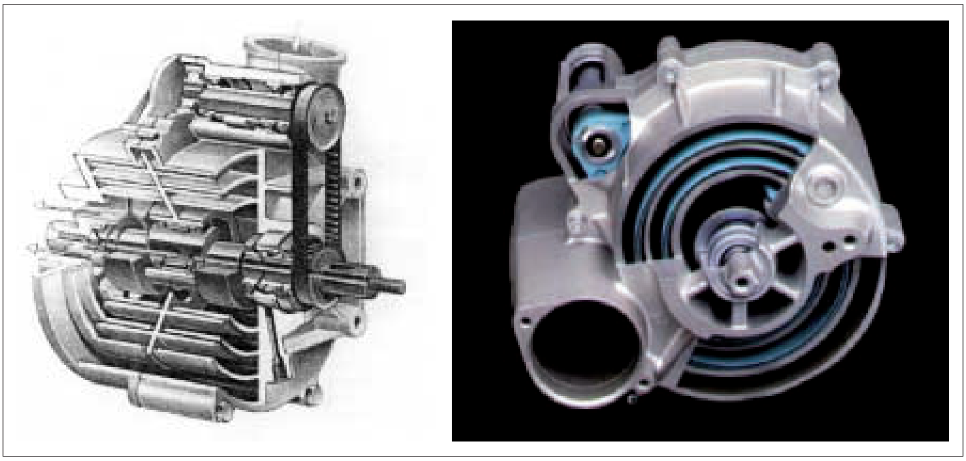

Figure 6.6: Spiral type supercharger realized as G supercharger [Source: VW]

displacer is driven by a main shaft which is connected to a crankshaft via a belt drive and is guided by an auxiliary shaft coupled with the main shaft by means of a belt. Both shafts have eccentrics so that the displacer does not rotate but only makes fast oscillatory movements. The air flows tangentially into the housing. It is enclosed between the spiral bars of the housing and the displacer and is transported towards the housing center, from where it reaches the intake system. The oscillatory movement leads to steadily diminishing volumes between the bars. The air is compressed and accelerated. Due to the considerable friction of the complex sealing elements and springs, which are located between the front sides of the displacer and the housing, G superchargers have a short service life if the high-wear components are not checked and replaced regularly. The G supercharger did not catch on for economic reasons as manufacturing costs (a scrap rate of up to 80%) repair and replacement costs were high. VW ceased production in the early 1990's.

6.2 Exhaust-gas turbocharging

To drive the charging unit, exhaust-gas turbocharging, **figure 6.7**, uses the hot exhaust gas from the engine, which is otherwise released into the environment with a high content of unused energy.

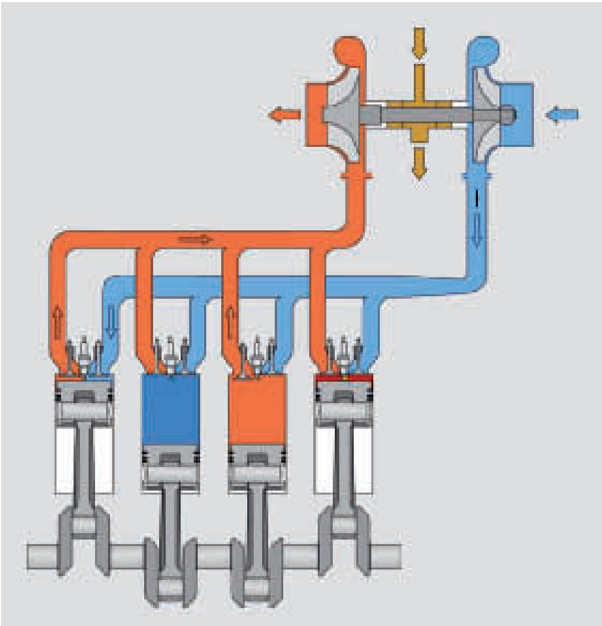

Figure 6.7:
Principle of exhaust-gas turbocharging [3]

Turbocharger and combustion engine are, therefore, coupled purely thermodynamically, rendering the following important advantages [1, 3, 5]:

- The fuel consumption of the turbocharged engine is lower when compared to an equally powerful naturally aspirated engine, because some of the otherwise unused exhaust energy contributes to engine performance.
- The turbocharged engine performs much better at high altitudes. The performance of a naturally aspirated engine is considerably reduced at high altitude as a consequence of the decreasing air pressure. In a turbocharged engine, the turbine power increases as there is a larger pressure difference between the almost constant pressure upstream of the turbine and the lower ambient pressure. The lower air density at the compressor inlet is partially compensated for, and the engine loses almost no power.

An exhaust-gas turbocharger consists of a compressor (centrifugal compressor) and a turbine, **figure 6.8**, connected to each other by a common shaft. The turbine converts some of the energy contained in the hot engine exhaust gas into mechanical work; the pressure and temperature of the exhaust gas decrease (polytropic, i.e. dissipative expansion). The mechanical work is then supplied to the compressor via the common shaft. In the running wheel of the compressor, the energy is initially transferred to the intake combustion air in the form of kinetic energy, and some of it is subsequently converted into pressure energy. During this process, the temperature of the intake combustion air increases. Apart from the mechanical and thermal losses, steady-state operating conditions show a power equilibrium between the turbine and the compressor.

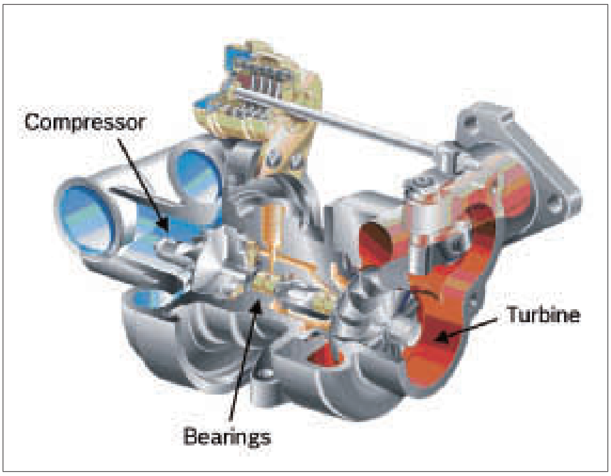

Figure 6.8:
Design of the exhaust-gas turbocharger

This power equilibrium results in a relationship which is also called the turbocharger main equation, applying the thermodynamic relationships for the compressor power, the turbine power and the isentropic enthalpy differences,

$$\pi_V = \frac{p_2}{p_1} = \left(1 + \frac{\dot{m}_T}{\dot{m}_V} \cdot \eta_T \cdot \eta_V \cdot \eta_m \cdot \frac{T_3}{T_1} \cdot \frac{c_{p,A}}{c_{p,L}} \cdot \left[1 - \left(\frac{p_4}{p_3}\right)^{\left(\frac{\kappa_A - 1}{\kappa_A}\right)}\right]\right)^{\left(\frac{\kappa_L}{\kappa_L - 1}\right)_L}$$

The main turbocharger equation shows for a particular inlet condition in the turbocharger (p_1, T_1) that the charge pressure ratio π_V is the higher, the greater the isentropic efficiencies for the turbine η_T and the compressor η_V, the smaller the mechanical losses (i.e. the greater the mechanical efficiency η_m) and the higher the turbine inlet temperature T_3 and the pressure p_3 upstream of the turbine. By reducing the mechanical losses, at otherwise identical conditions, a specific charge pressure p_2 is achieved even at smaller exhaust-gas back pressure p_3, which reduces the fuel consumption of the engine and leads to favorable conditions for the scavenging of the exhaust gases from the cylinder. For high overall efficiency of the turbocharger, coordination between the compressor wheel and the turbine wheel diameter is very important. The specification of an operating point in the compressor map results in a specific turbocharger speed. The turbine must be adjusted in such a way that it works in this operating range with the greatest possible efficiency.

It is possible to use the dynamic part ("pulse") of the escaping exhaust gases or to even it out. In constant pressure turbocharging the exhaust-gas pulses from the gas exchange are largely smoothed out by means of collectors with relatively large displacement. As a consequence, the turbine is charged with more or less constant pressure. In pulse turbocharging, however, the exhaust-gas pulses are maintained until the turbine inlet is reached by means of a suitable manifold design (volume as small as possible, separation or a suitable combination of the outlet ports up to the turbine inlet, i.e. firing order separation). Both variants – pulse turbocharging and constant pressure turbocharging – are implemented in the charging technology and have their advantages for specific applications. Constant pressure turbocharging is used in engines that do not need high excess torque for acceleration. Pulse turbocharging, in contrast, improves the torque yield in the lower speed range of the engine and is therefore preferably used to supercharge passenger car engines. The disadvantages of pulse turbocharging for the turbine, as compared to constant pressure turbocharging (poorer efficiency, lower discharge capacity, potential vibrational excitation of the blades), are more than compensated for by utilizing the pulse. For reasons of cost or space, less complex designs are frequently implemented in which the outlets lead to a common tube which is positioned upstream of the turbine and supplies the exhaust gases to the charger. The separation of the firing order, which is not consistent, reduces the potential of pulse turbocharging and has repercussions on the charge exchange. The potential for improvement through pulse turbocharging in the lower engine speed range is obvious; however, it cannot overcome the increasing discrepancy between the steady-state full load curve and transient behavior of turbocharged engines that goes hand in hand with the increasing degrees of supercharging.

In terms of thermodynamics, isothermal compression of the charge air would be the best solution (i.e. the temperature of the medium remains constant during the change of state). Isothermal process control is not possible under real conditions, though. Besides, the actual compression is not free from losses, i.e. it is not a reversible adiabatic process (final compression temperature $T_{2,s}$) but polytropic. As compared to reversible adiabatic process control, more work has to be used from pressure p_1 to p_2 during dissipative compression, which manifests itself in a higher compressor outlet temperature T_2 and is expressed by isentropic efficiency η_V, i.e. the ratio of isentropic $(h_{2,s} - h_1)$ and actual enthalpy difference $(h_2 - h_1)$ across the compressor.

$$\eta_V = \frac{h_{2,s} - h_1}{h_2 - h_1}$$

The higher final compression temperature T_2 leads to the charge air having less density. In order to counteract poorer cylinder charging that results from it, the charge air is carried to a charge air intercooler before it enters the engine so that the charge air can be cooled, to achieve an increase in charge density. This also reduces the thermal load of the engine and has a positive influence on the knock limit. The lower temperature level in the cylinder inhibits the formation of nitrogen oxide. However, the charge air intercooler causes a loss in pressure, which must be taken into consideration when designing the turbocharger. The cooling of the charge air is mainly influenced by the temperature of the coolant and the effectiveness of the cooling system, the heat having to dissipate into the environment through a closed circuit in the case of passenger cars.

Function and design of the exhaust-gas turbocharger

The centrifugal compressor of a turbocharger consists of a running wheel, a diffuser and a spiral housing. The charge air is drawn in axially and is accelerated to high velocity in the wheel (in the absolute system, i.e. for an observer stationary with regard to the housing). The air generally leaves the compressor wheel in a radial direction. A diffuser is connected right next to the running wheel and is formed from a compressor backplate and part of the spiral housing. The diffuser slows down the velocity of the air, largely without any losses, so that the pressure and the gas temperature both rise. Even for constant diffuser width, the radius increase in flow direction results in an increase of the flow cross-section and therefore a deceleration of the flow results. So far, bladeless diffusers with fixed geometry have generally been used for passenger car applications. They have a wide map range. The air is collected in the spiral housing, where the velocity continues to be reduced before the charge air reaches the compressor outlet, and this leads to another pressure increase. In terms of aerodynamics, the compressor flow is a delayed flow (in the relative system, i.e. for an observer positioned on the wheel). The channels of the compressor wheel are complex diffusers. The flow in the channels is in the direction of a positive pressure gradient. Compressor wheels for passenger car applications have a relatively high hub ratio (ratio between the compressor inlet and compressor outlet diameters), an adjusted number of blades, and thin blades bent backwards. To reduce the component stress during operation, the blades are attached diagonally to the hub.

The turbine of a turbocharger consists of a spiral housing, bladed or bladeless nozzles, and a turbine wheel. The exhaust gas from the engine is retained by the flow resistance of the turbine. The spiral housing is designed in such a way that the exhaust gas is directed to the nozzle as evenly as possible across the circumference. In the spiral housing and in the nozzle, some of the exhaust-gas pressure that has built up is converted into kinetic energy, i.e. the flow is accelerated and directed to the turbine wheel. The turbine wheel converts some of the enthalpy of the hot exhaust gas into mechanical energy that drives the compressor of the turbocharger via the shaft. Due to the mass flow and pressure ratios that arise, only centripetal turbines that receive airflows radially and expel them axially are used for passenger car turbocharging. In regard to their comparatively low moment of inertia, axial turbines would be quite interesting; the small mass flow, together with the Mach and Reynolds numbers necessary for an effective energy conversion, lead to very small flow cross-sections so that the boundary layer effects and clearance losses clearly limit the efficiency. In terms of aerodynamics, turbine flow is an accelerated flow; the channels of the turbine wheel are complex nozzles. The flow in the channels is in the direction of a negative pressure gradient. Certain flow phenomena, as can occur in compressors depending on the operating point (surge), cannot occur in a turbine. The turbine performance increases with increasing pressure gradients across the turbine. At higher engine speed, more exhaust gas is built up and the expansion work increases. Turbine performance also improves with increasing exhaust-gas temperature due to the higher energy content of the exhaust gas. The retaining behavior of the turbine is determined by the free flow cross-section at the crossing point from the inlet port to the spiral, or throat cross-section. If the throat cross-section is reduced, more exhaust gas is retained by the turbine, the turbine performance increases as a result of the higher pressure ratio and a higher charge pressure can be generated in the compressor. The throat cross-section is specifically set during the design of the turbocharger, i.e. when it is matched to the required performance of the engine. In addition to the throat cross-section of the turbine housing, the passage cross-section upstream of the wheel inlet also influences the flow rate characteristic of the turbine. The machining of the wheel contour makes it possible to adjust the cross-section and thus to influence the charge pressure. Contour enlargement results in a larger flow cross-section of the turbine. Turbocharger manufacturers usually offer turbine wheels of the same diameter with different contours within a production series; these can be made from the same blank.

Variable turbine geometry

Depending on the operating point, the free flow cross-section or the direction of the flow can be influenced by movable elements located in the spiral housing or the nozzle. While the turbocharger with "variable turbine geometry", or VTG, has established itself as the standard charging device for diesel engines, this technology is still a huge challenge for gasoline engines due to the high exhaust-gas temperatures. Besides the solutions for adjustable elements presented and examined here [6], in particular pivot mounted guide blades (VTG principle), **figure 6.9**, or an axially slidable sleeve (VST principle), **figure 6.10**, are the other main solutions.

The VTG turbocharger has a spiral housing with fixed geometry. Between the spiral housing and turbine wheel inlet, pivot mounted turbine guide blades are arranged so that the flow cross-section and the angle of approach to the turbine wheel can be changed. While straight guide blades were installed in the first VTG series turbochargers, today increasingly complex cambered blade profiles with much greater efficiency and favorable control behavior are used. The individual guide blades are connected with a ring by means of an adjusting lever charged by an actuator via a multi-linkage lever. The exhaust gas in modern

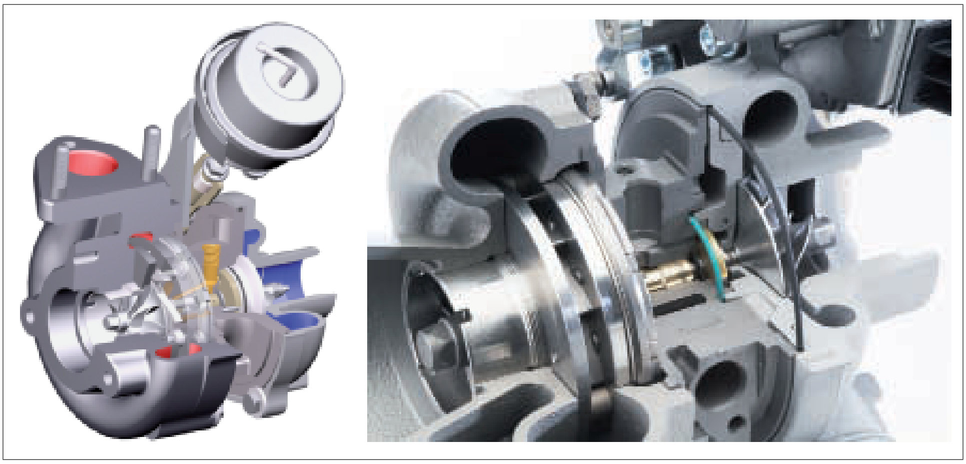

Figure 6.9: Turbocharger with pivot mounted guide blades (Variable Turbine Geometry / VTG)

gasoline engines can reach up to 1050 °C. The development goal is to reliably increase the working temperature of turbines with variable geometry to this value.

The constant, reliable movement of the guide blades in the hot exhaust-gas stream imposes high demands on the materials used and the design. Detailed adjustment of the tolerances of the individual components concerned within the turbine is indispensable. Irrespective of the size of the turbocharger, a certain clearance is required in order to guarantee that the blades will remain moveable during the entire service life of the vehicle. As a result, the clearance losses increase as the size decreases.

At low engine speeds, the flow cross-section of the turbine is reduced by closing the guide vanes. The charge pressure and therefore the torque of the engine increase as a result of the higher pressure difference between the turbine inlet and the turbine outlet. At high engine speeds, the guide vanes open. The desired charge pressure is reached for lower exhaust-gas back pressure, i.e. for a smaller turbine pressure ratio. The result is a favorable utilization of the engine's primary energy. During acceleration phases from low speeds, the guide blades close for higher accumulation levels and therefore provide plenty of excess power for the quick acceleration of the rotor. With increasing engine speed, the blades open continuously according to the run-up characteristic of the engine. The first turbocharger with pivot mounted guide vanes for gasoline engines with exhaust-gas temperatures of 980°C was introduced in series by BorgWarner Turbo Systems for the Porsche 997 in 2006 [7, 8].

A special feature of the original VST turbocharger is a twin flow turbine housing and a sliding sleeve, figure 6.10, which can be moved partially over the turbine wheel axially in the direction of the shaft axis and can thus directly change the flow cross-section at the turbine inlet. In the lower engine speed range, one of the two channels of the inlet spiral is completely covered, almost simulating a turbocharger with small turbine throat cross-section. From a certain operating point onwards, with increasing engine power, the second

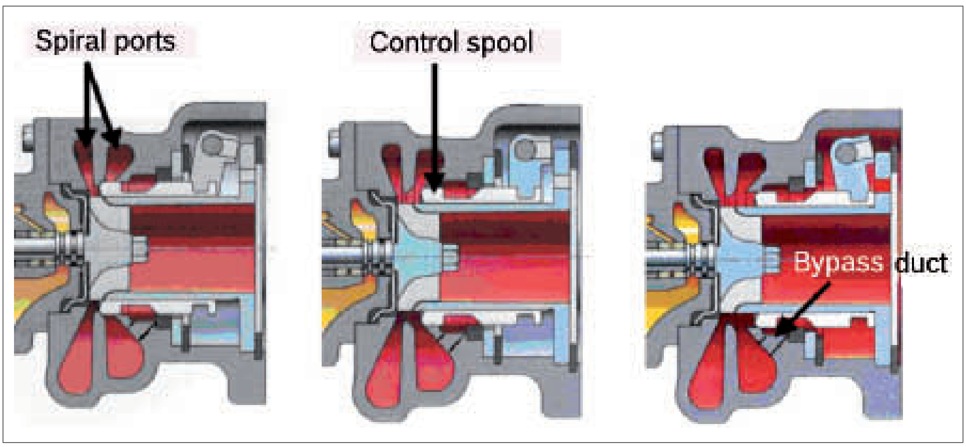

Figure 6.10: Axially slidable sleeve (VST) principle

channel is continuously exposed until it is completely open. When the amount of exhaust gas continues to increase, the sliding sleeve axially exposes a ring-shaped bypass duct. The sliding sleeve is directed by an adjusting fork which in turn is moved by an actuator by means of a lever system [3]. Further development of the VST principle in terms of optimized efficiency and control abandons the twin flow housing; a fixed radial guide blade is placed in the passage cross-section directly upstream of the turbine wheel to generate a directed flow.

Depending on the load and engine speed, the axially slidable sleeve increases or decreases the passage cross-section for optimal adjustment to the operating point. As in the original VST turbocharger, a bypass duct can be exposed using the sliding sleeve to decrease the basic charge pressure in the upper operating range. So far the VST principle has not been able to replace the principle of rotating blades.

6.3 Synergies between exhaust-gas turbocharging and direct injection in gasoline engines

As for modern diesel engines, the combination of direct injection and turbocharging in gasoline engines has great potential. In addition, gasoline engines have variable valve control as another important component. Therefore, the historical disadvantages of turbocharged gasoline engines – a compression ratio needing to be reduced due to knock tendency, efficiency disadvantages due to the enrichment required for the protection of components in certain operating ranges, and delayed response when accelerating from low engine speeds – can be effectively and sustainably counteracted. The combination of direct injection, turbocharging and variable valve timing opens up new possibilities for improving the engine process. Variable valve timing makes it possible to reduce the residual gas amount of the full load and to realize favorable torque behavior even at low engine speeds. If the scavenging slope is positive, i.e. the charge pressure is higher than the pressure at the turbine inlet, scavenging with fresh air can be achieved by setting a long valve overlapping phase; **figure 6.11**. In gasoline direct injection, the fuel is injected into the combustion chamber only after the exhaust valve has closed, which is why no unburnt hydrocarbon emissions are produced.

The mass flow rate increased by the greater air efficiency improves the response behavior of the turbocharger and has a positive effect on the dynamic behavior of the engine during load increase. Scavenging at low engine speeds leads to an improvement in the torque behavior, **figure 6.12**, as the cylinder can be scavenged almost completely and thus can be kept almost free from residual gas. As a result, the torque is built up more quickly during a load increase and dynamics is increased. BMW [9] additionally sets an air/fuel ratio of $\lambda = 0.9$. This leads to an optimal combustion speed (knocking and pre-ignition are

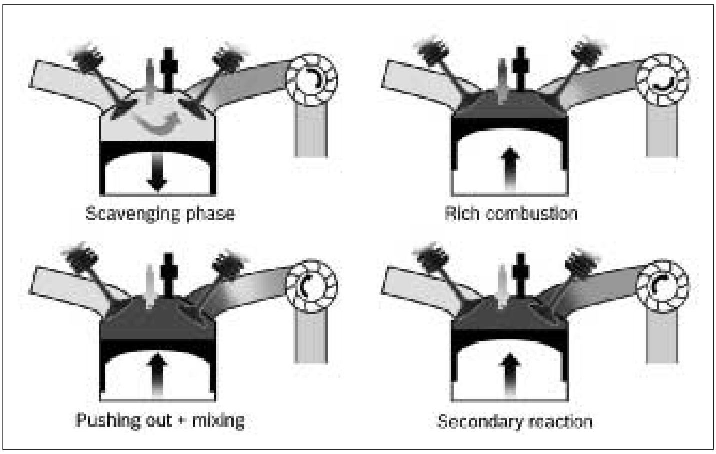

Figure 6.11: Principle of scavenging in the low-end torque range [9]

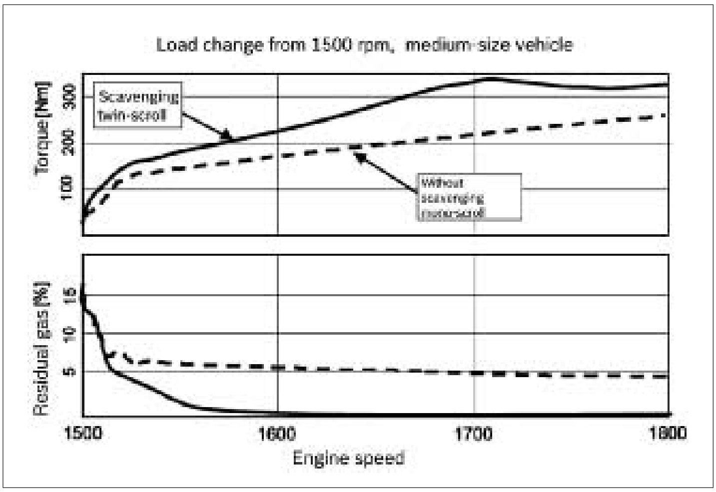

Figure 6.12: Torque build-up during scavenging [10]

avoided). In addition, the unburnt fuel of a cylinder reacts with the scavenged fresh air of another cylinder present in the manifold. This secondary reaction increases the enthalpy upstream of the turbine and leads to a stoichiometric air ratio upstream of the three-way catalytic converter, depending on the quantity of the scavenged fresh air.

Figure 6.13 shows the transient torque build-up at a constant engine speed of 1500 min^{-1}, starting from p_{me} = 2 bar up to full load without phase adjuster at the camshaft, with intake phase adjuster and with double phase adjuster.

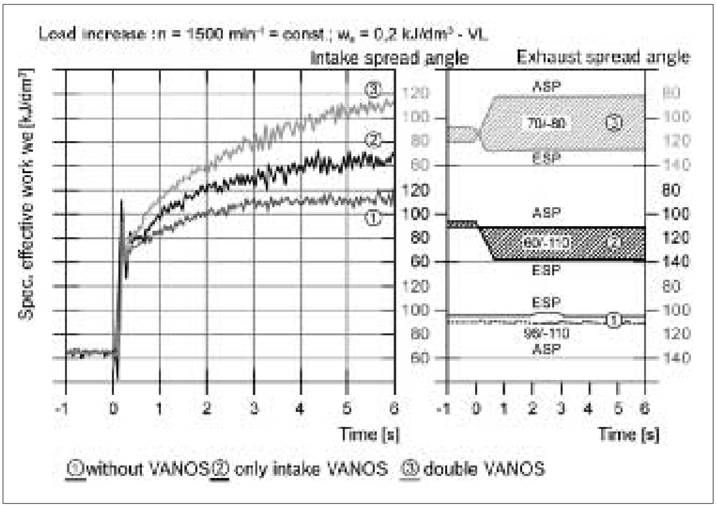

Figure 6.13: Dynamic behavior: variation of intake and exhaust spread angle [11]

Without the phase adjuster, the torque build-up is very slow and the maximum mean effective pressure remains on a low level. With the phase adjuster at the intake side, a much faster torque build-up and a higher absolute value of $p_{me} \approx 18$ bar are achieved. The use of a double-phase adjuster further improves the torque build-up and the steady-state absolute level reaches values of p_{me} over 23 bar for 1500 min^{-1}. The torque build-up is carried out almost as fast as in a naturally aspirated engine and thus suggests excellent dynamic behavior of the turbocharged engine with a double-phase adjuster.

Another effect of scavenging with fresh air is shown in **figure 6.14**. Due to the amount of scavenged air, the operating curve shifts to the right side in the compressor map, towards higher mass flows. The increased distance to the surge line opens up new freedoms for the selection of the turbocharger. For example, a larger compressor that has more favorable efficiencies in the nominal power range or offers more potential for power increase can be used.

The interaction of the technologies mentioned enables high cylinder charging and lower knock tendency as compared to intake manifold injection. The result is gasoline engines that have high torques even at low engine speeds while maintaining a high speed spread angle. From today's point of view, steady-state performance characteristic values are possible in the order of magnitude of 200 Nm/l and 100 kW/l with extremely responsive dynamic behavior; i.e. with regard to specific torque, the gasoline engine with direct injection can match the ambitious values of the diesel engine; with regard to specific performance, it clearly exceeds the values [11, 12, 13].

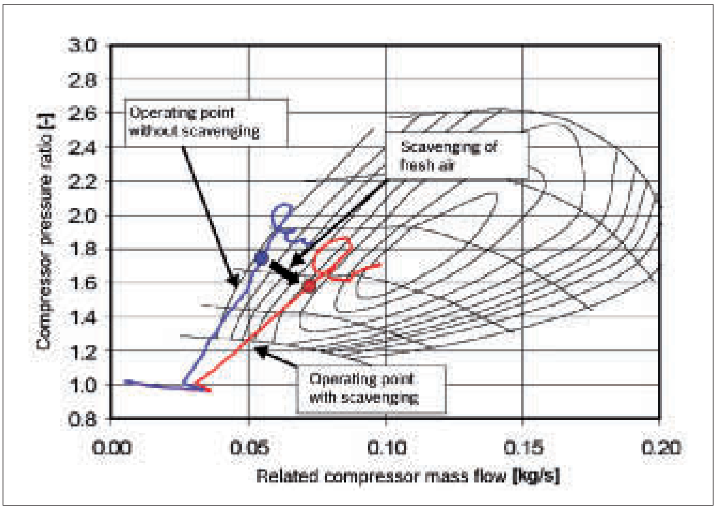

Figure 6.14: Shift of the operating points in the compressor map due to scavenging [10]

6.4 High pressure supercharging with an electrically driven supercharger

It is in particular the realization of attractive small-displacement gasoline engines that is inseparably connected with a high degree of boosting, among other measures. For such applications, the supercharging system must have a noticeable effect, also in the lowest possible engine speed range. The discrepancy between steady-state and transient behavior increases with increasing supercharged engine power output per liter, even if engine tuning and turbocharger design are optimal, i.e. during the acceleration of the vehicle, the steady-state full load curve is reached at higher and higher engine speeds. This increasing discrepancy can be covered by additional supercharging, whose operating power should be practically independent of or at least be decoupled from the operating state of the engine. Electrical energy is a good choice for this purpose. The development of an electrically assisted supercharging system, however, must always be seen in the context of the possibilities and limitations of the vehicle electrical system. Since the introduction of a 42 V vehicle electrical system with increased capacity has been delayed for various reasons, efforts are focusing on a modified 12 V vehicle electrical system with increased power. This development is currently driven by the increasing number of electrical high power loads, i.e. there is no direct relationship between this development and an electrically driven system. However, the supercharging system can benefit from innovation in the field of vehicle electrical systems. In this context, it should be pointed out that ultimately, electrically assisted supercharging can be classified as mechanical supercharging.

Electrically assisted exhaust-gas turbocharger

A possibility for electromotive assistance for exhaust-gas turbocharging is the integration of a suitable electromotor in the shaft of the turbocharger, for example between the turbine wheel and the compressor wheel. Specifically developed multi-phase synchronous motors with a power of 1.5 to 2 kW are proposed as electromotors, depending on the size of the turbocharger and the capacity of the vehicle's electrical system. In addition to the electromotor, power electronics is included as another component that transforms the current supplied by the vehicle electrical system accordingly. In the power units presented so far, the power electronics are spatially separated from the electrically assisted turbocharger, due to the high thermal and vibration loads. Correspondingly dimensioned cables connect the electromotor with the power electronics system. As in the conventional turbocharger, the shaft is directed by plain bearings lubricated with oil. The turbocharger is thoroughly cooled with oil or water. Depending on the implementation, compressor air is additionally used to cool the electromotor. The integration of the electromotor for an electrically assisted exhaust-gas turbocharger results in improved transient behavior in the operating points in which only little exhaust gas is available, despite the increase of the moment of inertia of the shaft assembly. A substantial improvement in the steady-state performance values is not possible, however, as improvements can be made only within the given compressor map limits through single-stage process control, for reasons related to the system. The electrically assisted turbocharger can also work like a generator, i.e. in suitable operating points or for a corresponding amount of exhaust gas, the electromotor works as a generator and supplies electrical energy to a suitable storage medium, for instance a capacitor battery (supercap), according to the efficiency chain. It remains to be clarified whether regeneration in passenger cars is technically and economically reasonable. In addition to the comparatively high thermomechanical load of the electromotor as an integrated component of the turbocharger – the electromotor will cover the entire operating range of a standard turbocharger even when it is not active – voltages are induced for the use of a synchronous motor with permanent magnets on the shaft even when the electromotor is not running.

Electrical booster

The electrical booster (eBooster) is a flow compressor driven by an electromotor; **figure 6.15**. The entire eBooster turbocharging system consists of the series connection of an appropriately designed conventional turbocharger and an electrically driven booster, i.e. the system works in two stages when the booster is active, which results in the overall pressure ratio from the product of the pressure ratios of the individual units.

The eBooster system reaches a higher charge pressure level at an earlier point in time in operating points in which little exhaust gas is available, which is due to the electrical drive (virtually independent of the operating state of the combustion engine) and the two-stage nature of the compression. The combination of two compressors, which can be optimally coordinated for their respective airflow rate ranges, results in an overall larger useable map width.

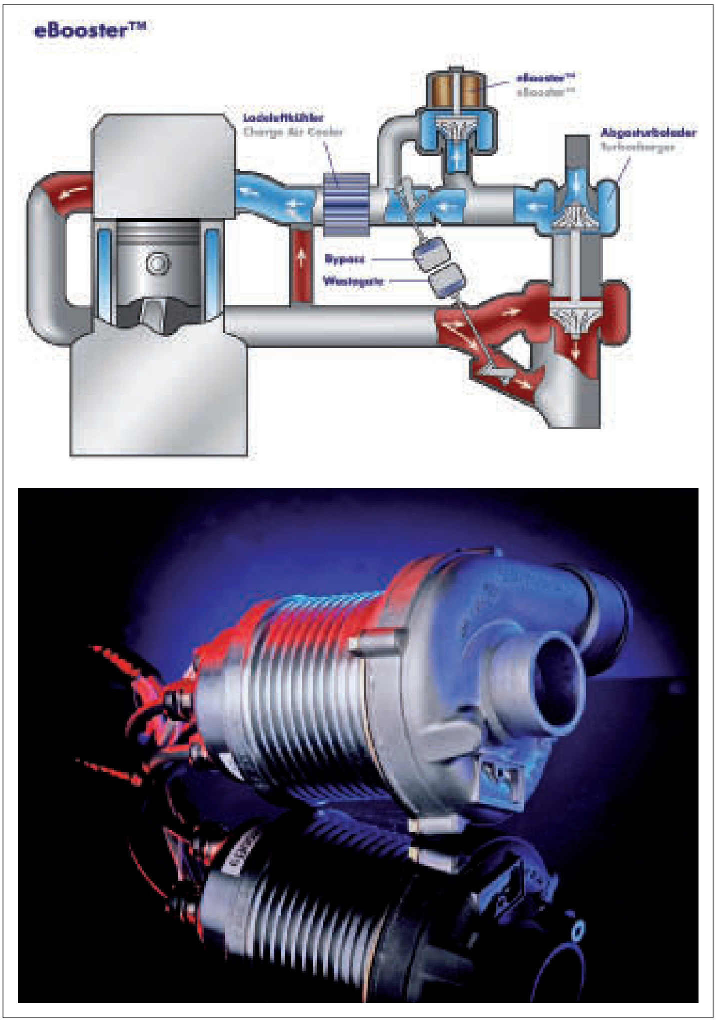

Figure 6.15: eBooster turbocharging system

Depending on the design of the turbocharger, a possible mode of operation of this system is an interconnected operation of eBooster and turbocharger below an engine speed of 2000 min^{-1} during start-up and acceleration, i.e. in transient operating phases in the lower engine speed range, whereas above this engine speed the turbocharger handles the air supply alone. If the required electrical power could be provided by the vehicle electrical system, then steady-state operation of the booster with a significant increase of the steady-state engine torque would be possible. Moreover, the booster can perform the function of a secondary air pump during cold start. The booster can be placed upstream or downstream of the exhaust-gas turbocharger. Placing the exhaust-gas turbocharger upstream offers greater flexibility with regard to the mounting position, whereas downstream placement enables shorter conduction paths. The system is completed with a bypass duct with a self-actuating or controlled flap that reduces the pressure losses in the intake port when the eBooster is not in action [14, 15]. To be effective, the eBooster must be able to produce a certain pressure ratio in a very short period of time. On the other hand, the electrical power consumption of the system should be as low as possible in order to remain within the capacity limits of the vehicle electrical systems currently available. This target conflict leads to a compromise in the design of the eBooster power unit with regard to operating speed and moment of inertia of the shaft or the wheel diameter of the compressor. Since the booster power unit is spatially separated from the exhaust-gas system, water or oil cooling is not necessary; the power unit can be realized with air cooling and the shaft can be guided via roller bearings with long-life lubrication. Besides, the power electronics of the electromotor can be integrated onto the eBooster power unit, which leads to very short electrical supply lines. The conceptual design of a suitable drive motor for the e-Booster depends on both the requirements with regard to the torque demand of the compressor wheel and the extremely short run-up time, which creates a quite significant additional torque demand for the acceleration of the rotating masses. The demand for a design that is both robust and cost-effective is another key focus [16]. The effectiveness of the eBooster turbocharging system especially for small displacement gasoline engines with high pressure supercharging has been proved in advanced development projects. It remains to be seen whether the system can establish itself in the market, including with regard to cost. The increasing electrification of vehicles that started some time ago may create the appropriate technical conditions.

6.5 Complex supercharging

In engine development, the following goals are paramount: starting torque as high as possible, good propulsion power, and high performance that complies with the legal emissions standards for pollutants and carbon dioxide. Good acceleration behavior requires a rather small turbocharger. This, however, leads to a high back pressure in the nominal power point, with high fuel consumption and lower performance as a result. High performance in the nominal power point along with fuel consumption being as low as possible requires the exhaust-gas back pressure to be as low as possible, i.e. a rather large turbocharger, which, would not have a very satisfactory acceleration behavior due to the shaft's relatively large mass moment of inertia, though. This target conflict is more and more solved by means of a complex, sometimes multi-stage interconnection of charging units. Additional supercharging virtually independent of the operating condition of the engine is considered as one option; see section 6.4.

Multi-stage supercharging

In multi-stage supercharging, **figure 6.16**, the supercharging of the engine is done via two smaller turbochargers connected in parallel instead of by one large turbocharger. In addition to the turbochargers, a multi-stage supercharging system includes several switching devices and sensors that facilitate a smooth change from a single turbocharger operation to a multi turbocharger operation. Only one turbocharger is active in the lower engine speed range. From a certain engine speed onwards, both turbochargers operate in parallel. Due to the fact that the air supply is distributed between two turbochargers, torque characteristics are responsive and nominal power is high. The significantly lower mass moment of inertia of the turbocharger shaft, as compared to a correspondingly designed single turbocharger unit, furthermore results in a very favorable response behavior by the engine. The system combines high performance with relatively low fuel consumption and impressive acceleration. In the lower engine speed range, all the exhaust gas and all of the charge air flow over only one turbocharger until the corresponding setpoint charge pressure is reached.

The turbine sequence valve and compressor sequence valve for the second turbocharger are closed; the second turbocharger is not active. If the desired charge pressure is built up and if more exhaust-gas enthalpy is available than is required to reproduce charge pressure, it is directed to the turbine of the second turbocharger which is connected in parallel and subject to acceleration through this step.

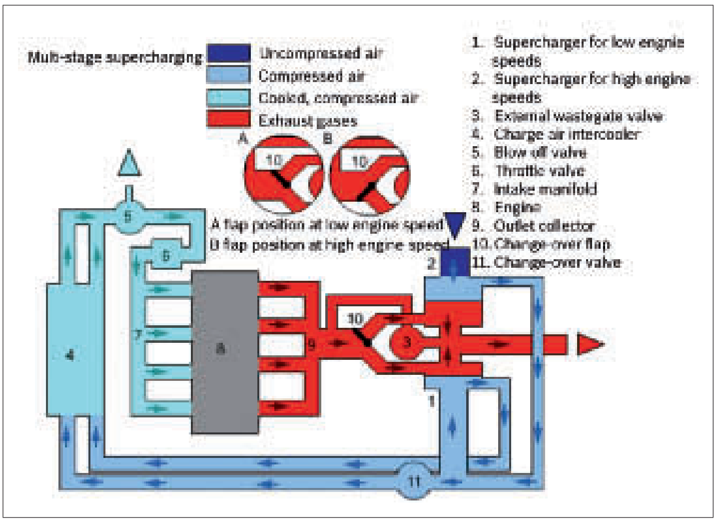

Figure 6.16: Principle of multi-stage supercharging

The compressor of the second turbocharger begins to deliver but the compressor sequence valve is still closed; the air mass flow being that is being delivered is slowed down via a relief valve and is introduced upstream of the compressor of the first turbocharger. The engine now works in controlled single turbocharger operation; the bypass valve is closed, the turbine sequence valve is operational. From a certain engine speed onwards, sufficient exhaust-gas enthalpy is available for both turbochargers and the second compressor to be connected. For this purpose, the turbine sequence valve is fully opened and the relief valve is closed. The pressure downstream of the second compressor increases rapidly and opens the compressor sequence valve. Now both turbochargers are operational and each handles approximately half of the engine's air supply. Excess exhaust-gas energy is expelled via a waste gate; the charge pressure is controlled by means of the bypass valve.

Two-stage supercharging

In contrast to multi-stage supercharging, two-stage supercharging connects the turbochargers in such a way that serial operation is carried out, at least in certain operating phases. Different connection variants have been examined for diesel engines [17] and it seems reasonable to apply this forward-looking approach to gasoline engines as well. In this extremely efficient system, turbo-chargers of different sizes are used, i.e. the flow rate behavior of the two turbochargers is different. As a result, the enthalpy gradient can be better utilized for identical mass flow. The advantage of this supercharging system as compared to the single-stage technique is the increase of the nominal power with a simultaneous improvement of the steady-state torque at low engine speeds and the acceleration behavior of the engine due to the rapid build-up in charge pressure. There is no "transition lag" when coordination and control are carried out in this way.

Combined supercharging: mechanical supercharger plus turbocharger

To improve the starting torque of turbocharged combustion engines, additional mechanical supercharging is possible. One way to reduce "turbo lag" is to use a combination of exhaust-gas turbocharging and mechanical supercharging. This system was used for the first time in 1985, in the Lancia Delta.

In the lower engine speed ranges, in which little exhaust-gas energy is available to drive the turbocharger turbine, the charge air is supplied by a mechanically driven compressor. The mechanically driven compressor can be placed either upstream or downstream of the compressor of the turbocharger. The complete supercharging can be handled by the exhaust-gas turbocharger only starting from an operating range in which sufficient exhaust-gas energy is available to generate the required charge pressure. During transition, the system works as two-stage supercharging. As soon as the exhaust-gas turbocharger is in a position to generate the charge pressure on its own, the mechanically driven supercharger can either be completely bypassed using a bypass valve or completely decoupled by means of a coupling.

Decoupling, e.g. via a magnetic coupling, has energetic advantages, as the drive energy of the mechanical supercharger is no longer needed due to this, but requires greater effort in terms of control engineering, mechanical durability and costs. In order to have sufficient charge pressure even at low engine speeds, it is necessary to make a correspondingly high transmission of the mechanically driven supercharger. This results in high engine speed gradients when switching the mechanical supercharger on and off, which in turn leads to a high mechanical load on the drive. With regard to the thermodynamic design of the exhaust-gas turbocharger in combination with the mechanically driven supercharger, advantages arise due to the fact that the turbocharger can be designed mainly for the medium and higher engine speed range. This leads to the use of relatively large turbochargers, which have good efficiency, especially for high mass flow rates, and thus enable a high maximum engine performance with favorable fuel consumption. In spite of the energetically unfavorable direct coupling of the mechanical supercharger to the crankshaft of the engine, very favorable fuel consumptions arise in the normal driving cycle, due to the optimal design of the turbocharger. Moreover, due to the mechanically driven supercharger, full advantage can be taken of downsizing as there is no longer any perceptible "turbo lag", even for high degrees of supercharging.

Despite the undeniably favorable characteristics of this supercharging system, the related manufacturing costs are regarded critically. Taking the measures required to improve the critical acoustics behavior of a mechanically driven positive displacement supercharger into consideration, significant technical costs arise, and these are reflected in the system price. Volkswagen's recently introduced VW 1.4 l TSI engine with combined supercharging impressively succeeded in demonstrating the technical potential; **figure 6.17**.

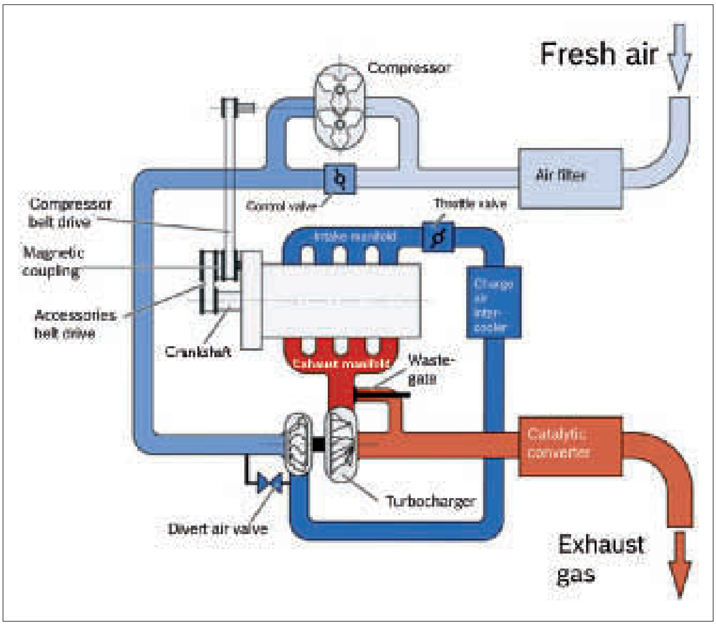

Figure 6.17: Principle of additional mechanical supercharging [18]

A mechanically driven Roots blower is connected upstream of the exhaust-gas turbocharger, the latter being linked to the engine via a clutch and a gearbox. The combination is coordinated so that the exhaust-gas turbocharger is always active and the mechanical supercharger is activated in the lower load and engine speed ranges; **figure 6.18**. Additional mechanical supercharging makes it possible to achieve both a high specific power (via turbocharging) and good response behavior at low engine speeds (via mechanical supercharging). However, the additional supercharging system clearly increases the complexity of the power unit and the coordination. The future will show how this system holds up against the other possible supercharging technologies, including with regard to system costs [15].

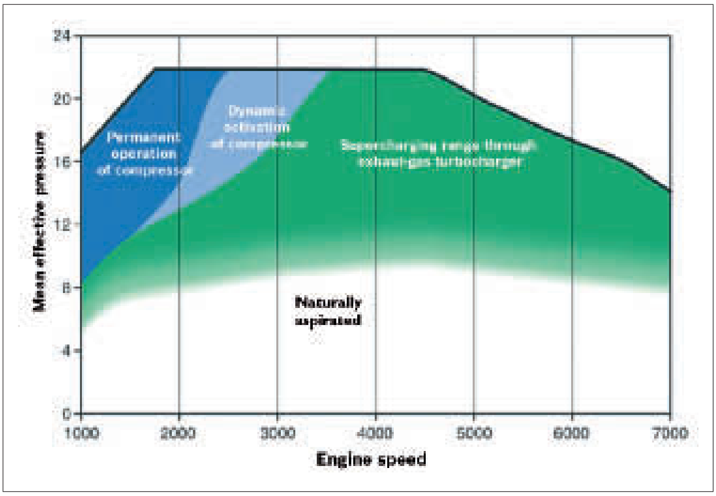

Figure 6.18: Distribution for additional mechanical supercharging [18]

Literature

[1] *Hiereth, H.; Prenninger, P.*: Aufladung der Verbrennungskraftmaschine. Wien, NewYork: Springer Verlag, 2003

[2] *Golloch, R.*: Downsizing bei Verbrennungsmotoren. Berlin, Heidelberg: Springer Verlag, 2005

[3] *Mayer, M.*: Abgasturbolader : Sinnvolle Nutzung der Abgasenergie. Landsberg: Verlag Moderne Industrie, 2001

[4] *Pischinger, S.*: Skript Kolbenarbeitsmaschinen. Rheinisch-Westfaelische Technische Hochschule Aachen, 2006

[5] *Zinner, K.*: Aufladung von Verbrennungsmotoren. Berlin, Heidelberg: Springer Verlag, 1985

[6] *Schmalzl, H. P.*: Aufladung von Pkw DI-Ottomotoren mit Abgasturboladern mit variabler Turbinengeometrie. Dissertation, TU Dresden, 2006

[7] *Gabriel, H.; Lingenauber, R.; Ramb, T.*: Der Turbolader mit variabler Turbinengeometrie (VTG) fuer den neuen Porsche 911 Turbo – Ein Meilenstein in der Ottomotorenaufladung. 11. Aufladetechnische Konferenz, Dresden, 2006

[8] *Sterner, A.; Hofstetter, M.; Kerkau, M.; Beer, M.; Ronneburger, R.; Knirsch, S.*: Die variable Turbinengeometrie fuer die ottomotorische Anwendung beim neuen 3,6 l Biturbo-Motor des Porsche 997 Turbo. 11. Aufladetechnische Konferenz, Dresden, 2006

[9] *Klueting, M.; Missy, St.; Schwarz, Ch.*: Potenziale des strahlgefuehrten Benzin-DI-Brennverfahrens in Verbindung mit Aufladung. 26. Internationales Wiener Motorensymposium, 2005

[10] *Reulein, C.; Kellerer, H.; Schwarz, Ch.*: Methodeneinsatz bei der Potenzialbeurteilung aufgeladener Verbrennungsmotoren. Motorprozesssimulation und Aufladung, Haus der Technik Tagung, Berlin, 2005

[11] *Ardey, N.; Klueting, M; Schmitz, K.*: Aufladung von direkteinspritzenden Ottomotoren – ein Ansatz zur Steigerung der effizienten Dynamik. 10. Aufladetechnische Konferenz, Dresden, 2005

[12] *Borrmann, D.; Brinkmann, F.; Walder, K.; Pingen, B.; Wojahn, J.; Behrends, P.*: Benzineindirektspritzung mit Turboaufladung – ein interessantes Downsizingkonzept. 11. Aachener Kolloquium Fahrzeug- und Motorentechnik, Aachen, 2002

[13] *Fraidl, G. K.; Kapus, P.; Piock, W.*: Otto-Direkteinspritzung mit Aufladung – Die Konkurenz zu dieselmotorischen Antrieben. 26. Internationales Motorensymposium, Wien, 2005

[14] *Hoecker, P.; Jaisle, J. W.; Muenz, S.*: Der eBooster – Schluesselkomponente eines neuen Aufladesystems von BorgWarner Turbo Systems fuer Personenkraftwagen. 22. Internationales Motorensymposium, Wien, 2001

[15] *Lang, O.; Habermann, K.;Wolf, K.; Pischinger, S.*: Anwendung der Zusatzaufladung bei abgasturboaufgeladenen Ottomotoren. 9. Aufladetechnische Konferenz, Dresden, 2004

[16] *Muenz, S.; Schier, M.; Schmalzl, H. P.; Bertolini, T.*: Der eBooster – Konzeption und Leistungsvermoegen eines fortgeschrittenen elektrisch unterstuetzten Aufladesystems. 8. Aufladetechnische Konferenz, Dresden, 2002

[17] *Guilain, S.; Lefebvre, A.; Doleac, L.; Schreiber, G.; Muenz, S.*: Optimization of a small two-staged turbocharged diesel engine. 11. Aufladetechnische Konferenz, Dresden, 2006

[18] *Middendorf, H.; Krebs, R.; Szengel, R.; Pott, E.; Fleiss, M.; Hagelstein, D.*: Der weltweit erste doppelaufgeladene Otto-Direkt-Einspritzmotor von Volkswagen. 14. Aachener Kolloquium, Aachen, Oktober 2005