MEAN VALUE MODELING OF A VARIABLE NOZZLE TURBOCHARGER (VNT)

Daniel DINESCU[1], Mohand TAZEROUT[2]

The paper offers an insight into the modeling of a variable nozzle turbocharger for automotive control applications. The emphasis is put on modeling the gas flows through the compressor and turbine and the corresponding isentropic efficiencies. The examined models are physically based regression models calibrated using the static maps provided by the turbocharger manufacturer.

Keywords: mean value modeling, variable nozzle turbocharger.

1. Introduction

In the last decade, turbocharging has become widely used as a means to increase automotive engine power.

Turbocharging uses some of the exhaust gas enthalpy (otherwise wasted) to increase the mass of air inducted in the cylinders. More air means that more fuel can be burned, which results in an increase in the power output of the turbocharged engine as compared to a similar naturally-aspirated engine.

A typical turbocharger consists of a compressor and a turbine mounted on the same shaft. The exhaust gas expelled from the cylinder spins the turbine, which drives the compressor. This machine sucks the ambient air, compresses it and then forces the much denser air into the intake manifold.

If the turbocharger is spinning too fast, the pressure in the intake manifold becomes very high and can eventually damage the engine. Variable nozzle turbines are equipped with a set of adjustable guide vanes positioned on the turbine stator. By changing the angle of the vanes, the turbine flow area and the angle at which the exhaust gas is directed at the turbine blades can be adapted in order to allow an optimum turbocharger operation over a large range of engine speeds.

The opening of the guide vanes is commanded by the electronic control unit (ECU) of the car. ECU controls many other aspects of engine operation: the instant and quantity of injected fuel, the opening and closing of the intake and exhaust valves, etc.

[1] Ph.D. student., Ecole des Mines de Nantes, France
[2] Prof., Ecole des Mines de Nantes, France.

Automotive control strategies are based on models which are computationally simple yet sufficiently accurate. A special type of control oriented models is represented by the mean value models. Their particularity is that the model variables (pressures, temperatures, flow rates, etc.) are represented by temporally and spatially averaged values.

This modelling approach is applied in the following sections to the two main components of the turbocharger: compressor and turbine.

2. Compressor Modeling

Compressor mean value models are usually based on static maps provided by the turbocharger manufacturer. These maps show compressor pressure ratio pr and isentropic efficiency η_{is} with respect to the corrected turbocharger speed N_{corr} and corrected air mass flow rate $\dot{m}_{corr}$. Compressor maps are scaled with respect to the standard conditions in order to allow an unbiased comparison between the performances of different turbochargers.

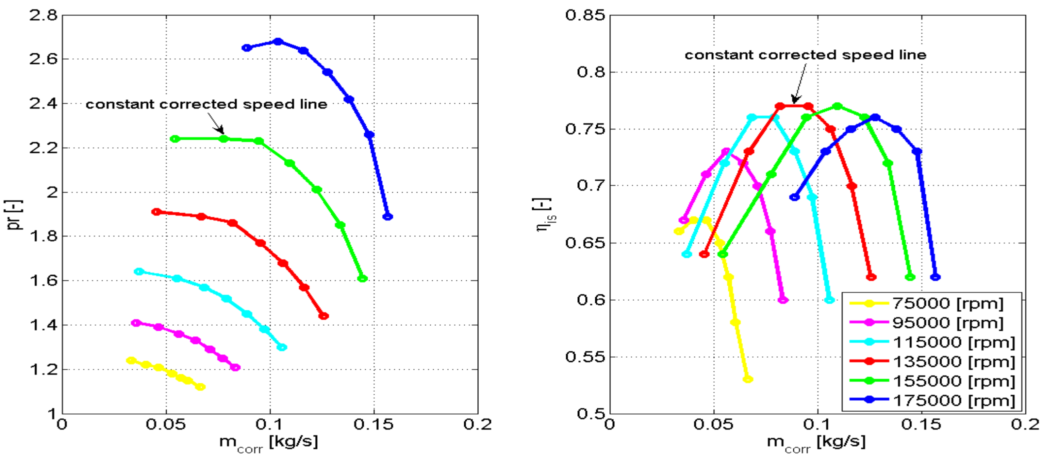

Fig. 1. Corrected Compressor Maps

The problem with compressor maps is that they cover only the medium to high turbocharger speeds, which correspond to the high speed-load range of the engine. For the normal engine operation situated in the low to medium speed-load region, the turbocharger can attain velocities as low as 10000 rpm. Hence the need to extrapolate the manufacturer maps to the very low speed domain.

Standard polynomial regression fails to produce reasonable results outside the mapped data region. In order to overcome this difficulty, the dimensional analysis is employed. This technique allows defining the following dimensionless numbers [1]:

- <u>flow coefficient</u> Φ expressed as the ratio between the flow velocity V and the blade tip speed U :

$$\Phi = \frac{V}{U} = \frac{\dfrac{\dot{m}_{corr}}{\rho \cdot \pi (D/2)^2}}{2\pi N_{corr} \cdot (D/2)}, \tag{1}$$

- <u>circumferential Mach number</u>:

$$Mach = \frac{U}{\sqrt{\gamma \cdot R \cdot T_1}}, \tag{2}$$

- <u>isentropic work coefficient</u> defined as the ratio between the isentropic compression work and the circumferential kinetic energy:

$$\Psi = \frac{c_p \cdot T_1 \cdot \left(pr^{\frac{\gamma-1}{\gamma}} - 1 \right)}{U^2/2}, \tag{3}$$

Using these dimensionless numbers, compressor maps can be put in a new form, which is more suitable for extrapolation.

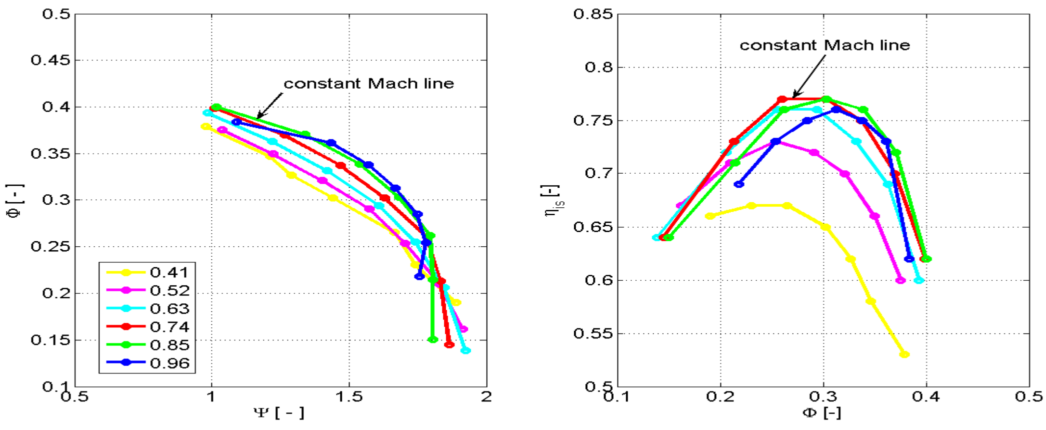

Fig. 2. Dimensionless Compressor Maps

Reference [2] noticed that the flow coefficient curves in Fig. 2 resemble to a quarter of ellipse. Based on this observation, Φ can be modeled as:

$$\Phi = a_1(Mach)\sqrt{1 - a_2(Mach) \cdot \Psi^2}\ , \tag{4}$$

Similarly, isentropic efficiency curves in Fig. 2 can be approximated by parabolas:

$$\eta_{is} = b_2(Mach) \cdot \Phi^2 + b_1(Mach) \cdot \Phi + b_0(Mach)\ , \tag{5}$$

Both models have coefficients expressed as functions of the circumferential Mach number. Consequently, relationships (4) and (5) can be used to expand compressor map to the low speed - low Mach number region as it can be seen from the figure from below:

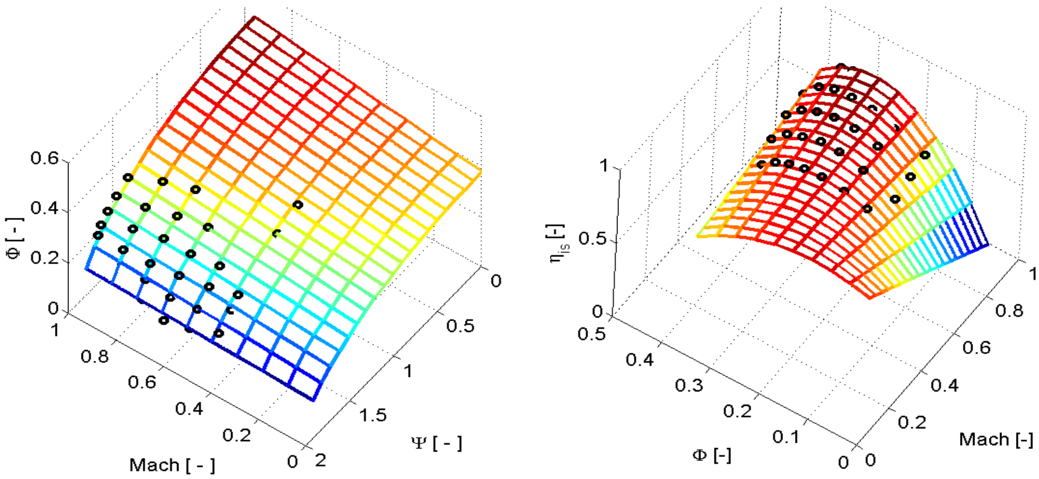

Fig. 3. Expanded Dimensionless Compressor Map

Despite its simplicity, the model showed a good precision when tested against experimental data measured on an engine test bench. The average error was about 10 % while the maximum error was approximately 20 %.

3. Turbine Modeling

Turbine performance modeling is also based on corrected static maps provided by the turbocharger manufacturer. The maps of variable nozzle turbines consist of pressure ratio pr, corrected mass flow rate $\dot{m}_{corr}$ and isentropic efficiency η_{is} values tabulated for different speeds N_{corr} and VNT positions.

The flow of the exhaust gas through the turbine can be estimated with a slightly modified version of the basic orifice flow relationships [3]:

$$\begin{aligned} & pr > pr_{cr} \\ & \dot{m}_{corr} = A_t(VNT)\frac{p_1}{\sqrt{R \cdot T_1}}\sqrt{\left[pr - g(VNT)+1\right]^{\frac{2}{\gamma}} - \left[pr - g(VNT)+1\right]^{\frac{\gamma+1}{\gamma}}} \\ & pr \le pr_{cr} \\ & \dot{m}_{corr} = A_t(VNT)\frac{p_1}{\sqrt{R \cdot T_1}}\sqrt{\left[pr_{cr} - g(VNT)+1\right]^{\frac{2}{\gamma}} - \left[pr_{cr} - g(VNT)+1\right]^{\frac{\gamma+1}{\gamma}}} \end{aligned} \quad ,(6)$$

, where the effective flow area A_t and theoretical zero flow pressure ratio g are modeled as polynomial functions of VNT. The term $(-g+1)$ is introduced in relationship (6) in order to compensate for the fact that the flow rate becomes 0 at a pressure rate greater than unity (See Fig. 4). Turbocharger speed N_{corr} has little effect on the flow through a variable nozzle turbine and is thereby neglected when parametrizing the corrected mass flow rate.

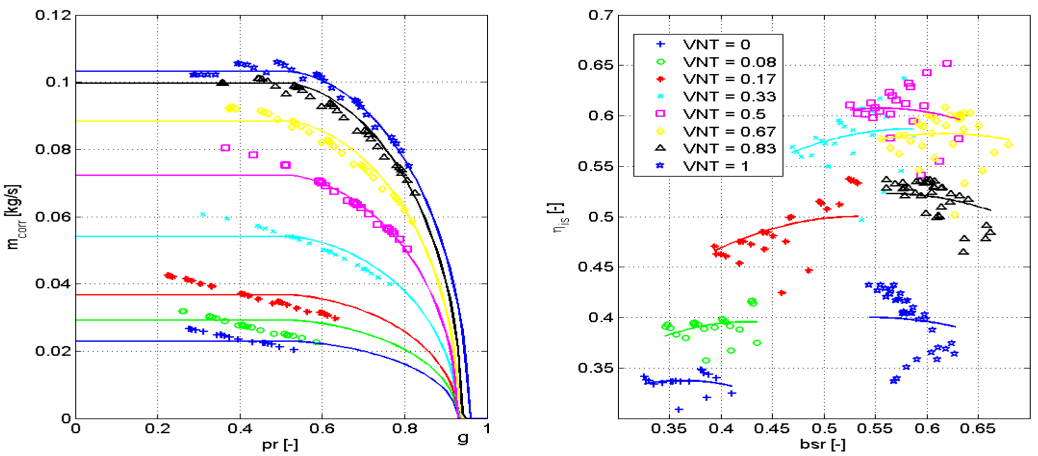

Fig. 4. Turbine Maps

Turbine efficiency η_{is} is usually correlated with respect to the blade speed ratio defined as the ratio between the blade tip speed U and the isentropic nozzle velocity C:

$$bsr = \frac{U}{C} = \frac{2\pi N_{corr} \cdot (D/2)}{\sqrt{2 \cdot cp \cdot T_1 \cdot \left(1 - pr^{\frac{\gamma-1}{\gamma}}\right)}}, \tag{7}$$

According to [4], η_{is} can be conveniently approximated with a quadratic function in bsr. This dependence can be elegantly written as:

$$\eta_{is} = \eta_{is,\max}(VNT) \cdot \left\{ 2\frac{bsr}{bsr_{opt}(VNT)} - \left[\frac{bsr}{bsr_{opt}(VNT)}\right]^2 \right\}, \tag{8}$$

, where the point of coordinates $(bsr_{opt}, \eta_{is,\max})$ represents the vertex of the parabola.

Since the mapped values cover all the range of VNT positions, there is no need to extrapolate turbine maps.

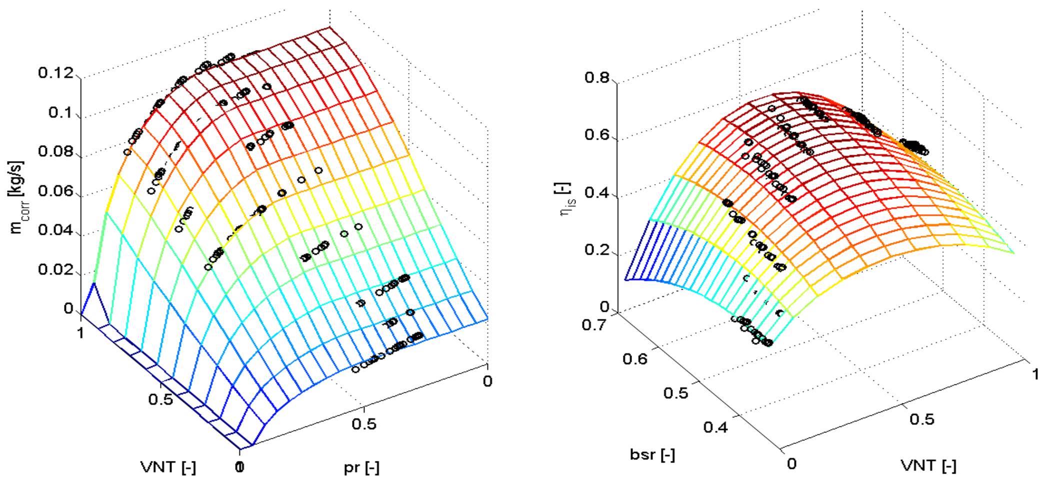

Fig. 5. Turbine Maps (2)

4. Conclusions

This paper demonstrates some of the techniques used for modeling the characteristics of an automotive turbocharger based on the static maps provided by the manufacturer. The examined models are physics based regression models developed in order to extrapolate the turbocharger mapped data over the entire engine operation range. The key element in successfully addressing this challenge is the use of dimensionless quantities.

The investigated models are general, simple and sufficiently accurate. They are specifically developed for engine control but can be also used in more complex models, which simulate engine performance and pollutant emissions.

The presented approach can be equally used for the modeling of other components of the vehicle system: engine, heat exchangers, fuel injection system, etc.

R E F E R E N C E S

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Nomenclature

Symbols

a, b		coefficients
A_t	[m^2]	turbine effective flow area
bsr	[-]	blade speed ratio
bsr_{opt}	[-]	optimum blade speed ratio
C	[m/s]	isentropic nozzle velocity
c_p	[J/(kg K)]	specific heat at constant pressure
D	[m]	blade diameter
g	[-]	zero flow pressure ratio
$\dot{m}_{corr}$	[kg/s]	corrected mass flow rate
$Mach$	[-]	Mach number

N_{corr}	[rps]	corrected rotational speed
p_1	[Pa]	inlet pressure
pr	[-]	pressure ratio
pr_{cr}	[-]	critical pressure ratio
R	[J/(kg K)]	gas constant
T_1	[K]	reference inlet temperature
U	[m/s]	blade tip speed
V	[m/s]	flow speed
VNT	[-]	guide vanes position

Greek Letters

γ	[-]	isentropic coefficient
η_{is}	[-]	isentropic efficiency
$\eta_{is,\max}$	[-]	maximum isentropic efficiency
ρ	[kg/m^3]	density
Φ	[-]	flow coefficient
Ψ	[-]	isentropic work coefficient