

Comparison of Lumped and Unsteady 1-D Models for Simulation of a Radial Turbine

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ABSTRACT

The physical 1-D model of a radial turbine consists of a set of gas ducts featuring total pressure and/or temperature changes and losses. Therefore, the wave propagation and filling/emptying plays a significant role if a turbine is subjected to unsteady gas flow.

The results of unsteady turbine simulation using the basic modules of generalized 1-D manifold solver in GT Power are demonstrated. The turbine model calibration parameters can be identified by means of 1-D steady model used in optimization code loop.

The examples of model results are compared to steady flow map predictions of turbine efficiency and engine pumping loop work.

The model may be used for prediction of turbine data in out-of-design points as presented in the paper. The other important role of a model, however, is an accurate evaluation of turbine parameters from pressure and speed measurements at an engine in operation.

INTRODUCTION

The simulation of unsteady and transient operation of turbochargers coupled to engine has become of high importance, especially in the case of automotive engines. Although even at engine steady operation the pressure pulses in an exhaust manifold cause turbine unsteadiness, the greatest care has been devoted to wave influence on an engine only and on turbocharger speed non-uniformity. The turbocharger turbine performance is currently simulated with waves in manifolds, but the turbine itself uses a quasi-steady, lumped-

parameters, map-based model. The turbine is being modeled as a boundary condition of a manifold, i.e., its wave or flow path length, mass/enthalpy accumulation capacity and the change of flow parameters during impeller revolution are simply neglected, e.g., ref. 1, 2, 3, 4, 6, 20. The comparison of the time scales characterizing exhaust manifold flows and waves and those in a turbine scroll, nozzle and impeller shows that a turbine should be modeled as a system with distributed parameters 26. Time scales of turbine unsteadiness cover different domains: large time scale characterizes transient response, medium one is suitable for exhaust pressure waves and channel passage in a turbine, the smallest one describes the time of interaction between impeller and nozzle vane flows.

Perfect 3-D models, as described in 16, used for turbine detailed aerodynamic modeling, have not yet been suitable for engine model coupling aimed to turbocharger matching and optimization, although they are becoming standard for time-consuming optimization of a turbine itself in nominal point of operation.

Experiments focused especially on pulse operation started with quasi-steady, out-of-design turbine performance – e.g., 10, but recently high-frequency unsteady turbine operation is measured, as well – ref. 7, 8, 9. The problem of these very sophisticated and demanding measurements is that the results suitable for 1-D approach – ref. 21 – cannot be duly evaluated without a suitable model. The local measurements of flow inside a turbine (e.g., in the gap between stator and rotor) are of the highest interest for comparison with 3-D simulation, but 1-D description needs space averaging of flow parameters (mass, energy, momentum) and the flow inside a turbine is inaccessible by spatially integrating probes

(see, e.g., the flow fields predicted in ref. 16). The comparison of unsteady measurements including instantaneous turbine power derived from compressor work and shaft speed changes and 1-D simulation using quasi-steady turbine maps as in ref. 11, 12, 13 or 14 brings about a conclusion, there is time to start the more profound simulation of in-turbine phenomena focused on the use in engine-matching process.

Though the more accurate turbine model has to be based on experiments it has to offer the natural extrapolation of a turbine map (based on physics) to high pressure ratios (supercritical) and a wide range of blade speed ratio (especially for prospective downsized engines with low number of cylinders per a manifold branch). Only if it fulfills these demands it is suitable for engine/TC simulation and matching. That is why 1-D model of a turbine should be adapted for these goals, respecting the predetermined results of steady operation from measurements or 3-D simulations in the best possible way.

The physical, to-1-D-reduced model of a radial turbine should consist of a set of gas channels featuring total pressure and/or temperature changes and losses. It has to describe an inlet volute with stepwise flow splitting to nozzles, mixing of flows at twin-scroll volute exit, nozzle blade flow, rotating impeller channels of general shapes and radial-to-axial bending, an outlet casing with a diffuser – if used, WG ports and valves, etc. Obviously, these demands cannot be fulfilled by a single-purpose model with sufficient flexibility.

The recent advanced in 1-D solvers, as 4, with high level of both integration and flexibility – like GT Suite component GT Power, ref. 17, 18, offer already

- Modules and solvers for 1-D “Pipe” with variable cross/section area, source terms for enthalpy and momentum, external acceleration (rotating pipe) and curvature.
- Modules for flow momentum mixing/ splitting (“Flow splits”).
- Modules for orifices with variable discharge coefficient/area and sensor/signal processor/actuator chains.

This much more general description remains yet unused. The reason for this consists in the second important problem, which forces engine designers to use the lumped-parameter, map-based turbine models: the lack of turbine data suitable for detailed 1-D model calibration. They are neither directly measurable nor directly available from 3-D simulations, as mentioned above. Moreover, the details of turbine design are the frequently classified know-how of a manufacturer. Nevertheless, the extrapolation of maps delivered by a turbine manufacturer to extreme blade-tip speed ratios BSR (u/c_s) using central streamline, steady flow turbine model was elaborated and successfully tested in ref. 21.

The remaining problem consisting in finding calibration parameters of a model usable in engine design practice has been solved by the best fitting model parameters to reality using optimization procedure (e.g., genetic algorithms for large sets of model parameters, ref. 23 and 26).

Therefore, the aims of the current contribution are focused on

- the simulation of interaction between exhaust pressure waves and a turbine (medium time scales);
- the extrapolation of a turbine map keeping the physical meaning of turbine operation description, especially for high level of pulsations in a single or a twin scroll layout, under influence of heat transfer, etc.

The tool developed to fulfill these goals is a prerequisite for understanding, and the evaluation of unsteady flow data, as in ref. 8, before full 3-D models are flexible enough for unsteady process simulation.

PRINCIPLE AND TOOLS OF 1-D RADIAL CENTRIPETAL TURBINE PERFORMANCE SIMULATION

The authors have already acquired positive experience with coupling advanced 1-D solvers to unusual complex models, e.g., ref. 22. This experience has been used to the realization of a current model described with details in ref. 26. The basic issue in understanding 1-D turbine features is a steady flow model described in full details in ref. 21 - Figure 1, which is based on algebraic equations and iterative solution.

The unsteady model uses modules with differential equations, presented in ref. 26, including the influence of centrifugal force in an impeller, which is applied if the channel streamline changes the radial coordinate of a module. The pipes substituting an impeller may be divided into sections with different orientation in space (e.g., for diagonal vanes).

STRUCTURE OF THE MODEL

The simplest unsteady flow model consists of several channels linked in a series - Figure 2. The linkage between them is realized by the simulation of nozzle-exit flow velocities using averaged, experiment-based exit angles relative to blade cascade, and loss or discharge coefficients of blade cascades. The flow transfer from standing, absolute space to impeller rotating channel upstream of impeller inlet and back downstream of impeller exducer is simulated by the appropriate change of flow stagnation states according to velocity triangles transformation.

The model must be amended – using experience from the calibrations of a steady case – by leakage description, otherwise it is not possible to obtain a good correlation between efficiency and reduced flow rate between model prediction and experiments - ref. 21.

During the transformation of algebraic steady flow model into the unsteady model, the main elements are substituted by pipes P or rotating pipes RP (split for solution of unsteady flow into finite volumes) and flow element connections, which can add throttling orifices between two pipes with defined discharge coefficients μ and cross-section area A , and flow splits FS, i.e., pipes with several outlets.

The examples of appropriate GT Power/GT Suite modules are less schematically presented in Figure 3, which will be described in more details later.

These are the following major problems that have to be addressed by the adapted modules:

- an unsteady rotating channel with unsteady flow inside has to be simulated (e.g., rotating pipe Turbine Impeller RP4 in Figure 2);
- total or stagnation state transformations stator-impeller and impeller-stator have to be applied with appropriate change of velocity, keeping static density of gas unchanged, which requires time changing the cross section of a channel (e.g., short pipes Stator-Impeller P3 or Impeller-Stator P4);
- the exit areas of blade channels may change further according to empiric blade exit angle, which depends on reduced turbine speed and pressure ratio as functions of time;
- the discharge coefficients of blade cascades are dependent on more variables than the standard

valve ones (which are used more frequently in boundary conditions); transonic limit is influenced by comparatively high losses especially in impeller channels due to secondary flow inside them; from those reasons the standard orifice (flow connection) discharge coefficients C_D are not suitable and the distribution of losses along the pipe in the form in a pressure loss C_P should be used;

- any supersonic flow should be avoided; it is especially important downstream of a turbine nozzle when the transformation module features cross-section rise and heat rejection; reaching just a sonic velocity in a nozzle throat would cause an undue Laval or Rayleigh nozzle effect;
- leakages at impeller shroud and hub (shaft sealing) have to be respected;
- flow split&joint elements should take momentum conservation into account (e.g., at an outlet of twin scroll sections); 1-D axial flow may feature additionally tangential velocity component bringing additional kinetic energy; their changes are subjected to angular momentum conservation (typically in an outlet diffuser);
- turbine torque must be integrated step-by-step (or finite-volume-by-FV) at any instant to correct the results for possible accumulation inside impeller channel; the resulting torque has to be transferred to a compressor shaft;
- last, but not least, the calibration parameters of the model must be found directly from available experiments and public turbocharger design data, otherwise the model cannot be used by engine designers.

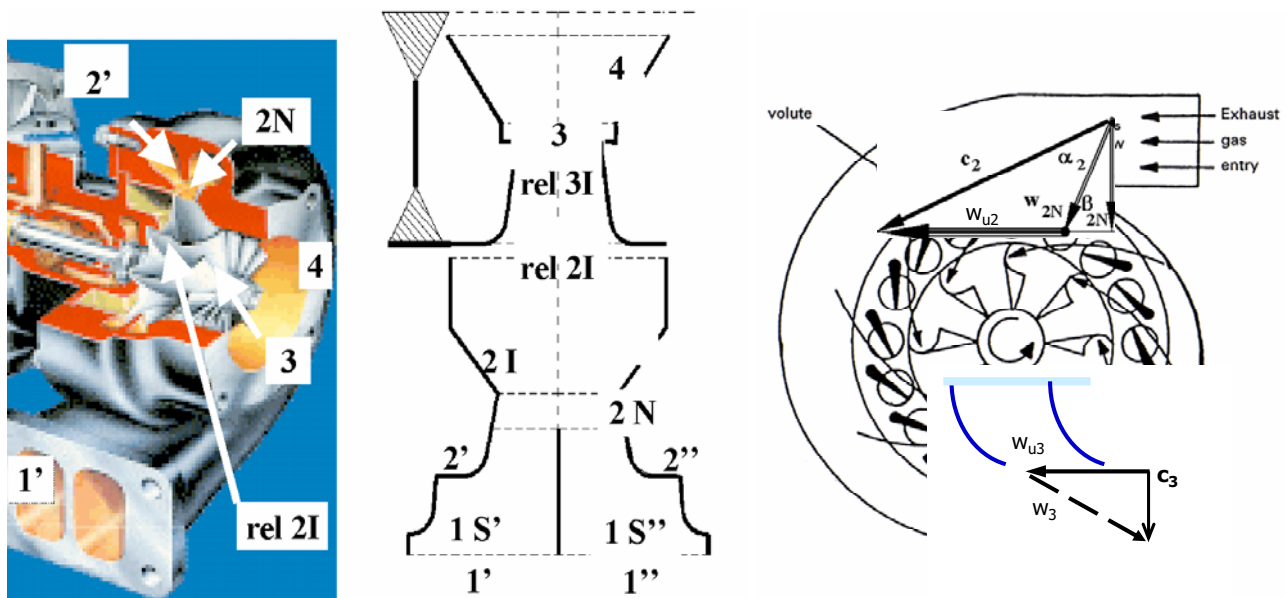


Figure 1. General scheme of a 1-D steady flow model of a radial turbine and transformations between steady and rotating coordinate system at impeller inlet and outlet

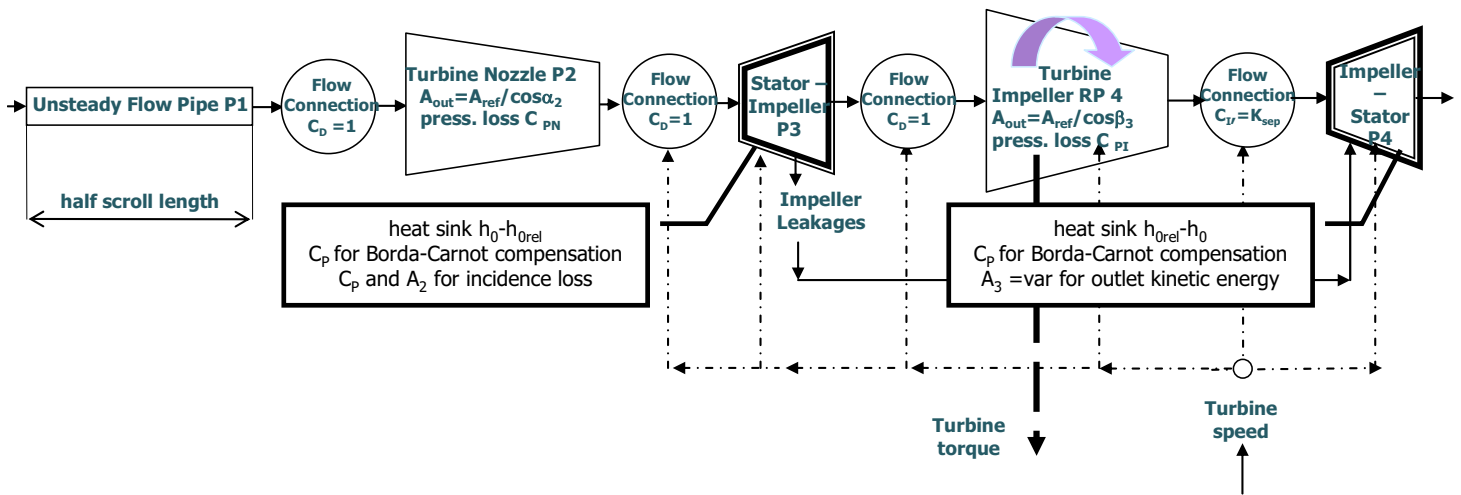


Figure 2. The simplest 1-D unsteady model of turbine components substituted by pipes and flow connections with additional source terms and sensors/actuators

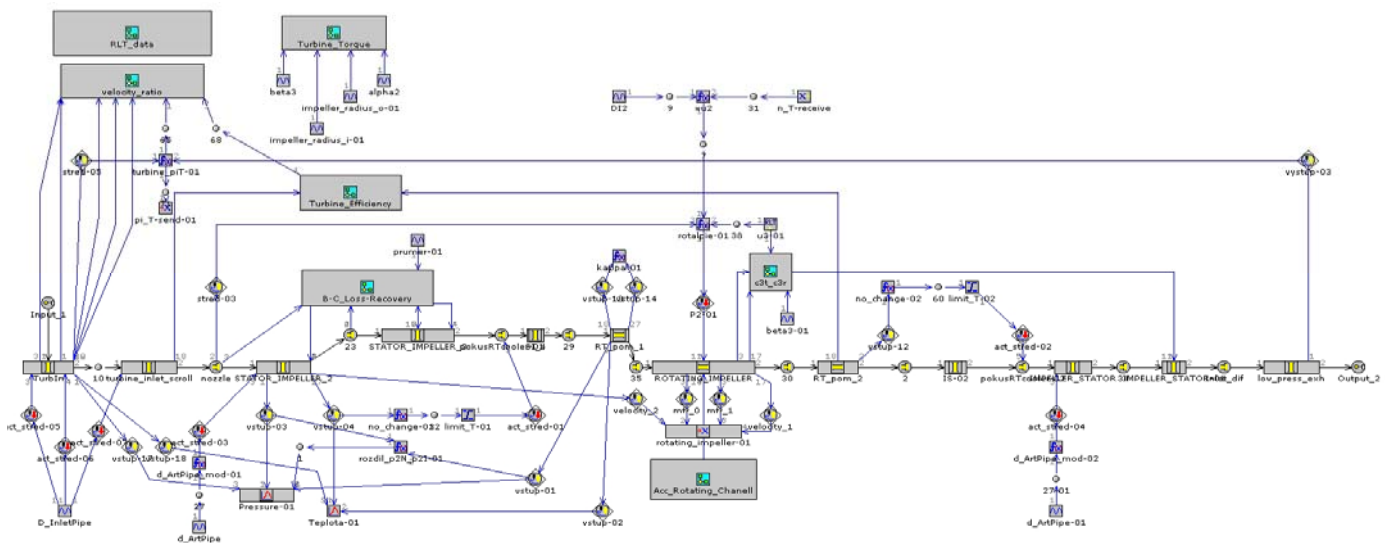


Figure 3. 1-D unsteady model of turbine components implemented to GT Power with additional source terms and sensors/actuators for stator-impeller and impeller-stator transformations together with impeller torque integration.

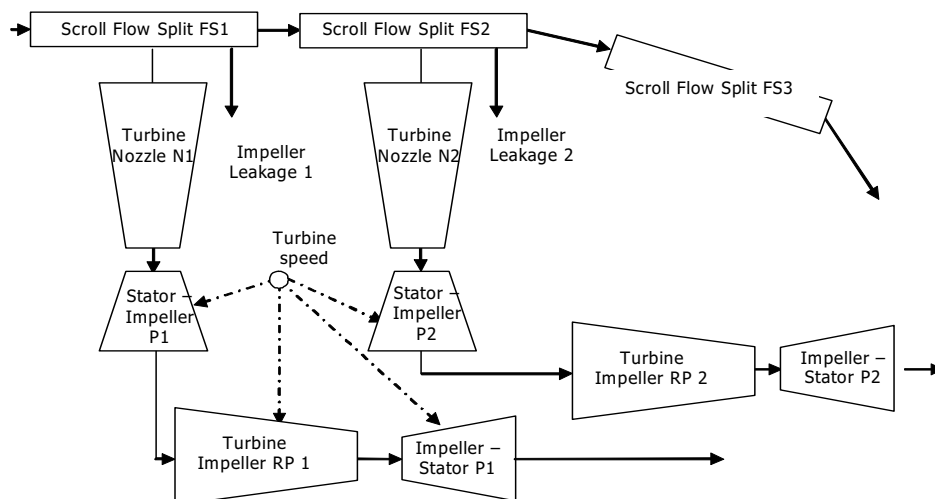


Figure 4 More accurate simulation scheme of a radial centripetal turbine with parallel branches in both stator and impeller. The general flow scheme is recalled above.

The treatment of individual requirements will be presented below.

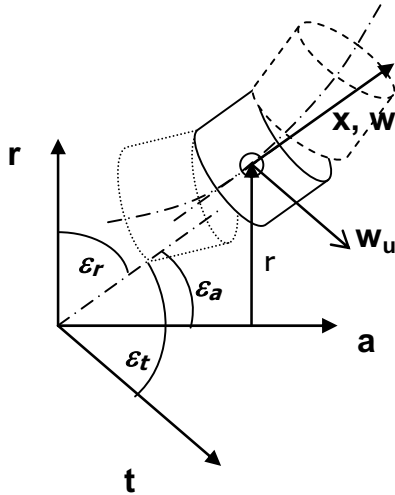


Figure 5. General scheme of a 1-D rotating finite volume (radius of rotation r , flow coordinate x)

CONSTRUCTION OF A MODEL USING GT-SUITE MODULES

Although the stated demands create a tough task, they can be fulfilled to a reasonable level by, e.g., GT-Suite and GT-Power. The advanced solvers features suitable interfaces in the form of additional source or sink terms and the possibility to define fictitious sensors and actuators to change the model parameters according to pre-defined signal processing, using, e.g., look-up tables. These possibilities avoid the requirements on source code changes before the whole procedure is de-bugged.

Rotation of a channel adds new source terms into momentum and energy conservation according to the following partial differential equations - 26

$$(1) \quad \frac{\partial(A\rho w)}{\partial t} + \frac{\partial(A\rho w^2)}{\partial x} = -A \frac{\partial p}{\partial x} - A\rho t + A\rho C$$

$$t = \lambda \frac{O}{4A} \frac{|w|w}{2}; \quad C = \omega^2 r \cos \varepsilon_r$$

$$(2) \quad \frac{\partial(A\rho e)}{\partial t} + \frac{\partial(A\rho wh_0)}{\partial x} = \dot{m} \omega^2 r \cos \varepsilon_r +$$

$$+ r^2 \rho A \omega \frac{d\omega}{dt} + \frac{O}{\sin \alpha} \alpha_Q (T - T_{st})$$

In the case of GT-Power, the source terms of the right hand side of energy conservation (2) should be added only, considering the potential energy in the field of centrifugal forces, the acceleration of flow due to non-uniform rotation of a turbine shaft and heat transfer to the ambient environment (air, cooling liquid, lubricating oil).

The centrifugal acceleration C has to be adjusted during the calibration according to the current experience. The adjusting coefficient remains constant for the whole turbine map.

The impeller channel may be profiled according to reality (if data are available) or its profile may interpolate between inlet and outlet flow areas. In this case, the real outlet area defined by outlet averaged (empirically calibrated) outlet angle should be applied (c.f. ends of pipes downstream of Turbine Nozzle or Impeller). The module of pipe with "flexible" walls is a standard component of advanced 1-D solvers, like GT-Power. The description of turbine geometry (e.g., splitting the impeller into segments according to Figure 5) must be done before the calibration is started.

The flow connections are used mostly to transfer the static and total states between pipes. If local flow separation occurs (typically at the shroud side of curved impeller channel) they are used to reduce the flow cross-section. Some additional effects of flow connections, like Borda-Carnot pressure recovery downstream of an abrupt change of cross section, c.f. Figure 6, should be treated in all cases they have no physical meaning, i.e., in the vicinity of transformation pipes. The pressure loss coefficient is a result of momentum equation integration

$$(3) \quad \Delta p = p_{throat} - p_{pipe} = C_{P,B} \rho \frac{w_{throat}^2}{2} =$$

$$= 2 \frac{A_{throat}}{A_{pipe}} \left(\frac{A_{throat}}{A_{pipe}} - 1 \right) \rho \frac{w_{throat}^2}{2} =$$

$$= 2 \left(1 - \frac{A_{pipe}}{A_{throat}} \right) \rho \frac{w_{pipe}^2}{2}$$

The experience from the currently tested cases has shown the correction to zero pressure recovery is reached after some calibration of Borda-Carnot $C_{P,B}$. The correction coefficient keeps the constant value for the whole turbine map. If sonic limit is reached, which is indicated by the critical pressure difference upstream of a nozzle and downstream of a pipe with increased cross-section area, the correction of Borda-Carnot correction has to be switched off.