The TSI with 90 kW – the expansion of the Volkswagen family of fuel-efficient gasoline engines

Abstract
With the 1.4 litre 125 kW TSI engine with direct injection and dual charging, Volkswagen was the first volume manufacturer to introduce with great success a downsizing strategy which reduces CO2 emissions while also increasing driving fun. The new 1.4 l TSI with 90 kW and a single-stage turbocharger starts off the next step in the implementation in the field of cost-effective volume engines.

The special technical features of the engine include the further development of the combustion process, simplification of the injector engineering, meticulously optimised turbocharger in conjunction with optimised charge exchange and a newly developed design for water-cooling charge air. Due to the moderate downsizing of the new 1.4 l 90 kW engine – about a 30 % reduction in displacement compared to the 4V naturally aspirated engine with the same torque – the new engine can be configured with an economical, one-stage turbocharger, fulfilling all the requirements for the introduction of TSI technology in Volkswagen’s volume segment.

The combination of the new TSI technology with the new Volkswagen 7-speed dual clutch gearbox with dry clutches makes possible an improvement of 22 % in fuel consumption compared to the predecessor Golf, completely attaining the target of 139 g CO2/km in the European driving cycle.

1. Introduction

Last year, Volkswagen introduced the dual-charged, direct injection 1.4 l 125 kW and 103 kW TSI engines in the Golf platform with great success. The combination of substantial low-end torque in the compact, highly charged engine and excellent fuel consumption led quickly to the world-wide acceptance of this ground-breaking engine technology for petrol engines.

The TSI downsizing technology in petrol engines, analogously derived from the TDI technology, exploits the power-loss advantage of low-displacement engines to achieve outstanding efficiency and fuel consumption. The combination with direct injection enables high compression and extensively avoids the losses in efficiency in charged MPI engines.

With a moderate degree of downsizing – about 30 % less displacement than 4-valve naturally aspirated engines – very good response behaviour was attained with a one-stage turbocharger. For extremely economical designs with high specific output like the 125 kW TSI, a second, mechanical, charger stage is required to provide excellent low-end pulling behaviour with the appropriately great output potential of 90 kW/l without “turbo lag”.

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The combination of these superior, torque-oriented engines with optimised 6-speed manual gearboxes and especially with the new 7-speed dual clutch gearbox, which is being used for the first time with this new engine, delivers excellent fuel economy and forms the core of VW's CO₂ strategy for petrol engines.

- Application of TDI solutions to petrol engine
- Breaking out of conflict between fuel economy and driving fun
- Downsizing
  - 2.0 l TSI 147 kW (- 29 %*) Turbocharger
  - 1.4 l TSI 125 kW (- 42 %*) Twincharger
  - 1.4 l TSI 88 kW (- 30 %*) Turbocharger
- Direct injection FSI enables compression ratio to be increased by one unit, e.g. from 9:1 to 10:1
- Turbocharging (e.g. in the Golf)
- Supercharging and turbocharging (e.g. in Golf GT Twincharger)

* displacement reduction vs. self-induced engine with comparable torque

Picture 1: Volkswagen’s TSI strategy

2. The EA111 engine family

The basis for the engine downsizing strategy is the EA111 engine family, characterised by the cylinder distance of 82 mm. The engine family, which is employed throughout the world, includes 3 and 4-cylinder engines having 2-valve or 4-valve technology. With displacements from 1.0 l to 1.6 l and specific outputs of up to 90 kW/l, the engine family is ideally positioned for the strong-selling engine segments of volume models. The significance of this engine family for achieving the fleet CO₂ target and for Volkswagen’s downsizing strategy is accordingly great.

The new 1.4 l 90 kW TSI engine has been developed for world-wide application and replaces the economical 1.6 l 85 kW naturally aspirated FSI engine [2] in all vehicles while demonstrating substantial increases in performance and still further significant reductions in fuel consumption.
3. Concept and developmental goal

The mechanics of the new 1.4 l TSI turbocharged engine are extensively based on the 125 kW TSI with dual charging and together they form a common modular building set. Important basic elements relevant to production are designed as uniform components to reduce complexity.

- Cast iron crankcase with open deck design
- Steel crankshaft
- Connecting rods

The main focus of attention during the development of the 1.4 l 90 kW engine in particular lay on the further reduction of friction and the economical realisation of the engine as a very robust volume model with low maintenance costs.

The further development of the combustion process as well as the redesign of the combustion chamber and intake port made it possible to do without tumble flaps, consequently making a substantial contribution to technical simplification and cost reduction. At the same time, fuel consumption, exhaust emissions, performance and smoothness of running are comparable to engines with tumble flaps.
The new TSI 1.4 l with 90 kW (front and rear views)

The compact 1.4 l engine was developed for broad application in all volume models of the Volkswagen Group. The possible applications range from the A0/00 class all the way to the Passat in the B class.

<table>
<thead>
<tr>
<th>Construction</th>
<th>1.4 l 90 kW TSI</th>
<th>1.6 l 85 kW FSI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valves per cylinder</td>
<td>4-cylinders, in-line</td>
<td>4-cylinders, in-line</td>
</tr>
<tr>
<td>Displacement</td>
<td>1390 ccm</td>
<td>1598 ccm</td>
</tr>
<tr>
<td>Bore and stroke</td>
<td>76.5/ 75.6 mm</td>
<td>76.5/ 86.9 mm</td>
</tr>
<tr>
<td>Stroke-to-bore ratio</td>
<td>0.988</td>
<td>1.136</td>
</tr>
<tr>
<td>Cylinder distance</td>
<td>82 mm</td>
<td>82 mm</td>
</tr>
<tr>
<td>Connecting rods</td>
<td>144 mm</td>
<td>138 mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>10.0:1</td>
<td>12.0:1</td>
</tr>
<tr>
<td>Nominal output</td>
<td>90 kW at 5.000 rpm</td>
<td>85 kW at 6.000 rpm</td>
</tr>
<tr>
<td>Maximum torque</td>
<td>200 Nm at 1.500-3.500 rpm</td>
<td>155 Nm at 4.000 rpm</td>
</tr>
<tr>
<td>Spez. torque</td>
<td>143.9 Nm/l</td>
<td>97 Nm/l</td>
</tr>
<tr>
<td>Fuel</td>
<td>RON 95</td>
<td>RON min. 95</td>
</tr>
<tr>
<td>Engine management</td>
<td>Bosch MED 17</td>
<td>Bosch ME7.5.20</td>
</tr>
<tr>
<td>Emissions standard</td>
<td>EU5</td>
<td>EU4</td>
</tr>
<tr>
<td>Gearbox</td>
<td>MQ200GA+ 6F, DQ200 7F</td>
<td>MQ200 6F, AQ250 6F</td>
</tr>
</tbody>
</table>

Picture 4: Technical data TSI 1.4 l 90 kW
As the success of the 1.4 l 125 kW TSI has already demonstrated, the combination of propulsion power and economical fuel consumption is extremely successful in the market. Therefore, the development emphasised the attainment of excellent torque in the frequently used lower engine speed range, motivating the driver to drive at lower engine speeds and simultaneously providing double benefits for the customer.

Due to the moderate goal of 90 kW with a specific torque of 144 Nm/l for the new 1.4 l engine, the chargers and the timing could be designed without compromise to attain good response behaviour at low engine speeds. In this way it was possible to attain such outstanding response even with a one-stage turbocharger.

Picture 5 shows a comparison of torque development of the new 1.4 l TSI engine and the 1.6 l naturally aspirated FSI engine. An increase in torque of up to 66 % was realised in the lower engine speed range. A maximum torque of 200 Nm is available at just 1.500 rpm. The design with extreme low-end torque offers the ideal conditions for the combination of this new engine with long gear ratios. The nominal output of 90 kW is already attained at 5.000 rpm and is constantly available to 6.000 rpm.

![Torque curve of the TSI 1.4 l 90 kW compared to the 1.6 l 85 kW FSI](image-url)
4. The combustion process and Fuel mixture formation

4.1 The combustion process

The combustion process of the 1.4 l 90 kW TSI, based on the 1.4 l 103/125 kW TSI engine, had to be thoroughly reworked due to the absence of the tumble flap as well as to the use of a one-stage turbocharger with a turbine temperature limit of 950°C. The development work focused on one hand, on the further development of the intake port in order to generate a flow inside the cylinder of the intensity necessary for the creation of a good mixture and fast, optimal fuel burn and on the other hand, the adaptation of the exhaust valve timing to produce excellent transient behaviour.

The engine has a stroke of 75.6 mm, a bore of 76.5 mm and a compact combustion chamber with the spark plug located centrally. The geometry of the wide, shallow recess in the piston deck could be adopted from the current series of the 1.4 l TSI. In spite of impressive numbers for full load in comparison to the competition – a specific torque of 144 Nm/l and a maximum mean effective pressure of 18.1 bar over a wide engine speed range of 1,500 to 3,500 rpm (see picture 5) - the engine could be designed for a compression ratio of 10.0:1 with RON 95 fuel and a maximum exhaust temperature of 950°C at charge pressures of up to 1.8 bar absolute.

4.2 The development of the intake port

The current production intake port of the 1.4 l TSI with tumble plate and tumble flap could be designed for charging without compromise. With the tumble flap closed, a very intensive tumble current is generated, which forms stably with only slight cyclic fluctuations [3]. In the absence of the tumble flap, an intake port had to be developed which generates the charge movement necessary for good homogenisation of the mixture and fast fuel burn. The inlet port of the 1.4 l 90 kW TSI was further developed on the basis of the standard TSI intake port with the goal of achieving a substantially higher degree of tumble. For this purpose, the new intake port leads up to the intake valve seat ring significantly flatter and more tangentially. In addition, for the first time in the EA111 engine family, port masking was introduced on the lower side of the intake port immediately before the seat ring in the direction of flow. As may be seen in picture 6, the flow breaks off at the edge of the casting in the port and, due to the flow over the upper side of the intake valve, a stable tumbling charge motion develops.
The intensive use of 3D simulation beginning in an early phase of the project enabled the creation of variant intake ports and finally, from six suggestions, the two versions with the greatest turbulent kinetic energy were selected and cast in cylinder heads. The turbulent kinetic energy is an effective measure of the energy for creating the fuel mixture and is the determining factor in the engine’s combustion behaviour. It is defined as follows:

\[ \text{TKE} = \frac{3}{2} u'^2 \]

where \( u' \) = turbulent fluctuation speed (irregular lateral movements perpendicular to the calculated main flow).

Picture 7 illustrates the turbulent kinetic energy (TKE at 2,000 rpm and full load) variations during the compression stroke of the production intake port with open or closed tumble flap in comparison to the two selected port. When the tumble flap is closed, the tumble macro flow collapses shortly before the compression stroke TDC into micro turbulence and very high TKE values are attained. To a lesser degree, this can be identified for the two tumble intake ports as well. In contrast, the production TSI port with the tumble flap open exhibits no increase in TKE at the end of the compression stroke. For port variant 1, which was implemented for production of the 1.4 l 90 kW engine, about 55% of the turbulent kinetic energy of the production port with the tumble flap closed could be attained for the compression stroke TDC.
As an example of the combustion behaviour of the new 90 kW TSI tumble port compared to a production port, picture 8 illustrates the elapsed time measured in degrees of crankshaft angle from ignition to the point of 50% fuel burn as a function of engine speed at full load. Up to the point at which the tumble flap opens at an engine speed of 2.500 rpm, fuel burn occurs about 4° crankshaft angle faster with the production port; on the other hand, the new tumble intake port is dramatically faster than the production TSI port with the tumble flap open. With the duration of combustion less than 27° crankshaft angle over the entire operating range, the 90 kW TSI tumble port demonstrates a short ignition phase as well as high combustion speeds. Especially at high engine speeds, a short combustion period is essential in order to hold fuel mixture enrichment to a moderate level at a maximum exhaust temperature of 950°C. At the same time, the pressure increase gradient always remains below the acoustically critical limit of 5 bar per ° crankshaft angle.

The defined breaking-off of the flow at the port masking leads to the formation of stably forming flow inside the cylinder. This represents a required condition for the very low cyclic fluctuations at part load as well as full load. At high loads in particular, the standard deviation of the indicated average pressure – as a measure of the cyclic fluctuations – lies below 0.5 bar.
4.3 The influence of exhaust valve timing

As the degree of charging increases, it becomes increasingly difficult to provide sufficient torque for driving off especially with single-stage turbocharging and a displacement significantly under 2.0 l. But with appropriate valve timing, the charge exchange can be optimised for the greatest possible torque at driving-off speeds. To do this, the exhaust valve timing is of crucial importance. However, at low engine speeds and high loads, the conditions with an exhaust cam shaft for which the duration of opening is longer than the firing interval (for 4 cylinders, 180° crankshaft angle) are not ideal. Due to the poorer cylinder charging and the tendency to knock because of the high temperature inside the cylinder, the full torque potential cannot be exploited. Effective purging is not possible due to the excessive pressure. Picture 9 illustrates this explicitly: with the 194°crankshaft angle camshaft, the intake manifold pressure persists at a low level as valve overlap is increased by advancing the intake camshaft.

Picture 8: Combustion time from ignition to 50 % fuel burn at full load
The conditions improve significantly with an exhaust camshaft with shorter duration. To be sure, the exhaust valve is not completely closed in this case either when exhaust valve for the subsequent combustion opens. But, in contrast to the variation with the exhaust cam shaft having longer duration, the backflow of residual gas can be almost completely eliminated. The remaining gas can be effectively purged in the valve-overlap phase. Therefore, the intake manifold pressure rises as the intake camshaft timing is advanced, while only a slight intake manifold pressure can be attained with the longer-duration exhaust cam shaft.

For the required potential output of 90 kW, the exhaust camshaft with the 180° duration offers the best compromise between response at low engine speeds and dethrottling at full load.

4.4 Fuel mixture formation

4.4.1 Injectors

As for the naturally aspirated FSI engines and for the 1.4 l 125 kW TSI, the injector is located on the intake side between the intake port and the cylinder head gasket level. A new type of multi-hole, high-pressure injector with 6 fuel jets (see picture 10) is being used for the first time; the spray cone does not have the typical round or oval shape, but rather, the possibilities for spatial distribution of the spray are optimally exploited. The strongest individual stream, formerly directed downwards towards the piston, has been moved upwards into the middle of the spray. This reduces the moistening of the piston during the very early start of injection at full load as well during as late points of injection shortly before compression stroke TDC in the double-injection catalytic converter heating mode. Consequently, the injection can take place earlier, lengthening the time for forming the fuel.
mixture and improving the homogenisation. The final result are very low hydrocarbon emissions. Furthermore, the introduction of fuel into the engine oil during cold operation is further reduced.

Picture 10: Injector spray design

To avoid undesirable interaction between individual streams, which can lead to an agglomeration of fuel droplets, two individual streams are directed outwards at a substantially greater angle and the uppermost, more strongly in the space between the valves (see picture 11). The injectors from Magneti Marelli also make it possible to construct the length-to-diameter ratio (l/d) of the fuel jets individually for each jet so the penetration can be optimised for each spray cone.

Picture 11: Position of spray cones in the combustion chamber
The figures on the right in picture 10 show in laser cross section (30 mm below the tip of
the injector and perpendicular to axis of injector) the distribution of liquid and vaporised
fuel in the fuel spray under catalytic converter heating conditions, employing the laser-
induced fluorescence (LIF) technique. It can clearly be seen that, in comparison to the
production TSI injector, the spray pattern is wider and asymmetric, with one stream in the
centre. The combination of good partial homogenisation and tuned penetration of the
individual streams form the basis for stable engine operation during double-injection
catalytic converter heating mode with the second fuel injection guided by the recess in the
piston deck with low raw exhaust emissions and extremely fast heating of the catalytic
converter.

4.4.2 High pressure Pump

A new one-cylinder pump is used for the high-pressure fuel injection. The special features
of the new high-pressure pump include full delivery in the no-voltage state (the reciprocal
actuation concept compared to the high-pressure pumps used up till now in the EA111
engine family and the integrated pressure-limiting valve with which the return-free fuel
system could be further simplified.

The geometric design of the high-pressure pump drive with four lobes on the intake
camshaft, 3 mm cam lift and 10 mm piston diameter enable extremely fast pressure
development during a cold start. Thus, during a cold start, just 0.5 seconds after the
beginning of the starter phase, the pressure in the fuel rail exceeds 60 bar. This design
makes the use of the retarded, high-pressure injection throughout the entire cold-start
temperature range both possible and sensible for the first time; a critical phase with post-
start pressure drops is impossible due to this design.

During a high-pressure start, the quantity of injected fuel injected could be substantially
reduced through improved fuel mixture preparation and injection during the compression
stroke. This is reflected in substantially reduced hydrocarbon emissions, in reduced fuel
consumption and likewise in a significant reduction in the introduction of fuel into the
engine oil. The start, optimised for emissions and fuel consumption, is complemented by a
double-injection process in the post-start.
Picture 12: High-pressure injection system with HPP3