

Analysis of Transient Operation of Turbo Charged Engines

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ABSTRACT

Transient operation of turbocharged engines is mostly optimised in the light of quickness of response and the provision of the demanded torque. The time from demanded boosted torque to delivered torque above the maximum torque provided by the natural aspirated torque value is known as turbo-lag. This could reveal as an issue for small gasoline turbo-charged engines with a displacement of 1.0ltr or lower. These small types of engines are moving more and more in the focus for automobile applications. To provide the required power and torque, gasoline direct injection and turbo-charging are helpful in order to enable a reduction of fuel consumption by both de-throttled operation over a large area of operation and improved thermal efficiency among others achieved by maintaining an appropriate compression ratio. However, the application of a turbocharger especially on small engines drives also the back-pressure at the exhaust, in turn affecting the knock sensitivity of the combustion and thus impacting efficiency and power negatively. To achieve low levels of back-pressure even on small engines, the size of the turbocharger is from an efficiency point of view to be chosen of a rather large scale. In turn, a larger turbocharger increases the turbo-lag. Thus, an appropriate matching process of the turbocharger is a careful balance between efficiency, maximum torque and transient response, and in the light of stringent downsizing, this thematic deserves closer attention. The paper presents a new approach to characterise the transient operation and response behaviour of turbochargers while applying a new combination of established and novel approaches. Firstly, the interaction between vehicle parameter and shifted gear during acceleration is presented. The difference between the development of the torque delivered during transient operation compared to the steady state torque curve gathered from a dynamometer test of the engine is elaborated.

Secondly, the turbo-lag is characterised by using the acceleration and corresponding jerk of the vehicle during an acceleration event. Based on these pre-requisites, the transient response of a C-car vehicle equipped with a small-engine is shown comparatively with a vehicle propelled by either a port-fuel injected natural aspirated gasoline engine or a turbo-charged diesel engine. Finally, a novel method to measure and analyse the turbo-lag using noise signals is presented. The advantage of this approach is that driveability and noise measurements can be done in parallel, leading to a reduction in set-up time and, moreover, allowing a more in-depth analysis of the interaction of transient driveability and attributes. Furthermore, the relation between subjective perception and objective measured data is demonstrated.

INTRODUCTION

The acceptance of turbo-charged engines given into customer's hands depends on both, low fuel consumption and good driveability. Since the first application of turbo-charging on internal combustion engines for automotive propulsion, the turbo-lag is an item of optimisation and driveability refinement. Hereby the turbo-lag is the time increment the turbocharger needs to build up sufficient pressure in order to take over the vehicle acceleration so far effected by the naturally aspirated engine operation only. Hence the brake mean effective pressure (*bme_p*) established by the internal combustion engine can be written as:

$$bme_p = bme_{pNA} + bme_{pTC}, \quad (1)$$

and, moreover, the total power output P_T can then processed with

$$P_T = M_t \cdot 2\pi n = (bmep_{NA} + bmep_{TC}) \cdot V_D \cdot n \cdot 0.5. \quad (2)$$

The turbo-lag divides the acceleration phase into two phases of acceleration of different level. This change is perceived as the turbo-lag. Subsequently, it defines an error state in terms of driveability as it affects the smoothness of vehicle's acceleration, mostly in longitudinal direction.

Thus, an ideal acceleration phase has a linear slope versus time, i.e. a constant jerk level with a short overshoot at the very beginning, i.e. 'kick'. This is also reflected e.g. in different driveability evaluation tools, e.g. [4]. However, in order to avoid time consuming instrumentation and calibration, simple longitudinal acceleration measurements can be carried out e.g. by employing GPS based transducers. Considering this in the light of downsizing and down-speeding, the impact of power and torque generation especially at low engine speeds requires a parallel assessment of the vehicle NVH, too [5]. In turn, this increases the complexity of instrumentation again. Followed by this, the request for a signal analysis metric was raised to combine the measurement of NVH and vehicle acceleration and turbo-lag respectively.

BASIC FACTORS AFFECTING TRANSIENT DRIVING-MANOEUVRES

THE ENGINE

The aim of a proper turbocharger matching leads to longitudinal dynamics as targeted of the propelled vehicle without showing turbo-lag while developing quickly high pressure rates in the charger with good efficiency [1]. In more detail, the transient response of a turbocharger, thus turbine and compressor can be characterised while reviewing the following three main equations and their controllable state variables of a turbo-charged engine system as depicted in Figure 1. First, the formulation of the conservation of the angular momentum yields the gradient of angular velocity $\dot{\omega} = 2\pi \cdot n$ of the rotor

$$\dot{\omega} = \frac{d\omega}{dt} = \frac{M_T - M_C - M_F}{J_R}. \quad (3)$$

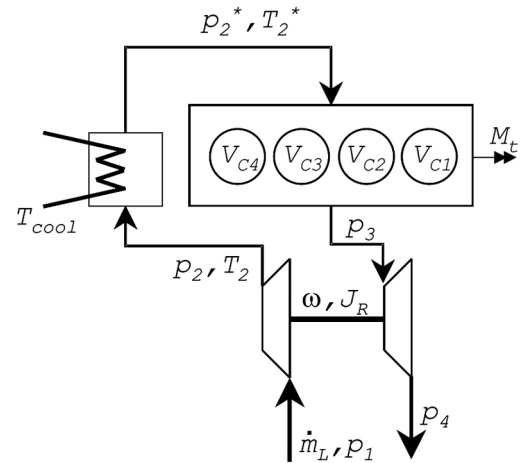


Figure 1. Basic schematic of a turbo-charged engine with intercooler

In equation 3 M_T represents the torque generated in the turbine, M_C the torque consumed by the compressor, M_F the losses due to friction, and J_R denotes the rotational inertia of the rotor-system. Hereby J_R depends on the size and the corresponding diameter d of the rotor-system as

$$J_R \propto d^5. \quad (4)$$

From a mechanical point of view a high rotational acceleration $\dot{\omega}$ can be achieved with a high level of work at the turbine, a low power consumption of the compressor, low level of friction and a low rotational inertia of the rotor-system itself. This can be achieved by a small size of the rotor system or light-weight materials as is e.g. a *TiAl*-rotor [6]. It is worthwhile mentioning that in reality the rotor can not be considered as a rigid part. The treatment of the rotor as a flexible body helps to analyse the dynamic and corresponding tribological characteristics of the rotor in its floating ring bearings in more detail [9]. Moreover, often the frictional loss in torque M_F in the turbocharger is already considered by the efficiency of the compressor, η_C .

Second, the thermodynamic factors are considered ([3]): one can write for the rotor-propelling M_T

$$M_T = \frac{\dot{m}_T}{\omega} \cdot \frac{\kappa_3}{\kappa_3 - 1} \cdot R_3 \cdot T_3 \cdot \eta_{is,3} \cdot \left[1 - \left(\frac{p_3}{p_4} \right)^{\frac{\kappa_3 - 1}{\kappa_3}} \right], \quad (5)$$

and for the torque-consuming compressor side (M_C):

$$M_C = \frac{\dot{m}_C}{\omega} \cdot \frac{\kappa_1 - 1}{\kappa_1} \cdot R_1 \cdot T_1 \cdot \frac{1}{\eta_C} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa_1 - 1}{\kappa_1}} - 1 \right]. \quad (6)$$

In the light of a good efficiency, thus targeting a high M_T and a low M_C , the discussion of [equations 5,6](#) delivers the following: M_T can be maximised while achieving a high gas mass flow of a high temperature, a low back-pressure level p_4 and a high pressure level of p_3 . Especially when looking at transient manoeuvres, the former mentioned factors can, if they are shaped appropriately by the controlling strategy on a temporarily basis, ‘kick-off’ the rotational speed of the turbocharger without affecting the fuel economy significantly [1]. By contrast, M_C should be a minimum while delivering a high level of p_2 . This can be achieved by a low level of pressure loss in the intake system before compressor accompanied by a low intake air temperature. Looking at the combustion system, the temperature of the intake gas flow after leaving the compressor should be minimised accompanied by an significant rise of its density. This can be a.o. addressed by an efficient intercooler. Considering transient events, the ε -approach can deliver a basic understanding [7], and, can in turn support the discussion of governing parameters:

$$\epsilon = \frac{T_2 - T_2^*}{T_2 - T_{cool}}, \quad (7)$$

with T_2^* denoting the intercooler outlet temperature of the charged-air, and T_{cool} representing the temperature of the incoming cooling fluid, i.e. air or water. A high efficiency of the intercooler is mandatory especially for gasoline engines to avoid abnormal combustion events (knock) and to support a minimum of power loss of the engine when being used at high ambient temperatures and altitudes.

THE POWERTRAIN AS INSTALLED

From a mechanical system's point of view a good transient response is a function of the stiffness of the torsional vibration system of the powertrain and the corresponding mounting system in the vehicle. High eigenfrequencies with a high level of damping and a low level of lashes are favorable in order to avoid shuffle and clonk noises [8]. It should be noted that the lower the total ratio number is, the higher is its shuffle-response frequency, and the lower is its amplitude.

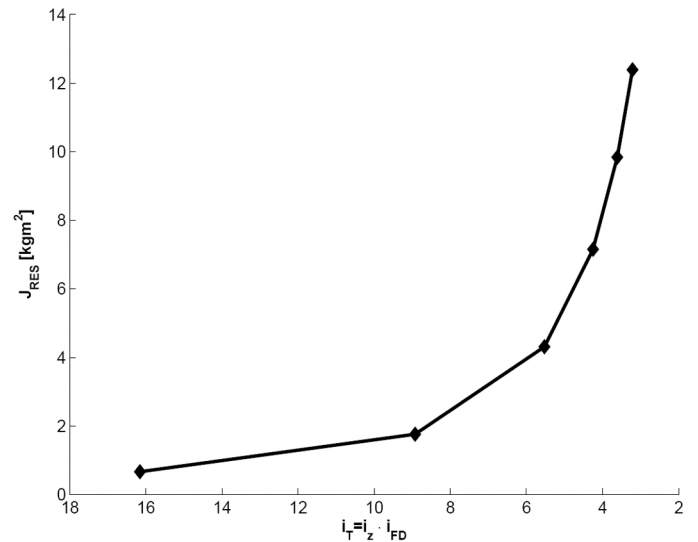


Figure 2. Acting inertia J_{RES} on flywheel for different total ratios i_T

Moreover, basic factors are the vehicle's mass, the effective diameter of the tyres and the total gear ratio that is actually employed. Whilst the total rotational inertia reduced to the flywheel J_{RES} that must be accelerated by the engine torque has a constant value for the engine side J_{Eng} , the vehicle's inertia J_{veh} reduced on the flywheel depends on the gear selected, i.e. the ratios of selected gear ‘z’ i_z and the final drive ratio i_{FD} :

$$J_{RES} = J_{Eng} + \frac{J_{veh}}{(i_z \cdot i_{FD})^2}, \quad (8)$$

while

$$J_{veh} = m_{veh} \cdot r_{dyn}^2. \quad (9)$$

Corresponding to [equation 8](#), the effect of vehicle's inertia reduced to the engine for different total gear ratios is depicted in [Figure 2](#).

It can be concluded that the inertia acting on the flywheel rises progressively with the decrease of the total gear-ratio selected. The result of this effect is depicted in [Figure 3](#).

After five seconds of full load acceleration started in each gear at near idle speed in first gear the engine gets to 6400rpm whereas in top gear the engine only revv's up to 1100rpm. The trajectory in terms of engine speed and load in lower gears is engine speed related due to the lower vehicle inertia reduced to the flywheel. It can be seen that the engine in first gear does not develop torque up to full load steady state conditions which is set by a infinite inertia on an engine