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Dipl.-Ing. Werner Bender, Dr. Bertold Engels

VTG Turbocharger for Heavy □
Duty Commercial Diesel □
Applications with High Braking □
Performance

Academy



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1. INTRODUCTION

The requirements of modern truck engines increase with each step of the emission legislation. In the same way the charging system as a part of the air management system has to be improved. This leads to more and more complicated solutions such as Regulated – Two – Stage (R2S), combinations of super- and turbochargers (e.g. eBooster) and turbos with Variable Turbine Geometry (VTG).

In particular the VTG is a solution of highest interest because it is able to fulfil most of the required development objectives for commercial diesel engines as

- Higher torque (boost pressure) at low engine speeds
- Better transient response (drivability)
- BSFC improvement
- Drive EGR at part and full load to reduce emissions

An additional advantage compared to other systems is that the installation dimensions are nearly the same as the standard (waste-gated) turbocharger.

Never the less this excellent tool for a controllable air management system, even after two decades of development never found access to the large truck fleets world wide! During all steps of emission legislation many customers showed interest and made tests with the product, but in the end nearly always a maybe less comfortable but cheaper solution was introduced to series production because VTG was a „ nice to have“ and not a „must“.

To launch the VTG, an additional feature to increase the benefit for the customers was required. As a new objective, the use of the VTG for engine braking was defined. The prospects of additional benefits for the customers were:

- Replacement of state-of-the-art engine brake systems (brake flap, Jake Brake)
- Reduction of total system weight
- Increase of brake power at low engine speeds
- Increase of maximum brake power
- Option of replacing retarders at high engine speeds

The requirements of several customers are summarised in the following charts:

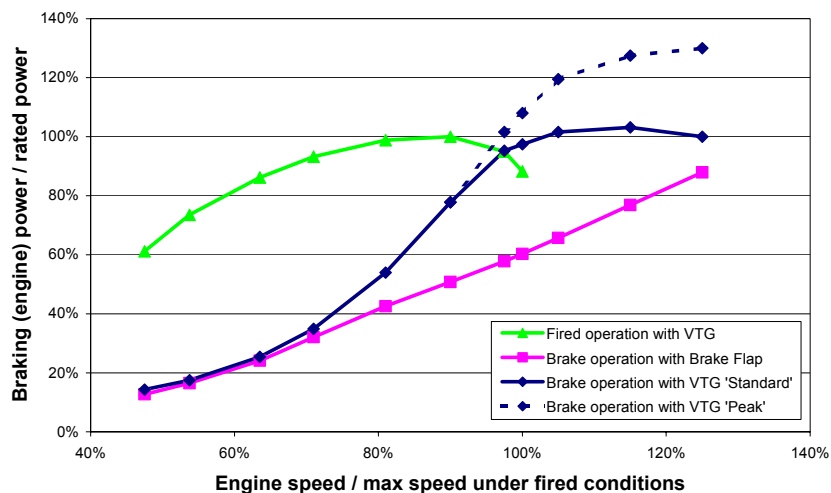


Figure 1a: Comparison of braking performance and rated power

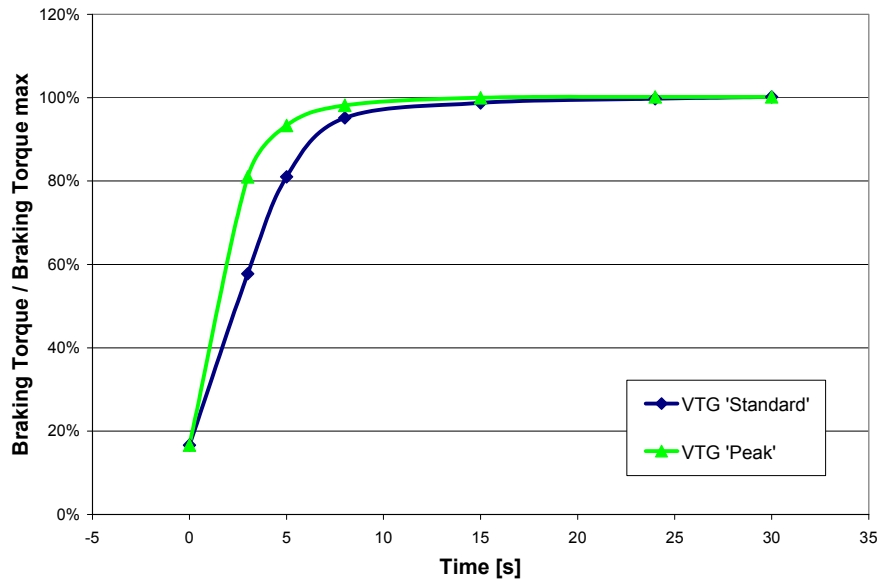


Figure 1b: Time to build up braking power

This means in figures:

- Ratio rated power / maximum brake power 1,00 (1,30)
- Increase of brake power at 60% - 90% of maximum engine speed 50 % (80%)
- Time to achieve 80% of brake power at 40% - 100% of maximum engine speed 5 sec (3 sec)
- Share of braking cycle to total operating time 10% (20%)

The first values show the average requirement, those in brackets the peak targets.

To achieve these target values for the engine, the turbocharger has to fulfil the following requirements under braking conditions:

- Mean back pressure of the turbine 5 bar (7 bar)
- Peak back pressure of the turbine 8 bar (11 bar)
- Time for one full stroke of the actuator 0,3 s (0,2 s)
- Number of actuations in life of engine (millions)
 - full strokes 0,6 (1,0)
 - part strokes 4,5 (9,0)

For this a new development was started in 1996.

2. DESIGN CONCEPT OF BORG WARNER'S VTG FOR CD

The VTG described below is based on a Standard turbocharger. The compressor side (1) is practically the same; the only difference in the core assembly (2) is the additional speed sensor (3), which is fitted to the center housing for permanent control of the turbocharger speed.

The actual VTG is based on a swing – vane design, where the performance and the mass flow of the turbine is influenced by changes of the flow angle to the wheel by means of moving the vanes.

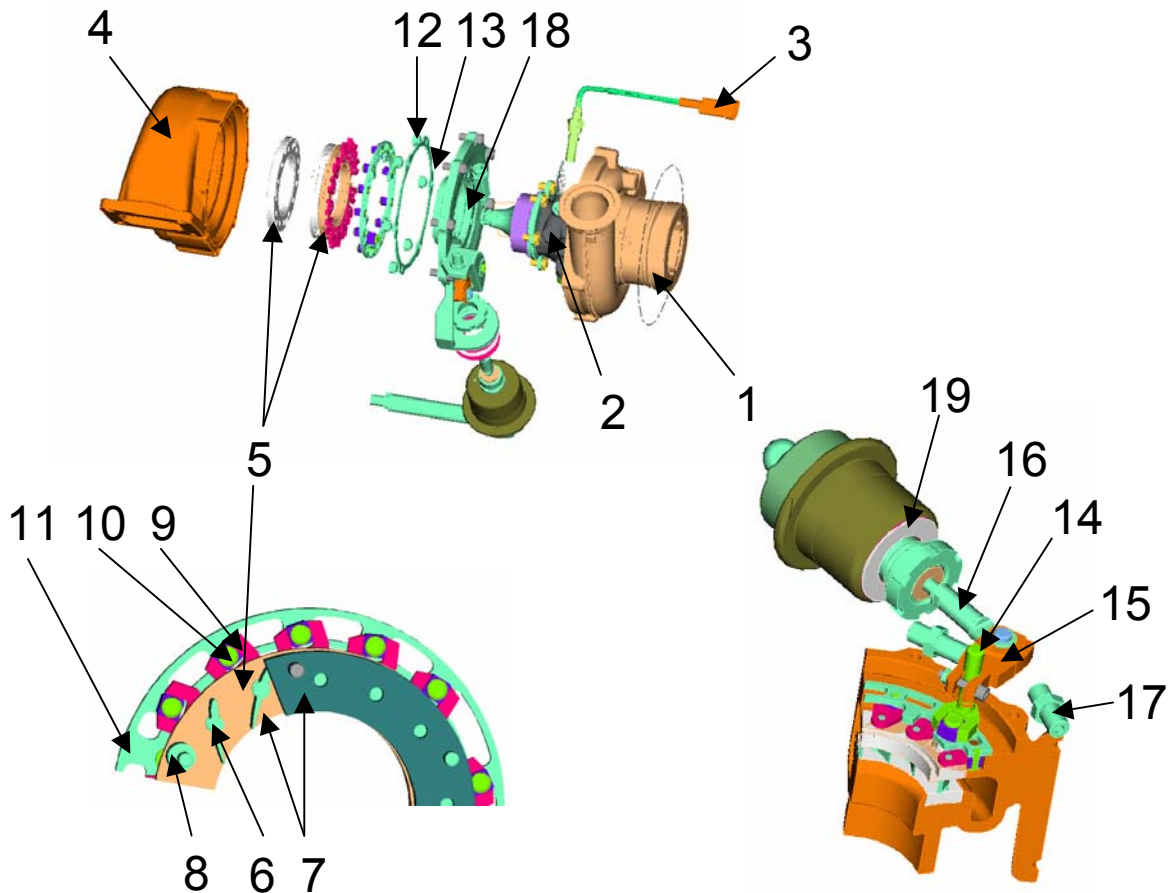


Figure 2: Exploded view to VTG turbocharger

The design features are as follows:

- Single flow turbine housing (4) for maximal flow
- Vane ring assembly (5) consisting of 13 dual – axis curved vanes (6), pivoted between an inner and outer vane ring (7). Three spacers (8) make sure that minimal axial clearances can be achieved.
- Vane levers (9) for moving the vanes designed as a fork. Adjustment is done by pivoted sliding blocks (10), which are positioned in the fork.
- Adjustment ring assembly (11) having 13 small sliding blocks for actuating the vanes and a large one on the opposite side to actuate the whole ring. The adjustment ring is centered by six rollers (12), which are equally spaced by a cage (13).

- Adjustment shaft (14), which transmits the actuating forces via adjustment lever (15) to the actuator rod (16). The fork connection to the large sliding block is similar to the small sliding block / vane fork interface. Two flats situated between a pair of bearing bushes accommodate the action of the adjustment lever.
- Adjustment lever connected to the actuator rod with a bolt and two curved areas to define the contact to the adjustable fixing screws (17) which define the stop for maximal and minimal vane position.
- Actuator flange (18) which closes the turbine stage and allows fixing of the core assembly. The flange also includes a mounting bracket for the actuator (19) and contains the bearing of the adjustment shaft and the stops.

3. TURBINE HOUSING / ACTUATOR FLANGE

3.1 Technical Challenges

The turbine housing and the actuator flange contain the whole VTG mechanism. Therefore, the prime task for these two parts is to avoid sticking of any of the moving parts. This must be assured at low and high temperatures and at transient temperature changes combined with extremely high back pressures, which occur simultaneously under braking conditions. Under the same conditions a second task is also important and difficult to achieve: To avoid losses by leakage, the two parts have to keep sealed at all contact areas.

If the design of the housing and the flange is not correct, a damage mechanism will occur as follows: Due to the high temperature changes between fired and braking mode the turbine housing will extend extremely. If the temperature distribution is inhomogeneous because the material distribution is, then high stress will occur at locations with high mass concentration. Additionally due to the high back pressure under braking conditions the parts will extend. A weak actuator flange could move like a diaphragm.

Under these cyclic stress conditions cracks will emerge and increase until the stress is released. This leads to distortion of the turbine housing and also of the vane ring assembly. In consequence of this either the vanes will stick or, if the actuator force ($> 1000 \text{ N}$) can keep them moving against the resistance force, all parts of the whole mechanism will be rapidly worn out.

The distortion of turbine housing and actuator flange will also lead to high leakage and hence braking power will decline.

3.2 Technical Solutions

Based on a measured temperature field a FEA analysis was done. The result showed that the expansion from the tongue area of the turbine housing to the opposite volute area was about 2.3 mm larger than in the perpendicular direction! Similar effects in axial direction could be shown at the tongue area compared to the free volute. The bending of the actuator flange from inner to outer diameter could be more than 1 mm.

The design of the turbine housing was optimized by reducing the material concentration in the tongue / inlet flange area and decoupling it from the volute. Radial arranged ribs stiffened the actuator flange. The simulation for the final design showed a maximal difference in expansion of 0.5 (0.8) mm.

Both parts have to be treated with a special annealed heat process to eliminate residual stress of the casting.

An additional option is the use of chromium – nickel based alloy.

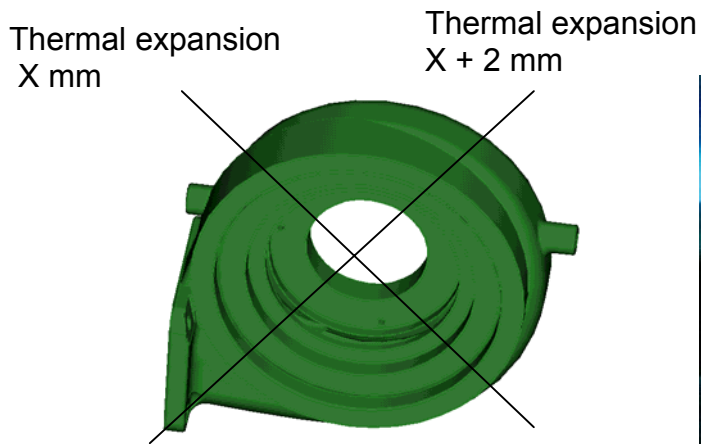
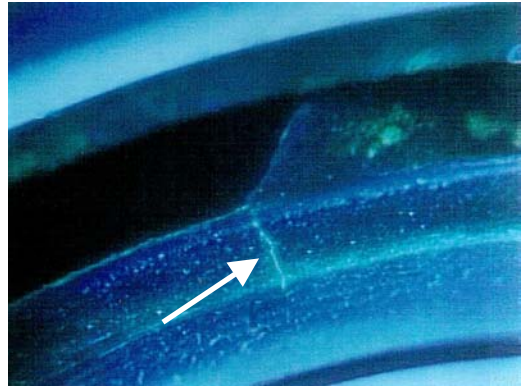


Figure 3a: Main expansion directions



Cracks after thermal shock test

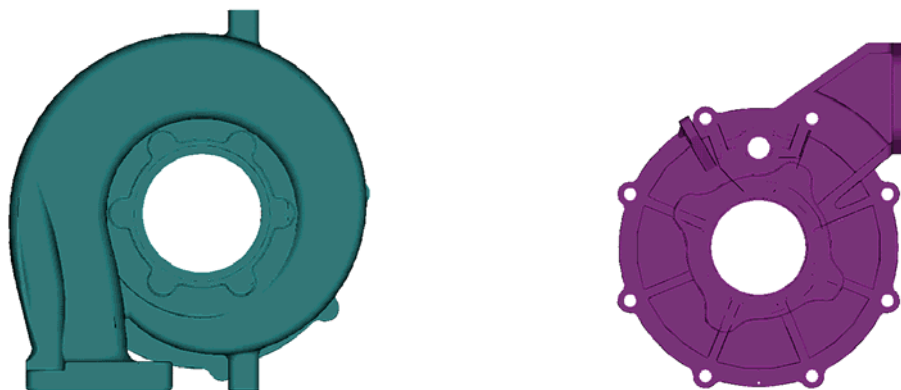


Figure 3b: Optimized design of turbine housing and actuator flange

3.3 Validation Programs and Results

To validate the improvement an accelerated durability test was defined. Two units were arranged as a tandem on a gas stand. A swing flap produced a cyclic load of 700 °C hot gas and 20 °C cold air alternately to both VTGs.

The result was outstanding: While the first design after only 150 hours testing had cracks in several areas up to 30 mm length and still growing, the final design had no cracks in the critical areas after 500 hours testing.

4. VANE RING ASSEMBLY

4.1 Technical Challenges

The vane ring assembly is the central component of the VTG. It has to create good performance both under fired and under braking conditions. The vanes are directly confronted with the exhaust flow, so they see the highest temperatures and the highest pressures including pulsation. Under these conditions the following critical items have to be considered:

- Fret on the axle of the vanes due to static gas forces
- Wear on the axle of the vanes due to dynamic gas forces
- High-frequent vibration-friction-wear caused by fast direction changes of the force in torque free vane positions

For the design of the vane will also basically generate the forces to all the other parts of the mechanism, it has to be considered very exactly.

4.2 Technical Solutions

The static forces are not a real problem, as long as the vane is freely movable in the vane rings and there is no enforced contact. This can be achieved if the positions of holes for the two axles of each vane are concentric. For this the inner and outer vane ring are machined in one operation.

The dynamic forces, especially for the first two or three vane downstream the tongue, are very critical. Combined with very hard sedimentations of adhesives of the fuel in the bore of the vane ring, abrasive wear can occur. There are two ways to improve this situation:

- Change the design of the tongue in such a way that the exiting forces are lower. This can be achieved by changing the distance to the vanes, the flow angle to the vanes or by distributing the forces in circumferential direction.
- Change the combination of material to better tribological features. So a material, which creates an oxide film on the surface will reduce sedimentations, and a partner with sulphur as a solid lubricant gives an additional improvement.

Both measures can be combined.

High-frequency vibration-friction-wear cannot be solved by tribological improvements. This will only decelerate the wear. To avoid wear, the high-frequency vibration has to be eliminated. This can be solved by a design where in every vane position the torque of the flow forces operates in one direction. For the fail-safe mode requires an open vane position, the pivot axle has to be positioned in front of the balance axle.

4.3 Validation Programs and Results

In this case the validation had to be done on the engine under real (braking) conditions. A high cyclic program with a share of more than 30 % braking was run for 750 hours. The first design showed in this test wear of up to 1 mm on the axis of the vanes. The results were satisfying with a unit, which included all above mentioned measures and a vane design shown in figure 4b. The position of the pivot axle made sure that there is no change in force direction, but the actuating forces got very high. Figure 4c shows the design that gives the best compromise between direction and magnitude of the force.

Overlapping of next vane

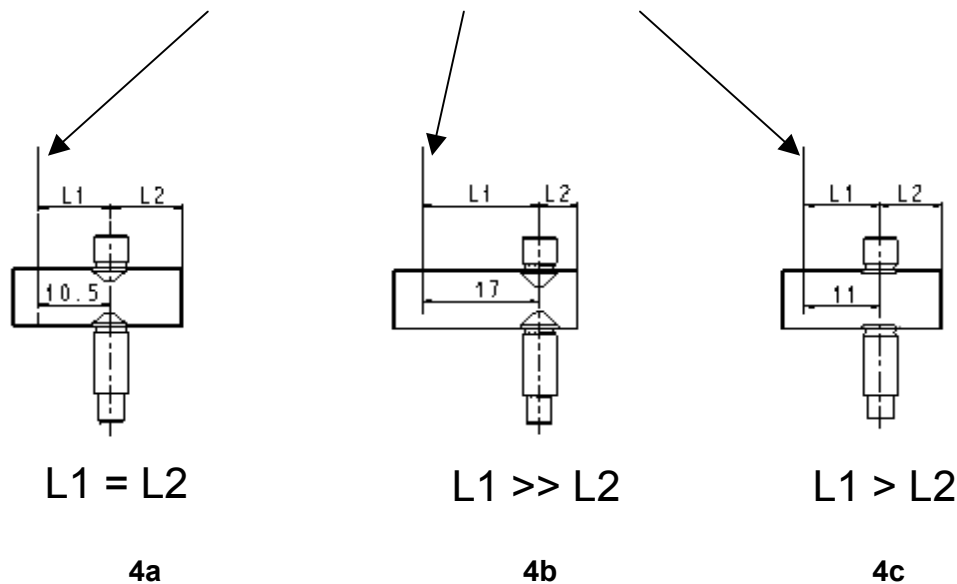


Figure 4a – 4c: Comparison of vane designs

5. ADJUSTMENT RING ASSEMBLY

5.1 Technical Challenges

The adjustment ring assembly is the link between the vane ring assembly and the adjustment shaft. It has to make sure that the actuating forces are transmitted with low hysteresis and no wear and that the vanes all have exactly the same position especially in nearly closed positions for braking mode.

This results in two main challenges:

- To achieve low hysteresis and no wear the adjustment ring has to be as weight reduced as possible.
- To ensure exact vane positions the ring has to keep its flat and circular shape even after welding of the pins for the sliding blocks.

5.2 Technical Solutions

It is clear that both challenges can only be solved together. It is easy to reduce the area of the full ring disc to the minimum of the required contact areas. Also the manufacturability is no real problem.

But the welding of the pivot pins is tricky. Since hot spots cause distortion, a low energy welding process is required. Even with electron beam welding the required flatness and roundness is not easily achievable. Only in a very close collaboration with the manufacturer of the ring and the welding company the process and the design can be advanced to the target.

5.3 Validation Programs and Results

The mass optimized adjustment ring design is shown in figure 5. The weight reduction was about 40 % compared to the basic full disc version.

With an electron beam process which includes compensation spots on the opposite side of the welding of the large sliding block, flatness and roundness of < 0.35 mm is achievable. In the validation process the adjustment ring assembly passed the same engine test as the vane ring assembly successfully.

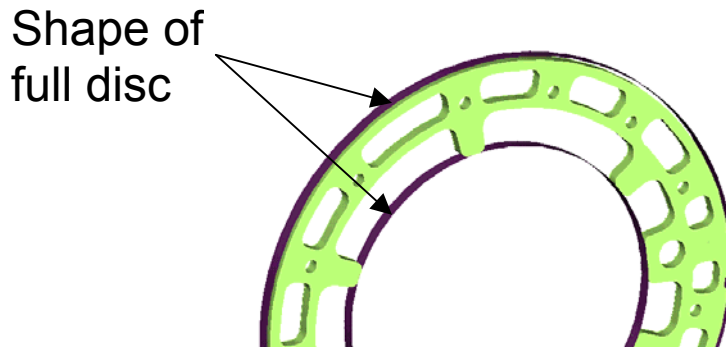


Figure 5: Optimized design of adjustment ring

6. ACTUATOR

6.1 Technical Challenges

The actuator of a VTG has to fulfil a much more demanding specification compared to wastegated turbochargers in commercial diesel:

Feature	VTG	Wastegate
• actuation stroke	24 mm	4-5 mm
• control pressures	6,5 bar	3 bar
• spring forces	2000 N	450 N
• full strokes	1 million	20.000
• part strokes	8 million	700.000

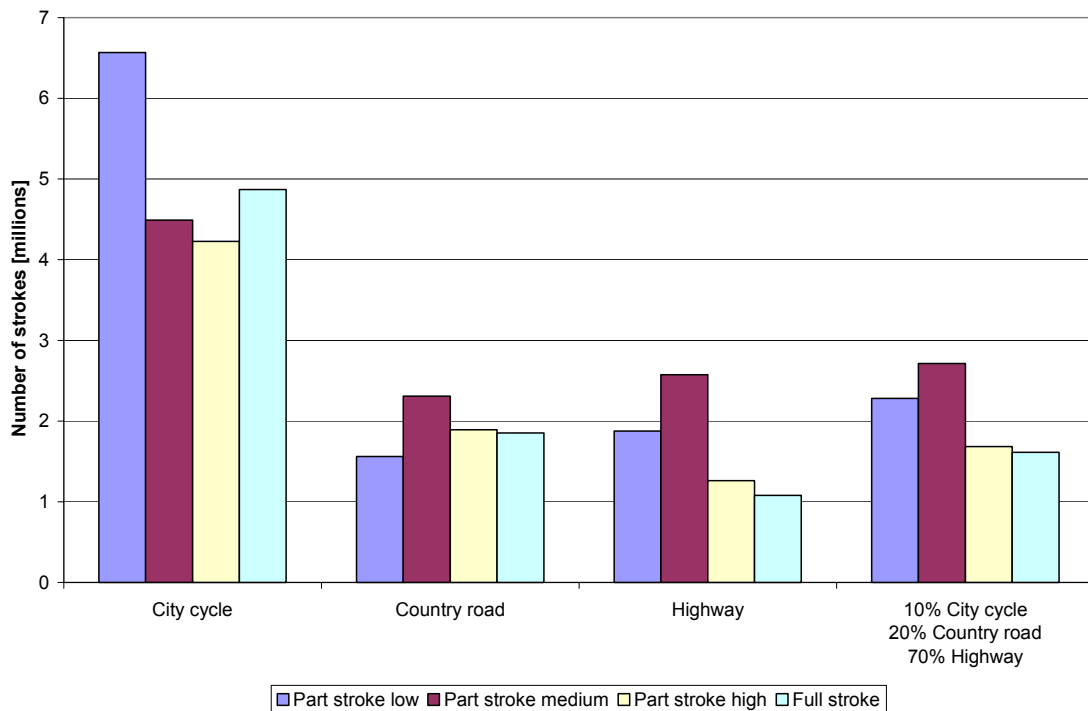


Figure 6: Measured actuator cycle for heavy-duty truck

6.2 Technical Solutions

The higher pressure and the higher spring forces require a more solid design. This is achievable with a similar design as for wastegate but with a thicker metal sheet for the cover.

The real challenge is the lifetime of the diaphragm with strokes of up to 24 mm. The production process with the fabric on top of the silicon (lay up) as used for the standard wastegates gives only a lifetime of 500.000 up to 800.000 full strokes. To increase the lifetime by a factor of about 10, a totally new manufacturing process for the diaphragm is required. The cloth diaphragm consists of a special fabric, which is positioned in the middle of the elastomer, which allows homogeneous stress behaviour. Pleats, which are the basic reasons for failures of the diaphragm, are postponed fundamentally. Hence life is increased!

6.3 Validation Programs and Results

To validate the improvement an accelerated durability test was defined. Several complete actuators were arranged in a test rig and operated under the following conditions:

- test pressure 0 – 7 bar
- test frequency 0.8 Hz
- test temperature 120 °C
- actuation stroke 24 mm

A lifetime of minimum 4.5 million full strokes was achieved. A conversion to a mixed cycle as shown in figure 6 gave a safety factor of 1.3. An option is a linear operating actuator, which gives additional 20% of lifetime.

7. PROSPECTS OF THE MARKET/FUTURE DEVELOPMENT STEPS

The above described development features are part of a comprehensive validation program which exceeded 15.000 hours of gas stand and engine testing and 2.000.000 km on trucks. Currently more than 300 trucks increase field experience day by day. A series application will be launched in Europe early in 2002; SOP for a matching with EGR that fulfils EPA 02 in the US will be October 2002.

Concerning the prospects of the market for the VTG in the view of the upcoming emission limits Euro 4 and Euro 5, the following statements can be made:

- None of the engine manufacturers knows today which will be the main route to fulfil these legislations: EGR, SCR or PM-filter
- All of these possibilities could be well supported by VTG

EGR: VTG generates sufficient back pressure to drive EGR at full load

SCR: VTG enables conversion at part load by increasing exhaust temperature

PM filter: VTG enables regeneration of filter by increasing exhaust temperature

- The drivability will be improved fundamentally in all cases

This means that the VTG is an interesting tool also for future applications. Now the durability issues have been solved, to increase its attraction future development steps will focus on cost reduction and efficiency increases. Cost reduction items are simplified designs and new production processes as for example metal injection moulding (MIM). Efficiency can be increased primary by reduction of the losses caused by all the clearances of the mechanism and especially of the vanes. Besides a reduced BSFC, an increased braking performance at medium engine speed would allow a replacement of existing large and permanent operating

retarders by smaller, discontinuous working units which would be a huge cost reduction of the system.

8. SUMMARY

- VTG for Commercial Diesel is now a well developed charging system
- Durability under braking conditions has been validated
- VTG is also an interesting tool to fulfil the next emission legislation steps
- There is a big potential in further development steps to increase the benefit to the customer

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BorgWarner Turbo Systems
Worldwide Headquarters GmbH
Mannheimer Strasse 88
67292 Kirchheimbolanden / Germany
Phone: ++49 (0)6352 75 33 0
Fax: ++49 (0)6352 75 33 99

3K-Warner Turbosystems GmbH
Mannheimer Strasse 85/87
67292 Kirchheimbolanden / Germany
Phone: ++49 (0)6352 403 0
Fax: ++49 (0)6352 403 1866

BorgWarner Turbo Systems Ltd.
Euroway Industrial Estate
Bradford BD4 6SE
West Yorkshire / UK
Phone: ++44 1274 684 915
Fax: ++44 1274 689 671

BorgWarner Turbo Systems
PO Box 15075
Asheville, NC 28813/USA
Phone: 001 828 684 4000
Fax: 001 828 684 4114

BorgWarner Automotive Brasil Ltda.
Estrada da Rhodia Km 15
P.O. Box 6540
13084-970 Campinas-SP / Brasil
Phone: ++55 19 3787 5700
Fax: ++55 19 3787 5701

Hitachi Warner Turbo Systems Ltd.
3085-5 Higashi Ishikawa Saikouchi, Hitachinaka-shi
Ibaraki-ken
312-0052, Japan
Phone: +81 (0) 29-276-9388
Fax: +81 (0) 29-276-9397

www.turbos.bwauto.com

