pulsed air supply is carried out via an engine cylinder head (=pulse generator) on which is collected a pressure wave at the outlet of a single cylinder. The pressure profile provided by the experiment produces a pulse with a pulse length fraction close to φ = 1/3 in all of these cases. An idealized pressure wave (15) with a pulse length fraction φ = 1/3 is set to excite the models by pressure, and the stationary pressure and the unsteady amplitude are equals to the measured mean values of each one.

In the part below, the comparison between the turbine measurements and the models results is presented. Currently, the rotational velocity of the turbine rotor is assumed to be fixed (at 85 000rev/min) regardless of the solicitations; as shown by Marelli *et al.*(16), this assumption can be considered as possible, although it depends on the validity of the turbine dynamic qualities (such as the rotor inertia for example) and on solicitations subjected to it. The compressor model integration is a part of my future work. There is two cylinder head velocities which they were tested, 1000 and 2000 rev/min. The results of this works are presented on the figure 7 to 8.

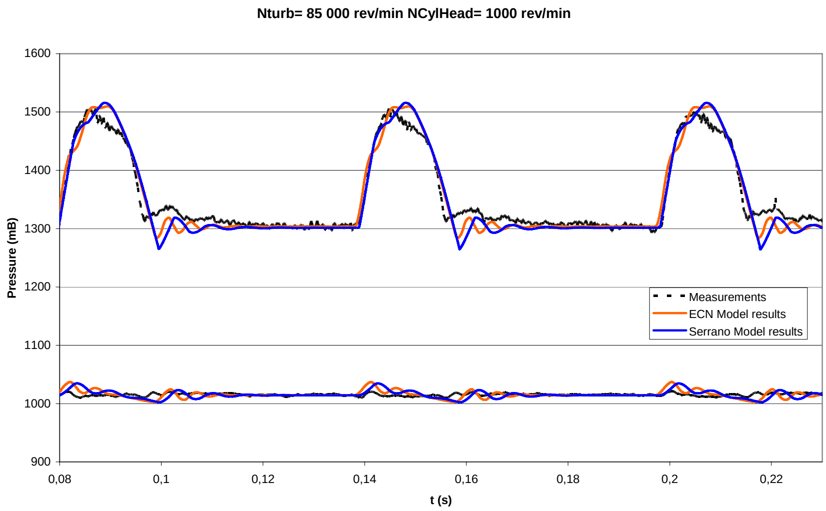

Figure 7. Comparison of measured and modeled inlet and outlet turbine pressures.

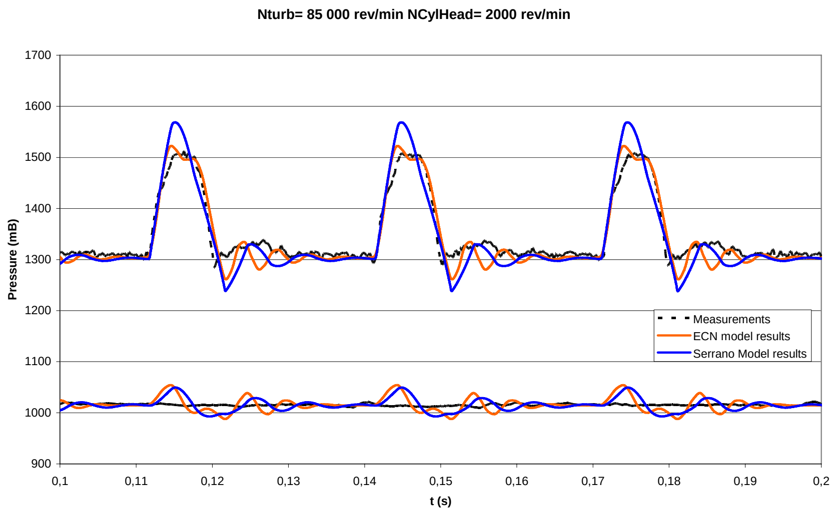

Figure 8. Comparison of measured and modelled inlet and outlet turbine pressures.

The comparison between the modelled and the measured results for a VGT turbine was carried out. Figures 7 and 8 show an example of the results obtained for different pressure wave frequencies. Figures 7 and 8 show the comparison between the modelled and measured pressure–time histories results for turbine inlet and turbine outlet. Figure 7 shows a good agreement of the calculations of both

models with measurements. The ascendant phase of the pressure rise is very well evaluated. However, regarding the amplitude of the pressure wave and the general shape of the upper third, there is a slight difference between the measurements and calculated values. This is may be due to the use of an idealized pressure wave as excitation source of the system. The new model seems to better reproduce the peak form of the pressure signal having a plateau. During the main pressure signal, the phase shift between the pressure signal at the turbine intake and the one located at the turbine exhaust is nonexistent at this small pressure level, thereby the turbine models work in quasi-steady conditions as a single nozzle. Then, regarding "residual pressure waves" reflected by the system at the turbine inlet, the presence of a phase shift between the models predictions must be noted. Moreover, it is also found on the pressure waves at the turbine outlet which are in phase with the inlet ones. Figure 8 shows that the difference between the two models is even more pronounced. The model from Serrano *et al.*(5) slightly overestimates the values of the expansion ratio across the turbine while the new model seems to better reproduce the pressure signal at the turbine intake, both qualitatively and quantitatively.
On the whole, there is a good agreement between the experimental values and those obtained from each model.

CONCLUSION

A new model suitable to modelling variable and fixed geometry turbines, in zero or one-dimensional fluid dynamics codes has been developed. This model consists of a volume representing the volute and, a “nozzle” representing the stator and the rotor which reproduce the pressure expansion ratio across them. The part abusively named "nozzle", uses a direct interpolation method of the extrapolated turbine data map. The extrapolation method is inspired from literature works which incorporates physical laws and some extra parameters. This method permit to predict with a good accuracy the turbine performance on a very large range of pressure expansion ratios and rotor velocity, which is critical for 0D and 1D models. Then, the interpolated values are integrated in the Naviers-Stokes system of equations and implemented in the one-dimensional gas dynamics code.
In order to validate the turbine model, results were compared to tests performed on a turbocharger test bench and to an other reputed model from Serrano *et al.*(5). The conclusion of this comparison was that these models are able to reproduce the fluid-dynamic behaviour of the turbine with good accuracy. The laboratory model seems to have a slightly better pressure prediction than the other model. But other tests must be done in order to corroborate this assumption. The laboratory model permits to reproduce the turbine behaviour under unsteady flow condition by using the maps supplied by the manufacturer as the only input data.
The future developments will attempt to verify and improve the turbine efficiency predictions which represent a blank in the design of most of turbine models.

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NOMENCLATURE

Acronyms

VGT	:	Variable geometry turbine.

Notations

$\dot{m}_{\mathrm{Red}}$	:	Corrected mass flow rate. $\dot{m}.\sqrt{T_e^*}\big/P_e^*$ $\left(kg.K^{1/2}\right)/(s.MPa)$
N_{Turb}	:	Rotor rotation speed (rpm)
P	:	Pressure (Pa).
T	:	Temperature (K).
h	:	Specific enthalpy ($J.kg^{-1}$)
S	:	Cross section area (m^2)
γ	:	Specific heat ratio. (C_P / C_v).
η_{Ts}	:	Total-to-static efficiency.
τ, τ_P, PR	:	Pressure expansion ratio ($P_e^*\big/P_s$).

Subscripts and superscripts

*	:	Total value.;
$_e$	:	Turbine inlet conditions.
$_s$	:	Turbine outlet conditions.
$_{is}$	:	Isentropic.
$_{Crit}$	:	Critical conditions (for a choked flow).
$_{crg}$	:	Corrected value.
$_{eff}$	:	Effective.