Thermodynamic advantages and challenges of a parallel sequential twin boosting system on a high efficiency 1.0 l CNG engine

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ABSTRACT

The introduction of the Worldwide Harmonized Light Vehicles Test Cycle (WLTC) and the implementation of Real Driving Emissions (RDE) increase the requirements for automotive powertrains. The limits for CO_2 and pollutant emissions are continuously reduced which leads to increasing challenges for powertrains to be compliant to regulations and still offer a decent drivability. Reducing engine displacement and increasing the compression ratio, while simultaneously increasing the specific engine power, is one major strategy to reduce the fuel consumption. At the same time scavenging and fuel enrichment have to be omitted in order to fulfil future emission legislations.

The high knock resistance of compressed natural gas (CNG) enables an efficiency increase of spark ignition engines. Without the knock limitation of gasoline a significantly higher compression ratio (CR) can be applied in combination with a considerably increased boost pressure before knocking inhibits an optimum combustion phasing. While the increased CR directly leads to a higher efficiency, the higher boost pressure enables to increase the specific power and therefore the downsizing capability of the engine. With a higher degree of downsizing a small engine can replace a larger engine at the same power level. Due to the lower displacement of the downsized engine pumping and friction losses are reduced and thus lead to an improved engine efficiency.

Raising the specific power and torque leads to an increased demand of boost pressure which cannot be provided by a conventional mono-turbocharger configuration. Therefore, a parallel sequential boosting system, comprising of two thermodynamically identical wastegate turbochargers is developed. One of the turbochargers is operated permanently. The second turbocharger is activated by a continuously variable valve lift system when required. In mono turbo mode one small turbine supports favorable transient response and maximum low end torque. The performance target in parallel twin turbo mode at rated power is to exceed 110 kW/l. An integrated exhaust manifold with split exhaust ports, a compressor recirculation valve and a shut off valve complete the boosting system.

The following paper describes the thermodynamic advantages and challenges of a parallel sequential twin turbo boosting system. Based on calibrated 1-D engine models detailed studies are conducted. The result is an intelligent switching strategy and an optimally sized turbocharger for a constant high torque level during the transition between mono and twin turbo mode. Compared to common turbochargers a very low mass flow rate and high thermodynamic efficiency are the main requirements for the compressor and turbine design. Measurements on the hot gas test stand and on the engine dynamometer prove the high performance capability of the parallel sequential boosting system applied to a dedicated 1.0 l CNG engine.

ABBREVIATION

BMEP	brake mean effective pressure
BSAC	brake specific air consumption
BSFC	brake specific fuel consumption
CA	crank angle
CAT	catalytic converter
CNG	compressed natural gas
ECU	engine control unit
GasON	Gas-Only internal combustion engine
LET	low end torque
MF	mass flow
MFP	mass flow parameter
MW	map width
PFI	Port fuel injection
PR	pressure ratio
RAAX	radial axial
RP	rated power
TC	turbocharger
TDCF	top dead center firing
tiVVA	twin independent variable valve actuation
tiVCT	twin independent variable cam timing

1 THE GASON ENGINE CONCEPT

The Gas-Only internal combustion engine (GasON) project is part of Horizon 2020, an EU Research and Innovation program that promises more breakthroughs, discoveries and world-firsts by taking great ideas from the lab to the market [1].
One of the major aims of the GasON project is to develop a CNG-only engine that is able to comply with 2020+ CO_2 emission targets and simultaneously improving engine efficiency and vehicle performance also with regard to its CNG range capability. To fulfil all the objectives of the project, the target is a 20 % CO_2 emission reduction in NEDC cycle compared to the current best in class CNG vehicle 2014 [2]. This reference vehicle is a Ford Grand C-Max with a turbocharged 1.6 liter engine with port fuel CNG injection.
The GasON engine layout is based on a Ford 1.0 liter 3-cylinder engine. To achieve the CO_2 emission reduction several advanced technologies are applied [3]:

- Downsizing the original 1.6 litre 4-cylinder [4] to a 1.0 litre 3-cylinder engine with high compression ratio CR=13 to fully exploit the superior knock resistance of CNG with its RON number > 120 [5]. Reinforcement of engine structure and components to stand the higher specific engine load and peak pressures of 160 bar while minimizing the friction drawbacks.

- Change to CNG Direct Injection system (CNG DI) [6] [7] to avoid the volumetric disadvantages of PFI. The gas injection into the ports would displace a huge amount of the trapped air which leads to an increased boost pressure demand.

- Introduction of a twin independent variable valve actuation (tiVVA) system on both intake and exhaust side; on the intake side for reduction of pumping work and advanced gas exchange control, on the exhaust side mainly for the sequential activation of the turbo chargers.

- Development and application of an innovative parallel sequential twin boosting system to enhance low end torque and transient response while simultaneously increasing the specific power. This twin turbo approach requires a modified cylinder head design with separated exhaust ports and manifolds.

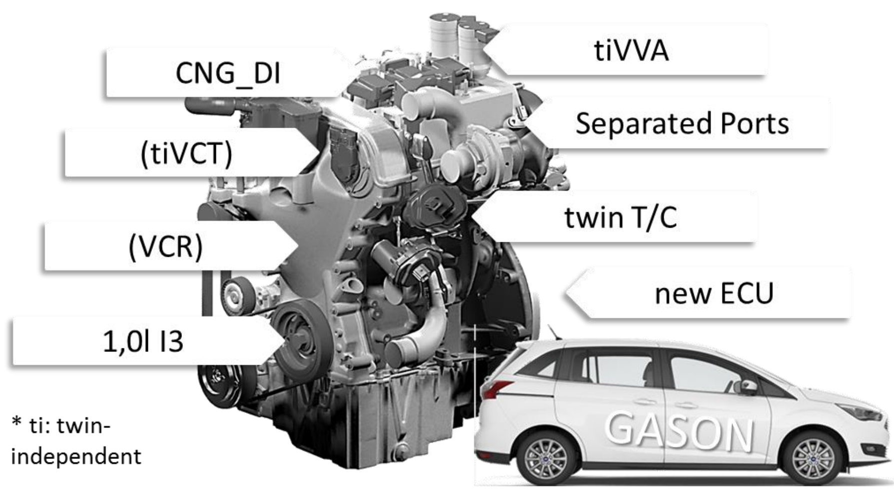

Figure 1: GasON engine concept [3]

2 PARALLEL SEQUENTIAL TWIN BOOSTING SYSTEM

2.1 APPLICATION SPECIFIC TURBOCHARGER DESIGN

Due to the application dependent, individual engine requirements Continental develops specific turbochargers for each application. This enables an optimum adaption of the thermodynamics of the turbocharger to the internal combustion engine. The ongoing optimization of thermodynamics and structural mechanics improves the overall performance of this turbocharger technology [8].

Figure 2 shows the thermodynamics design process of the turbocharger. In the first step during the matching process an optimal compressor and turbine map is predicted which would lead to an ideal engine performance. By means of computational fluid dynamics (CFD) and finite element method (FEM) a corresponding compressor and turbine stage are developed. The 3-D design is aligned with the customer in order to meet the packaging constraints. Every prototype is measured on the hot gas test before it is delivered to the customer. This experimental review is an integral part of the process. Deviations between matching, CFD layout and measurement are evaluated. Thereby, the prediction quality is improved and the number of engine tests is reduced.
The interaction of the turbocharger with the internal combustion engine, especially with the center of combustion, can be evaluated by a 1-D gas exchange model with knock modeling. An accurate prediction of the brake specific air consumption is achieved.

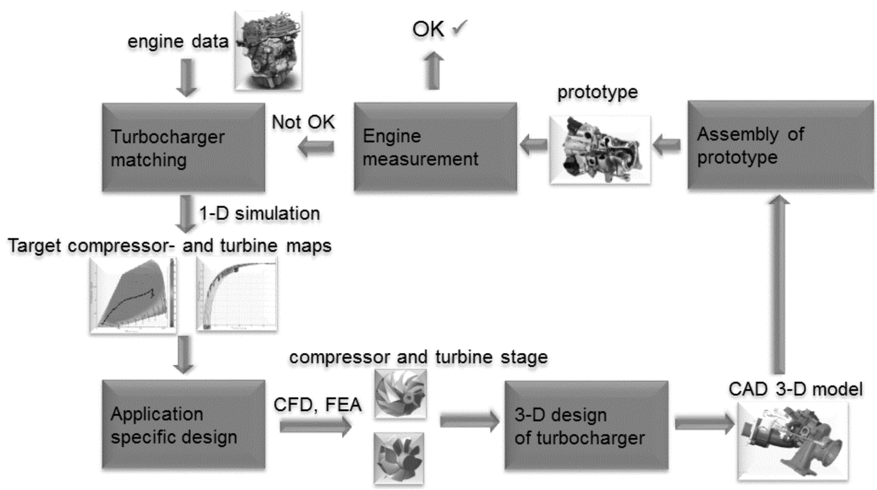

Figure 2: Application specific turbocharger design process [8]

The Continental process of application specific turbochargers is applied to develop the compressor and the turbine stage of the boosting system for the high requirements of the GasON engine. The thermodynamic advantages of a parallel sequential twin boosting system and the challenges of the high efficiency 1.0 litre CNG engine are described in the following section.

2.2 TURBOCHARGER MATCHING

The engine simulation and turbocharger matching is done by using a 1-D gas exchange model. Therefore, the first step is to calibrate the existing engine model of the well-known spark ignited 1.0 l inline three cylinder Ecoboost engine [9] to steady state dynamometer test results of a mono turbocharged engine operated with CNG fuel. This engine measurement data, recorded with a brake mean effective pressure (BMEP) of 25 bar and a rated power of 92 kW, correlated well to the simulated air mass and the intake manifold pressure as well as to the predicted turbine inlet pressure, Figure 3.

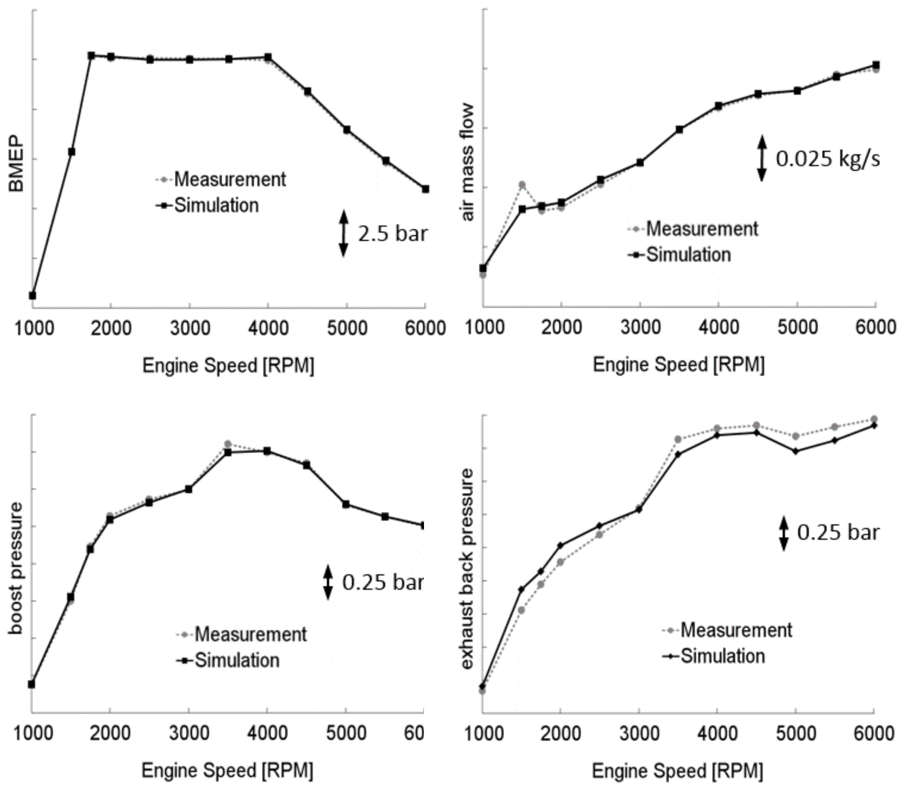

Figure 3: Results of 1-D engine model calibration

The next step is the implementation of a twin boosting system to the calibrated engine model. In the pre-matching phase different boosting systems (parallel sequential, mixed sequential, serial) are compared. The matching target of using two identical thermodynamic layouts for the two turbochargers resulted in a parallel sequential twin boosting system. With this twin turbocharger configuration the maximum engine performance is achieved. The parallel sequential twin boosting system is set up according to the scheme shown in Figure 4. The turbochargers are connected to two 3-to-1 integrated exhaust manifolds. All three exhaust valves on the left side of their combustion chambers are connected to the permanent operated turbocharger TC1, all three exhaust valves on the right side are connected to the second, intermitted operated turbocharger TC2. A fully variable valve lift system enables an optimal switching between single and parallel mode. The compressor shut off valve and the compressor recirculation valve help to rev up turbocharger TC2. Both turbochargers include electrical wastegates for boost pressure control.

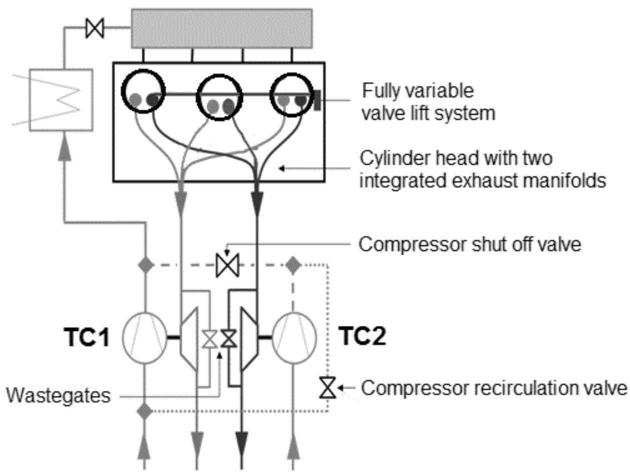

Figure 4: Scheme of parallel sequential twin boosting system on 3 cylinder engine [4]

The performance targets are increased to a BMEP level of 30 bar at an engine speed range from 1500 RPM to 4000 RPM and a specific power of 110 kW/l at 6000 RPM. Additional matching boundaries are listed in the following table:

Performance targets and matching boundaries

Twin turbochargers	indentical thermodynamic layout
Low end torque	240 Nm / 30 bar @ 1500 RPM
Specific power	110 kW/l @ 6000 RPM
Compressor speed margin	10 %
Maximum exhaust temperature	1050 °C
Exhaust Lambda	1.0 (no fuel enrichment)
Max. scavenging	1 % O_2 upstream CAT
Max. cylinder pressure	160 bar
Max. compressor outlet temperature	220 °C

The maximum scavenging rate is limited to 1 % O_2 concentration upstream of the catalytic converter. An optimization of intake and exhaust valve timing was conducted to achieve the high torque level with a low amount of scavenging. The ignition control sets the spark advance to a minimum which is limited by the maximum cylinder pressure of 160 bar. Due to the high efficiency engine concept with a compression ratio of 13 the brake specific air consumption (BSAC) at full load is increased.

Both, the low scavenging rate and the high BSAC result in a high boost pressure demand. The compressor pressure ratio increases from low end torque (LET) to rated power (RP) from 2.7 up to 3. Therefore one of the main targets of the compressor matching is to move the area with maximum compressor efficiencies to high pressure ratios. To ensure the speed margin of 10 % at rated power, a maximum compressor speed of 300,000 RPM was applied. The compressor map width is increased to obtain the required surge margin at 1500 RPM and 30 bar BMEP. The complete full load lug line, i.e. all operating points in mono turbocharger mode from LET 1500 RPM to 2500 RPM and all operating points in twin parallel sequential turbocharger mode from 3000 RPM to RP 6000 RPM, are in areas of high compressor efficiencies, Figure 5.

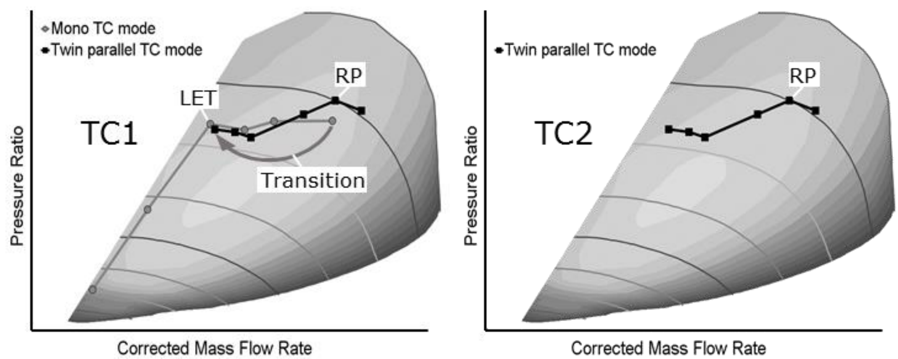

Figure 5: Compressor map - full load in mono and twin parallel TC mode

On the exhaust side of the engine, the pressure loss of the exhaust gas aftertreatment system creates a high pressure level post turbine. To achieve the requested turbine power with the lowest possible exhaust back pressure the turbine matching focuses on high efficiencies at high turbine blade speeds. The turbine mass flow capacity has to be extraordinary low. It is limited by the first operating point in twin parallel mode at 3000 RPM. When the right exhaust valves open (Figure 4) and the exhaust mass flow is separated on both turbine stages, enough turbine pressure ratio and enough turbine power has to be provided to reach the high boost pressure that is necessary for a 30 bar BMEP full load without torque reduction.

The transition at full load between mono and twin turbo mode is between 2500 RPM and 3000 RPM engine speed. The change of the compressor operating point in the permanently running turbocharger TC1 during the transition is shown in Figure 5. Up to 2500 RPM the wastegate of TC1 is used to control the boost pressure, the wastegate of TC2 and the compressor shut off valve remain closed. The right exhaust valves are continuously opened by the VVA system starting at 2500 RPM. At the same time the recirculation valve at TC2 opens to rev up TC2. When the boost pressure after TC2 reaches the value of the boost pressure TC1, the compressor shut off valve starts to open while the recirculation valve closes, Figure 6. Henceforward both turbochargers TC1 and TC2 run in parallel mode to deliver the air mass flow demand at engine speeds up to 6000 RPM.

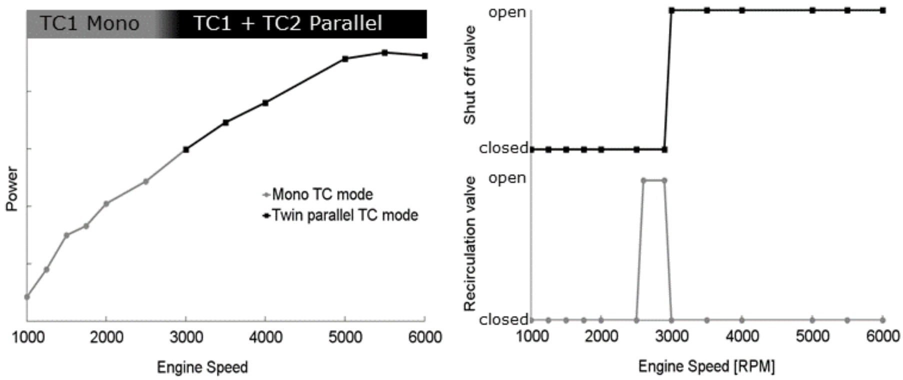

Figure 6: Transition between mono and twin TC mode

The high performance targets and the matching boundaries of the GasON engine concept can only be fulfilled with a parallel sequential twin boosting system. A single stage turbocharger would not reach the high torque and power level. In Figure 7 a single turbocharger matching is compared to the matching result for compressor and turbine of the parallel sequential twin boosting system. A single stage turbocharger for a 1.0 l CNG engine already has a low mass flow. The twin turbocharger matching results in a mass flow that is 30 % lower both on compressor and turbine side. The relative compressor map width has to be similar to a single stage turbocharger in order to fulfill the large spread between low end torque and rated power. To reach the required compressor and turbine efficiencies at such low mass flows is one of the biggest challenges for this aerodynamic design.

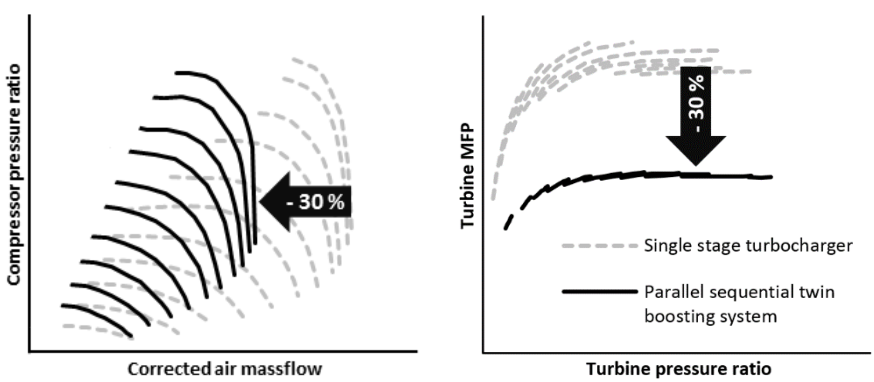

Figure 7: Single vs. twin turbocharging

2.3 COMPRESSOR UND TURBINE DESIGN

The low turbine and compressor capacity leads to a small turbocharger layout. However such a low size layout is limited by different boundary conditions. Small size turbomachines limit the Reynolds numbers and increase thermal losses. These physical characteristics already restrict the thermodynamic performance. The size of tip clearances, which result from the bearing concept, leads to a negative effect on efficiency. Additionally rotational speed is limited by the size of the available bearing housing. The diameter of the compressor wheel could not go below a certain limit, as long as a minimal circumferential speed is needed to achieve the required pressure ratio.

The consequence of low compressor diameter specific capacity is a narrow annular channel with short blades, which brings an adverse clearance to passage height ratio. Avoiding this fact, the channel width is simply increased by implementing a full bladed compressor wheel without splitter blades and furthermore an increased number of blades. In addition the back sweep angle is increased, which requires a further passage width enlargement at the trailing edge to achieve the required circumferential absolute flow velocity component for the pressure ratio.
A result of these arrangements of passage height enhancement is a comparatively low flow coefficient both at impeller inlet and outlet, which means the meridional velocity component is low. Thereby the level of flow losses is reduced on the one hand and on the other hand the level of the kinetic energy transfer is reduced as well. However, the missing component of energy transfer is supplemented by the large radial extension of the impeller. The results of the low inlet diameter correspond to the outlet diameter of the impeller. Despite the named arrangements of channel height enhancement the inlet diameter still remains low in relation to the outlet diameter of the available layout. The radial extension brings an increase of circumferential blade speed, which delivers an energy transfer effect of the centrifugal field on the flow within the compressor rotor. Hence, a high radial extension causes a high centrifugal effect and high energy rise. The effect of the centrifugal field on the flow in the impeller in general has a large positive effect, because it is effectively loss-free.
Furthermore, the back the sweep angle enhancement leads to an absolute velocity reduction at the diffuser inlet and thereby lower diffuser losses and lower diffusion in the diffuser. The degree of reaction increases and the diffusion in the impeller decreases. Conclusively, the described arrangements lead to the reduction of over tip leakage, reduction of flow losses within the impeller and the diffuser, whereby the difficult preconditions regarding performance could be re-adjusted with the available layout.

The precondition on the turbine side is the same as on the compressor side and accordingly a similar approach is adopted. The stage capacity in relation to the rotor diameter is lower compared to common layouts. Therefore, in terms of annular channel width enhancement, blade numbers and their angles were increased to reduce tip leakage. But in contrast to the compressor, the rotor inflow angle is determined by the volute geometry at the turbine. Furthermore, the capacity of the stage is adjusted by the volute size in a certain range. The volute size or the A/R ratio respectively was minimized for achieving the required low stage capacity.
A secondary effect of the small volute size is a comparatively high rotor inflow angle with a large tangential velocity component and a low degree of reaction. However, both properties have a negative effect on efficiency, if certain limits are exceeded. The measures for rotor channel width enhancement and volute size reduction have to be well balanced, to achieve the optimal performance.

Beside the application design of the compressor and turbine wheels and volutes for optimum performance, the 3D-design of the turbocharger inlets and outlets has to meet the packaging constraints.
Connecting two turbochargers with two electrically actuated wastegate systems on the 1.0 liter GasON engine with a relatively small cylinder head is a challenging task. At the same time additional package constraints given by the connection of air and exhaust pipes, the oil and water supply for the bearing housing as well as the engine periphery and the vehicle chassis have to be fulfilled. Therefore, an overlapping turbine housing flange design is applied in order to provide enough sealing surfaces and to keep a compact turbine housing, see Figure 8. On the other hand it is necessary to keep the same rotational direction for both turbochargers because it enables the usage of identical center sections i.e. identical turbine and compressor wheels and identical bearing system. Consequently, same rotational direction facilitates cost reduction potentials under mass production conditions.

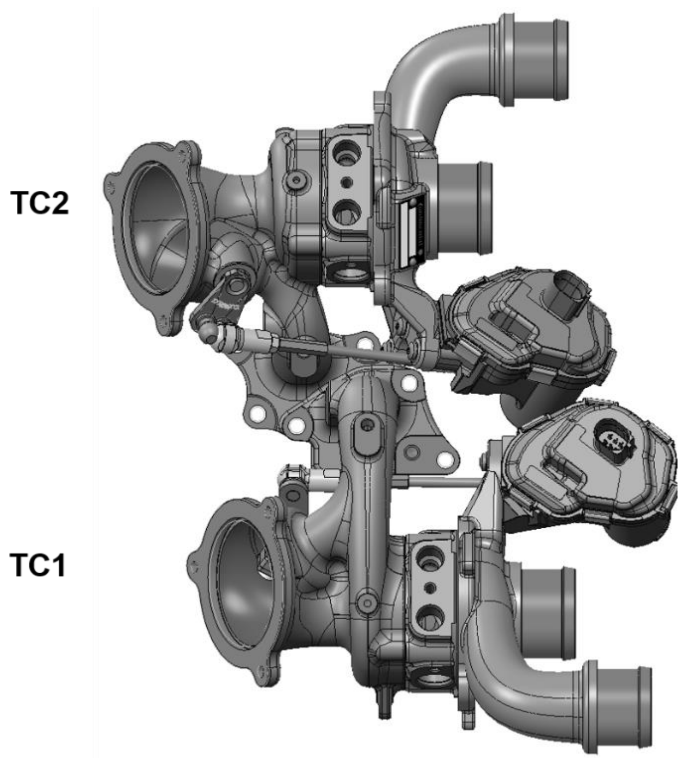

Figure 8: Upper and lower turbocharger assembly of parallel sequential twin boosting system

Due to the fact that the lower turbocharger is operated permanently and the upper one only is activated on demand in twin parallel mode, a special oil sealing system is applied to prevent oil leakage when the upper turbocharger is idling during the mono mode. Therefore, there is no need to shut down the oil supply circuit when only the lower turbocharger is running. Both turbine housing exit flanges are placed in the same plane and connected with one flexible adapter. On the lower turbocharger, which is running permanently, the wastegate system is oriented in a way to enable fast catalyst heating in cold start conditions. The resulting application design is shown in Figure 9.

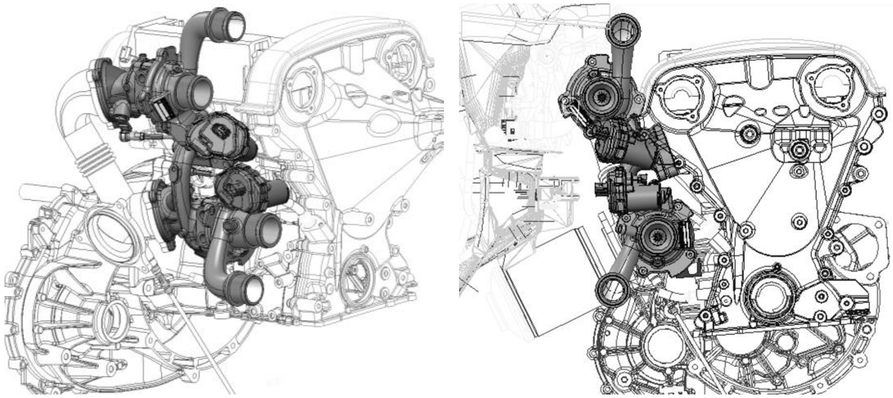

Figure 9: Packaging constraints for the twin boosting system

3 TURBOCHARGER AND ENGINE PERFORMANCE MEASUREMENT

3.1 TURBOCHARGER PERFORMANCE

To evaluate the performance the turbocharger is measured on the hot gas test stand. The measurement data is compared to a high performance 2.0 l GDI turbocharger with conventional range of massflow and pressure ratio. The GasON engine has got a large spread between the low end torque with low scavenging and therefore low air mass flow and the rated power operating point. Figure 10 shows the map comparison of the GasOn compressor and a compressor of a 2.0 l engine with increased diameter and massflow. The maximum massflow of the GasOn compressor is reduced by 60 % and the compressor ratio is increased up to 3.8. The isentropic compressor peak efficiency is on the same level compared to the compressor of the 2.0 l engine.

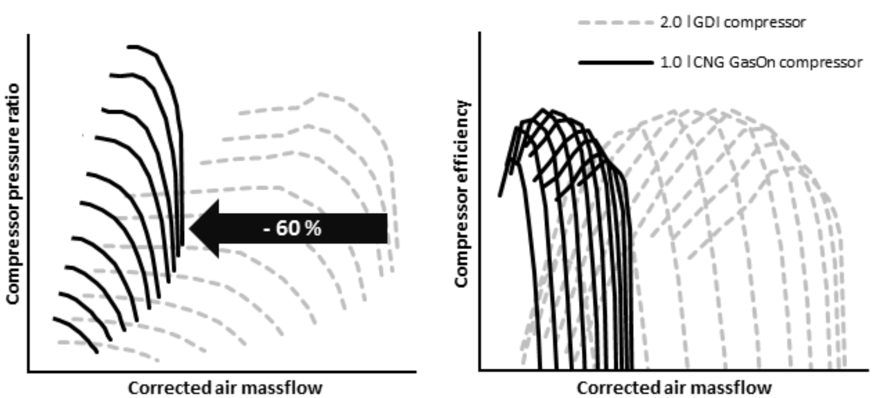

Figure 10: Compressor performance comparison

The turbine performance comparison shows a 65 % reduced massflow compared to a turbine of a 2.0 l GDI engine. The difference in the measured turbine efficiency consisting of the isentropic efficiency of the turbine and mechanical efficiency of the turbocharger is about 9 % points, Figure 11.
Due to the small turbine diameter, the required radial and axial clearances between turbine wheel and turbine volute have a notable adverse impact on the turbine efficiency. The percentage of friction loss in the bearing system of the turbocharger and the heat transfer in the long turbine inlet compared to the whole turbine power is higher compared to a conventional sized 2.0 l GDI turbocharger turbine. These three effects lead to a measured turbine efficiency that is lower compared to commonly sized turbine stages. Even so, for this extremely small mass flow the turbine efficiencies are high.

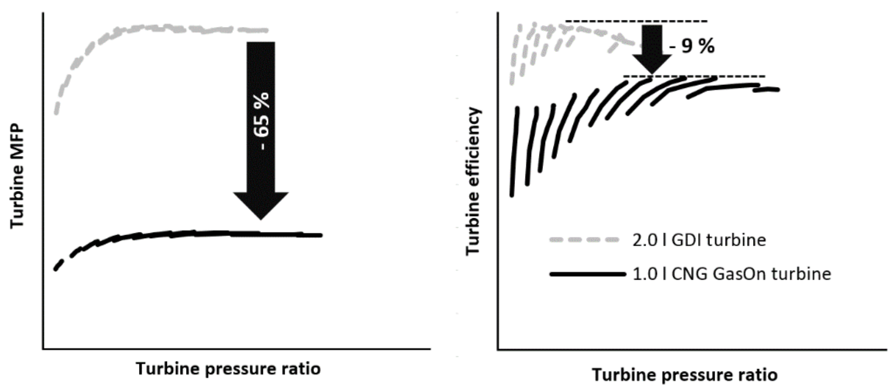

Figure 11: Turbine performance comparison

Overall, due to the high thermodynamic performance of the turbocharger all steady state engine targets and boundaries of the GasON engine are fulfilled or even exceeded. The engine measurement results are shown in section 3.2.

Although the design focus is on high efficiency for the GasON application, this turbocharger has got a low inertia. The shaft wheel assembly inertia of the turbocharger is shown in the turbocharger scatter band of FEV GmbH in Figure 12 [10]. The inertia is below the scatter band indicating benchmark level inertia due to the innovative and continuously improved RAAX turbine technology [8]. This low inertia further improves the fast transient response of the engine.

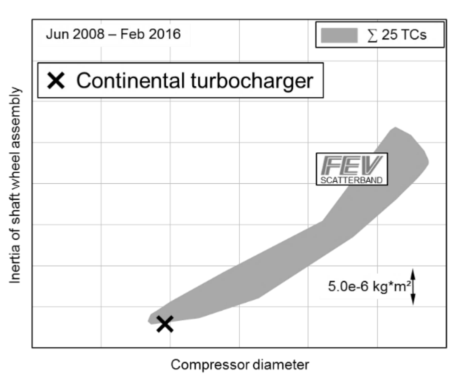

Figure 12: Inertia of shaft wheel assembly [10]

3.2 ENGINE PERFORMANCE

The engine measurements confirm favorable thermodynamic performance of the GasON engine with its parallel sequential twin boosting system compared to a single turbocharged CNG engine. The maximum BMEP level is increased by 30 % from 23 bar up to 30 bar. The LET is reached on engine speed of 1500 RM. The specific power at 6000 RPM is raised by 20 % from 97 kW/l to 117 kW/l.

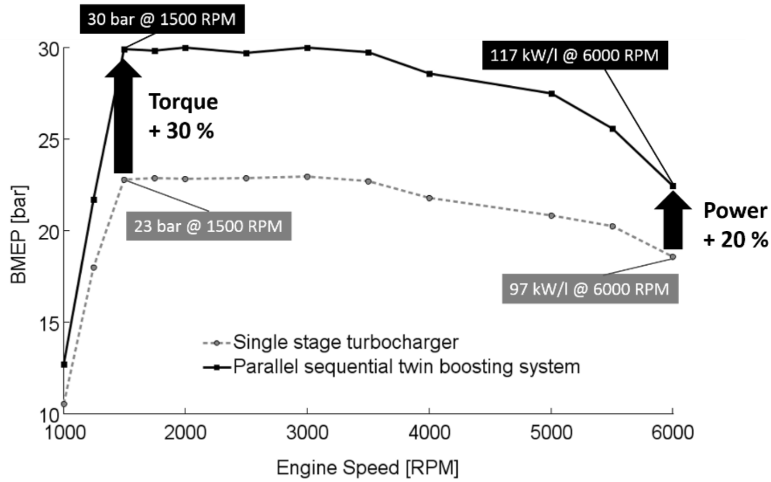

Figure 13: Full load performance comparison

The GasON engine is able to reach the LET and rated power performance shown in Figure 13 with the parallel sequential twin boosting system. Additional engine measurements are carried out to investigate the knocking behavior of the engine and to optimize the transient operation during the transition from mono to twin parallel turbocharger mode.

Due to the high torque level and the maximum scavenging rate of 1 % O_2 concentration upstream of the catalytic converter the boost pressure has to be increased to a level of 3 bar for the GasON engine, Figure 14.

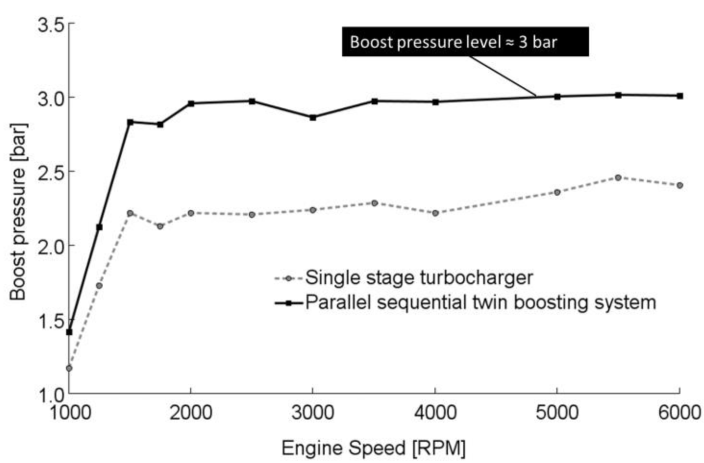

Figure 14: Boost pressure at engine full load

4 CONCLUSION

The GasOn concept is a 1.0 liter CNG DI engine equipped with a parallel sequential twin boosting system. A fully variable valve actuation system in combination with a compressor shut off valve and a compressor recirculation valve is used to switch between mono and twin turbocharger mode. Both turbochargers comprise electrical wastegates for boost pressure control. In order to achieve the same drivability compared to a 1.6 liter engine, challenging performance targets and matching boundaries are derived, leading to increased thermodynamic requirements in the design of the compressor and turbine stage while simultaneously meeting the packaging constraints.

With an application specific turbocharger design the low compressor and turbine capacity is adjusted to the matching targets and high efficiencies are achieved. Measurements on the hot gas test stand proof the high thermodynamic performance of the turbocharger.

The engine measurements show that the predicted low end torque operating point with 30 bar BMEP at 1500 RPM can be fulfilled while limiting the scavenging rate to a maximum of 1 % O_2 concentration upstream of the catalytic converter. The rated power target of 110 kW/l was reached and even exceeded. The parallel sequential twin boosting system delivers a boost pressure level up to 3 bar over the complete full load lug line.

Additional engine and vehicle tests are scheduled to demonstrate the CO_2 emission reduction compared to the best in class CNG vehicle.

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