

Volumetric Efficiency and Pumping Torque Estimation and Compressor Recirculation Control of Turbocharged Engines

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ABSTRACT

This paper presents three sub-models of production Engine Management Systems software for turbocharged engines; 1) new load variable for volumetric efficiency modelling, 2) new pumping torque model, where 1) and 2) both solve non-monotonic behavior of traditional techniques for active boost controlled engines, 3) Compressor Recirculation Control requiring only the compressor surge-line for calibration.

INTRODUCTION

The Engine Management System (EMS) of a turbocharged engine must include additional estimation and control logic unique to the additional hardware (HW) and associated physical processes as compared to a naturally aspirated engine. [1] described model-based boost control and exhaust state estimation of turbocharged engines. This paper presents some relatively smaller sub-models and control algorithms that are however equally required for EMS software (SW) for turbocharged engines.

Some of the traditional EMS sub-models used for naturally aspirated engines are not adequate for a turbocharged engine due to a fundamental change in engine boundary condition behavior during load variation. This paper covers two such examples where traditional load variables suitable for the Volumetric Efficiency (VE) and Pumping Torque models for a naturally aspirated engine are no longer applicable to a turbocharged engine with active waste-gate control. The presented solutions to these problems are generic and therefore work for any air induction type. Because their solutions are based on

a more detailed description of the governing phenomena, their performance is better.

Furthermore, this paper describes the scheduling strategy of the compressor recirculation valve. The placement of the valve is shown in Figure 1. Its purpose is to open, and thus short-circuit the compressor inlet and outlet, during tip-out conditions, in order to avoid compressor surge and large boost pressure spikes that would otherwise occur in the volume between the compressor and throttle. This avoids unpleasant NVH and reduced compressor longevity, reduces stress on the induction plumbing, and reduces the turbo speed drop thus improving response on a subsequent tip-in. The strategy presented here requires only the compressor characteristic data from the turbocharger manufacturer for calibration, and works under all ambient conditions due to its model-based approach.

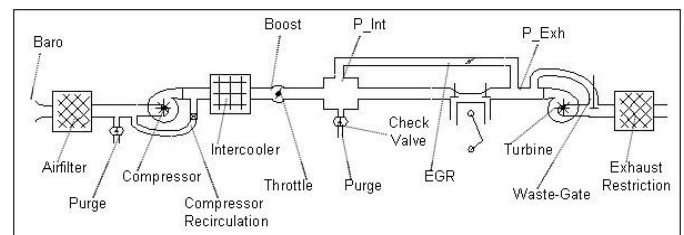


Figure 1: Turbocharged engine layout including the compressor recirculation path.

Editorial Note 1: Boost control is achieved by the waste-gate which can be either actively or passively controlled [1]. Thus, “active waste-gate” and “active boost control” are used interchangeably. Boost is defined here as the pressure downstream of the compressor and intercooler and upstream of the throttle.

Editorial Note 2: Active and Passive Waste-Gate are referred to as AWG respectively PWG. They will also refer to data presented for engines of such type.

- GT-Power® : Model of 2.2liter I4 2valve PFI naturally aspirated
- AWG : 1.8liter I4 4valve PFI Turbo with Active Waste-Gate Control
- PWG: 2.2liter I4 4valve PFI Turbo with Passive Waste-Gate Control

Table 1: Engines referred to in VE and Pumping Torque sections.

VOLUMETRIC EFFICIENCY ESTIMATION

MODEL DEDUCTION

Background

A wide range of approaches to model the airflow or Volumetric Efficiency of an engine have been reported in the literature, e.g. [2], [3], [4] and [5]. However, probably the most common approach in production vehicles is the speed-density equation using VE tabulated vs. engine speed and load. A common VE load dependency is engine pressure ratio (Pr_{eng}^{tot}) which will also be referred to as $Pr1$:

$Pr1$ = engine pressure ratio =
exhaust manifold pressure / intake manifold pressure

Figure 2b shows that a turbo charged engine with a passive waste-gate has the same characteristics as a naturally aspirated (NA) engine in that;

- VE increases monotonically with increasing load
- engine pressure ratio decreases monotonically with increasing load

Figure 2a shows that this is not the case for a turbo charged engine with active waste-gate;

- VE increases monotonically with increasing load
- engine pressure ratio decreases with load increasing until the lowest load with wide open throttle (WOT), and then increases again for further increasing load

Therefore, $Pr_{eng}^{tot} = Pr1$ will not work as a VE model load dependency for an active waste-gate turbo charged

engine because there is not a unique relationship between load and engine pressure ratio.

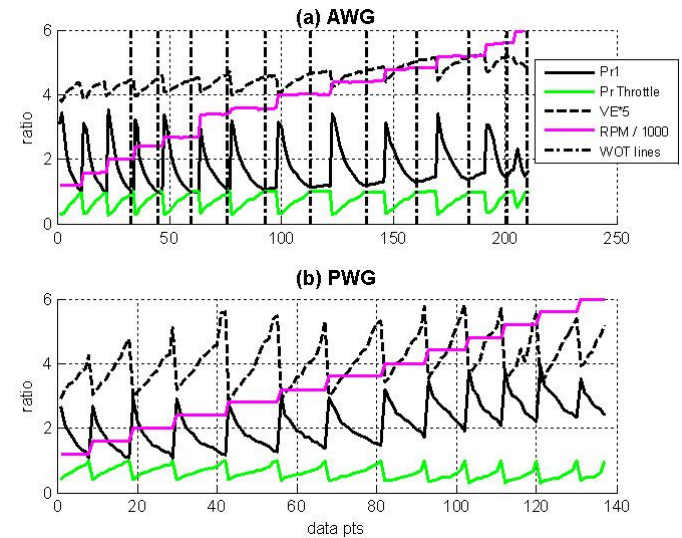


Figure 2: Comparison of VE and pressure ratios for both active and passive Waste-Gate turbocharged engine. Data points arranged for increasing RPM and increasing load.

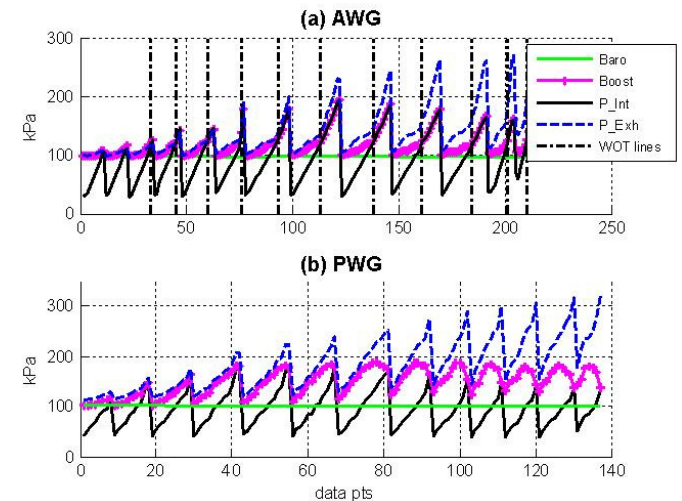


Figure 3: Comparison of intake and exhaust pressures for both active and passive Waste-Gate turbocharged engine. Data points arranged for increasing RPM and increasing load.

Deduction

A possible solution is to not use the traditional VE tabulation approach which requires the combination of the intake and exhaust pressures into one single load variable and instead use statistical models where such variable combination would not be required. However, it is recognized that the tabulation approach is widely considered practical in the production environment, and therefore it is worthwhile to find a solution that allows its continued use for active waste-gate turbo charged engines.

Therefore, the goal is to define a VE load dependency which both reasonably combines the effect of intake and

exhaust pressure into one variable and which is suitable for real-time EMS SW implementation.

The reason intake manifold pressure (P_{Int}) is not used as a VE load variable is because it does not take into account the effect throttling has on VE which results in incorrect VE modelling at altitude. Instead the engine has traditionally been thought analogous to a nozzle and the pressure ratio introduced as the load variable. Since the pressure ratio includes the exhaust manifold pressure estimate which is proportional to the barometric pressure, altitude compensation was achieved, and experience with naturally aspirated engines at altitude has not disagreed.

The engine-nozzle analogy described by a single pressure ratio is now found to be an over- simplification in the case of the turbo charged engine with active waste-gate.

The reasons VE changes with load are the same reasons the residual gas concentration changes with load; back flow ($\dot{m}_{BackFlow}^{tot}$);

$$\text{Eq. 1} \quad VE \propto 1/\dot{m}_{BackFlow}^{tot}$$

The main contributors to total back flow ($\dot{m}_{BackFlow}^{tot}$) are back flow from the cylinder to the intake manifold during the early part of the intake valve open period ($\dot{m}_{BackFlow}^{Cyl2Int}$) and that caused by back flow from the exhaust manifold to the cylinder ($\dot{m}_{BackFlow}^{Exh2Cyl}$). Additionally, negative scavenging will occur especially for large overlap cam timing where the exhaust manifold pressure causes backflow into the cylinder and onwards into the intake manifold.

$$\text{Eq. 2} \quad \dot{m}_{BackFlow}^{tot} = \dot{m}_{BackFlow}^{Cyl2Int} + \dot{m}_{BackFlow}^{Exh2Cyl}$$

The pressure ratios responsible for these back-flow contributions can be approximated as:

$$\text{Eq. 3} \quad \dot{m}_{BackFlow}^{Cyl2Int} \propto Pr^{Cyl2Int} \cong \frac{P_{Cyl}^{@IVO}}{P_{Int}}, \quad \dot{m}_{BackFlow}^{Exh2Cyl} \propto Pr^{Exh2Cyl} \cong \frac{P_{Exh}}{P_{Cyl}^{@IVO}}$$

This shows why it is natural to apply the convenient assumption that the total back flow amount can be described by Pr_{eng}^{tot} dependency since:

Eq. 4

$$Pr_{eng}^{tot} = Pr^{Cyl2Int} * Pr^{Exh2Cyl} = \frac{P_{Cyl}^{@IVO}}{P_{Int}} * \frac{P_{Exh}}{P_{Cyl}^{@IVO}} = \frac{P_{Exh}}{P_{Int}}$$

This simplification is however only reasonable if the proportionality factor between pressure ratio and back-flow for Cyl2Int and Exh2Cyl are comparable.

In order to better understand how VE changes with intake and exhaust pressure, GT-Power® simulations were performed on a naturally aspirated (NA) model where the exhaust restriction and intake restriction (throttle) was varied. The simulation allowed pushing operation to regions not realistic on an actual engine for the purpose of understanding. Such a VE surface was created for three different engine speeds and five different cam phase positions.

Figure 4 shows an example of the VE surface created by GT Power® data for 3000 RPM and nominal cam position. It should be noted that the engine model used was for a traditional non-camphasing engine and that retarding the intake cam by 10deg CAM is equivalent to no overlap. A naturally aspirated GT Power® model was used because it was readily available and requires less simulation time than a turbo engine model while still describes the relevant understanding of the effect of intake and exhaust pressure on VE.

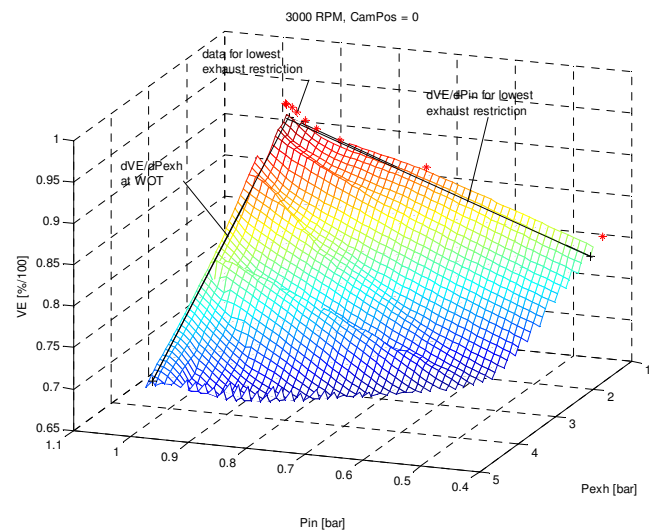


Figure 4: GT-Power® simulation of VE for a naturally aspirated engine at 3000RPM and nominal cam phasing.

Figure 4 shows that the effect of back pressure is relatively independent of intake pressure. Therefore it is reasonable to expect these trends to continue for $P_{Int} > 1\text{bar}$ (boosted operation region of turbo charged engine). By comparing the gradient of VE change to intake vs. exhaust pressure change, one can get an idea of what a reasonable relationship between intake and exhaust pressure should be for the load variable.

Table 2 lists $dVE/dP_{Int@NominalExhRestriction}$, $dVE/dP_{Exh} @ WOT$ and their ratio for different engine speeds and cam phasing. It is found in all cases that the dependence of VE on intake pressure is significantly stronger than exhaust pressure (ratio > 1).

RPM	CamPos	dVE / dP_Int for lowest exh resistance	dVE / dP_Exh at WOT	ratio
-	deg Cam	%/bar	%/bar	-
1600	-10	23	-12	1.84
1600	0	9	-6	1.45
1600	10	8	-6	1.27
1600	20	86	-43	2.02
1600	30	258	-138	1.86
3000	-10	27	-5	4.99
3000	0	25	-6	4.56
3000	10	34	-11	3.03
3000	20	53	-24	2.22
3000	30	83	-58	1.43
5200	-10	9	-6	1.39
5200	0	18	-7	2.43
5200	10	17	-9	1.96
5200	20	27	-13	2.07
5200	30	36	-27	1.34

Table 2: VE gradients for post processed GT-Power® data.

In Figures 5 & 6 the measured VE data for the AWG and PWG engines are overlayed the GT Power® VE surface. Engine speeds for the measured data were chosen close to the GT Power® simulation.

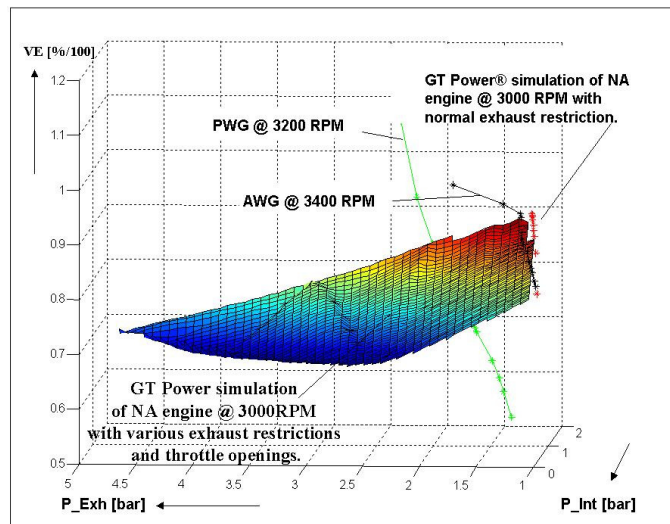


Figure 5: Measured VE for active and passive boost control engines overlaid GT Power® data, 3D-view.

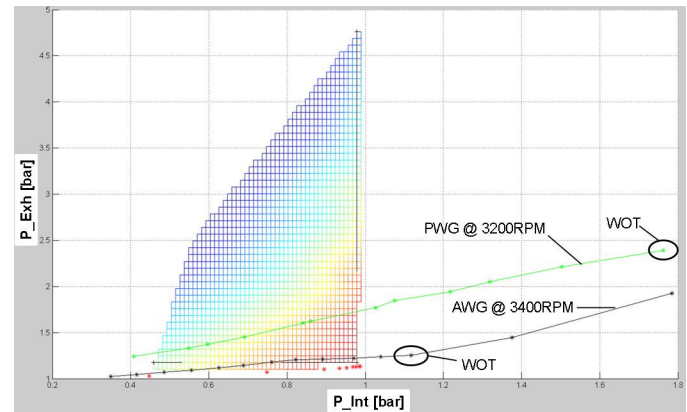


Figure 6: Measured VE for active and passive boost control engines overlaid GT Power® data, 2D-view of Figure 5.

It is clear that the AWG engine behaves much like a naturally aspirated engine as far as exhaust pressure and VE change until WOT. After this point the exhaust pressure increases substantially with further increasing load due to the closing of the waste-gate, but VE still rises due to increase in P_{Int} , although at a smaller rate. Conversely, the PWG engine does not reach WOT until maximum load and the exhaust pressure increases proportionally with intake pressure throughout the load range since the WG is closed until WOT. However, VE still increases throughout the load range because exhaust back pressure affects VE less than intake pressure. This is further illustrated in Figure 7 where VE is looked-up in the GT-Power® created VE surface using the intake and exhaust pressures measured on the PWG engine. This also shows that even though exhaust pressure rises rapidly with load on the PWG engine, VE still rises with load because exhaust pressure affects VE less than intake pressure.

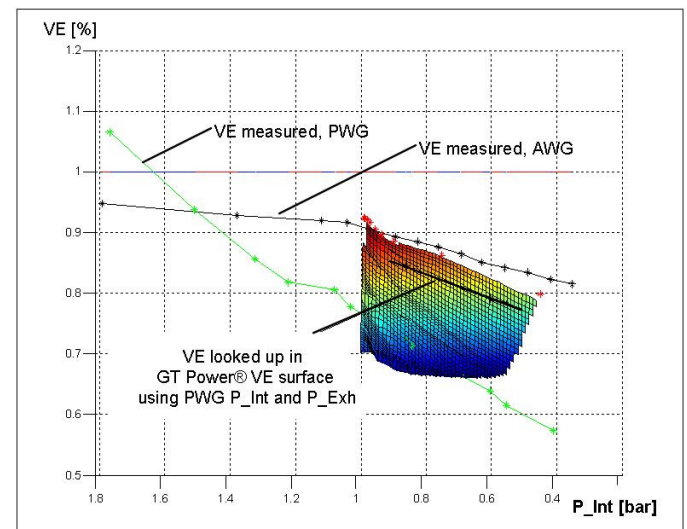


Figure 7: VE comparison.