

Different Speed Limiting Strategies for 50cm³ Two-Wheelers and Their Impacts on Exhaust Emissions and Fuel Economy

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ABSTRACT

Usually the power output of 50cm³ two wheelers is higher than necessary to reach the maximum permitted vehicle speed, making engine power restriction necessary. This publication deals with different power restriction strategies for four-stroke engines and their effect on exhaust emissions. Alternative power limitation strategies like EGR and leaning were investigated and compared with the common method of spark advance reduction to show the optimization potential for this certain engine operation conditions. From these tests, a substantial set of data showing the pros and cons in terms of emissions, combustion stability and fuel economy could be derived for each speed limiting technique.

INTRODUCTION

Many test campaigns and development projects regarding the segment of 50cm³ two wheelers were conducted at the Institute for Internal Combustion Engines and Thermodynamics of Graz University of Technology (IVT). One main conclusion resulting from all of these projects is the significant impact of vehicle speed limiting to 47 km/h, due to the high percentage of driving at the speed limit during the ECE R47 test cycle. This impact can be even higher in real world conditions. Previous papers published by IVT [3] pointed out the adverse effects on exhaust emissions and fuel economy of four-stroke engines associated with the common way of engine power restriction by ignition retarding. An alternative way of power restriction, the use of full load EGR, had already been presented. This paper is based on these investigations, leading them one step further to a more detailed point of view. Included is an additional method of power restriction - leaning of air/fuel mixture. The desired result was to evaluate different speed limiting techniques, showing all advantages and disadvantages of each method. To get a sufficient separation of single effects on engine performance, exhaust emissions, and efficiency, a representative prototype engine was equipped with a fuel injection system and operated on the engine test bench. One main point of interest is the estimation of emissions in the vehicle test cycle. Thus the emission measurements on the engine dyno were conducted with a CVS (Continuous Volume Sampling) system. Based on such a sampling method the direct estimation of test cycle emissions is

possible in a proper way. With this experimental setup the different power reduction strategies, i.e. ignition retarding, exhaust gas recirculation and lean burn operation were extensively investigated using many different setups for each method. Hence the best possible values for efficiency and exhaust emissions could be derived for each system.

TEST CARRIER AND EXPERIMENTAL SETUP

A representative vehicle was chosen based on the market situation in Europe in the field of 50cm³ four-stroke engines. This vehicle has already been used as a test carrier for the investigations published in JSAE 2009-32-0069 [3]. It is powered by an air cooled 2-valve four stroke engine with carburetor for mixture preparation. It is also equipped with a catalyst and a secondary air system and homologated according to the ECE R47 EURO 2 standard. Power transmission is through a CVT (Continuous Variable Transmission) and a secondary gear. The specifications are shown in table 1.

Table 1: Test vehicle, engine and fuel specifications

vehicle	
type	scooter
weight	95 [kg]
power transmission	Continuous Variable Transmission with secondary gear
rear wheel diameter	423 [mm]
secondary gear transmission ratio	13.124 [-]
engine	
working principle	four stroke, spark ignited
displacement	49 [cm ³]
compression ratio	11 [-]
gas exchange	2 valves, OHC
mixture preparation	carburetor (constant depression, 19 mm)
cooling	fan cooled
exhaust gas aftertreatment	1 catalyst, secondary air system
homologation	EURO 2
fuel	
type	gasoline
RON	98
density	740.4 [kg/m ³]
calorific value	42.24 [MJ/kg]
percent carbon	83.68 [Wt.%]
percent hydrogen	13.38 [Wt.%]
percent oxygen	2.94 [Wt.%]
C : H : O ratio	0.52 : 1 : 0.0138

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JSAE 2009-32-0069 [3] showed the influence of exhaust gas recirculation on combustion and exhaust emissions in one specific point corresponding with the maximum velocity of the test vehicle. The conducted tests showed big potential compared to power restriction done by retarding spark. This publication expands the investigations to gain more diversified information about power restriction methodologies. Therefore the engine was run in several operating points covering different transmission setups and showing the maximum values of power reduction during WOT operation. Since the main goal of these investigations was to get substantial data about the influences of air/fuel mixture and exhaust gas recirculation on the combustion process, and the untreated emissions as well as their interactions, the tests were conducted on the engine test bench. The engine was equipped with a prototype cylinder head designed at IVT. This cylinder head is characterized by its completely revised intake port, increased compression ratio, improved cooling and an integrated system for exhaust gas recirculation. The port design was supported by 3D-CFD simulation and tests on flow test bench to reach the best possible rates of charge motion together with an increased cylinder filling. By means of these modifications the engine was able to deal with the intended high EGR rates and lean combustion experiments. During the previous test campaign, an integrated EGR system including a rotary control valve was used, to show the possibility of full integration in a mass production engine. This system was calibrated to the required amount of EGR at one specific operating point. For the present experimental setup, an external recirculation pipe system was used since many different rates of EGR and experiments with cooled exhaust gas were intended. To enable an accurate adjustment of all designated operating points the original carburetor was replaced by a fuel injection system.

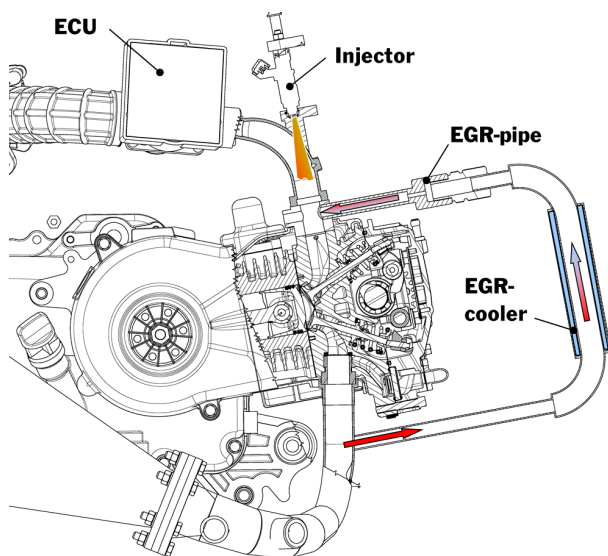


Figure 1: Fuel injection system and EGR device

Figure 1 illustrates the configuration including the injector and the exhaust gas recirculation device. Fuel injection is controlled by an ECU integrated in the throttle body. The injector is a standard automotive mass production part and operates with a pressure of 5 bar. For the purpose of adapting the injector as well as integrating an exhaust gas recirculation pipe, a new intake system was manufactured.

Ignition was also controlled by the ECU. Additionally, a prototype exhaust system was built in a modular way with the objective of investigating different catalysts. For the purpose of showing effects of speed limiting methodologies on untreated emissions, the original catalyst was replaced by the same type of carrier without catalytic coating. By means of this setup, the backpressure of the exhaust system stays almost the same, ensuring that there is no unintentional side effect on the gas exchange.

The volumetric flow of EGR was adjusted by replaceable throttles with different diameters. This system was chosen instead of a control valve because it allows a more accurate adjustment using calibrated diameters. As pointed out in the picture, the external EGR system was designed to enable the cooling of recirculated exhaust gas. Therefore a water jacket was built around the EGR pipe supplied with cooling water. The intensity of cooling was controlled by adjusting the mass flow of water.

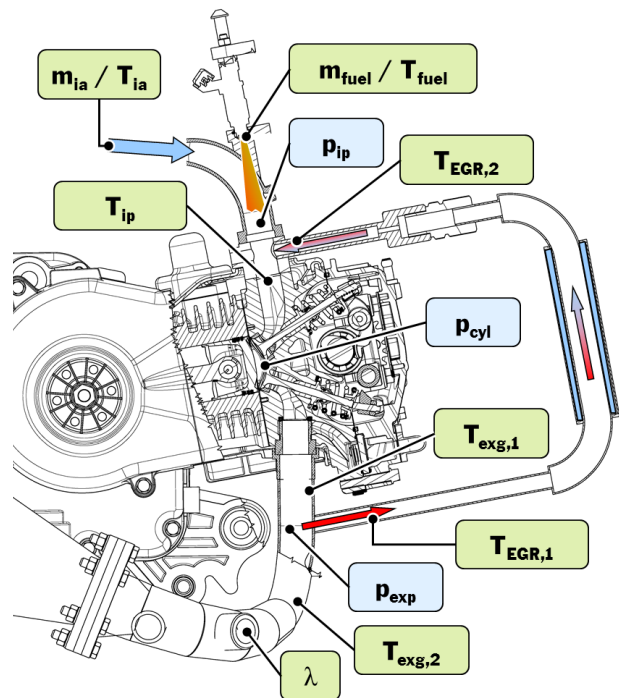


Figure 2: Description of measurement setup

Figure 2 illustrates an excerpt of variables measured on the engine test bench. To gain detailed information about the combustion process, cylinder pressure was recorded over crank angle as well as the pressure in the intake and exhaust pipes. The mass of intake air was measured by means of an air mass flow meter and the fuel mass flow with a fuel balance. For the mass balance calculation in case of introducing exhaust gas to the intake port, all temperatures in the intake system and the exhaust system as well as in the EGR pipe were measured. Additionally, the engine was equipped with a broad band λ -sensor in the exhaust to view and adjust air/fuel-ratio.

p_{ip} indicated pressure within the intake port [bar/°CA]

p_{cyl} indicated pressure in the cylinder [bar/°CA]

p_{exp} indicated pressure within the exhaust port	[bar/°CA]
m_{ia} mass flow of intake air	[kg/h]
m_{fuel} fuel mass flow	[kg/h]
T_{ia} intake air temperature	[°C]
T_{fuel} fuel temperature	[°C]
T_{ip} intake gas temperature after EGR supply	[°C]
$T_{exg,1}$ exhaust gas temperature after exhaust valve	[°C]
$T_{exg,2}$ exhaust gas temperature after EGR extraction	[°C]
$T_{EGR,1}$ temperature of recirculated exhaust gas before EGR cooler	[°C]
$T_{EGR,2}$ temperature of recirculated exhaust gas after EGR cooler	[°C]
λ measured λ -value	[-]

As mentioned in the introduction, exhaust emissions were measured from the tail pipe by means of a CVS system, which is normally used on vehicle test benches. The intention behind this choice of measurement methodology was to get the same dilution as on the chassis dyno, being able to establish a direct correlation between the emission behavior on the engine test bench and vehicle emissions during the homologation cycle. Also, this offers the possibility of sampling particulate matter from the dilution tunnel.

TEST PROCEDURE

To correlate data with the previous publication, the basic vehicle was driven on a chassis dyno test bench to calculate the correlation between engine speed, torque, and vehicle velocity.

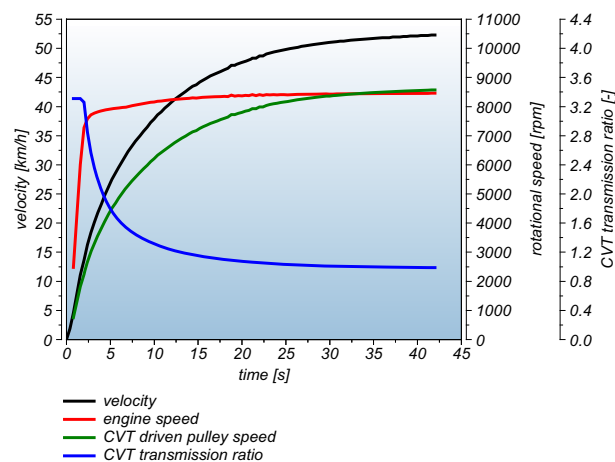


Figure 3: CVT transmission characteristics of the test vehicle

Figure 3 shows the results of this calculation, which served as input data for the current measurement campaign. For

detailed information about the calculation process, refer to JSAE 2009-32-0069 [3]. For this basic power transmission setup, the correlating engine speed at the vehicle speed limit of 47 km/h is about 8400 rpm. Based on that, the range for the intended tests was defined to start from 7000 rpm going up to 8400 rpm.

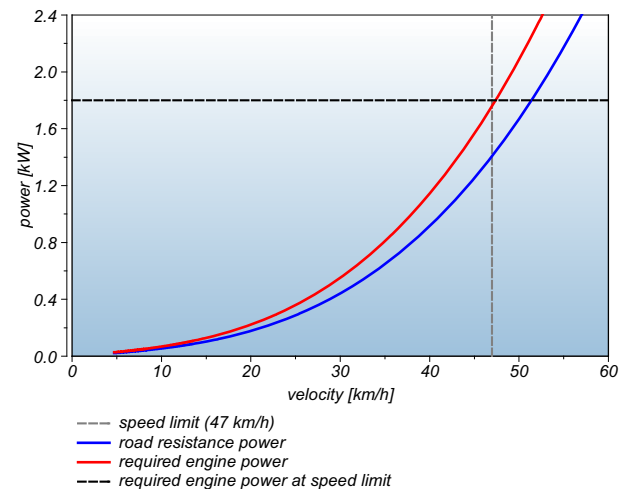


Figure 4: Road resistance and engine power demand

The road resistance curve that was evaluated for the test vehicle is displayed in Figure 4. It shows the rear wheel power demand over vehicle velocity. Based on this data the corresponding engine power requirement was calculated using an overall transmission efficiency (CVT and secondary gear) of 0.8. This efficiency value is derived from an engine/vehicle test bench comparison. It can be seen that the power on the crankshaft should be ~ 1.8 kW to maintain a vehicle velocity of 47 km/h. Using this specified power value instead of torque enables to bypass a dependency on engine speed. Thus it can serve as a target over the whole defined speed range.

As a basis for comparison, the first tests represent the most common way of power reduction by ignition retarding. To gain more information about the effects of ignition retarding, a whole map was derived using different ignition timing values over a range of engine speeds. Having this information, it is possible to estimate different scenarios for maximum vehicle speed and corresponding engine speed, being suitable for different transmission setups and vehicle requirements. The air fuel mixture was adjusted to a constant λ -value of 0.9.

In a second step the effects of using a leaning strategy for power restriction were investigated. This methodology is already in use in some vehicles equipped with fuel injection. Again, a whole map was generated, displaying the effects of λ -variation over engine speed. Spark advance was not modified during the tests on lean operation. The values of the unthrottled case were used.

After these tests, showing the capabilities of existing speed limiting techniques, investigations on exhaust gas recirculation were conducted. EGR rates were adjusted by means of different throttle diameters. Thus the engine was operated with different EGR throttles over the specified speed range to derive a map of power reduction over engine speed and EGR rate. Similar to the tests with ignition

retarding, the λ -value was kept constant at 0.9. To see the potential of this methodology in combination with a leaning strategy, the engine was also operated with EGR using a constant λ -value of 1.0.

First, ignition timing was kept constant at the values of the unthrottled case. Since there is an increase of combustion duration and ignition delay to be expected under operation with EGR, the influence of optimizing ignition timing for both cases ($\lambda=0.9$ and $\lambda=1.0$) was also investigated. As a consequence of improved combustion, EGR rates had to be increased when using optimized ignition timing to maintain the required power reduction.

One main objective of the measurements was to evaluate strategies for speed limitation on small capacity four stroke engines that could be transferred in mass production vehicles in quite a simple way. Therefore the focus was set on using EGR without conditioning, which would be an easily implementable solution [3]. Furthermore, the impacts of cooling the recirculated exhaust gas were investigated to elaborate the additional benefits that can be derived by using such a system, representing the theoretical optimum.

RESULTS

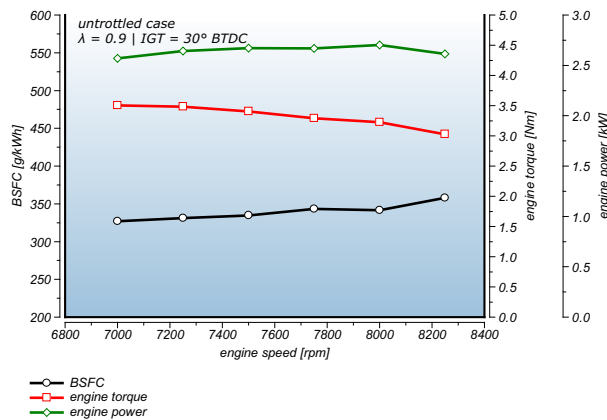


Figure 5: Performance data of basic engine with prototype parts and EFI

The graphs of Figure 5 are demonstrating the unrestricted performance data over the speed range defined above, using the engine setup described in the previous section. λ was kept constant at a value of 0.9.

In the further context, the results of the main methodologies for power restriction will be discussed to show the most significant trends:

A Reduction of spark advance

B Leaning

C₁.....EGR at $\lambda=0.9$ / unconditioned

C₂.....EGR at $\lambda=1.0$ / unconditioned

C₃.....EGR at $\lambda=0.9$ / conditioned

Subsequently the most promising solution will be presented:

D.....EGR at $\lambda=1.0$ / unconditioned /

optimized ignition timing

This setup has been derived in consideration of best possible efficiency in combination with low emissions.

Reduction of spark advance

(A)

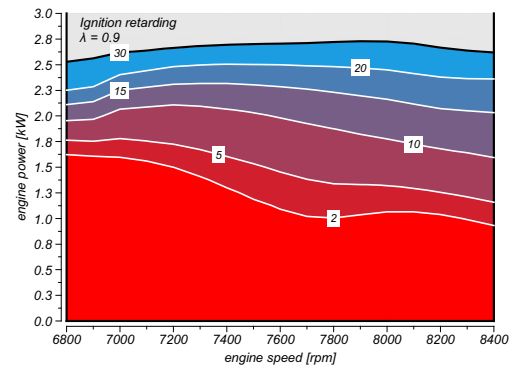


Figure 6: Engine power map for different values of spark advance (°CA BTDC)

The derived map for engine power with different values of spark advance is shown in Figure 6, whereas the maximum line shows the basic performance curve with a spark advance value of 30° before top dead centre. It can be seen that a power reduction to 1.8 kW requires very late ignition, between 5° and 10° BTDC. Spark timing for constant engine power of 1.8 kW over the considered speed range was extracted from this measurement matrix.

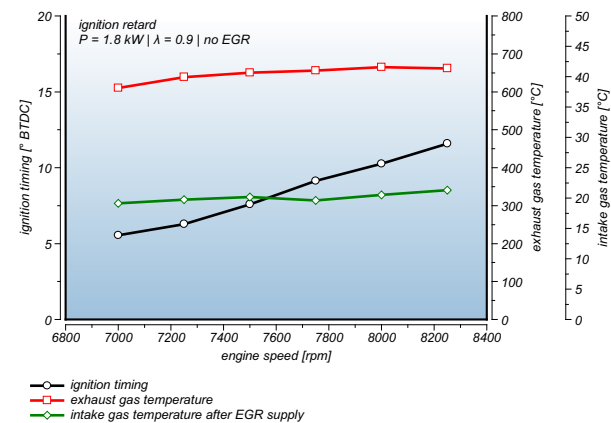


Figure 7: Ignition timing and intake/exhaust gas temperatures for a constant power output of 1.8 kW at $\lambda=0.9$

Figure 7 displays the required spark advance reduction to maintain constant power output. Due to the late ignition timing, the exhaust gas temperature gets very high. The intake gas temperature measured after the EGR supply (T_{ip}), (zero EGR in this case) is also displayed, serving as a basis for comparison with the EGR measurements.