

Advanced Numerical/Experimental Methods for the Analysis of a Waste-Gated Turbocharger Turbine

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ABSTRACT

In the paper the results of an experimental campaign regarding the steady characterization of a turbocharger waste-gated turbine (IHI-RHF3) for gasoline engine application are presented. The turbine behavior is analyzed in a specialized test rig operating at the University of Genoa, under different openings of the waste-gate valve. The test facility allows to measure inlet and outlet static pressures, mass flow rate and turbocharger rotational speed.

The above data constitute the basis for the tuning and validation of a numerical procedure, recently developed at the University of Naples, following a 1D approach (1D turbine model - 1DTM). The model geometrically schematizes the entire turbine based on few linear and angular dimensions directly measured on the hardware. The 1D steady flow equations are then solved within the stationary and rotating channels constituting the device. All the main flow losses are properly taken into account in the model. A preliminary tuning is carried out based on the characteristic map measured for a completely closed waste-gate valve.

A 1D schematization of the experimental test rig is implemented within the commercial GT-Power™ software, including the turbine, the waste-gate and the upstream and downstream circuits. In particular, the turbine model itself consists of a sequence of pipes that schematize each component of the device (inlet/outlet ducts, volute and wheel). The pipes dimensions are automatically provided by the geometrical module of the 1DTM. In the turbine schematization, the "wheel-map", estimated by means of the above mentioned 1DTM, is utilized. The latter does not take into account phenomena and losses occurring in the volute and in the intake/outlet ducts, as well.

A refined modeling of the waste-gate system reveals to be necessary to correctly reproduce the swallowing capacity of the by-pass port. The waste-gate model is tuned against the experimental findings for a completely closed turbine wheel and various settings of the by-pass port.

Finally, once the 1D models of the turbine and of the waste-gate system are independently tuned, their behavior under parallel flow is analyzed. The proposed numerical and experimental results demonstrate the mutual interaction of the two components. In addition, it is put into evidence that turbine and waste-gate circuit cannot be considered as two systems working in parallel under the same pressure ratio, as usually assumed in 1D codes.

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INTRODUCTION

It is widely recognized that turbocharging is a very useful technique commonly applied in both spark ignition and compression ignition engines to improve the efficiency [1, 2, 3]. Advanced technologies, such as variable nozzle, multi-stage turbocharging, high and low pressure loop EGR systems, assisted turbocharging and turbocompounding, are emerging to control pollutants emissions and maintain fuel economy. Moreover, an effective management of the turbocharging system gives the possibility to achieve a suitable engine torque curve and an excellent vehicle drivability [4].

The engine-turbocharger matching can be successfully carried out by means of 1D models [5, 6, 7, 8]. The classical quasi-steady approach, based on the employment of compressor and turbine characteristic maps (provided by the manufacturers or directly measured) is usually followed.

In order to improve turbo-matching calculations, turbine behavior should be investigated with various waste-gate valve openings. On the other hand, turbocharger manufacturers usually provide information about turbine characteristics only for a fully closed waste-gate valve. As a consequence, in the 1D engine simulations, only the turbine map with the by-pass port fully closed is employed, while the waste-gate flow contribution is estimated through a proper orifice working in parallel to the turbine under the same overall expansion ratio.

This assumption does not appear to be completely acceptable for a small automotive turbocharger since a substantial by-pass flow contribution commonly verifies, often greater than the mass flowing through the turbine rotor. Significant interactions between the two flow components probably take place within the turbine volute casing and lead to considerable modifications in mass flow rate and efficiency characteristics. According to the flow distribution, the different expansion ratio experienced by the turbine wheel and the waste-gate valve may lead to considerable modifications in the overall mass flow rate map [9,10].

In order to analyze the turbine behavior, since several years a specialized components test facility has been operating at the University of Genoa [11,12]. The test rig is fitted with specific devices which allow for investigations under steady and unsteady flow operations both on single components and engine subassemblies. Several experimental studies on automotive turbocharger turbines, also fitted with different regulating devices (waste-gate valve and variable geometry systems), have been performed [9,10].

In the present paper, a small in-series waste-gated turbocharger turbine (IHI RHF3) is experimentally characterized at the test rig under steady operation at various waste-gate valve openings. In addition, the swallowing capacity of the sole waste-gate valve is measured, by closing the turbine wheel inlet.

In the recent year, a 1D turbine model has been proposed by the University of Naples, (1D turbine model - 1DTM) [13,14]; it is based on the solution of the 1D steady state flow equations within the stationary and rotating channels constituting the device, starting from a reduced set of geometrical data. A direct modeling of main phenomena and losses is performed, also referring to correlations derived from the literature. The model is tuned with reference to the turbine map measured with a fully closed waste-gate valve.

The experimental apparatus, including the turbine, the waste-gate circuit and the upstream and downstream piping, is modeled in the GT-Power™ environment. In particular, the turbine modeling follows an innovative methodology based on the schematization of each component of the device (inlet and outlet pipes, volute and wheel) by means of ducts of proper dimensions. The latter, usually unavailable or hard to directly measure, are automatically provided as an output by the above recalled 1DTM. The GT-Power turbine model also includes the characteristic map of the sole wheel, ranging over the entire operating domain of the device, that is estimated by means of the 1DTM.

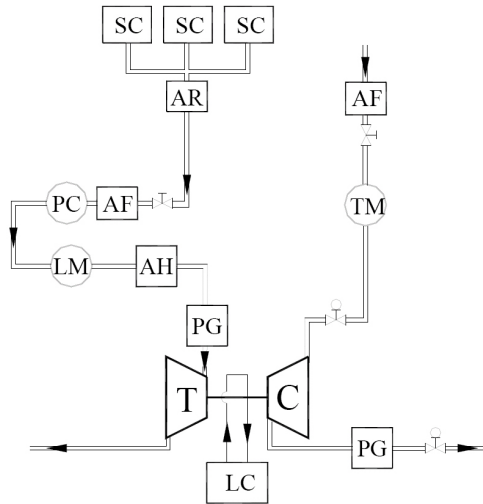
The waste-gate valve system is modeled through a pipes network including proper objects able to correctly describe the losses related to the flow deviation experienced by the gasses passing through the by-pass port. The valve itself is schematized as an orifice, whose equivalent diameter depends on the valve opening. After a tuning, the experimental swallowing capacity of the sole by-pass circuit is accurately reproduced by the refined model.

Finally, once the independent tuning of the turbine and waste-gate system models is performed, the operations under parallel flow are analyzed and the numerical results are compared with the experimental findings. Then, the proposed refined methodology is also compared with the standard modeling of a waste-gated turbine used within a commercial code, showing an higher accuracy in the prediction of the measured data.

EXPERIMENTAL SETUP

The experimental investigation is carried out at the turbocharger test facility of the University of Genoa, schematically shown in Fig. 1. A more detailed description of the test bench is presented in [11,12]. Three electrical screw compressors provide a total mass flow rate of about 0.6 kg/s at a maximum pressure of 8 bar. An electric heater station allows to increase turbine inlet temperature up to 750 °C, depending on the turbocharger size. However, the experimental tests here presented refer to an inlet air temperature of about 80 °C; this choice prevents water from condensing and freezing at the turbine outlet. The measured quantities are scaled based on the conventional non-dimensional groups to properly take into account for the density variation of the flow entering the turbine. Even if the experimental investigation is performed

under “cold operations”, both the turbocharger and the connecting pipes are thermally insulated to properly estimate the adiabatic behavior of the device.



AF	Air Filter	PC	Pressure Control
AH	Air Heater	PG	Pulse Generator
AR	Air Reservoir	SC	Screw Compressor
C	Compressor	T	Turbine
LC	Lubricating Circuit	TM	Thermal Mass Flow Meter
LM	Laminar Flow Meter		

Fig. 1. Schematic layout of University of Genoa turbocharger test facility

Since in the considered test rig the compressor is used as a brake for the turbine, in order to extend the turbine mapping, the pressure at the compressor inlet is properly modulated, both at values higher and lower than the ambient condition.

Two different devices are available to reproduce typical pressure oscillations induced by periodic intake and exhaust valve opening in an engine. In a first configuration, the pulsating flow is generated by diametral slot rotating valves driven by a variable speed motor. Another available layout is fitted with a variable valve actuation cylinder head that produces a controlled pulsating flow upstream the turbine or downstream the compressor. However, the results reported in the paper refer to an experimental activity developed under steady state operating conditions.

Measurements are performed by an automatic data acquisition system, using interactive procedures in LabVIEW™ environment. Different parameters are measured:

- Static pressure: this quantity is measured at different measuring sections located along the turbine and the compressor circuits through strain-gauge transducers characterized by an accuracy of 0.15% of the full scale.
- Total temperature: air temperature levels are measured through platinum resistance thermometers (Pt 100 Ohm) with an accuracy of ($\pm 0.15^\circ\text{C} + 0.2\%$) of measured value.

- Turbocharger rotational speed: the rotational speed is measured by an eddy current probe mounted close to the compressor wheel allowing to detect rotor revolution with a full-scale frequency precision of 0.009%.
- Air mass flow rate: turbine mass flow rate is measured through a laminar flow meter placed upstream the turbine, characterized by an accuracy of $\pm 2\%$.
- Waste-gate valve setting: a rotary variable resistance sensor is used to measure the opening degree of the waste-gate valve in terms of the angular position of the related driving lever (α_{WG}). Angular position measurement uncertainty is about ± 2 degrees.

The activity regards a typical automotive turbocharger (IHI RHF3) coupled to a small-size spark ignition engine; its single entry nozzleless radial flow turbine is fitted with a waste-gate valve.

In a first stage, turbine performance are fully characterized for different rotational speeds and waste-gate valve openings. Then, the turbine outlet is coupled to a pipe including a dividing wall extended to a length of about 150 mm before a mixing section (Fig. 2); in this way, the separation of the flow contributions coming out from the wheel and from the waste-gate port continues up to the mentioned mixing section. It is thus possible to analyze the rotor and by-pass swallowing capacities when working alone or together. In particular, the waste-gate flow contribution is estimated thorough a hot wire fibre film probe located in the corresponding portion of the outlet pipe. In order to take into account for possible modifications of the flow field induced by the valve plate, the anemometric system is calibrated against the previously mentioned laminar flow meter for various waste-gate settings. The wheel flow contribution is not directly measured, but can be estimated as the difference between the overall mass flow rate and the waste-gate contribution.

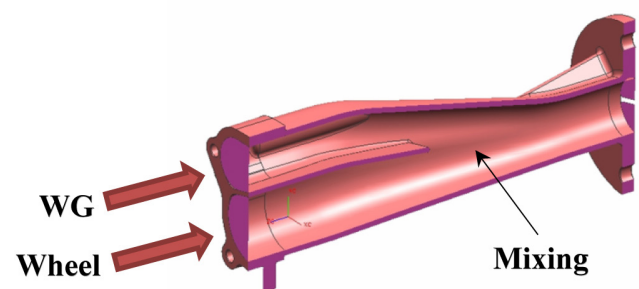


Fig. 2. Turbine outlet pipe including the diving wall

The experimental campaign includes a preliminary characterization of the sole waste-gate circuit carried out by clogging the wheel, for different valve plate settings ($\alpha_{\text{WG}} = 8^\circ, 20^\circ, 35^\circ$ and 60°). The measurement of the turbine characteristic for a fully closed waste-gate valve and for a reduced rotational speed of 4000 rpm/ $\sqrt{\text{K}}$ is then performed.

In the second stage of the experimental campaign, the wheel and waste-gate swallowing capacities under parallel operation are investigated for the previously listed angular positions of the by-pass valve.

For all the tested operating conditions, the performance maps are evaluated with reference to the pressure and temperature conditions measured in the sections upstream the turbine and downstream the mixing station of the outlet pipe.

1D TURBINE MODEL

The turbine model includes a geometrical module that provides the data required to solve the 1D flow equations inside the different ducts composing the device. In particular, the geometrical schematization is constituted by 5 pipes: the inlet duct, the volute, the nozzle, the wheel and the outlet duct. Fig. 3 shows the 1D schematization of a waste-gated turbine, while Fig. 4 draws the main stations along the fluid path.

In the following, the 1D turbine model is recalled, including a brief description of the geometrical schematization, of the 1D flow equations holding in stationary and rotating ducts and of the boundary conditions and flow losses. A more detailed model discussion can be found in [13,14].

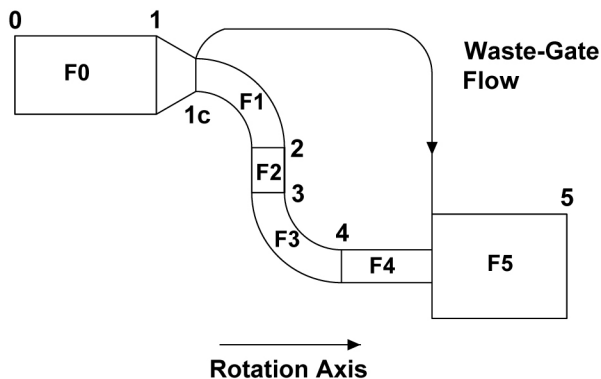


Fig. 3. Waste-gate turbine 1D model

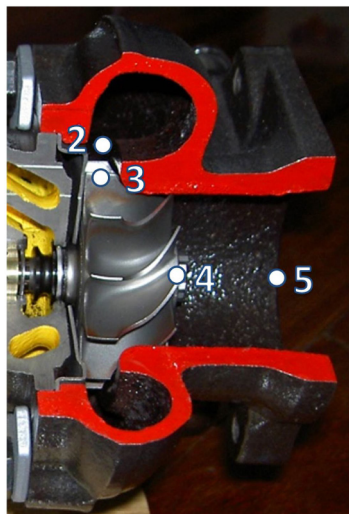


Fig. 4. Turbine main stations

GEOMETRICAL MODEL - The turbine is schematized as follows (Fig. 3):

- F0 - Inlet pipe (from section 0 to 1) of constant diameter;
- F1 - Volute (from section 1 to 2): it is divided in two parts; the first is a converging cone followed by a linearly variable diameter duct representing the circumferential (1c-2) part of the volute. An equivalent length is considered as half length of the volute centerline; the outlet cross section area is imposed equal to the nozzle inlet one;
- F2 - Vaneless Nozzle (from section 2 to 3): it is considered as a converging pipe of a constant height in the radial direction;
- F3 - Wheel (from section 3 to 4): consisting of Z rotating pipes, Z being the number of blades. The profile of the blade-to-blade duct is assigned in terms of blade angles, rotation angles and radii at the inlet and outlet. A numerical procedure is developed to reproduce the 3D wheel geometry based on the previously listed data. The equivalent 1D profile of the blade-to-blade duct (local blade angle, radius, cross section area and wetted perimeter), to be employed in 1D flow equations, is automatically provided by the rebuild geometry;
- F4 - F5 Outlet pipe (from section 4 to 5) of constant diameter; it includes an additional pipe where the mixing between by-passed gas and expanded through the wheel one occurs.

1D FLOW MODEL - Stationary ducts flow model - The steady 1D flow in the stationary ducts (i.e. the inlet pipe, the volute, the nozzle and the outlet duct) is described by the following equations:

$$[\mathbf{F}]_x = \mathbf{S}_C \quad \mathbf{F} = \begin{Bmatrix} \rho c \\ \rho c^2 + p \\ \rho c H_C \end{Bmatrix} \quad (1)$$

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

The terms ρ , c , p , $H_C = c_p T_0 = c_p T + c^2/2$ in the system (1), (2) represent density, absolute velocity, static pressure and total enthalpy per unit mass, respectively. The source term, $\mathbf{S}_C$, accounts for the duct area variation, $\alpha = 1/\Omega \, d\Omega/dx$, the friction coefficient, f , and the rate of heat exchange, q .