In-vehicle Optimization of
2-stage Turbocharging for Gasoline Engines

Felix Buchner¹) Stefan Wedowski²) Andreas Sehr³) Sandra Glück⁴) Christof Schernus⁵)
¹) FEV Japan Co., Ltd., 1008 Burex Kojimachi, 3-5-2 Kojimachi, Tokyo 102-0083, Japan
²) ³) ⁵) FEV Motorentechnik GmbH, Neuenhofstr. 181, 52078 Aachen, Germany
⁴) Institute for Combustion Engines, RWTH Aachen University, Germany
Presented at the JSAE Annual Congress on May 18, 2011
Received on July 7, 2011

ABSTRACT: Future downsizing for gasoline engines will lead to higher boost ratio and specific torque than realized with todays engines. In the present paper, the vehicle integration of a 2-stage-turbocharging system into a middle class vehicle is discussed, starting with the adaptation of the combustion system to high boosting ratio by means of CFD supported Charge-Motion-Design-Process. Furthermore, the layout of the control concept had been supported by 1D-simulation of the overall engine/vehicle-system. Results from this work will be presented and compared to vehicle measurement data, leading to an assessment of optimization potential regarding drivability and discussing the sensor and actuator concept.

KEY WORDS: (Standardized) Heat Engine, Spark ignition engine, turbocharger [A1]

1. INTRODUCTION

Downsizing of SI engines for CO2 emissions reduction has been established as a trend in the market. This trend was enabled by the development of direct injection for SI engines, resulting in new degrees of freedom concerning injection strategies and control of gas exchange of boosted engines. Concepts for new engines with significantly reduced displacement in future powertrains are therefore in discussion.

The supplementation of current 6-cylinder by smaller 4 cylinder engines will increase the level of specific power above 100 kW/l. In addition a dynamic torque characteristic for comfortable driving according to the customers demand needs to be realized. Furthermore, a high specific power output can be used also for high performance versions in smaller vehicles, which leads to a reduction of cost and development effort by concentration on a limited number of base engines in the production portfolio. This strategy requires boost pressure levels which are higher than in todays charged engines.

The development of such a downsized powertrain with „high pressure charging“ leads to new challenges. In this paper, tools and methods which are used to fulfill these challenges are discussed.

2. TARGETS AND MOTIVATION OF 2-STAGE CHARGING

The study was based on an initial target definition of 120 kW/l and corresponding high torque between 1500 and 5500 rpm, as shown in Fig. 1 in comparison to the scatterband of todays turbocharged engines.

Fig. 1: Target of 2-stage boosted engine
The impact of such a target torque curve on the charging system is shown in Fig. 2.

Fig. 2: Limits of 1-stage turbocharging
The full load curve of an engine with 90 kW/l and state-of-the-art low-end-torque requires the complete width of a typical compressor map. At low engine speeds and maximum torque, the steady state operation points are close to the surge line of the compressor map. At rated speed high efficiencies and sufficient high altitude and control reserve has to be protected for. The operation area of today’s mass production radial compressors is insufficient for an engine layout with 120 kW/l and high low-end-torque as shown by the two open circles in Fig. 2. The compressor characteristic is optimized towards an increased map width which results in drawbacks concerning efficiency. The transient response for such a large turbocharger due to the high air flow will be unacceptable.

Also with regard to turbine behaviour, it would not be feasible to realize the high torque requirements at low speeds in combination with acceptable exhaust back pressure at rated speed range.

Therefore, a charging system with two boosting devices need to be chosen. In Fig. 3, different charging systems which are in discussion are compared with regard to package and effort like e.g. design complexity, transient torque build-up and maximum specific power.

![Fig. 3: Evaluation of different charging systems](image)

The reference is a standard fixed geometry turbocharger with waste gate, which is able to fulfill the state-of-the-art torque requirements with relatively low effort. Next improvements are “Advanced-1-stage” solutions, which may include e.g. variable turbine geometry or other measures. These chargers enhance the performance while increasing the necessary effort for mass production, however are still limited due to the above mentioned reasons.

Systems with electrically or mechanically driven compressors, either of positive displacement or turbo machinery type are shown above. Both hybrid systems result in an improved transient behaviour and allow a power oriented layout of the turbocharger. The electrically driven concept even offers package advantages due to its degrees of freedom concerning positioning compared to a drive-belt driven compressor. However its major drawback is still the limitation of today’s electrical systems.

A combination of a mechanically driven compressor and turbocharger is quite well known from a number of studies and also found its way into series production. Nevertheless, the compressor should be disengaged in part load and rated power for fuel consumption reduction due to lower friction. Finally, the 2-stage turbo charging system is a promising concept and already in series production in diesel engine applications. It offers high potential with regard to power output. The main challenges are the appropriate layout of the turbochargers, an efficient combustion process in spite of high boost pressure levels and realization of good transient response while managing the package situation.

Due to this evaluation, it was decided to investigate solutions for this concept in the feasibility study discussed here.

### 3. LAYOUT OF CHARGING GROUP

For the layout of the charging group, the commercially available one-dimensional code GT-Power by Gamma Technologies was used, the simulation could be based on a well-matched model of the FEV 1.8 ltr baseline engine.

The focus of the layout was of course the matching of available turbochargers. For both stages, different wheel diameters and housing shapes as well as control concepts were investigated. Also, available turbochargers which are already in the market should be used. As result of the layout phase two turbochargers were defined, which enable the realization of the required boost pressure and were promising concerning good transition between the two stages, as the operation lines in Figure 4 show.

![Fig. 4: Compressor map: Steady-state full load](image)
While the steady-state operation line seems to be located unfavourable at first sight, the transient operation line shows that during acceleration the high pressure turbocharger is taking over more work due to the slow spool up of the low pressure stage, and reaches higher pressure ratios at higher flow rates while operating in the area of optimum efficiency, see Figure 5.

![Vehicle Acceleration](image)

**Fig. 5: Compressor map: vehicle acceleration**

The layout of the low pressure stage was carried out with focus on minimum back pressure and good fuel efficiency at rated power. The large dimension of this charger would result in the delayed spool up during acceleration.

After investigating alternative turbine bypass arrangements, it was decided to carry over the control concept known from in-series diesel applications. This includes the complete bypassing of the high pressure stage on turbine and compressor side and actuation of the low pressure turbine by a standard waste gate in the high power operation range. The high pressure turbine control is realized in form of a bypass flap dimensioned for the complete rated power mass flow, the high pressure compressor bypass is actuated pneumatically. Its active opening is part of the control concept, but it is done by a pure open-closed switching.

### 4. ADAPTATION OF THE COMBUSTION SYSTEM

The realization of the high specific power and torque required an adaptation of the combustion system. The chosen concept requires a high charge motion intensity which improves the mixture homogenization in order to gain a fast combustion for improved knock resistance and prohibiting the chance of irregular combustion.

The development of the intake port flow and the incylinder charge motion is performed by the CMD process\(^3\). Three different intake port variants were designed and assessed by this methodology. These ports are the base intake port geometry (base), a version with medium tumble intensity (MTP) and a port with high tumble intensity (HTP). Figure 6 shows the incylinder flow field at the end of the suction stroke.

![In-Cylinder Flow Field](image)

**Fig. 6: In-Cylinder Flow Field and CMD prediction**

The base geometry does not show a developed tumble motion up to this crank angle, as seen also from the non-existence of a tumble vortex centre. Compared to this, the MTP variant generates a more developed tumble with upwards directed motion above the piston on the exhaust side. A further increase in tumble intensity is obtained with the HTP port variant. The flow field shows a significant tumble upwards motion and a tumble centre in the middle.

The evaluation of the flow field at the end of compression with respect to convection and turbulence characteristics enables a prediction of burn duration and burn delay based on specific correlation between characteristic quantities\(^4\). This approach is used here to assess the combustion potentials of the intake port variants in the given thermodynamic conditions of high pressure charging. The results in Figure 6 show that with increasing tumble intensity a reduction in burn delay and burn duration is observed. This phenomenon occurs especially at lower engine speed.

A further challenge for high pressure charging is an adequate mixture formation, since the spray – flow interaction is significantly affected by the high charging degree and the larger amount of fuel injected. Different combinations of intake port and fuel injectors are investigated on the FloTec engine. FloTec is an optically motored engine, which can be operated with a metal flow box as cylinder head, and enables a broad optical access through a quartz glass liner. On the FloTec optical diagnostics such as PIV, Mie scattering and LIF provide detailed information on flow, injection and mixture formation, without the necessity to design and set-up a costly...
transparent single cylinder engine.

Figure 7 shows the design of the FloTec and Figure 8 a sequence of high speed Mie scattering images of the direct injection at a low engine speed of 1500 rpm. The tumble charge motion moves the fuel spray, which is initially captured by the piston, out of the piston bowl and transports it towards the centre of the combustion chamber, where the fuel finally evaporates. The presented sequence does not show insufficient mixture formation, such as high fuel droplet concentration in direct vicinity to the chamber walls.

5. RESULTS COMBUSTION SYSTEM STUDY

During engine test bench investigations and with also with a prototype application in a vehicle, the feasibility of the high pressure charging concept could be proved, especially the target torque values between 1500 to 2000 rpm could be achieved without any irregular combustion phenomena. For a characterization of the combustion system, figure 6 shows the peak pressure position as function of final compression pressure, which is calculated from compression ratio and cylinder mass derived from boost pressure and volumetric efficiency.

The left part of the scatter band consists of values from NA engines and PFI turbocharged engines database without scavenging gas exchange. Engines running with high octane fuel are located at the right border of this scatter range. Turbocharged direct injection engines with significantly reduced residual gas fraction and therefore reduced knocking tendency, are shown as the separate area. At the borderline of this area, the base 1-stage charged version as well as the 2-stage engine version can be found.

The values from both engines are located on a line of constant quality of knock resistance relative to boost pressure, which could be achieved by the optimization of the combustion system as discussed before. Nevertheless this graph shows improvement potential, as of course, the absolute position of combustion shall be developed towards optimum peak pressure position. With regard to the feasibility of a high pressure charging concept, the achieved results were evaluated positive, as there can be expected further potential in detailed optimization of gas exchange and mixture formation.
6 PACKAGE

The feasibility of the concept concerning package was also be studied regarding design and digital mock up of a vehicle package solution, using a Ford Focus medium class vehicle. The design is characterized by a very compact exhaust manifold, which directly is assembled to the high pressure turbine housing. The turbine bypass is incorporated in the same housing, and actuated pneumatically.

![Vehicle integration of 2-stage turbocharging](image1)

**Fig. 10:** Vehicle integration of 2-stage turbocharging

![Prototype turbo charger design](image2)

**Fig. 11:** Prototype turbo charger design

It is opening the bypass to the low pressure turbine, which is located above the high pressure stage. As can be seen on the Fig. 10, the complete charging group could be implemented into the vehicle without any significant modification of the vehicle front. This preliminary design status of the study proved the feasibility. The full potential of the system will be exploited only after further optimization also of the design, for example of cold side piping and actuation components. This design optimization is shown in Fig.11.

7. TRANSIENT PERFORMANCE

Achieving the steady state targets is insufficient to validate a turbocharged engine concept. Important is an acceptable transient behaviour and good driveability. Especially in the case of a 2-stage system, the interaction and control of both stages has to be investigated and evaluated.

Fig. 12 shows the turbocharger speed for each stage during acceleration in third gear, from simulation as well as from vehicle measurement. Proper correlation of simulation and testing is only possible if simulation can be based on a well matched model in regards to engine, turbocharger control and vehicle.

The small turbocharger spools up directly after drive pedal actuation. The gradient of the turbocharger speed is approximately 100,000 rpm/sec. The low pressure stage spools up more slowly due to inertia and energy conversion in the high pressure stage.

![Vehicle Acceleration 3rd Gear](image3)

**Fig. 12:** Transient behavior of 2-stage turbocharging

Reaching the target boost pressure of about 2.6 bar absolute after three seconds the high pressure turbine bypass is opened, leading to a reduction of turbine speed. Following this, the boost pressure is controlled, as can be seen in the speed curves.

Fig. 13 shall compare the boost build-up of the 2-stage system to several 1-stage systems. From the FEV database, measurements of boost pressure build up during vehicle accelerations in third gear, starting from different engine speeds, were analyzed.
Fig. 13: 2-stage concept evaluation

In the upper diagram area the maximum achieved boost pressure during the acceleration, in the lower half the boost pressure gradient in bar/sec is plotted. Time reference for the latter is the time between reaching natural aspirated full load, typically about 0.2 seconds after start and reaching 90% of maximum boost pressure has been chosen.

Even with the pre-optimized concept status, the advantage of the 2-stage system below 2000 rpm is obvious, taking into account the high level of maximum pressure. It is clear that the boost level of 1-stage systems is reached much quicker.

8. CONCLUSIONS AND OUTLOOK

The realization of high pressure charging for SI engines to achieve a specific power above 100 kW/l was performed. The new challenges concerning the charging system layout and combustion system development were successfully solved. This enables the possibility to replace large displacement engines for fuel consumption reduction or to enlarge an engine family by a high power version. It could be shown within this feasibility study, using advanced CAE based optimization tools as well as thoroughly matched engine and vehicle simulation models, that these challenges can be dealt with.

A 2-stage turbocharging system was designed and integrated in a demonstrator vehicle. The engine performance and control concept could be verified showing that the combustion system has the potential to accept high boost pressure levels with acceptable combustion stability without irregular combustion.

Nevertheless, the study also pointed out the road to further optimization, namely an optimized design and control calibration for improved transient response.

9. ACKNOWLEDGEMENTS

The authors wish to thank the “Bundesministerium für Wirtschaft und Technology” of the Federal Republic of Germany and also Borg Warner Turbo Systems for the strong support during the study.

REFERENCES

(1) Habermann, K. et.al. Maßnahmen zur Verbesserung des Anfahrdrehmomentes bei aufgeladenen Ottomotoren, Aufladetechnische Konferenz Dresden 2002
(2) KREBS, R., et al.: Neuer Ottomotor mit Direkteinspritzung und Doppelaufladung von Volkswagen, MTZ 66, 11/2005
(3) SCHMITD, A., et al.; Charge Motion Design Prozess™ - Neue Wege zur Auslegung der Ladungsbewegung und Integration in den Entwicklungsprozess, 8.Internationales Stuttgarter Symposium - Automobil- und Motorentechnik / 11.-12.März 2008
(4) WIESE, W., PISCHINGER, S., ADOMEIT, P., EWALD, J.: Prediction of Combustion Delay and Duration of Homogeneous Charge Gasoline Engines based on In-Cylinder Flow Simulation, SAE Paper 2009-01-1796