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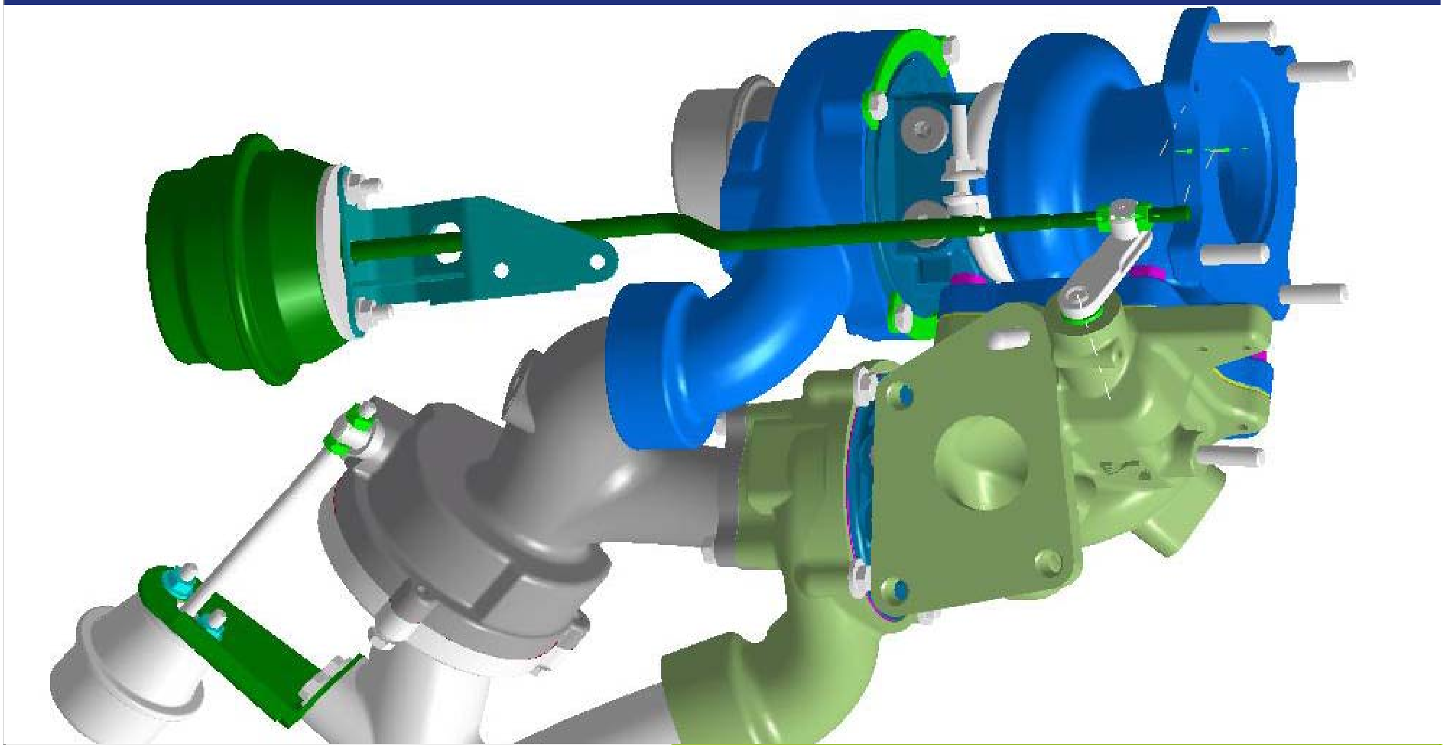
Regulated Two-Stage Turbocharging for gasoline Engines *(2010)*



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1 Motivation

On the way to fuel efficient, economic and high-performance engines, turbocharging used in Diesel engines has also become established with gasoline engines. The demands for gasoline engines are different though. In the past the utilization of turbocharging to enhance the performance was subject to compromises in the low end torque, because of the necessity for larger turbochargers. The customer demand for high-performance engines that are simultaneously responsive at start-up was accounted for by the ongoing development in turbocharging and other engine related components, however the limits became increasingly evident.

Figure 1.1 provides an overview of contemporary turbocharged gasoline engines with respect to specific power and specific torque at 1500 RPM. Two Diesel engines with VTG and regulated two-stage turbocharging are also shown.

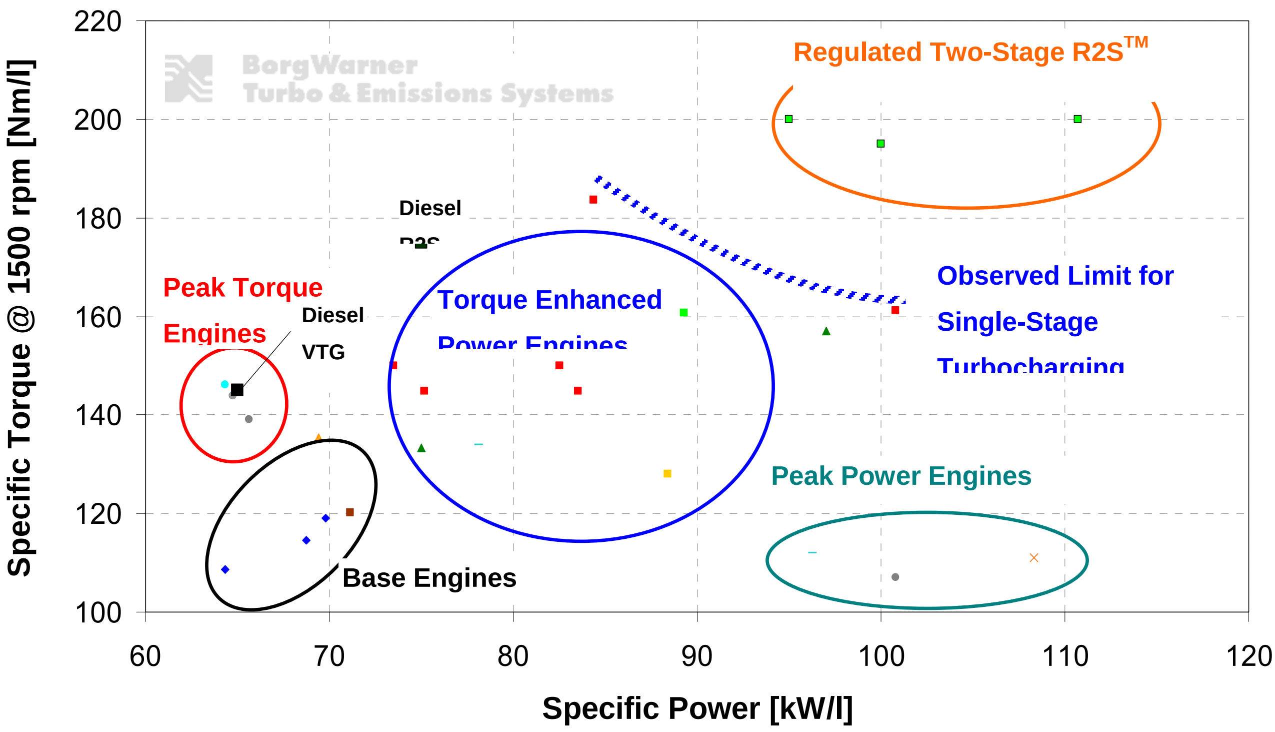

Figure 1.1: Trade-off on turbocharged gasoline engines between specific power and specific torque at 1500 RPM

From the figure a clustering of the engines into determined characteristic ranges becomes evident. In the smaller specific values range, preferably cost-effective engines ("Base Engines") are to be found that utilize technologies such as port fuel injection, simple mono-scroll turbines and only one or even none inlet cam phaser. In the higher specific engine power range, a group of high-power engines ("Peak Power Engines") can be found that sacrifice low end torque for high performance targets. The next group can be seen on the left of the image; it features direct injection and typically one cam phaser to achieve an increased low-end torque at low specific power ("Peak Torque Engines"). A modern Diesel engine with VTG, shown for the purpose of comparison, delivers almost equal performance values thus showing the potential of gasoline engines for higher specific powers.

The next group in the centre of the figure demonstrates an easing of the conflict of goals between specific low-end torques and specific powers; these engines are equipped with direct injection, cam phasers on the inlet and outlet valves as well as single-scroll turbines ("Torque Enhanced Power Engines"). By the combination of the stated technologies a "scavenging" charge cycle and thus a significant increase in torque at low engine speeds is possible. A further increase is possible and is design-dependent with 4 and 6 cylinder in-line engines using separated exhaust gas routing in the exhaust manifold and twin-scroll turbine housings. This technology enables an improvement in the preconditions for a "scavenging" charge cycle through avoiding of exhaust-gas pulsation interference of neighbouring cylinders and can significantly increase the low end torque [1]. The Diesel engine included for comparison with regulated two-stage turbocharging exhibits a specific low-end torque in the upper range of this group; the specific power is however limited to a value of approx. 75 kW/l in comparison to the gasoline engines shown. Further synergy potential by the combination of multi-scroll turbine housings with variable turbine geometry is possible and the subject of ongoing investigation [2].

Figure 1.1 shows that with the current engines with one-stage turbocharging the specific low-end torque as well as the specific power is limited to values of about 180 Nm/l (at 75 kW/l) up to 160 Nm/l (at 100 kW/l). Demands for a further increase in the low-end torque, resulting from further downsizing or an extension of the engine power range to be covered within an engine family, can be implemented with single-stage turbocharging only to a very limited extend.

The regulated two-stage turbocharging can solve the conflict described above. Concept studies on different gasoline engines turbocharged in this way have shown that low-end torques exceeding 200 Nm/l and simultaneous power densities of more than 110 kW/l can be delivered.

In the following the design, control strategy and the operational behaviour of this type of 2-stage turbocharging on a DI gasoline engine are described in detail.

2 Regulated two-stage turbocharging (R2STM) on a DI gasoline engine

2.1 Thermodynamic aspects of supercharged gasoline engines

In order to realize the specific torque and power objectives of more than 200 Nm/l and 100 kW/l as described in chapter 1 as well as compliance to future CO_2-emission limits with the assistance of the shift in the load points, gasoline engine turbocharging concepts with more than 20 bar break mean effective pressure (BMEP) are increasingly becoming the focus of automobile manufacturers. Single-stage charging systems reach their thermodynamic limits in such applications.

The turbine mass flow rate or the performance factor of a turbine in unregulated operation depends mainly on the engine speed n_M [3] (Equation 1):

$$\frac{\dot{m}_T \cdot \sqrt{T_3}}{p_3} = \frac{V_H \cdot i}{R_L} \cdot \left[1 + \frac{1}{L_{min} \cdot \lambda}\right] \cdot \lambda_a \cdot \frac{\sqrt{T_3}}{T_{2S}} \cdot \frac{p_{2S}}{p_3} \cdot n_M \quad \text{[Equation 1]}$$

At low engine speeds and consequently low turbine mass flow rates, the representation of an effective torque build-up of a small turbine with low absorption capacity is necessary, as in accordance with the first turbocharger main equation (Equation 2), a correspondingly higher pressure before turbine p_{3t} must be built up for the necessary pressure ratio π_V:

$$\pi_V = \frac{p_{2t}}{p_{1t}} = \left[1 + \frac{\dot{m}_T}{\dot{m}_V} \cdot \frac{c_{p,Abg}}{c_{p,L}} \cdot \frac{T_{3t}}{T_{1t}} \cdot \eta_{is,V} \cdot \eta_{is,T} \cdot \eta_m \cdot \left(1 - \left(\frac{p_4}{p_{3t}}\right)^{\frac{\kappa_{Abg}-1}{\kappa_{Abg}}}\right)\right]^{\frac{\kappa_L}{\kappa_L - 1}} \quad \text{[Equation 2]}$$

In principle, gasoline engine combustion helps providing the necessary turbine power due to stoichiometric combustion resulting in high exhaust temperatures and high turbine mass flow rate; however, the turbine pressure ratio $\pi_T = p_{3t} / p_4$ and consequently the turbine size remain the main influential parameters.

At rated power small volute housings with low A/R-ratios lead to undesirably high pressure levels before the turbine thus negatively affecting the gas exchange of the engine. High residual gas content, retarded ignition angle and the requirement for enrichment are the result and will not only lead to an increase in fuel consumption, but to a limitation of the power, should the enrichment be above a significant level. This power limitation is dependent on the width of the compressor map, which is a limiting factor for single-stage charging systems due to the high necessary air mass flow rate for the current power objectives. This factor is exacerbated by the existing high spread in delivery rate with gasoline engines, requiring the use of larger turbines for the rated power range. The increasing power density currently observed and the associated increase in turbine size is at the expense of transient engine operating response with single-stage turbocharging systems [4].

The transient response depends, according to the principle of conversion of angular momentum for the turbocharger (Equation 3) on the turbocharger speed n_{ATL}, but particularly on the available surplus exhaust gas enthalpy content for the turbine or the system, as well as the moment of inertia of the rotating parts Θ_{ATL}:

$$\frac{dn_{ATL}}{dt} = \frac{1}{4\pi^2 \cdot \Theta_{ATL} \cdot n_{ATL}} \cdot \left(P_T - P_V - P_R\right) \quad \text{[Equation 3]}$$

Single-stage turbochargers designed for high specific powers have a noticeable delay in transient operation due to their relatively large polar moment of inertia and the large turbine absorption capacity. This is disadvantageous and undesired because of the desirable high acceleration response from low speeds. The problem described becomes particularly evident on heavy vehicles as well as in extreme conditions such as high altitudes or pulling out on steep gradients.

Regulated two-stage turbocharging (R2S™) does not only offer the solution for this conflict of goals but also utilizes the benefits of the dedicated turbocharger size for a certain operating range. Because of the small design of the high-pressure stage and the compressors connected in series, R2S systems can realize the highest charging pressures at low engine speeds, so that the objective of 200 Nm/l from 1500 RPM can be achieved. As a result of the low polar moment of inertia and its high bank up ability, the high-pressure stage enables a significant increase of the low-end torque and the dynamic response.

The design of the low-pressure stage is undertaken for the respective target power (> 100 kW/l) and the additional reserve requirements. Compared to the Diesel engine, the higher variation in delivery rate of the gasoline engine can be more easily considered, and the low-pressure compressor at the rated power point can be operated at good levels of efficiency. The high flow coefficient and low bank up behavior of the low-pressure turbine supports scavenging of the cylinders, which leads to lower residual gas content and a good fuel economy levels also at higher loads.

A two-stage regulated turbocharger assembly as a system component of the engine is dependent on an environment, which can realize the implementation of the generated high charging pressures in break mean effective pressure (BMEP), and that eases, if possible, the problem areas of supercharged gasoline engines. The knocking problem requires in addition to a reduced compression ratio and direct injection special measures that reduce the residual gas content in the cylinder to a minimum. 4 cylinder concepts with conventional cam timing configuration and 4-in-1 type exhaust manifolds have a system-related disadvantage. The gas exchange is affected by the outlet pressure pulse of the cylinder next in the firing order. The high pressure level before the high-pressure turbine causes high residual gas content in the low-end torque range, retarded ignition point and late BMF (burnt mass fraction) so that the potential of the higher charging pressure cannot fully be exploited. As a direct consequence, 3 cylinder concepts with two-stage regulated charging systems are viewed as a useful alternative [5] to the 4 cylinder engine. Here, active scavenging strategies with the aid of cam phasers at the inlet and outlet valves can be used to get the associated positive effects on low residual gas content in the cylinder for good low-end torque. But even on 4 cylinder engines, variable valve timing can significantly reduce the residual gas content and makes a good combination with both single and 2-stage turbochargers [1].

2.2 Single-stage turbocharging – design and limitations

The design and adaptation of a one-stage turbocharger is frequently undertaken with the demands that the target torque is reached as quickly as possible at a defined rated power, see Fig. 1.1. Transferred to the compressor map on the one hand means a very steep rate at the surge line that limits the compressor performance characteristic to low mass flow rates. On the other hand, the choke line as the limitation for high mass flow rates should be far enough away from the surge line. With the required rated power, the maximum throughput of the compressor is defined and according for this size the design-dependent available characteristic map width. With the use of adequate valve overlaps and an existing positive pressure drop between the inlet and outlet side, using a “scavenging” charge cycle, the engine operating curve can be shifted, particularly for low engine speeds to higher mass flow rates, and thus the torque can be significantly increased [1]. By corresponding adaptation of the turbine, the effective torque range can be shifted to lower engine speeds, and thus the achievable air mass flow variation between maximum torque and rated power can be increased a little.

On the turbine side, with the series industrialisation of variable turbine geometry (VTG) [6], the prerequisite for a solution suitable for gasoline engines for increasing the mass flow rate variation is provided. Studies indicate that this technology facilitates considerable increases in the startup torque [1].

On the compressor side, measures for extending the characteristic map width as well as for increasing the achievable pressure ratios were examined, where the focus of the measures concentrated on enhancing the pressure ratios and should be discussed in more detail.

In order to achieve the necessary charging pressures for supercharged gasoline engines with one-stage charging, modified technology compared to today's state-of-the-art is necessary. An increase of the charging pressure can generally be achieved in three ways [7]:

- Increase of the circumferential speed by increasing the turbocharger speed
- Increasing the impact of the rotor outlet flow through a larger blade exit angle
- Generation of swirl on the rotor inlet by using an inlet guide or outlet guide vanes in the diffusor

An increase of the boost pressure by increasing the rotor speed causes greater mechanical stress because of the higher centrifugal force. In particular, the increased stresses on the compressor impeller can act unfavorably on the lifetime of the compressor impellers, based on cast aluminum alloys used up to this point. This could be partly counteracted by series introduction of milled compressor impellers, which can bear higher loads because of the homogeneous microstructure of the raw parts used (forged circular blanks).

A further increase in the service life is possible by the use of higher-quality materials, such as titanium alloys, but this enhancement is subject to a more complex manufacturing process and higher costs. Increased material densities, such as those with Titanium alloys, lead to increased moments of inertia with an impaired transient response, should the existing geometry be retained. Furthermore, an increase of the pressure ratio can cause outlet temperatures that exceed the admissible material temperatures of the materials currently used in the compressor and the downstream components and require the use of more suitable materials.

Measures with a modified blade exit angle, generation of swirl in the rotor inlet as well as outlet guide vanes in the diffusor are described and have been evaluated in [7] and should therefore not be studied further.

To conclude, after careful considerations of the above mentioned modifications the one-stage compressors do not appear to be suitable for providing large air mass variations in combination with high pressure ratios that are required by highly downsized turbocharges gasoline engines.

2.3 Regulated two-stage turbocharging (R2STM) – design and regulation

The limitations of single-stage turbocharging shown in the previous chapter lead to deliberations concerning the introduction of regulated two-stage turbocharging, which can significantly increase the charging pressure over the entire engine speed range. An advantage with this charging process is that in the two-stage regulated turbocharging range, the respective stage loading compared to one-stage turbocharging can be reduced.

Hereby, a second charging stage designed for a lower flow rate ensures the required increase in charging pressure with low engine speeds, hereinafter referred to as a high-pressure turbocharger. This is situated upstream (compressor side) or downstream (turbine side) of the main charging stage, so that regulated two-stage turbocharging can be employed at low flow rates. With this arrangement, the startup torque is represented decisively by the high-pressure stage, which in addition to the lower space requirement also offers a considerably reduced moment of inertia for the rotating parts and thus provides the best preconditions for transient states [7]. So for the low-pressure turbocharger a compressor with a higher maximum flow rate can be used.

The associated design reserves can be used either for representation of a higher power or for compliance to extended application reserves for critical applications (operation at high-altitude, see Chapter 3.4). In figure 2.3.1, the set-up of the turbocharging system and its actuators can be seen.

The control of the charging group is operating point-dependent controlled using the turbine bypass, waste-gate and compressor bypass control elements. The turbine bypass determines the share of the required total turbine power for the turbocharger, which the high-pressure turbine generates. At the lowest speed and at full load, the turbine bypass is practically closed, so that the exhaust gas passes both turbocharger stages and the air is compressed in two-stages on the compressor side. As the engine speed increases, the low-pressure compressor can increasingly contribute to the required total pressure ratio, until ultimately the high-pressure turbocharger is deactivated by completely opening the turbine bypass on the exhaust side. At this point the compressor bypass is opened, and the high pressure compressor is bypassed to reduce throttling losses.

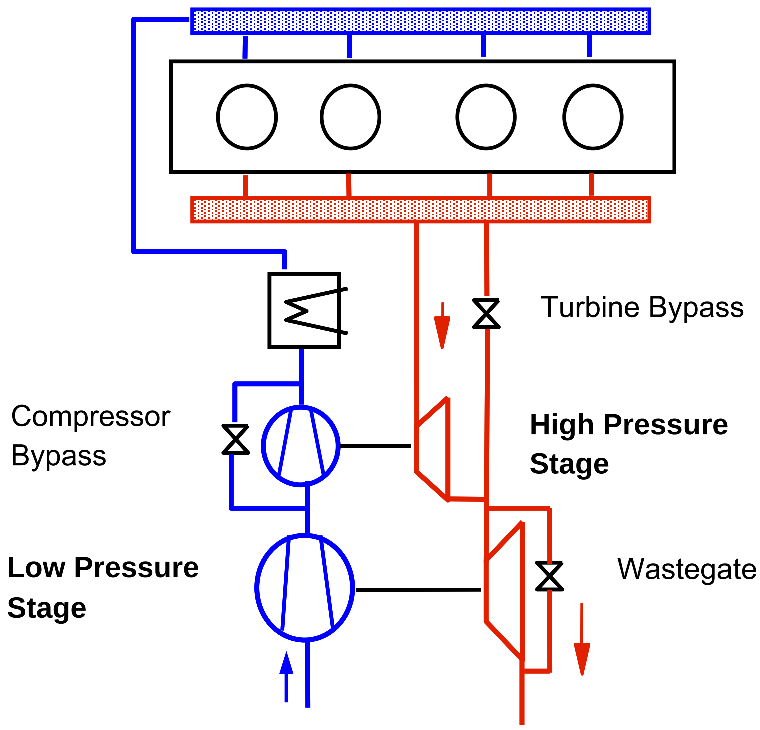

Figure 2.3.1: Diagram of two-stage regulated charging

A possible actuator control in the operating characteristic of a gasoline engine is represented schematically in the following figure 2.3.2.

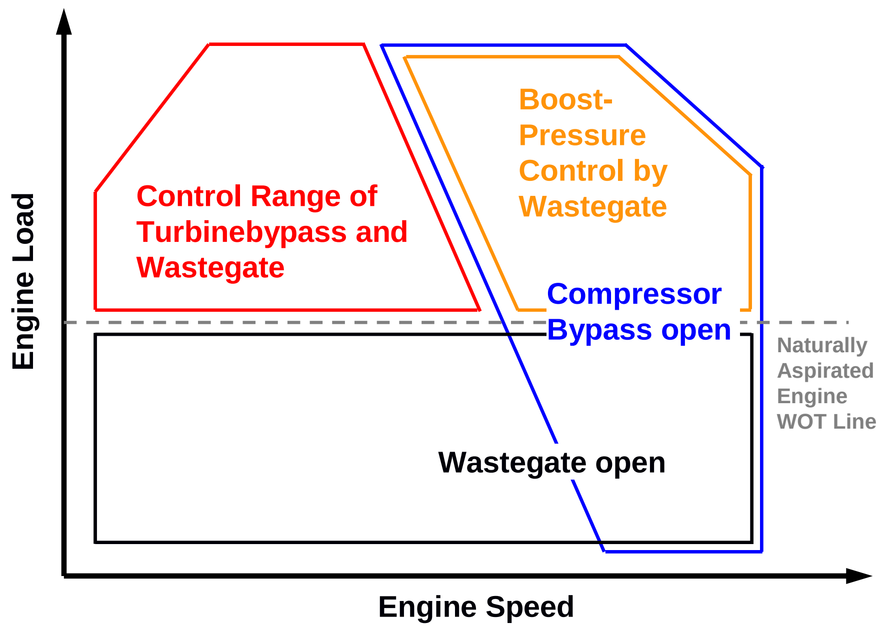

Figure 2.3.2: Control strategy of a regulated two-stage turbocharger system in the map of a gasoline engine

The control strategy of the actuators aims to keep the charge-cycle losses of the engine to a minimum; the results shown in chapter 3 were measured according to this strategy. From the illustration it is evident that the high-pressure stage is not required from medium speeds on and only the low-pressure stage provides the required boost pressure. In order to reduce losses, the waste-gate is opened below the naturally aspirated full load line, so that boost pressure is generated above this line only.

The interaction of the actuators is mainly dependent on the selection or pairing of the charging components and will be examined more closely in the following chapter.

2.4 Design of regulated two-stage turbocharging

During the course of these investigations, the design of a two-stage regulated turbocharging system for a direct injection gasoline engine for delivery of a specific torque of 200 Nm/l from 1500 RPM and a specific engine power of 100 kW/l was undertaken. A further demand was the consideration of high-altitude reserves of 1000 m, whereby the torque and power objectives equal to those at sea level were to be retained.

In the first step, the low-pressure compressor was selected for the required power, taking into consideration that the compressor at its rated power point is sufficiently far enough away from the choke line and that it can be operated in the good isentropic compressor efficiency ($\eta_{is,V} \approx$ 75%) range (see Fig. 2.4.1). The map of the selected compressor shows sufficient potential to increase the circumferential speed and accordingly the charging pressure, in order to deliver the required high-level operation with a reduction in power.

The BMEP plateau of 25 bar requires a high overall compressor pressure ratio of $\pi_{V,total} > 2.7$, which can be served solely in a speed range of n = 3000 – 4500 RPM by the low-pressure compressor, just like the required pressure ratio in the rated power range. At speeds < 3000 RPM and the associated low flow rates, the low-pressure compressor alone cannot maintain the required pressure ratio because of the limitation by the surge line. In order to hold the BMEP level or to enable the necessary torque curve, the high-pressure compressor must increasingly deliver a share of the required total pressure ratio, whereby the share of the low-pressure compressor must be considered with a certain safety margin to the statically on the gas stand determined surge line.

Furthermore, Fig. 2.4.1 shows that the full load operating points in the compressor map are within the area of moderate efficiencies. This is due to the fact that the compressor design is primarily intended for one-stage high-pressure usage and not for a two-stage system. The position of the isolines of constant compressor efficiency can be optimized by adaptation of the compressor geometry (outlet angle, number of blades, etc.), as well as measures on the diffusor and casing for the two-stage application examined here (see chapter 2.2).

The response of the two-stage turbocharging system with highly transient acceleration processes is also mainly influenced by the choice of compressor and makes the selection of a suitable control structure necessary. The low-pressure compressor must be able to support the dynamic acceleration of the high-pressure compressor at the switchover point without itself having to surge or cause a dip in the boost pressure. Good controllability requires a certain amount of overlap of the compressor map as well as coordination of the dynamic response of the compressors to one another. For a good transient response it is not simply sufficient to only optimize the high-pressure turbocharger for acceleration. Configuration of the regulated two-stage turbocharger just for stationary aspects would leave a considerable amount of potential unused.

The selection of the turbine size with one-stage turbocharger systems is characterized by the conflict of goals between acceleration or low-end torque response as well as acceptable pressures from the turbine and the associated consequences for gasoline engine charge cycles, see chapter 2.1. This conflict is practically solved using two-stage regulated turbocharging. On these systems, the focus of the turbine side considerations is concentrated mainly on the selection of suitable turbine pairs. On the one hand, the respective operating range should be optimally covered by the corresponding stage, and on the other hand, a favorable switchover point should be guaranteed in order to enable both good stationary and transient operating conditions. A small low-pressure turbine shifts the switchover point in the direction of low engine speeds, allows less scope for corresponding high-altitude operation or leads to bad fuel consumption values in the rated power range. Inversely, by using large low-pressure turbines that allow the high-pressure turbine to remain operational for longer at full load, so that unfavorable pressure ratios before the high-pressure turbine can result in higher levels of residual gas content as well as poorer fuel consumption values in the mid engine speed range. Considering these aspects, for this design a switchover point in the range from n = 2600 – 2800 RPM has been selected.

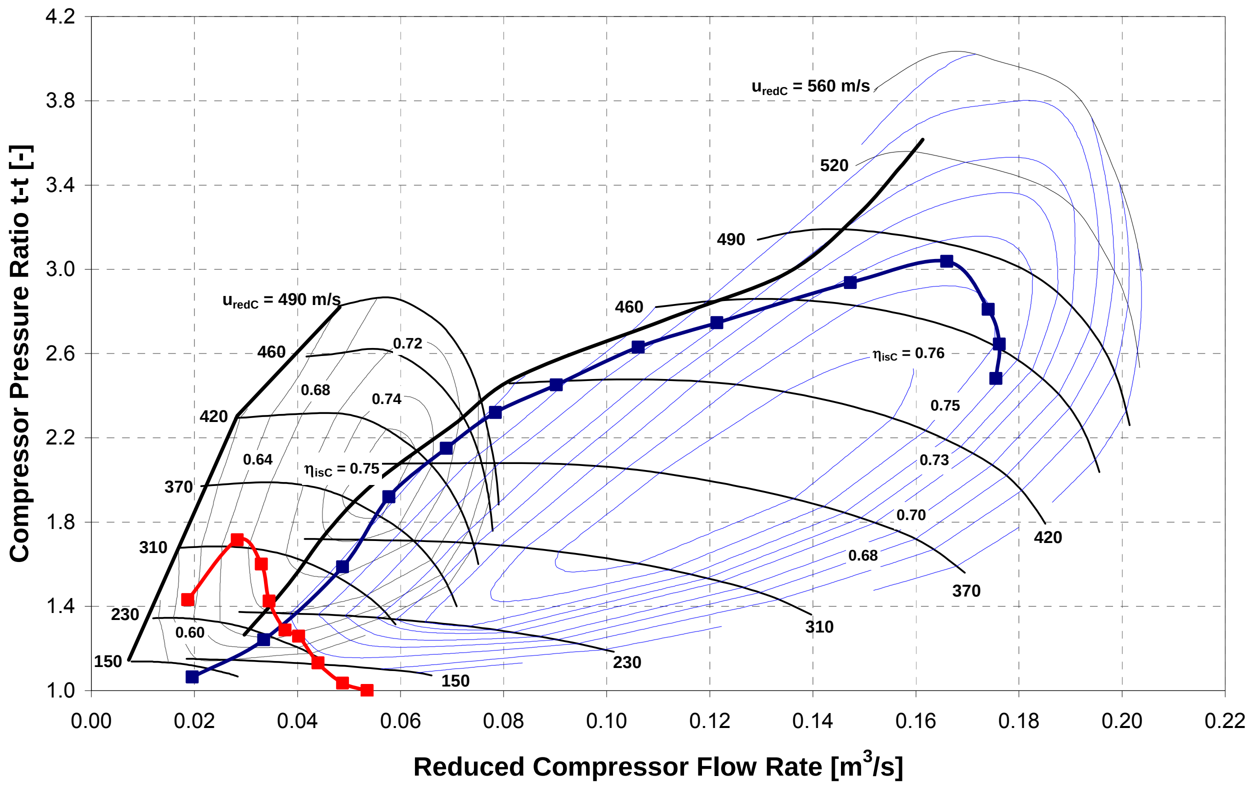

Figure 2.4.1: Matching of the regulated two-stage turbocharging system in the maps of the high- and low-pressure compressor

2.5 The regulated two-stage turbocharging in high-altitude and hot conditions

The cylinder charge and therefore the power of normally aspirated engines decline because of the lower air density at high altitudes. The same applies for the operation in warm ambient temperatures. One big advantage of turbocharged engines is that this reduction in power can be compensated for to a large degree, as the design reserves of the compressor are in a position to provide the necessary higher pressure ratio required to deliver nominal charging pressure. The turbocharger will operate at increased speed, and the operation point will shift in the compressor map in the direction of larger pressure ratios and volume flows. Up to which height or temperature the power of the engine can be kept constant before the turbocharger reaches its speed limit, is a parameter that is part of the matching strategy of the turbocharger to the engine. Different specifications exist depending on manufacturer and vehicle type.

A matching strategy frequently used is to keep the power constant up to a height of 1000 m and an ambient temperature of 40 °C. The full load curve for such extreme operation in comparison to the standard conditions for the engine under investigation can be seen in figure 2.5.1 for a specific power of 85 kW/l.

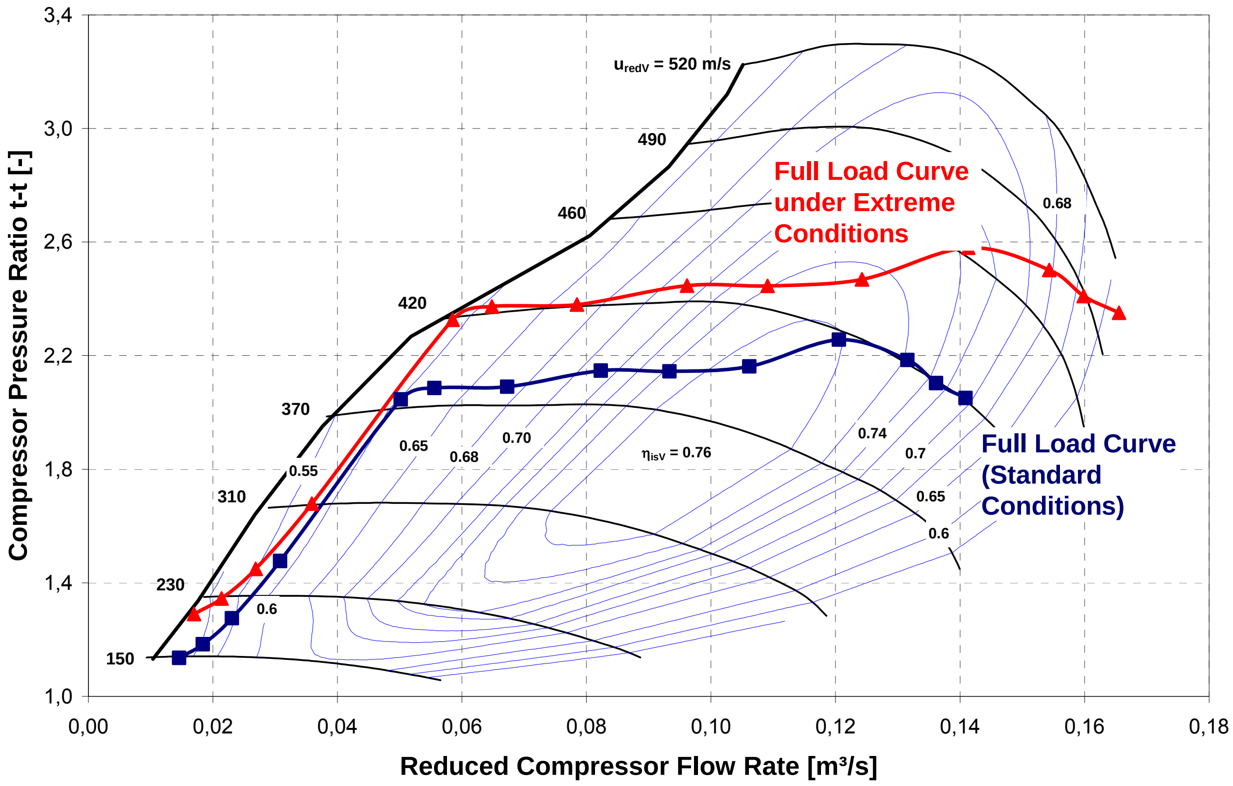

Figure 2.5.1: Full load curve of a single-stage turbocharger in the compressor map, standard conditions and extreme conditions

It is clearly evident that the turbocharger reaches its operating limits under extreme conditions. On the one hand the surge limit is reached, on the other hand the rated speed for the flow rate limit is achieved. The last operating point is also on the speed limit, which presents an operating risk. The calculated application or safety reserves in the basis design are no longer sufficient under these operating conditions. A design of this type would not be practical for series applications without a clearly power limitation in extreme conditions. A larger charger would remedy the situation and would include all the known disadvantages of a comparable delayed response and operation within the area of moderate efficiencies as well suffering to provide enough boost pressure at low engine speeds. A further result of the operation under extreme conditions can be seen in the following figure 2.5.2.

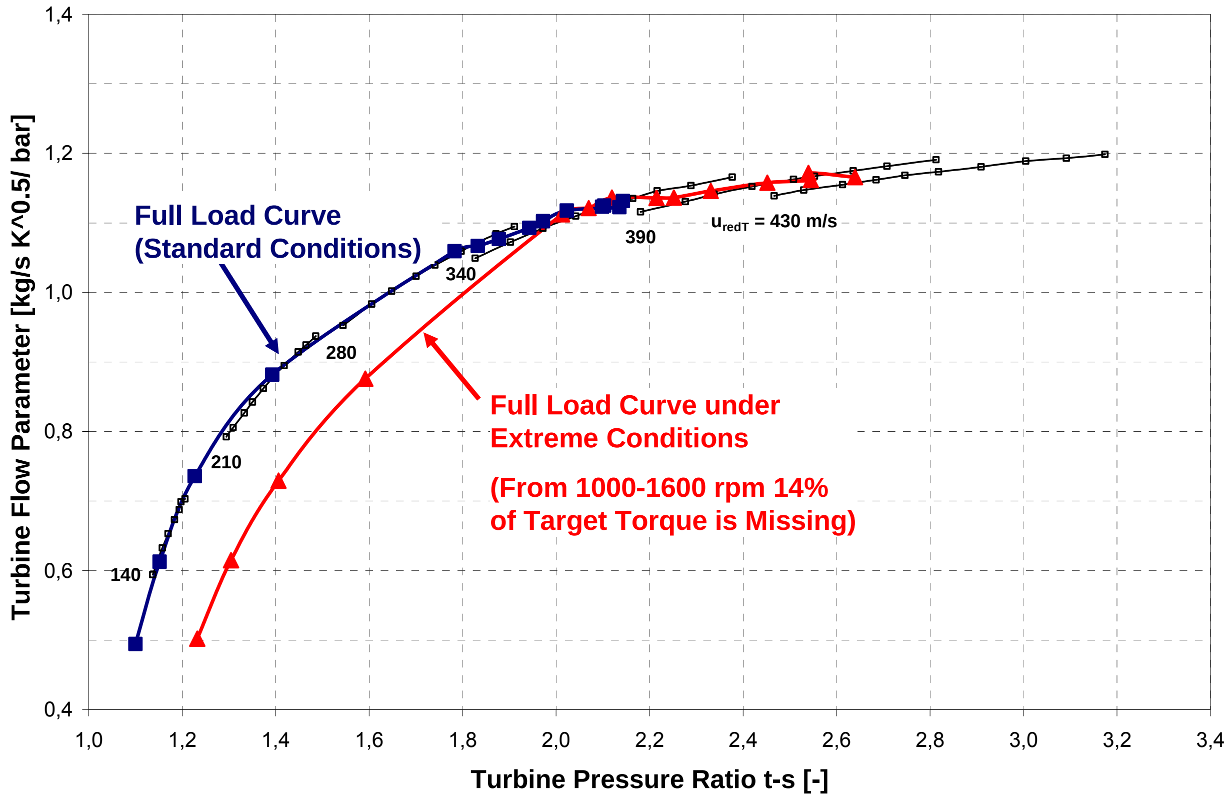

Figure 2.5.2: Full load curve of a single-stage turbocharger in the turbine map, standard conditions and extreme conditions

Here both the engine characteristics for standard and extreme design conditions in the turbine map are shown. It is evident that the flow rate curve of the selected turbine under extreme conditions in the speed range from 1000 – 1600 RPM cannot be achieved, i.e. the turbine appears to be “too large” and the required turbine power cannot be achieved. This leads to a reduction of the attainable engine torque of about 14 % in the described speed range, where the additional problem of rated power limitation is associated with a tangibly lower startup torque.

In comparison, the regulated two-stage R2S system performs decisively better, as can be seen in figure 2.5.3. As the operating range of the high-pressure stage (characteristic on the left of the graphic) is not fully exploited in standard operation, high-altitude operation or an increased ambient temperature only has a minor effect on the operation of the turbocharger. This means that the steady state response of the engine does not measurably worsen.

Also in the characteristic of the low-pressure stage, the extreme operation conditions are shown to be uncritical. The operating points at higher engine speed are shifted to higher turbocharger speeds analog to a single-stage design. The comparably larger turbocharger, however, allows considerably higher flow rates, so that even when operating under so-called extreme operating conditions, there are sufficient application and safety reserves available, and the operating points are still within the range of the operating limits.

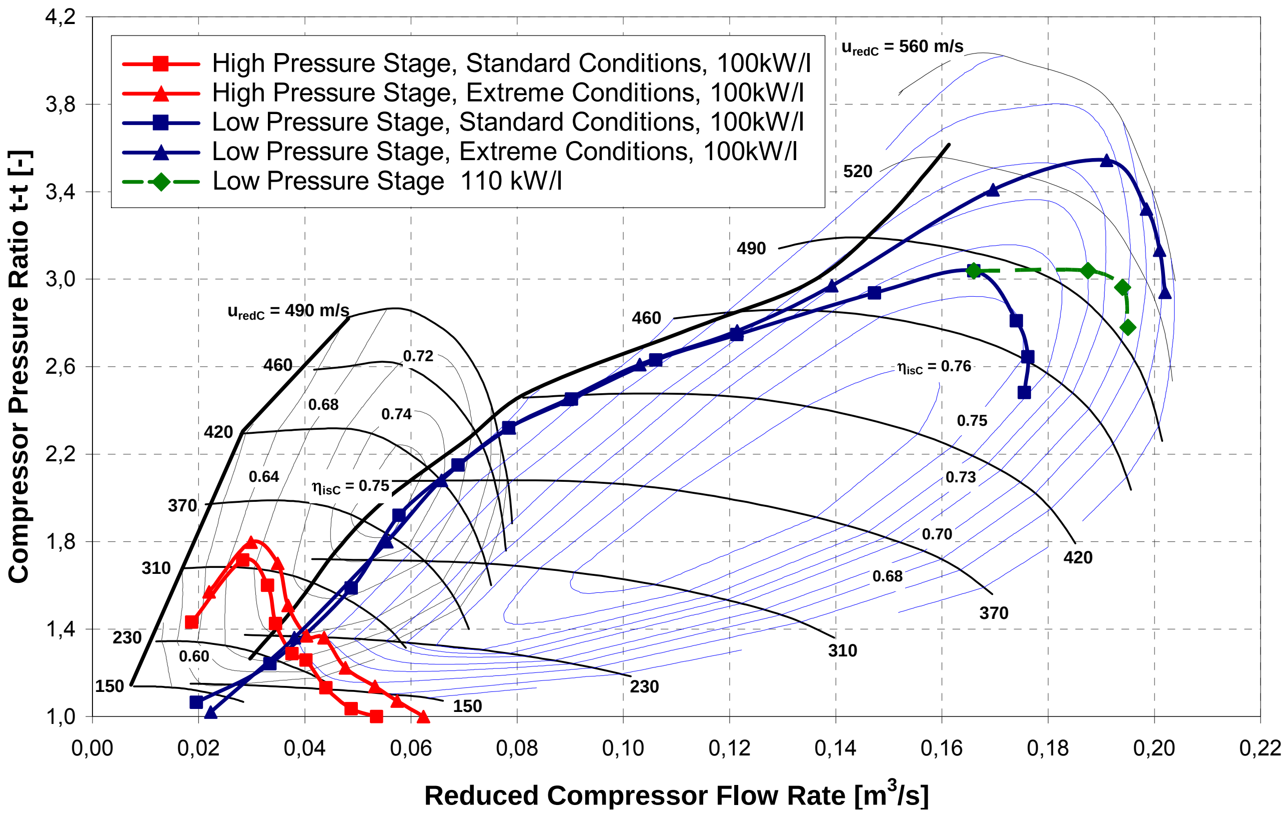

Figure 2.5.3: Full load curve for a R2S system in the compressor map, standard conditions, extreme conditions and higher rated power

The design shown in figure 2.5.3 is well suited for use in practice and the rated power of the engine would be available at an ambient temperature of 40 °C up to a height of 1000 m. With regard to fuel consumption, a moderate rise can occur, as the extension of the regulated two-stage operation leads to higher engine speeds and corresponding higher afflux of the smaller sized high-pressure turbine.

Should the design reserves not be used for high-altitude and hot conditions, they can be used alternatively to deliver a higher engine power. The additional characteristic curve (dotted line) in figure 2.5.3 indicates the potential for a matching to 110 kW/l for a high power variant.

2.6 Design and function

The 2-stage turbocharging system can be designed under different priorities. In this case the emphasis was put on a compact design that can be easily integrated in the typical engine compartment of a B-Class vehicle. This compact design incorporates the arrangement of the high-pressure turbocharger directly on the connection flange of the exhaust manifold with the downstream low-pressure turbocharger. In figure 2.6.1, an overall view of the turbocharger with the schematic representation of the gas ducting as well as the control elements is shown.

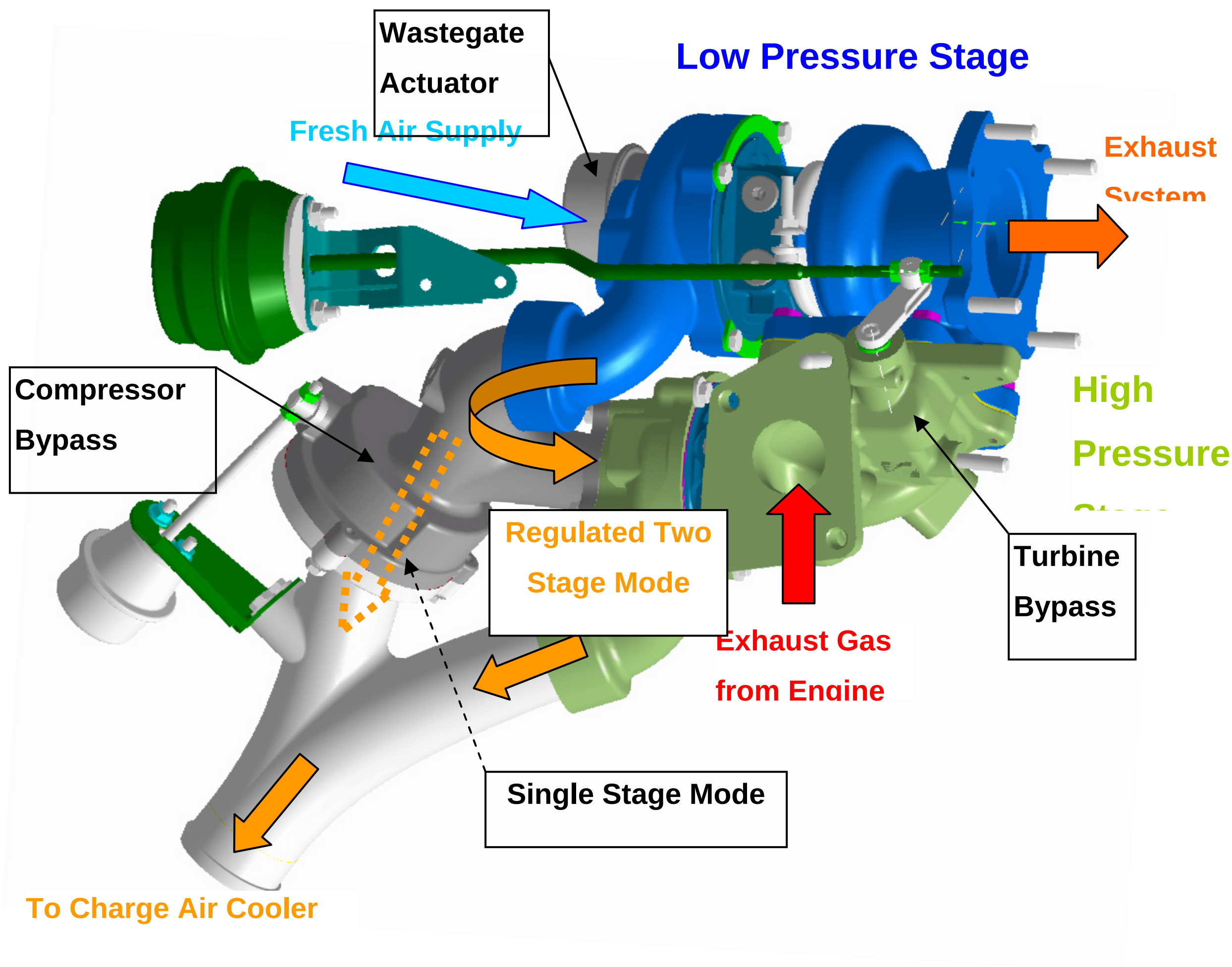

Figure 2.6.1: Overall view of the regulated two-stage turbocharger system

Using this figure, the gas ducting with regulated two-stage turbocharging is indicated, i.e. the turbine and compressor bypass as well as the waste-gate are closed, and the operating medium is compressed or expanded in two stages. Because of the dynamically active gas forces, the actuator of the turbine bypass is dimensioned to be larger than the other two actuators. The optimised design and dimensioning of the actuator guarantee good controllability of the turbine power in order to also safely enable transient acceleration conditions. With respect to the design and the kinematics of the turbine bypass, the knowledge gained in the Diesel area was used to assure the necessary functions. If the high-pressure stage is not required, the turbine and compressor bypass are opened, and the low-pressure stage takes over the charging pressure generation.

On the compressor end, an actively switched compressor bypass was used for the investigation presented here to examine further degrees of freedom. Alternatively, a self-actuating compressor bypass used as standard in the Diesel area can be used. This enables a simplification in the design of the charging air circuit with practically identical functional properties and leads to easier package conditions. The following figure 2.6.2 shows a different view of the turbocharger.

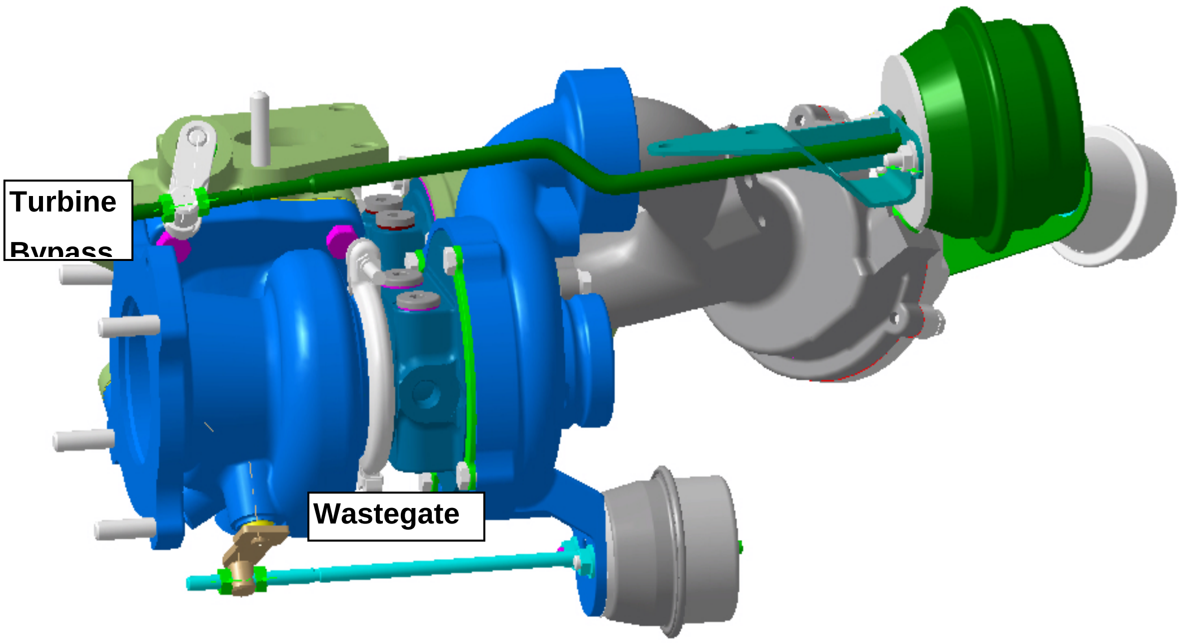

Figure 2.6.2: Regulated two-stage turbocharger system, view of the low pressure stage

In this figure, the compact construction of the turbocharger assembly can be seen. Due to the arrangement of both turbochargers above one another, only a slight additional amount of fitting depth compared to conventional turbochargers is required, and it is thus suitable for longitudinal and transverse engine installation.

2.7 System optimization for regulated two-stage operation

In chapter 2.4 the influencing parameters for the adaptation of an R2S system to an engine have already been discussed, and the position of the full load curve of the engine in the compressor performance characteristics has been shown. It becomes clear that a compressor designed and optimized for one-stage operation cannot optimally meet the demands when used in a R2S system. As the high-pressure turbocharger only contributes at medium to high loads and up to moderate engine speeds to the charging pressure generation, it is only used in the lower left area of the compressor map under steady state conditions, that is to say with low pressure ratios and flow rates. It is obvious that to optimize the high-pressure compressor, the area of maximum efficiency is to be shifted to lower pressure ratios and the efficiency in the lower characteristic map region is to be increased in total, as can be seen in the left characteristic in figure 2.7.2.

Characteristic for the operation of the low-pressure stage in the R2S system is that a wide variation in the pressure ratio is required. As the low-pressure stage has to provide the charging pressure for the target torque up until rated output without the support of the high-pressure stage from moderate speeds, the operating points are shifted to higher pressure ratios in comparison to the one-stage design. In consequence, the characteristic for a R2S application should enable high pressure ratios and exhibit high degrees of efficiency in the upper characteristic range. This is illustrated in the right section of figure 2.7.1.

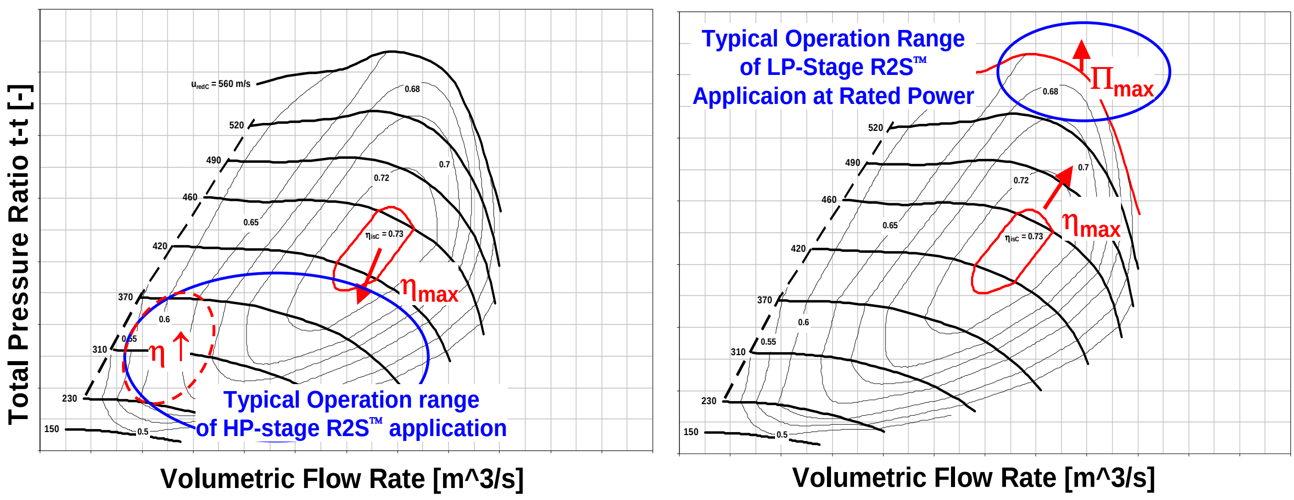

Figure 2.7.1: Compressor optimisation for the R2S system

The characteristic of a HP-compressor optimized to these demands in comparison to a compressor for one-stage applications is shown in figure 2.7.2.

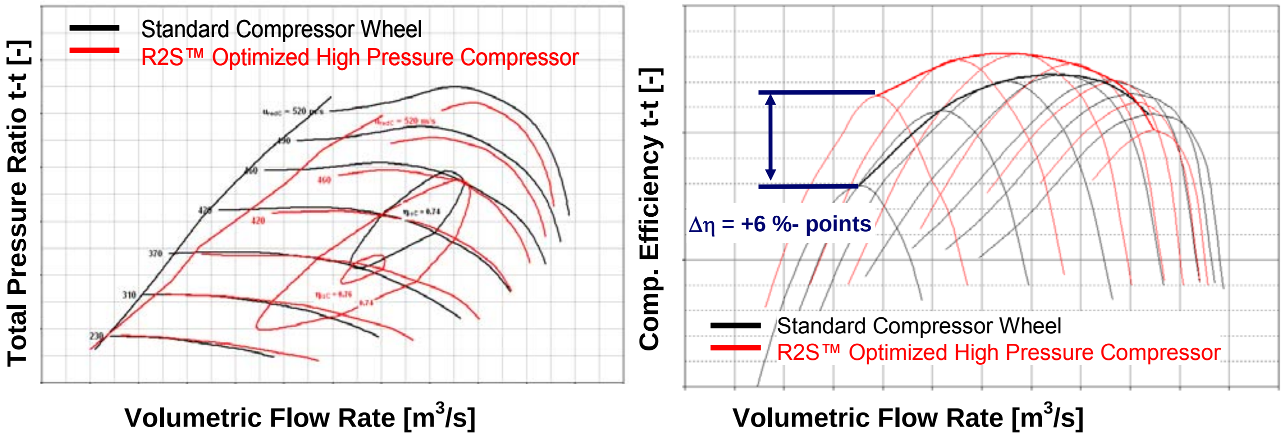

Figure 2.7.2: Map of a compressor optimized for operation as a high-pressure stage in the R2S system

It is obvious the surge limit is shifted moderately towards higher flow rates and the map is a little narrower, what is not considered as a drawback for this application. However, the efficiency is increased and retained at a high level in a wide range, especially in the area of small pressure ratios which is most important for the high pressure turbo. This will be immediately recognized in a better transient behaviour at low engine speeds, is beneficial for low pumping losses and therefore leads to reduced fuel consumption.

3 Test engine and measurement results

3.1 Test engine

The tests were performed on a modified gasoline engine with direct injection and camshaft phase adjusters on the inlet and outlet sides. Table 1 gives an overview of the most important engine characteristics.

Engine Type	-	Inline-4-cylinder
Firing Order	-	1-3-4-2
Combustion System	-	Gasoline Direct Injection
Valvetrain	-	DOHC, Cam Phaser on Intake- and Outlet Camshaft
Spec. Power	kW/l @ min^{-1}	100 @ 5500
max. Mean Effective Pressure BMEP	bar @ min^{-1}	25 @ 1500-4500
Turbocharging System	-	Low-Pressure Stage: K04 High-Pressure Stage: KP35

Table 1: Technical data of the test engine

The adaptation of the engine-specific setting parameters on the regulated two-stage turbocharging occurs via access to the required setting variables of the control device. The overall system of engine and turbocharger is operated under defined adjusting conditions in order to guarantee reproducible results: The following test boundary conditions were defined:

- Break mean effective pressure (BMEP): Target is 25 bar, to be achieved as early as possible
- Air ratio: $\lambda = 1$ (as far as possible)
- Maximum exhaust gas temperature: T_3 = 950 °C (enrichment if required)
- Spark timing: For efficiency-optimum for maximum BMEP or knock limited
- Minimum air ratio: $\lambda = 0.75$ (power reduction if necessary)
- In the case valve timing variations: control of the air ratio by lambda sensor upstream of catalytic converter
- Latest 50% BMF (burnt mass fraction) point: 35°KW n.OT

3.2 The regulated two-stage turbocharging used at full load

In a first step, a comparison is made between a single-stage and regulated two-stage turbocharger set-up. Essential components of the engine (compression ratio, valve timing and duration, exhaust manifold, exhaust system) are identical in order to facilitate a direct comparison of the turbocharging system. The cam phasers have been set for a minimum valve overlap. A comparison with the same power target values was not performed for the reasons specified in chapters 2 and 3.3. Selected results are represented in figure 3.2.1.

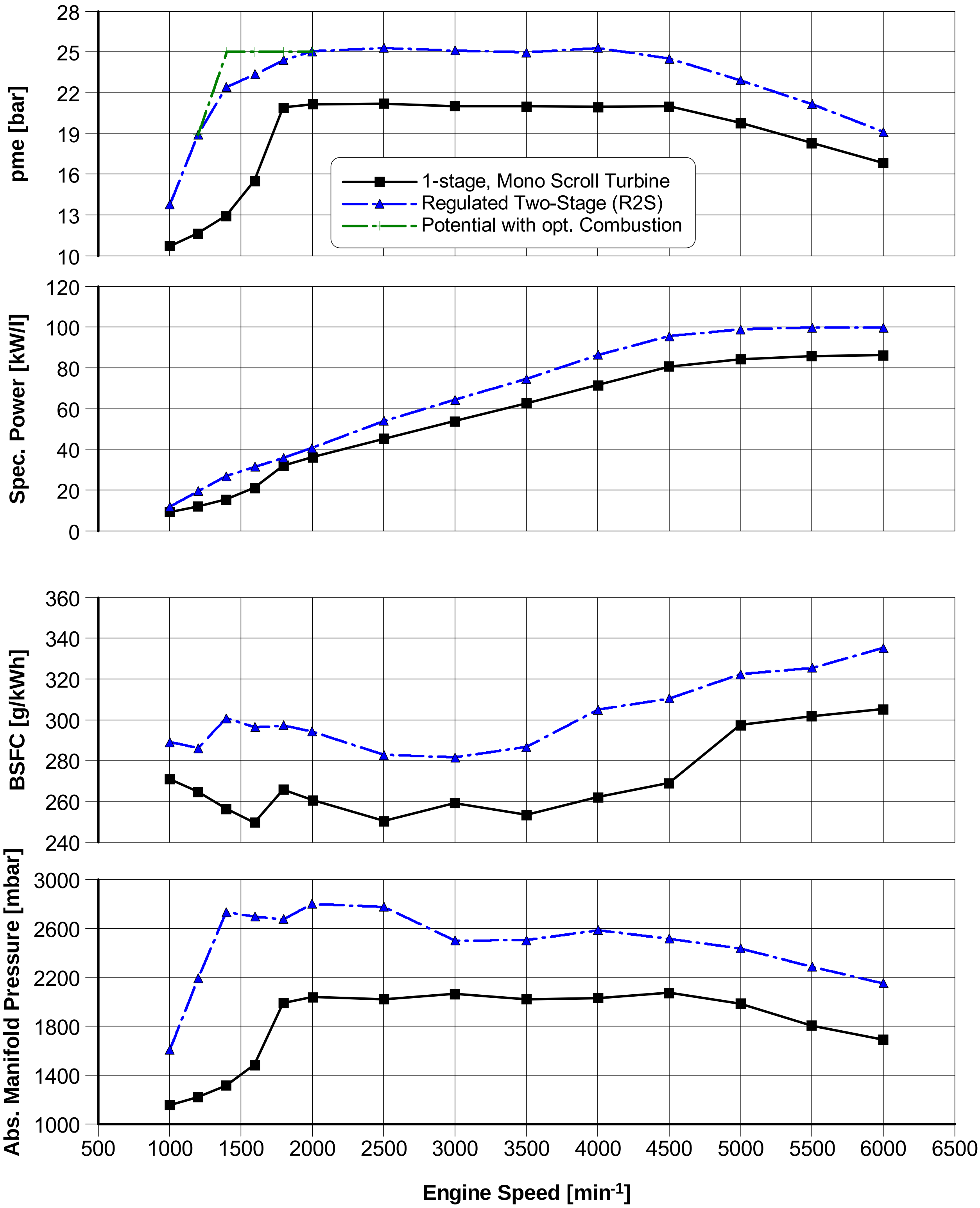

Figure 3.2.1: Comparison between one-stage and regulated two-stage turbocharging at full load

It can be seen that with the R2S system a significantly higher BMEP is achieved. The dip in the progression between 1400 RPM and 1800 RPM results from an NVH related retard of the combustion. Without this NVH limitation, it can be assumed from the torque gradient between 1000 RPM and 1200 RPM that the target BMEP would be achieved at a speed of about 1400 RPM. From a speed of 2000 RPM, a BMEP level of 25 bar has been achieved, whereby a further increase is not limited by the turbocharging, but rather in the current case it is limited by the base engine. Together with the desired increase of BMEP, there is an associated simultaneous increase of the design power from 85 kW/l to 100 kW/l. This becomes apparent in a moderate increase of the specific fuel consumption that results from the necessary ignition timing retard with a higher BMEP, and which follows the known sequence in the engine characteristic. It can be assumed that with an extensive adaptation of the engine (charge cycle components, combustion process) to the increased charging rate, a reduction of the specific fuel consumption can be achieved. The observation of the absolute intake manifold pressure clearly shows the superiority of the R2S system over one-stage basis charging. From a speed of 1000 RPM, an absolute intake manifold pressure of about 1600 mbar can be delivered increasing up to 1400 RPM to a peak value of about 2750 mbar and thus offering the best preconditions for a high low-end torque. Together with the progression of BMEP it is easily recognizable that for effective implementation of higher charging pressures in high BMEP, an optimization of the design of the overall engine system with turbocharging is necessary. Between a speed of 1400 to 2500 RPM, a practically constant absolute intake manifold pressure of about 2800 mbar is delivered. The further progression becomes clear in the observation of the stage pressure ratios from high and low-pressure compressors in figure 3.2.2.

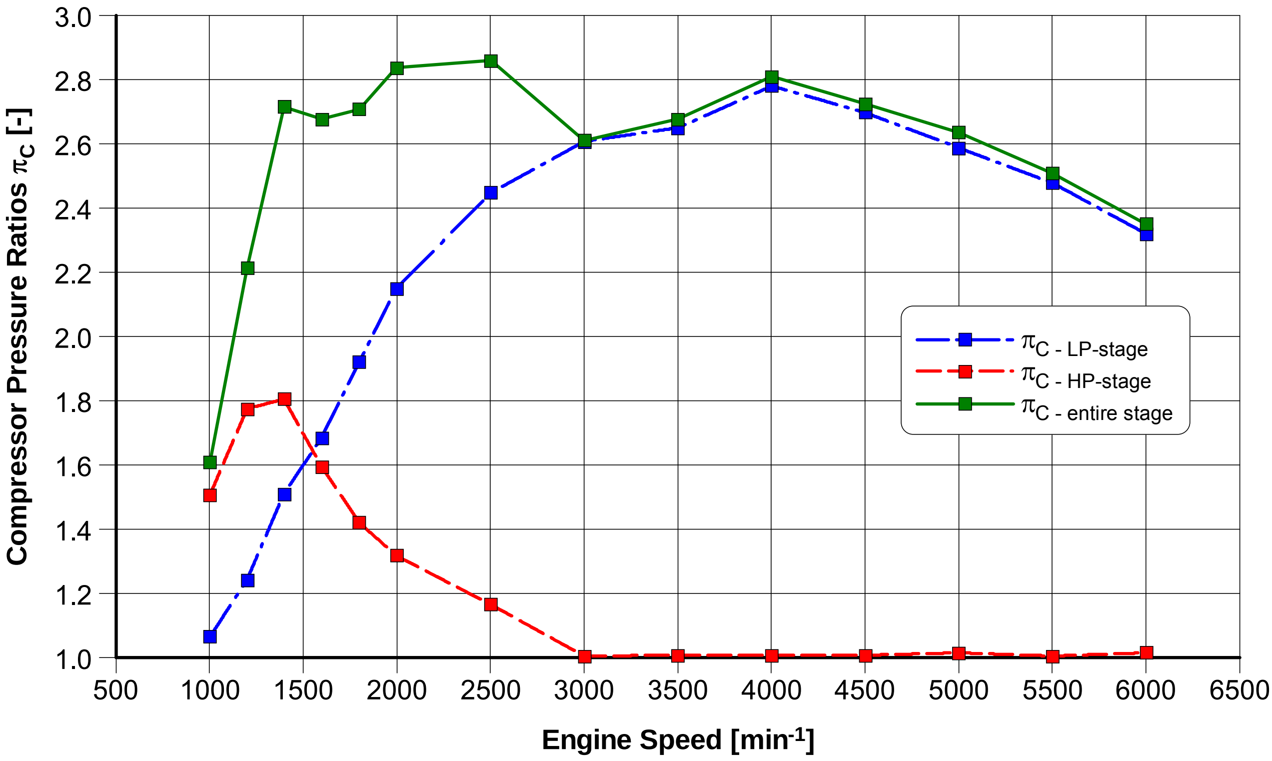

Figure 3.2.2: Compressor stage and total pressure ratio of the regulated two-stage turbocharging system at full load operation

From a speed of 3000 RPM on, the low-pressure compressor alone can provide the necessary charging pressure, and the downstream high-pressure compressor can be bypassed. Below this speed, the high-pressure compressor must contribute a maximum pressure ratio of 1.8 in this configuration. At speeds below 1600 RPM, the share provided by the high-pressure compressor prevails significantly, whereby the support particularly at low speeds is clearly evident. This progression shows that with a conventional compressor used in a high-pressure stage only a low share of the entire compressor characteristic available is utilized (see chapter 2.7). Furthermore, it becomes evident that the high-pressure compressor can provide a visibly higher contribution, should the design require higher target values.

The progression of the turbine stages and the overall expansion ratio is shown in figure 3.2.3.

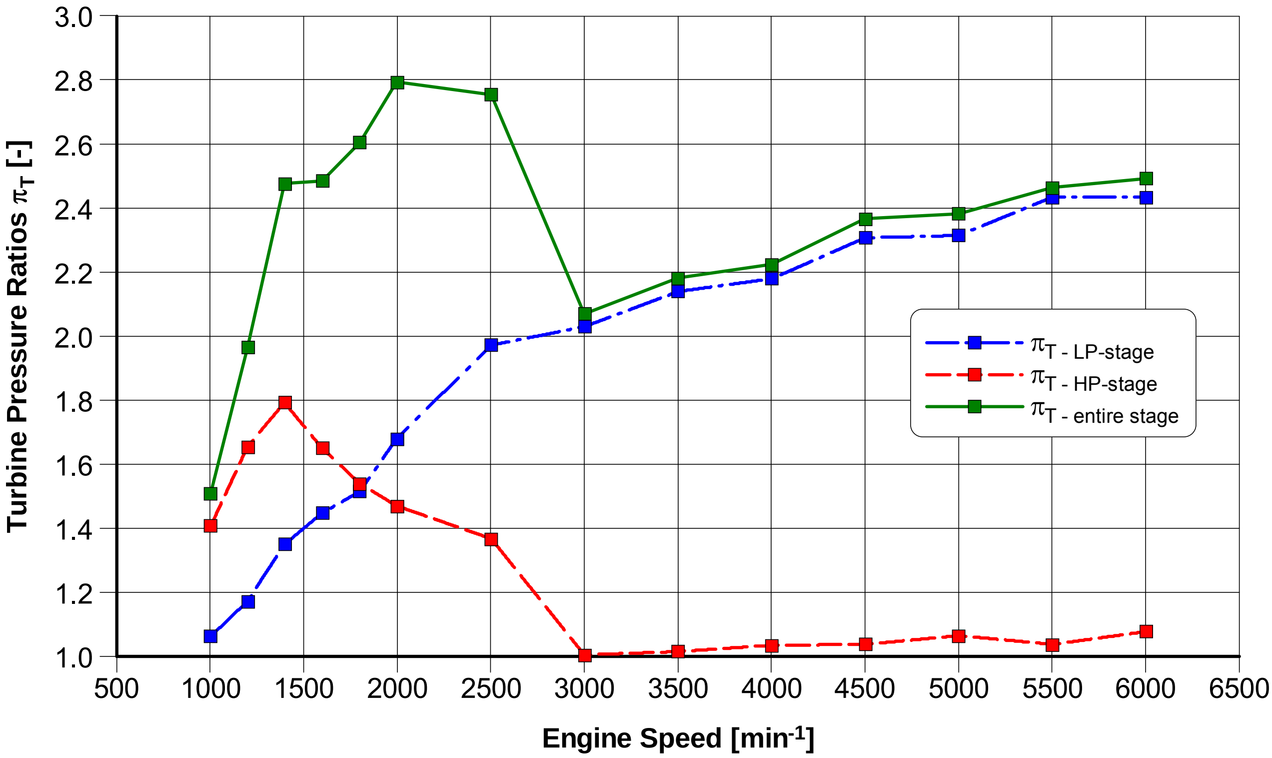

Figure 3.2.3: Turbine stage and total pressure ratio of the regulated two-stage turbocharging system at full load operation

The progression of the turbine expansion ratio shows a similar response to the compressor side. With regulated two-stage charging operation, the high-pressure turbine contributes a higher share of the necessary expansion ratio up to about 1800 RPM, until the flow rate matches the characteristic of the larger low-pressure turbine. On the turbine end, it must be considered that by regulating the power of the high-pressure turbine by opening the turbine bypass in stages with rising speeds, a respective mass flow bypasses the high-pressure turbine and can provide direct expansion assistance on the low-pressure turbine. From a speed of 3000 RPM, the expansion occurs using a single-stage via the low-pressure turbine. Because of the associated reduction on the turbine end pressure build-up behaviour, the exhaust gas pressure is reduced significantly in comparison to the regulated two-stage charging operation, which becomes visible from the progression of the overall expansion ratio. The lower share of the high-pressure turbine on the overall pressure ratio at medium to high engine speeds results from the structural-dependent arrangement of the high-pressure turbine casing and turbine bypass, where gas flows through with an opened turbine bypass, before the exhaust reaches the low-pressure turbine (also refer to chapter 2.6). Because of this arrangement, the rotating parts of the high-pressure charger achieve a basic speed level even though the bypass is opened, which on the one hand can be beneficial with quick changes in load; however, on the other hand it supports the tribological function of the bearing system. Here too it becomes evident that with a maximum expansion figure of 1.8, the characteristic map width of the high-pressure turbine is only slightly affected. In the context of the available series and contour stages, this can be considered for an optimization with the selection of the turbine/compressor pairing.

3.3 Regulated two-stage turbocharging at full load with timing adjustment

In the next step, the interaction of regulated two-stage turbocharging with variable timing using camshaft phase adjusters is examined. For better understanding, in the following figure 3.3.1 the valve events with the resulting overlap areas are shown.

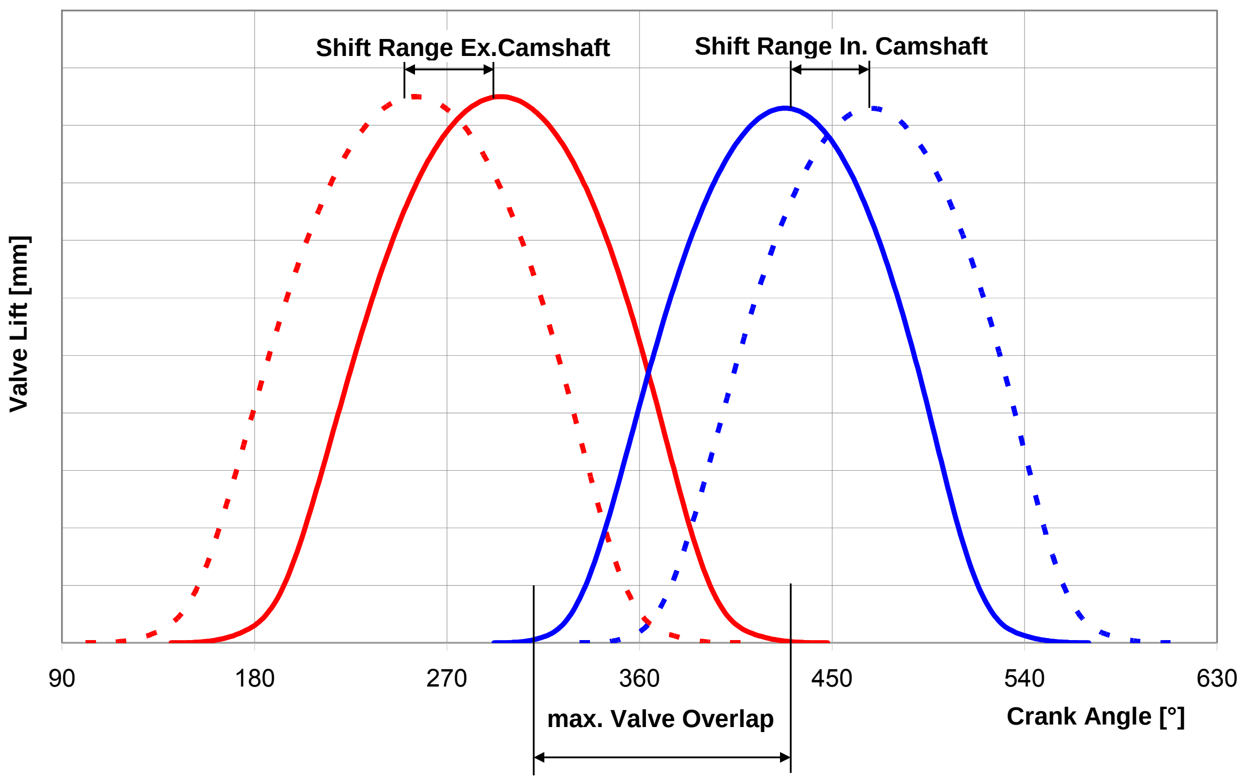

Figure 3.3.1: Valve events with minimum and maximum valve overlap

It can be seen that with this timing configuration the resulting valve overlap areas can be varied from high overlap to minimum overlap (dashed line). The associated effects on BMEP and fuel consumption are shown in the following figure 3.3.2.

The use of camshaft phase adjusters for generating maximum valve overlaps has different effects: At low speeds and maximum overlap, a significant reduction of BMEP is evident; from a speed of about 1100 RPM, BMEP can be increased by up to 2 bar.

This progression is surprising insofar as with one-stage turbocharging, significantly higher BMEP increases are known with the use of phase adjusters [1]. A more detailed analysis not displayed here has shown that starting from operation with minimal valve overlap with ever increasing overlap, a “local minimum” of BMEP is achieved first, before it again rises with an increasing overlap. This leads to the conclusion that an overall system optimization can be reached by adjustment of valve timing and duration, see also [1], [5].

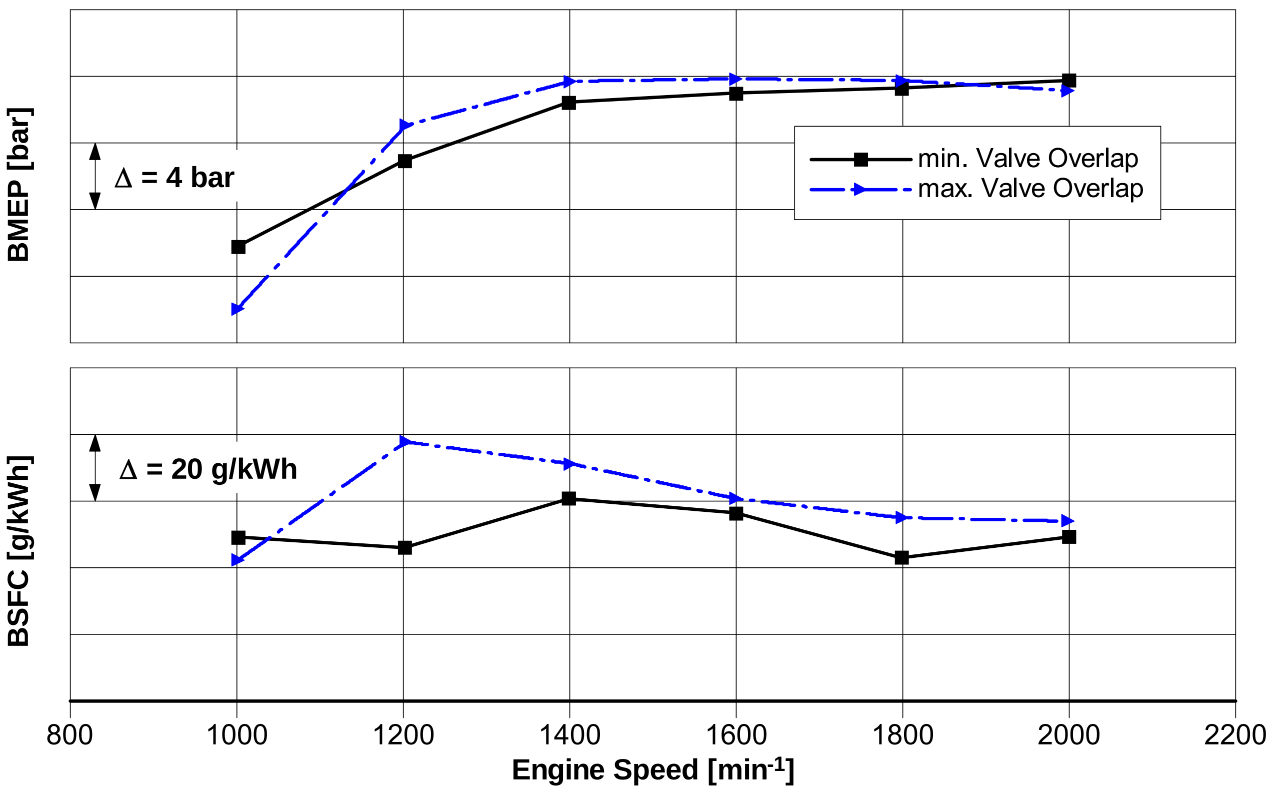

Figure 3.3.2: BMEP and specific fuel consumption with minimum and maximum valve overlap of the regulated two-stage turbocharger system at full load

A similar progression is evident with the specific fuel consumption, with the exception of a reduction at 1000 RPM, that is otherwise increased by up to 30 g/kWh. An analysis of the cause for this behaviour is made possible by observing figure 3.3.3.

The progression of the BMEP of the engine charge cycle clearly shows disadvantages by operation with maximum valve overlap of up to 0.8 bar. The increased charge cycle leads to a shift of the load point with corresponding effects on BMEP and fuel consumption. This behaviour is enhanced by the progression of the pressure drop across the engine (pressure difference from intake manifold up to the turbine inlet).

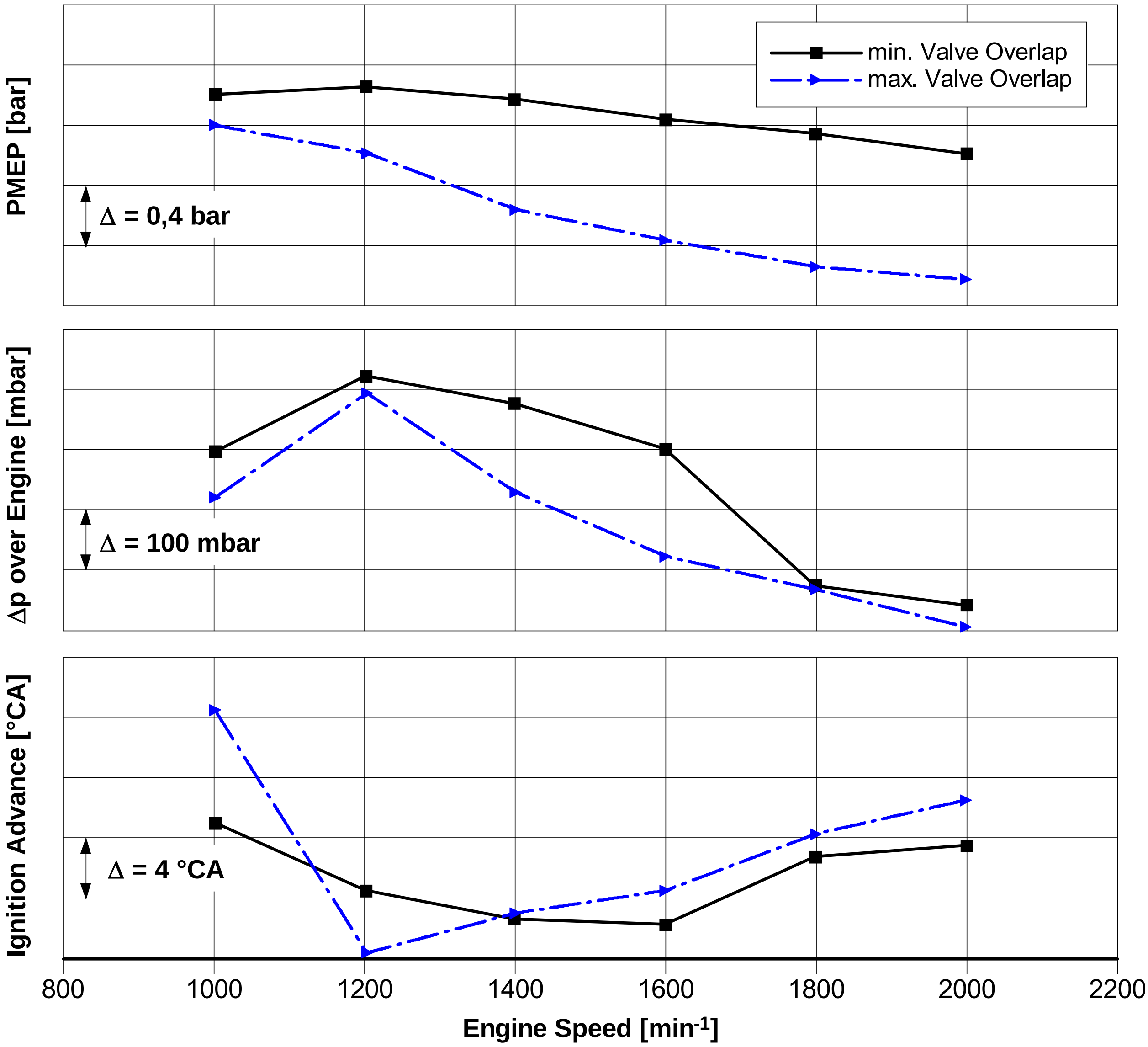

Figure 3.3.3: Pumping losses (PMEP), pressure gradient across the engine and spark advance with minimum and maximum valve overlap of the regulated two-stage turbocharging system at full load

At maximum valve overlap, a constant decrease in the direction of negative values is also recognizable here. The ignition advance shown indicates a large ignition advance angle with the exception of 1200 RPM, indicating more favourable thermodynamic conditions at the ignition point. Clearly – irrespective of the reduced pressure difference – the opening cross-sections at maximum valve overlap enable scavenging of the residual gas and thus a reduction of the charge temperature.

In conclusion, by using cam phasers a relatively moderate increase of BMEP can be achieved at lower engine speeds. Obviously, the regulated two-stage turbocharging reacts less to these valve timing variabilities than a one-stage system.

4 Further potential of regulated two-stage turbocharging

In chapter 2.6 it was already shown that the design reserves can be used for increasing the engine power further instead of using them to maintain power and low-end torque under extreme conditions. In figure 4.1 possible power variations with the engine displacement as a parameter are shown.

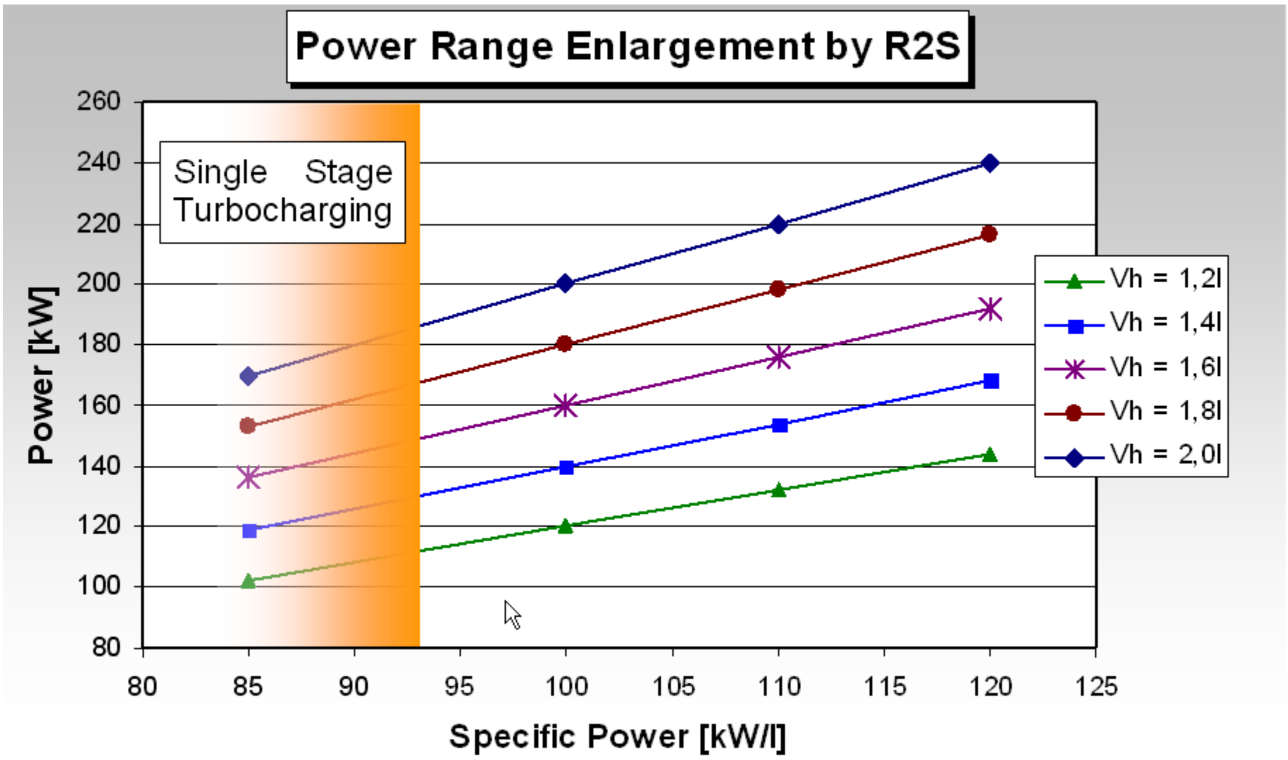

Figure 4.1: Extension of the power range as a function of engine displacement and specific power

It becomes evident that an impressive power variation of gasoline engines is possible by inventing regulated two-stage turbocharging. On the basis of a specific power of 85 kW/l the power can be extended to 110 kW/l and even a further increase up to 120 kW/l is not limited by the turbocharger system.

Two-stage turbocharger systems are the basis for a high degree of downsizing. The system consists of standard components so that a wide range of charger sizes is available. With a single-stage turbocharger representing the basis an attractive power range within an engine family can be realized thus avoiding higher cylinder numbers and, to a certain extend, larger displacements.

Because of the beneficial properties of regulated two-stage turbochargers applications can be identified that offer further customer benefits:

- Combination with high and/or low-pressure EGR to reduce emissions and exhaust temperatures and a reduction in material requirements
- Combination with Miller/Atkinson process (reduction of effective compression ratio)
- Combination with alternative knock-resistant fuels (example: Compressed Natural Gas) for realization of highly-efficient engines [8]

5 Summary

The investigations undertaken show that in the trade off between low-end torque and achievable rated power to realize higher degrees of downsizing in tubocharged gasoline engines, one-stage charging systems reach their limits. An extension of the performance beyond a certain degree can only be achieved by the use of regulated two-stage turbocharging.

The results achieved with an R2S turbocharging system of this type can be summarized as follows:

- Even with a rated power of 100 kW/l high intake pressures and BMEP can be reached even at lowest engine speeds (absolute intake pressure of 2,8 bar more than realistic at 1400 rpm)
- Power loss and disadvantages in low-end torque can be avoided under extreme conditions (operation in height and hot ambient conditions)
- With reduced reserves for extreme conditions there is potential to increase the specific power to 110 kW/l and more
- With an overall system optimization (adaptation of the basie engine, valve timing, cam phasers with adequate operating range, turbocharger system) a further increase in the target values, especially fuel consumption, is possible

Furthermore, the regulated two-stage turbocharger features the following properties when used with gasoline engines:

- Significantly higher pressure ratios possible compared to supercharging devices
- Good NVH behaviour
- Suitability for low pressure EGR (resistance to contamination has been proven with Diesel engines)
- Use of optimized but standard components possible to reduce expenses associated with the development and to improve reliability
- Vibration and wear free engagement of the high-pressure stage due to gas side coupling

To get the best performance out of regulated-two-stage charging systems a careful optimization of the individual components is required. The demands on the tubochargers in such a charging group clearly exceed the demands on the standard component. By participating from the continuous generic development on regulated-two-stage charging systems for Diesel engines the thermodynamic and mechanical properties could be enhanced for gasoline applications as well.

Regulated two-stage turbocharging is a further contribution from BorgWarner for the realization of future high-performance, low-emission and especially low fuel consumption gasoline engines.

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