

Layout of a Charged Power Sport Engine

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ABSTRACT

The main target for the development of power sport engines is and will be in future the increase of the power to weight ratio. However, the reduction of carbon dioxide emissions is getting more and more important as future legislation and increasing customer demands ask for lower fuel consumption. One possible technology for CO₂ reduction which is widely used in automotive applications is downsizing by reducing the engine capacity and increasing the specific power by charging strategies. Focusing on power sport applications, like motor-cycles, the automotive downsizing technologies cannot be transferred without major modifications. The essential difference to automotive applications is the extraordinary response behavior of today's motor-cycles, as well as the large engine speed spread. Additionally, packaging and cost reasons exclude the direct transfer of highly complex automotive technology, like two-stage charging, cam-phasing, etc. to motorcycle applications.

This publication treats the conceptual design and layout of a charged 800cm³ two-cylinder motorcycle engine with the demand of excellent drivability, power and response. In detail, the paper deals with the layout of an exhaust turbo charging system for an inline two cylinder engine where the achievement of an appropriate response behavior is of particular importance. As reference for the transient behavior a turbo charged automotive engine has been chosen. The benchmarks for the transient behavior have been a state of the art naturally aspirated motorcycle on one hand and an automotive TC engine on the other hand. Based on these targets, a previously presented simulation strategy [1] for the selection of the different hardware components has been applied. Stationary and transient 1D-CFD simulations as well as 3D-CFD and coupled simulations have been performed and various modifications on the engine and the periphery have been carried out for prototyping.

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INTRODUCTION

The reasons for the development of this charged two cylinder engine are various. First and foremost the proof of automotive downsizing strategies for power sports applications and in detail for motorcycles should be given. Therefore the torque characteristic of the downsizing base, BMW R1200GS, and the response of the basic 800cm³ NA engine are chosen to validate the investigations. The second reason is the improvement of the performance of an existing engine and the attendant possibility to create or extend a modular engine platform with the help of a charger. That means to use as much equal parts as possible for several performance classes and diverse applications. That leads to a significant reduction of production costs.

The previously presented thermodynamic potential study and efficiency analysis [1] delivered, regarding the requirements of power sports engines, a clear statement calling for the use of an exhaust turbo charger for all further investigations. Although a mechanically charged concept offers several advantages it was not further investigated cause of lower efficiency and challenges regarding to the torque layout over wide engine speed range. [Figure 1](#) shows a comparison of a load step at constant engine speed, 3000rpm / 3bar to WOT, for an automotive turbo charged (TC) (solid line) engine and the basic engine with an available aftermarket TC-kit (dashed line). It can be observed that the aftermarket TC shows an unacceptable slow transient IMEP build up and that demonstrates the need of detailed

investigations when applying an exhaust turbo charger on an existing motorcycle engine.

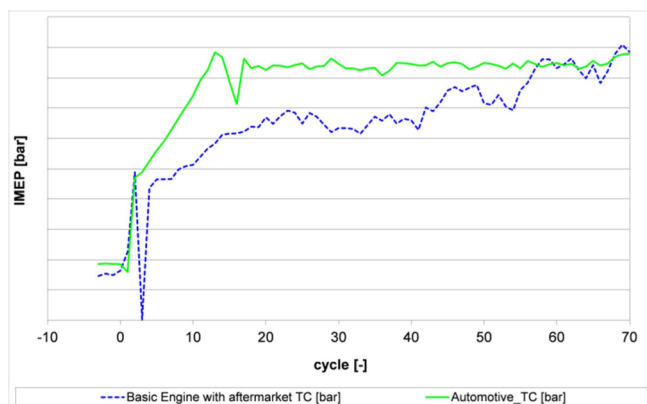


Figure 1. Load step at constant engine speed: basic engine with aftermarket TC and automotive TC [1]

The automotive TC engine (Figure 2) has been chosen as reference for the comparison of the transient behavior. Additionally the engine of the BMW R1200GS has been chosen as benchmark for the torque characteristic. The target is to exceed the torque characteristic of this Motorcycle (see Figure 2) with the help of the charged 800cm³ engine. Of special interest is the low end torque. In Figure 2 the torque curve of the BMW R1200GS (solid line) and two target torque characteristics of the prototype engine are shown. The first one (TC_characteristic_A, dashed line) has high low end torque and constant torque over a wide engine speed area like a typical automotive characteristic. The second one (TC_characteristic_B, dash dotted line) was purposely reduced at low engine speeds to generate a potential improvement of drivability. The relevant low engine speed area for drivability should be calibrated in the vehicle. It is visible that the torque curve of the TC engine exceeds the characteristic of the BMW R1200GS nearly over the whole engine speed range.

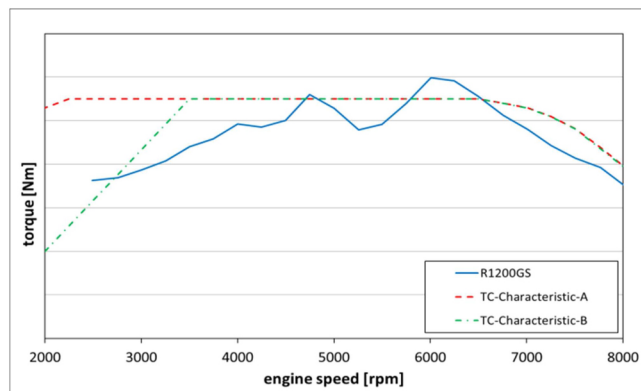


Figure 2. Torque characteristic of a BMW R1200GS and the target of the TC prototype engine

Table 1 shows the engine data of the basic NA-engine for this project. An 800cm³ gasoline inline two cylinder engine has been chosen for the development of the turbo charged motorcycle engine. The chosen concept implicates a few challenges [2], as there is, for example, an unsteady mass flow of a 2-cylinder engine to the turbo charger turbine. Nevertheless it has been selected after an extensive investigation process [1]. In the first step the engine has been equipped with an intake manifold fuel injection and has been prepared at the same time for gasoline direct injection. The investigations with gasoline direct injection are not part of this publication. When using this engine in combination with a turbo charger, some boundary conditions as well as limitations have to be considered. For example is it necessary to reduce the compression ratio, to change the intake port and therefore the cooling jacket. Additionally exhaust temperature limitation of 920°C and a maximum cylinder peak pressure of 105bar have to be regarded.

Table 1. Engine data of the basic NA-engine for the development of the TC-engine

displaced volume	800 ccm
number of cylinders	2
stroke	75.6 mm
bore	82 mm
compression ratio	12.5:1
number of valves/cylinder	4

Development Targets

The main targets for the development of this engine are the response behavior, the drivability and of course a torque target characteristic. Of special interest, for motorcycle applications and especially for charged motorcycle engines, is the response. The benchmark for TC engines is a state of the art automotive engine with a single turbo charger. Therefore measurements of various load steps at constant engine speed and with defined driving resistance have been performed. One example, load step at constant engine speed 3000rpm/ 3bar IMEP to WOT, for the bench mark engine and the basic 800cm³ NA engine can be seen in Figure 3. The NA-engine (red dashed line) delivers a better response (about 4 engine cycles) than the automotive TC-engine (green solid line, about 14 engine-cycles). With the help of the basic NA engine the characteristic of the ideal transient behavior (grey dash dotted) for the prototype engine has been defined. As the desired curve cannot be achieved, the transient behavior of the automotive TC-engine is a good bench mark for this application considering all limitations.

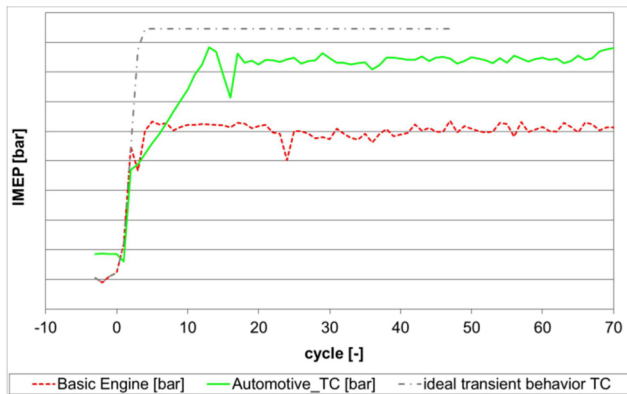


Figure 3. Examples of response for original NA-engine and automotive target

All simulations for the layout of the different engine components and test bench measurements have been performed on the basis of a predefined torque respectively power characteristic (Figure 4). These characteristics have been presented in [1], which dealt with the specification of the charging concept for this publication. The visible torque target characteristic is similar to a charged automotive characteristic. For motorcycle engines, the high torque at lower engine speeds, for this application from 2000rpm to 4000rpm, is atypical. Therefore this target has to be validated in the vehicle regarding drivability.

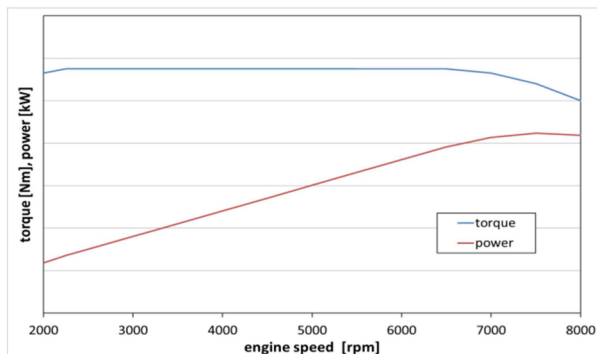


Figure 4. Power and torque target for further investigations

ENGINE MODIFICATION

As mentioned before, several modifications on the basic engine are necessary to prepare it for the turbocharger application. The three main modifications, which are presented in this publication, are the reduction of the compression ratio, the changes at the intake port to stabilize the combustion, and the changes at the cooling jacket because of the preparation for the GDI. The changes on the cylinder head, cooling jacket and intake port have been performed with the help of 3D-CFD simulation. In the next three chapters only the final results of these modifications are presented.

Additional to these modifications, the oil circuit had to be adapted to ensure the oil supply for the turbo charger. Therefore it is necessary to bleed oil from the regular circuit and guarantee a pressure less backflow of the oil into the oil pan. For initial tests with MPFI, no changes on the piston are necessary, except for the changes for the reduction of the compression ratio (see chapter REDUCTION COMPRESSION RATIO).

Reduction Compression Ratio

Table 2. Comparison compression ratio of the different engine configurations

Engine configuration	Compression ratio	
Original NA engine	12.0	-
charged MPFI engine	9.0	-
charged GDI engine	9.5	-

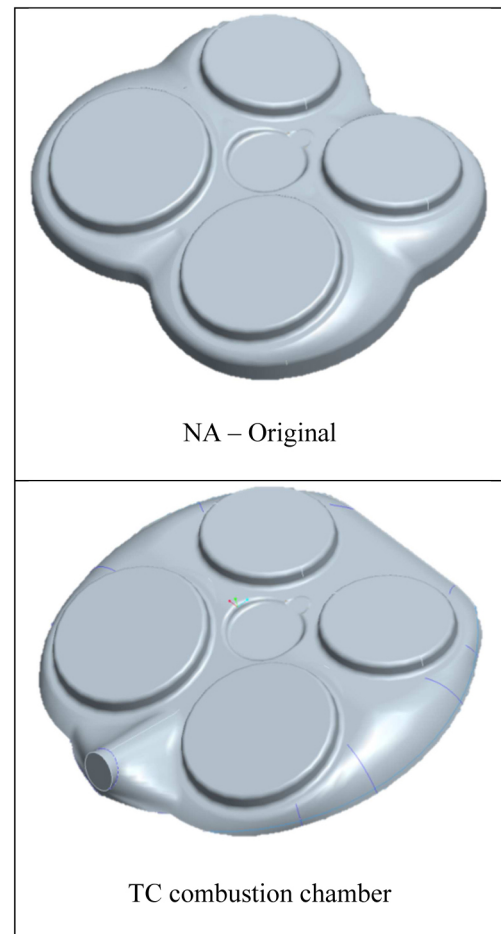


Figure 5. Combustion chamber of the original NA engine and the modified DI-TC engine

For turbo charging, a reduction of the compression ratio compared to the NA engine is necessary. In Table 2 the

defined compression ratios for original NA engine and MPFI / GDI charged engine configuration are visible.

The decrease of the compression ratio to the defined values has been realized by increasing the volume of the combustion chamber. Therefore several adaptations were necessary. For example the combustion chamber height has been increased and squeeze volumes have been reduced. Additionally a volume caused by the implementation of the GDI-Injector has been added to the new combustion chamber. Figure 5 shows two different versions of the adapted combustion chamber. In the top graphic the original NA combustion chamber and in the lower the modified TCGDI version with all mentioned changes is visible.

For the test bench measurements of the TC MPFI engine and the TC GDI engine the same cylinder head should be used. For the GDI configuration the TC combustion chamber offered a satisfactory decreased compression ratio of 9.76. To reach the target value for the TC MPFI compression ratio of 9.0 is the modification of the piston bowl sufficient.

Adaption Intake Port

The boundary conditions for the design of the intake ports are a circular throttle body unit on one hand and the increase of the charge motion on the other hand. Figure 6 shows a section through the final intake port layout with GDI injector and fuel rail. To improve the charge motion a tumble intake port design has been chosen. Therefore the port angle has to be increased to improve this tumble motion. This angle is limited by the GDI injector, which has been placed under the intake port. This results in a compromise with an intake port angle of 60° and an injector angle of 20° . Therefore it will be possible to use standard automotive injectors for all GDI test bench investigations.

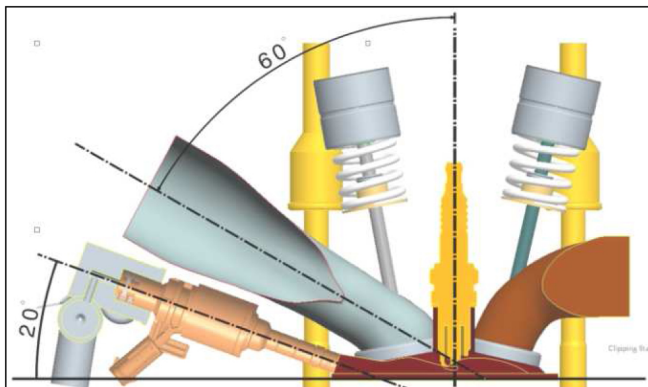


Figure 6. Final intake port layout with positioned injector and fuel rail [8]

Additional to the design activities various 3D-CFD simulations have been carried out. A steady state flow simulation with different valve lifts has been used to verify the potentials of different port designs. Parallel to the 3DCFD simulations flow box measurements have been performed to evaluate the simulation results. Figure 7 shows the

comparison of the tumble momentum and the flow coefficient for three different port designs, original NA port design (red dashed line), modified design V1 (green solid line) and the modified design V6 (orange dash dotted line) from the 3D-CFD simulation. The original port design is a filling port and for this reason it delivers the best flow coefficients and the worst tumble momentum over the valve lift. Both modified designs offer much more tumble momentum by having only a bit worse flow coefficient. Because of the importance of the tumble momentum the modified design V6 is the best compromise for the usage in a turbo charged engine. This design delivers the best tumble momentum and has up to 5mm valve lift nearly the same and over 5mm only a bit worse flow coefficients as port design V1.

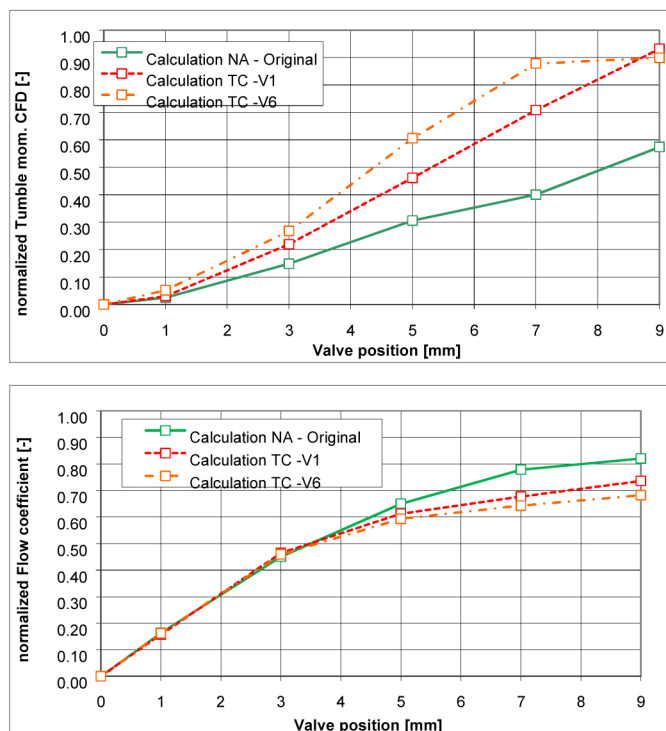


Figure 7. Flow coefficient and tumble momentum for different intake ports

Cooling Jacket

The cooling jacket has to be modified because of changes in the cylinder head for direct injection and the implementation of the tumble intake port. Figure 8 shows the modifications on the cooling jacket. The disconnection under the intake port (green circle) is necessary because of the implementation of the GDI injector. The changes effected by the new intake port design can be identified in Figure 8 by the golden surface. All modifications and redesigns have been analyzed with the help of 3D-CFD simulation to avoid critical surface temperatures or cavitation in the cooling jacket.

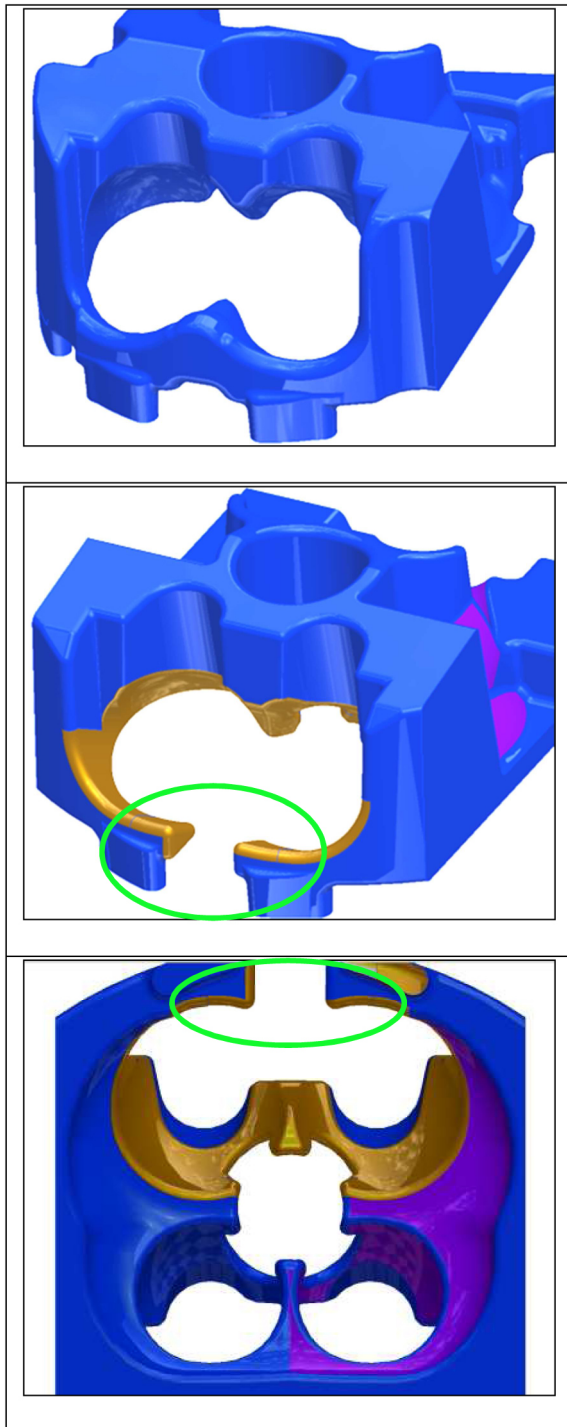


Figure 8. Modification cooling jacket for direct injection (middle and lower picture); original (upper picture)

The 3D-CFD simulation of the thermal flux between the cylinder head and the cooling jacket has been carried out with the simplification of constant wall temperatures. With the help of this simplification is a fast and accurate verification of the critical areas possible.

The main parameters for the analysis of the cooling jacket are the surface temperature and the saturation pressure

difference. This pressure difference is used for the detection of areas with high cavitation risk. Additionally to these parameters the flow situation, the temperature difference as well as the pressure difference between inlet and outlet are analyzed. The modified cooling jacket offers a thermal problem which was solved by an adaption of the transfer ports through the cylinder gasket. Figure 9 shows the pressure drop and the temperature difference between inlet and outlet for the original cooling jacket, the first (GDI version 1) and the final (GDI version 4) GDI version. The better results for the final design come, for both diagrams, primarily from the adaption of the transfer ports.

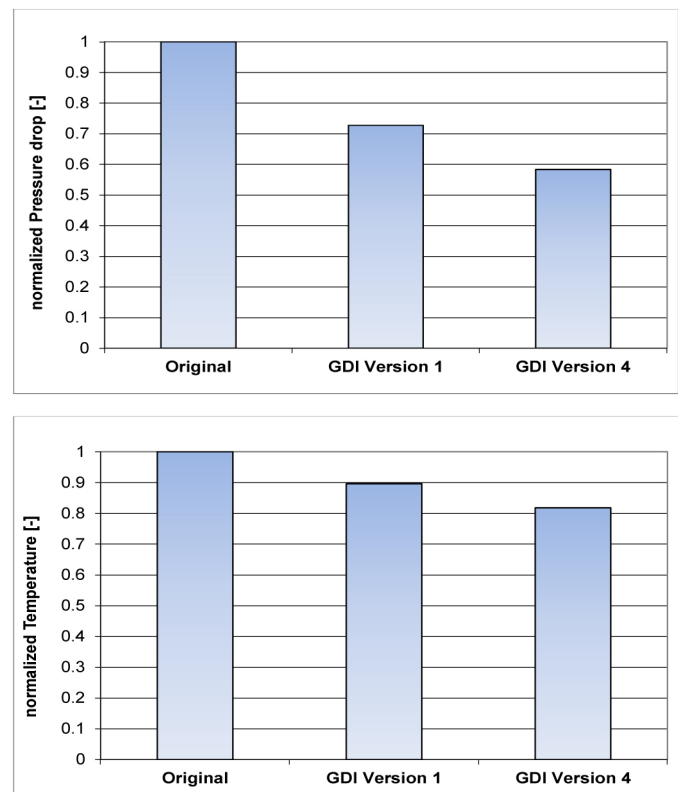


Figure 9. Temperature and pressure drop for different cooling jacket versions

In Figure 10 are surface temperatures of the GDI version 1 (upper picture) and GDI version 4 (lower picture) visible. The Version 1 shows higher temperatures around the changes for the GDI injector (red circles upper picture). To overcome this temperature rise the above mentioned transfer ports are placed directly under these high temperature areas (black arrows lower picture) to increase the discharge and therefore decrease the surface temperature. The lower picture shows the achieved improvement of the temperature behavior (black circles lower picture) with the help of all performed optimizations. It is visible that the temperature is more homogenous distributed and offers continuous raise from the inlet to the outlet.