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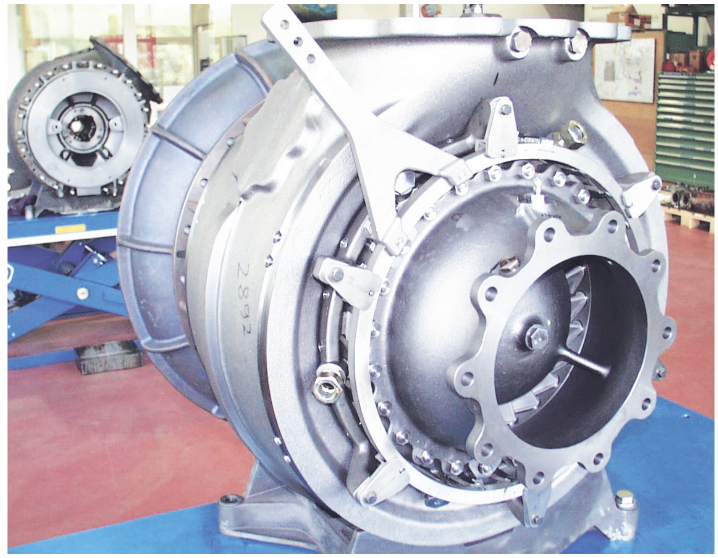
Field experience with variable turbine geometry on ABB turbochargers

Abstract

This paper presents findings from field experience with ABB turbochargers with integrated variable turbine geometry (VTG), i.e. nozzle ring modules with movable vanes. The first serial radial type turbocharger from ABB with VTG, the TPS57, was released for production in 1998. In 2004 VTG technology was also introduced for axial-type turbochargers. In the meantime, over 700 radial-type turbochargers and over 110 axial-type turbochargers with VTG are in operation. Of the radial-type turbochargers, 28 units have passed the 50’000 running hour mark, while the first of the axial turbochargers have also run up over 25’000 running hours. The field experience gained with this population is continually analysed and an overview of the past 10 years with over 1000 standard overhauls is given in this paper.

Also described is an innovative evaluation method for the in-depth analysis of the mechanical behaviour of the movable parts of a variable turbine geometry module running on an engine. This method combines an analytical evaluation of the VTG blade movement with measuring data collected on running engines. Presented results from these evaluations show a good correlation between calculations and engine measurements. As a result, reliable prediction of the wear behaviour of the VTG blade bearing parts has become possible.

Finally, the paper highlights the potential of VTG-technology in combination with emission reduction measures, such as exhaust gas recirculation (EGR) and selective catalytic reaction (SCR), required to satisfy the tightening environmental regulations according to IMO Tier III on 2-stroke and 4-stroke diesel engines.



Design

The working principle of the initial TPS VTG unit is shown in Fig. 1 with reference to a single guide vane. The VTG consists of several blade units, aligned in a circumferential direction so as to form the complete guide vane assembly upstream of the radial turbine wheel (not shown in Fig. 1). The adjustable guide vane (1) is supported by two guide bushes (2) positioned within the bearing housing (3) of the turbocharger. The angular positions of the VTG blades are set by adjusting levers (4) which are rotated around the blade centre line by moving the external slotted ring (5) in a circumferential direction. The slotted ring is usually positioned by means of electro-pneumatic actuators.

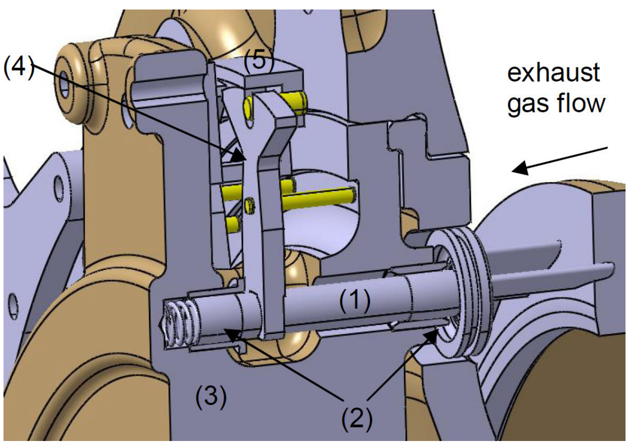
During the initial development phase of the VTG,the unit was tested on selected gas engines in order to identify possible weaknesses and obtain first field experience with prototypes. As the VTG mechanism is supposed to be adjusted during the engine’s operation, relative movements between the VTG parts are in any case inevitable. These small movements could cause high wear friction (fretting) on any contact surfaces of the different VTG parts and lead to deterioration of the unit. Understanding the relevant mechanisms is a key part of the VTG design process and is essential in order to fulfil the service requirements (exchange intervals) and ensure the operational integrity of the product.

Qualification of moving parts

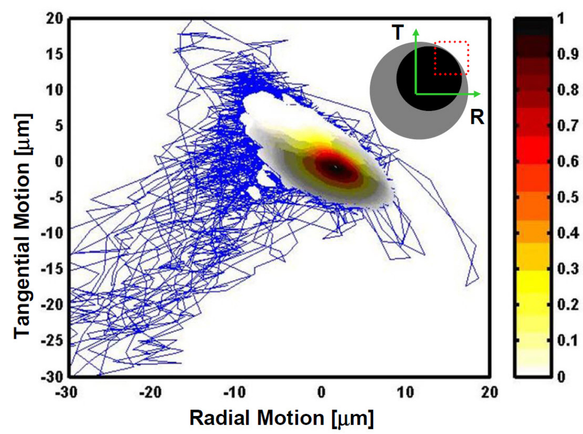
Today, TPS57 and TPS61 turbochargers are optionally equipped with VTG technology that has been optimised in several steps in the last few years. These developments are all based on analytical and experimental studies as well as field experience.

Key VTG know-how was gained from time-resolved measurements of the guide vane motion as well as pressure pulses on the blade profile of the VTG unit at full engine load. As wear (fretting) is always a function of relative motion and associated friction forces the dynamic movement of all VTG parts needs to be measured for the optimized design (Fig 2). The pressure fluctuations occurring in the exhaust gas channel of the engine and turbine inlet as well as the vibration of the turbocharger were measured by means of heat resisting displacement sensors as well as miniature piezo-electric pressure transducers. The calibrated capacitive sensors (two plane measurement) cover a range of 100 microns at a resolution of 5 microns in order to capture the radial and tangential motion of the shaft within the two guide bushes. Additional sensors in the axial direction capture the remaining degrees of freedom of the different VTG parts.

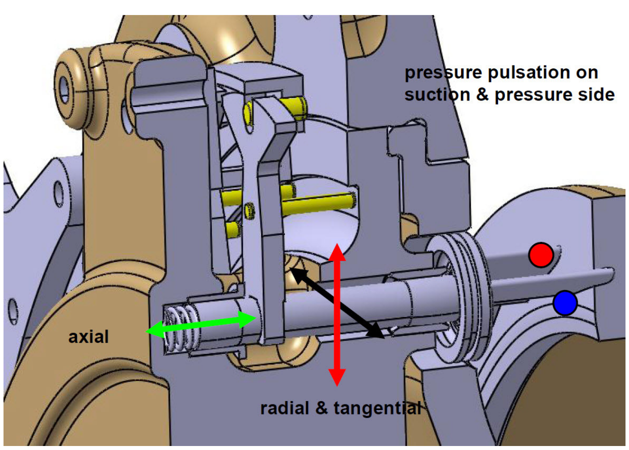
Based on these measurements, the unsteady shaft motion is then associated with the turbocharger's vibrations as well as the pressure fluctuations on the blade profile (pressure and suction side) of the VTG, as plotted in Fig. 3, for the initial version of the TPS VTG design. The diagram is plotted at full engine load for a measuring time of 0.5s and shows the effective path (blue line) of the guide vane shaft's axis within a representative guide bush clearance of 0.1mm. The probability function of the shaft's axis position is added as a contour plot. A probability of 1 denotes



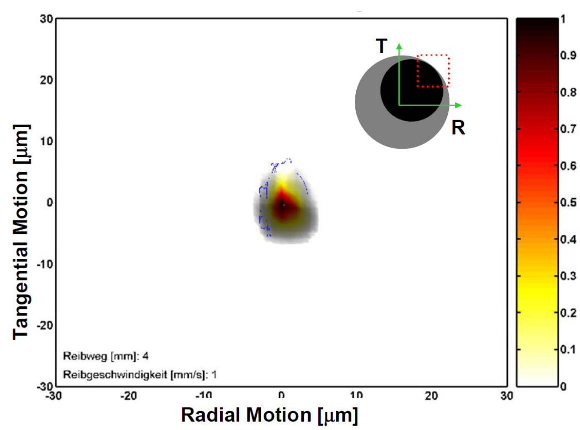
1 Initial design of TPS-VTG turbocharger.



3 Motion of VTG shaft within guide bush at full engine load within 0.5s (initial design).



2 Time resolved measurement of VTG blade motion and pressure pulses on blade profile.



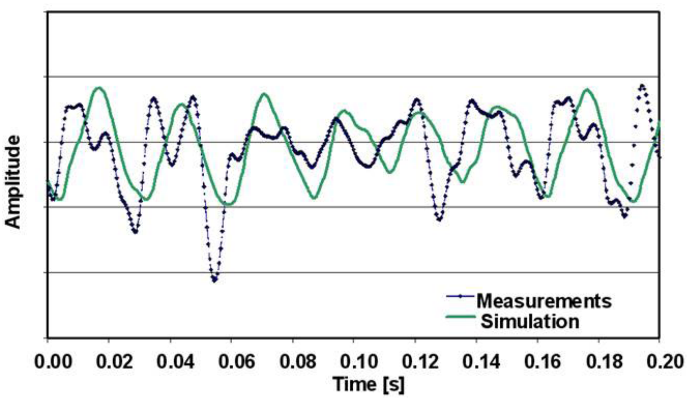
4 Motion of VTG shaft within guide bush at full engine load within 0.5s (final VTG design).

the most probable position of the axis within the guide bush for a given load on the VTG blades (in this case in the upper right hand corner). The blue line reveals a strong relative motion of the shaft within the guide bush with associated high level of wear. This pattern can lead to strong fretting within the VTG unit, causing functional failures if it is not taken into consideration in the VTG design.

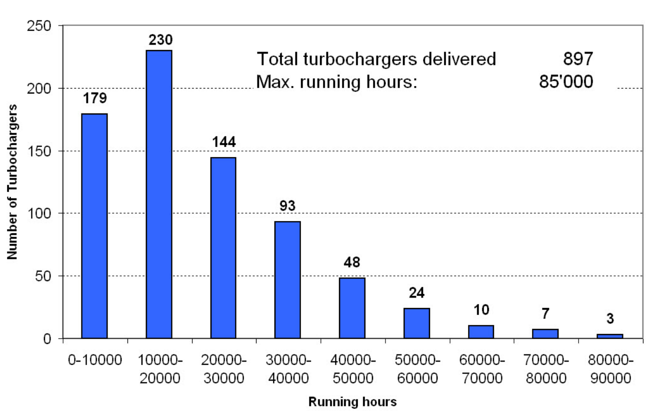
Consequently, different improvements were applied and tested to reduce the guide vane movement. With the final TPS VTG design a motion pattern according to Fig. 4 has been achieved with considerably reduced movements and thus less wear. The final design also achieves the required service exchange interval and ensures the mechanical integrity of the entire unit.

Parallel to the measurements, an analytical model for calculating the friction losses was defined. The model consists of a spring-mass-damper which was calibrated with the measured parameters in order to extract realistic damping factors. The graph in Fig. 5 shows the correlation between the measured and calculated guide vane movement. The damping factor has been adjusted accordingly. The measurements show higher order amplitudes that are not fully captured by the simple spring-mass-damper model. The first order effect is, however, well predicted by the model and is sufficient for the following evaluation.

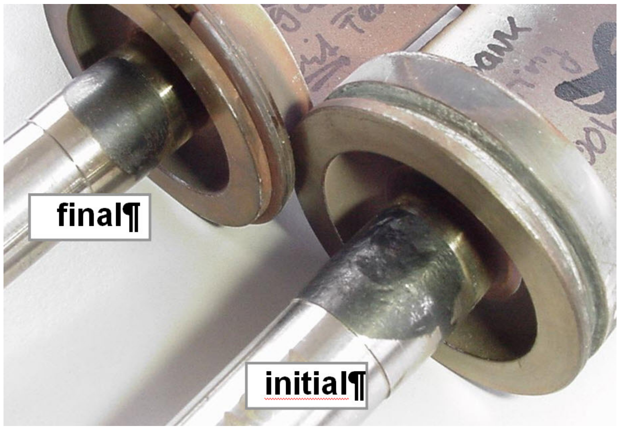
The non-linear partial differential equation is solved by means of numerical tools in order to deduce a representative friction force between the shaft and the guide bush as well as the friction power. As wear is a function of friction power, the calculation enables a quantitative definition of service intervals for the TPS-VTG. Hence, the calibrated vibration model can be applied to any engine application for deducing service intervals as a function of engine operating conditions.



5 Comparison of VTG shaft motion / simulation versus measurements.



6 Field experience of radial type VTG.



7 Wear (fretting) on shaft of VTG guide vanes after 2'900h running hours for different VTG designs.

Field experience

A total of 897 TPS-VTG turbochargers (two frame sizes, TPS57 and TPS61) have been installed on different gas engine types. Of these at least 700 are already in commercial operation. 28 units have accumulated over 50’000 running hours (Fig. 6). Over 1000 turbocharger service jobs have been performed in different countries around the world. Together, the large number of accumulated running hours and extensive service experience provides a very representative amount of field experience for radial VTG turbocharger technology.

Finally, a comparison of wear for two different VTG guide vane shafts is shown in fig. 7: On the right is a blade from the initial VTG design and on the left one from the final design. Both blades have accumulated 2’900 running hours on a gas engine at full load. A remarkable reduction in wear on the shaft of the final design can be observed. Fretting problems on the contact surface to the guide bush as well as material defects on the shaft are avoided. This result is in line with the conclusions derived from the comparison of shaft motion in Fig. 3 with that in Fig. 4. Since wear can also occur on the adjusting lever of the VTG blade if the relative motion between the different VTG parts is not taken into account, a key target in the design of the VTG is to minimize any relative motion between the movable parts. Thus, various design versions as well as material optimisations have been performed for the TPS VTG design and subsequently examined in several field experiments and qualification tests. The available field experience has finally proven that the applied measures result in a robust and reliable design.

Axial type VTG: working principle and qualification

Working principle

The TPL65VA (Fig. 8) is based on the fixedgeometry TPL65 and contains many proven components and subassemblies – e.g. the compressor side, the bearing system and the turbine wheel – of this reliable platform. Also, the field experience that was available from the earlier TPS turbocharger was considered in this design, and will be discussed in a later part of this paper. The mounting dimensions are the same as for the standard TPL turbocharger, ensuring a high degree of interchangeability between the fixed and variable turbine geometry TPL turbocharger.

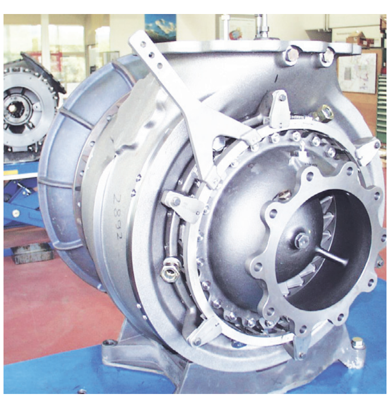
The variable turbine geometry has been integrated in a single module (Fig. 9) to enable it to be fitted to the turbocharger as one unit.

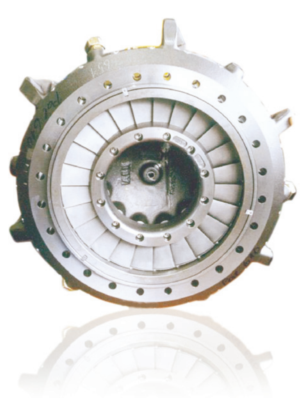
The nozzle vanes are supported by two bearings in the VTG support ring. Each nozzle vane is connected to the actuating ring by means of a vane lever. The ring itself can rotate within six supports, so that external movement of the actuating lever will cause a proportional rotation of the vanes.

Performance

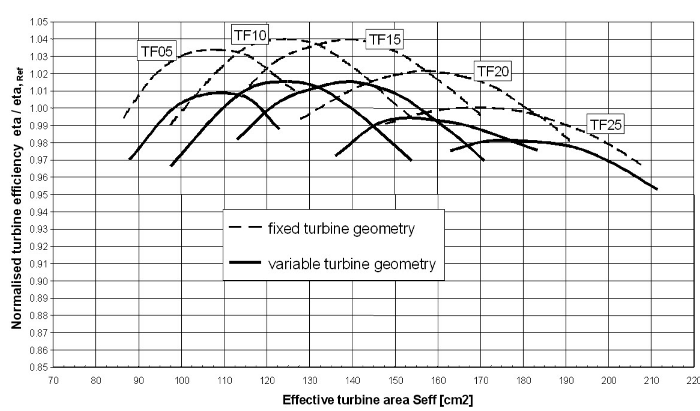
Performance measurements were carried out on the TPL65VA turbocharger for a wide range of flow rates. Turbine efficiency is between 1% and 3% lower than with the TPL65 turbocharger with fixed geometry (see Figure 10).

These losses are due to the sealing air and the inherent gaps in the variable turbine geometry. The more the nozzle blades are closed, the higher these losses will be. In general, the low level of losses points to the soundness and efficiency of the concept.





8 **TPL65VA,** 9 **VTG module for the TPL65VA**



10 Turbine efficiency of the TPL65 with fixed and variable turbine geometry PIT=2.5 (TFxx refers to the stagger angle of the turbine blades).

Mechanical qualification

To ensure reliability, all new turbocharger series developed by ABB Turbo Systems Ltd must undergo a stringent qualification test program at the company's own turbocharger test laboratory in Baden/Switzerland. The most important tests on the variable turbine geometry are:

- Thermal cycling test on the turbine casings and all static parts of the VTG module
- Mechanical cycling tests on the variable geometry components
- Blade strain measurements

Starting and stopping of the diesel or gas engine cause material temperature gradients in the turbocharger. Experience shows that transient temperature differences can produce high stresses in the hot casing parts which may lead to deformations and/or cracks and subsequent failure. To reduce thermally induced stress, ABB Turbo Systems Ltd carried out finite element analyses during the design process, and these were used extensively in the case of variable turbine geometry.

To experimentally verify its mechanical integrity, the TPL65VA turbocharger was subjected to 1,000 cycles in 10 minutes at turbine inlet temperatures between 270 °C and 650 °C. Subsequent inspection of the turbocharger showed that the casings and the VTG module were free of cracks and that the entire variable turbine geometry mechanism was fully operative.

Mechanical cycling tests of the variable parts

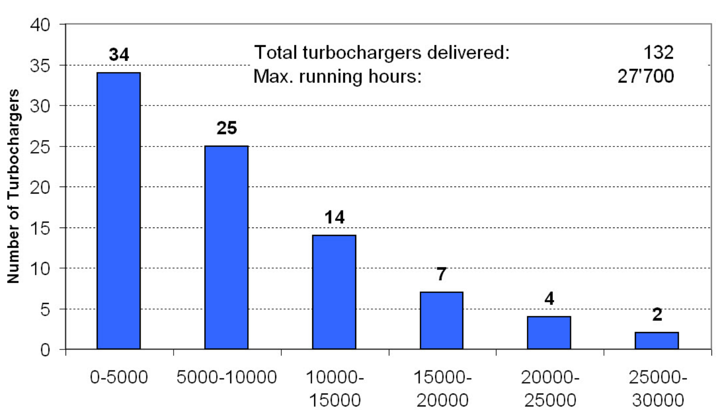
The nozzle vanes were rotated through their full range by means of external movement of the actuating lever. 200,000 cycles were accumulated at a turbine inlet temperature of 700 °C. A subsequent inspection showed no significant wear or deformation. The variable turbine geometry retained its full range of movement. The required force at the actuating lever remained constant.

Blade strain measurements

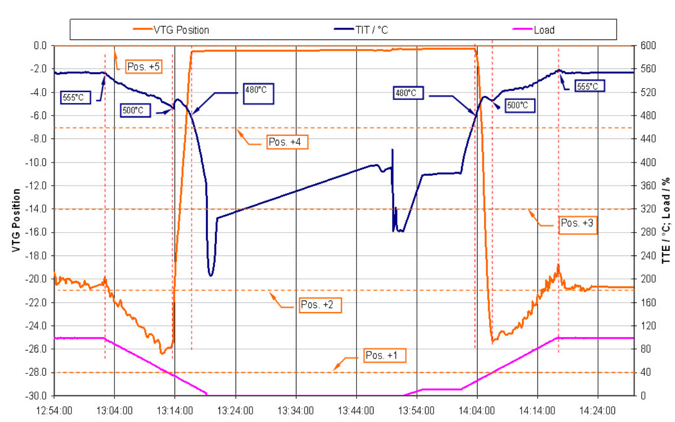
Blade vibration is one of the major issues in the turbomachinery industry. Vibrations were thus measured on the turbine blades and on the nozzle vanes of the TPL65VA turbocharger. The measured strains are within the safety limit. Resonance endurance tests were also conducted: the turbocharger was kept running at the critical speed for the turbine blade and for the nozzle vanes until at least 1.5*10 cycles had been accumulated. The blades and the vanes were subsequently checked and found to be free of cracks.

Cleaning cycles

If a diesel engine is run on heavy fuel oil the turbine stage needs to be cleaned at regular intervals in order to remove deposits. One way to clean the turbine stage is to inject water. This causes material temperature gradients, which can lead to cracks and deformations. The TPL65VA on the test rig was subjected to 1000 cleaning cycles with water injection. The turbine inlet temperature was 500°C. Subsequent inspection showed that the VTG module was free of cracks and the entire variable turbine geometry was fully operative.



11 Field experience of axial type VTG



12 VTG position, TIT and load over time

Field experience with the Axial type VTG

Field experience

The VTG for the axial turbochargers became a serial product in 2004. Since then 132 turbochargers have been delivered. Of these at least 110 are already in commercial operation and for 86 the cumulated running hours are known. Axial turbochargers equipped with a VTG module have been used mainly on gas engines.

Fig. 11 gives the number of turbochargers and the corresponding running hours. 34 turbochargers have accumulated fewer than 5000 running hours and 25 turbochargers have been running for between 5,000 and 10,000 hours. At least 14 turbochargers have collected up to 15,000 running hours and 7 turbochargers up to 20,000 running hours. Four turbochargers have been running for up to 25,000 hours, while two turbochargers have already run up over 25,000 hours of operation. The units which have been running longest have accumulated nearly 28,000 running hours.

Within its so-called CPEX program (Customer Part EXchange), ABB Turbo Systems Ltd regularly inspects and overhauls the VTG modules. The inspections of the axial type VTG started in 2008. In the meantime 15 VTG modules with operating hours of up to 8,000 have been disassembled and investigated. Generally, it can be stated that the VTG modules are in good condition and show a low level of fowling due to their application on gas engines.

So far the operating experience of the VTG modules shows that they function with a high degree of reliability.

However, the introduction has not been completely trouble free. During the initial phase of the introduction of the VTG module, i.e. validation of prototypes in the field, two issues had to be solved:

1. Individual blades showed wear at their tip surface exceeding an acceptable level after 6,200 running hours. In order to avoid abrasion of the tip surface in the serial VTG module, the tip surfaces are today coated with a hard wear resistant coating.
2. The second issue – deformation of the relief ring – was first detected after 3,500 running hours and in another case after 6,200 running hours. The relief ring is located between the blades and the support ring carrying the blades. Its function is to protect the support ring from the hot gas and thus to reduce the thermomechanical loading of the support ring FE calculations revealed that the thermal enlargement of the relief ring during start-ups and load changes was restricted, resulting in a deformation of the part. By adapting the clearances at the relevant locations in the VTG module a deformation of the relief ring could be eliminated.

An axial turbocharger with VTG module now running in the field is currently being monitored online. This turbocharger is running on a gas engine. Among the parameters being monitored, most of which are unrelated to the VTG, is the control of the blade position. Fig. 12 shows the VTG position (orange line), the engine load (magenta line) and the turbine inlet temperature (TIT, blue line) over time. The graphs are shown for a shut-down from full load and a run-up back to full load.

The position of the VTG is varied between +1 (closed position) and +5 (open position). At full load and TIT ~ 550°C the VTG position is at +2. On gas engines the VTG position is often controlled by the TIT. During a load reduction, and thus a reduction in TIT, the turbine area is reduced. During the shutdown the VTG is opened to the maximum open position (+5). After the start-up at very low load the VTG position is again closed fully. Uploading, and thus increasing the TIT, results in the VTG opening again to position +2.

Design features adapted from radial type VTG

The VTG module for radial turbochargers of type TPS was introduced as a serial product in 1998. The field experience gathered with the radial type VTG was the basis for the development of the axial type VTG. In this chapter some examples are given of how the experience from the TPS turbocharger could be used in the TPL-type.

1. The material for the blades in the radial type VTG is a heat resistant nickel alloy with high resistance against abrasion. As the field experience with the TPS was positive, the same material is used for the blades of the axial type VTG. To increase the durability further the tip surface of the blades was later additionally coated to reduce abrasion.
2. Due to different temperatures along the shaft of the VTG-blades, angle errors can occur between the two bearing positions of the shaft. This could result in tight movement or even jamming of the blades. To avoid jamming a cambered design is used in all VTG-types for the bearing surfaces on the shaft.
3. The material for the static bearing sleeves in the supporting ring is the same as that already used in the radial type VTG. A good combination of materials is vital as no lubricants can be used in the area of the bearings due to the high material temperatures.

4. In the axial direction the blade of the VTG is pressed against the inner ring by a spring surrounding the shaft. The dimensioning of the springs in the axial type VTG also benefited from the experience collected with springs applied to the radial type VTG.

Application fields for the VTG Technology in 2-stroke engines

Overview

Electronically controlled 2-stroke engines are opening up new application fields for the VTG technology. The variable turbine nozzle ring gives extra flexibility to the optimization of engine operation for a number of reasons, and will be discussed in the following chapters.

VTG to compensate for changes in the operating conditions

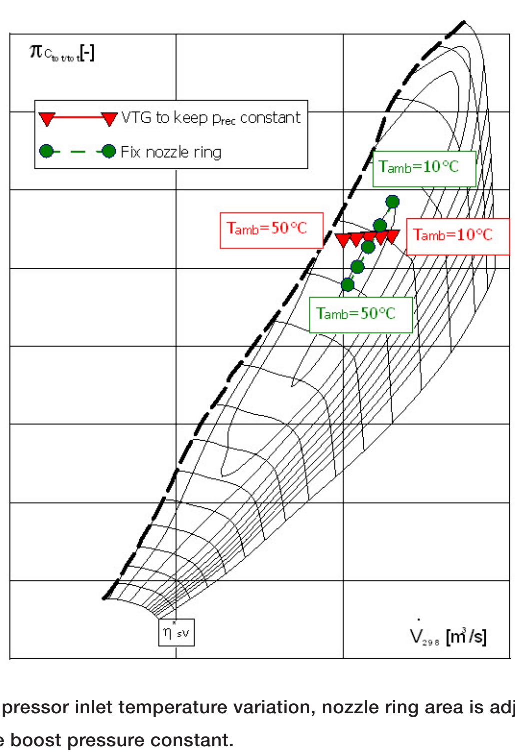
Changed ambient conditions, increased losses in the gas path and deterioration of components combined with reduced efficiency affect engine operation. These factors can increase the operating costs and reduce component lifetime due to higher thermal or mechanical loading.

Fig. 13 shows an example of the compressor inlet temperature variation in the range 10°-50°C for a 2-stroke engine. A variable nozzle ring gives the flexibility needed to keep the boost pressure constant as long as a sufficient surge margin is available. With a fixed nozzle ring, operation at a low temperature increases the maximal cylinder pressure significantly, while a high temperature increases the engine thermal load and reduces the air/fuel ratio during combustion.

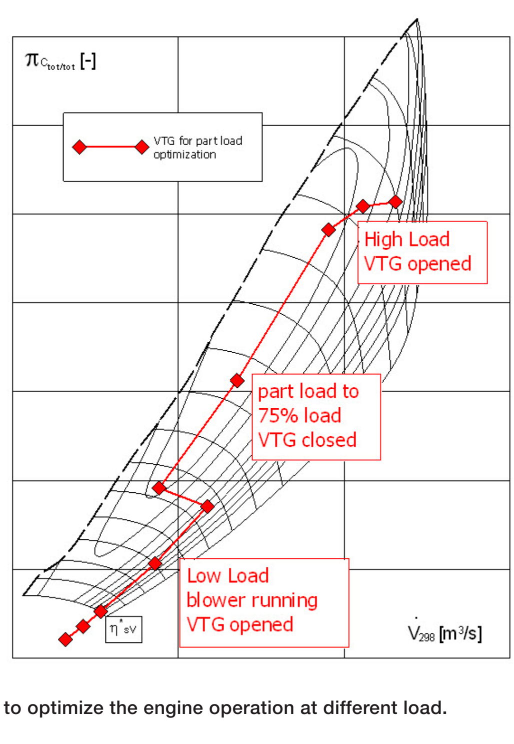
VTG to optimize the engine operation at different loads

The demands on the turbocharging system vary as the engine load changes. In this section, a possible VTG control strategy for a part load optimized 2-stroke engine is described:
The engine operating range is divided into three main areas, see Fig. 14:

a) At very low load with running auxiliary blower(s), the VTG is completely opened to increase the engine scavenging.
b) At part load up to 75% load, the VTG is closed to get a higher boost pressure and hence reduce fuel oil consumption and the engine's thermal load.
c) At high load the VTG is gradually opened to control the maximal cylinder pressure.



13 Compressor inlet temperature variation, nozzle ring area is adjusted to keep the boost pressure constant.



14 VTG to optimize the engine operation at different load.

Future application combined with SCR and EGR technologies

Additional technical measures will be necessary to fulfil the emission requirements according to IMO Tier III. To comply with this new legislation NOx levels need to be reduced by 80% compared to IMO Tier I. Engine builders are therefore currently evaluating various technologies, such as SCR (Selective Catalytic Reactor) and EGR (Exhaust Gas Recirculation).

A drawback of both of these technologies is that CO2 emissions and running costs are increased. As the tight IMO III limits for NOx emissions will be enforced only in the so called Emission Control Areas (ECA), the capability to switch between two operating modes must be considered:

1. Operating mode for low NOx emissions, which have a local impact in ECA areas.
2. Operating mode for low CO2 emissions, which have a global impact and reduced running costs on the open sea.

VTG in combination with SCR technology

a) Transient operation in IMO III mode

Because of the low exhaust gas temperature, SCR reactors for 2-stroke engines are likely to be installed before the turbine of the turbocharger. The thermal inertia of the SCR reactor changes the characteristics of the turbocharging system. During load changes the energy from the exhaust gases at the engine outlet is delivered to the turbine with a time delay. As a result, a system instability called “hunting” can occur. VTG offers the possibility of improving the transient behaviour and controlling the “hunting”.

b) Switching between IMO II and IMO III mode:

The engine can be run with the lowest possible CO2 emissions and running costs on the open sea by bypassing the SCR catalyst. As an additional benefit this measure will extend the exchange interval of the catalyst material.

The two modes exhibit different pressure losses in the gas path. With a VTG the turbine nozzle area is controlled to keep the scavenging pressure constant. The adjustment can also be made during operation, as deposits in the catalyst increase the pressure losses.

As the engine is operated in IMO II mode for a longer time, the temperature of the SCR reactor may be low when it is turned on again. To prevent a rapid drop in the turbine inlet temperature when the IMO III mode is reactivated, the SCR bypass valve needs to be opened progressively. VTG technology can provide additional variability in the turbocharging system for a smoother and faster switch back to IMO III mode.

VTG in combination with EGR technology

For this evaluation a high pressure EGR system is considered. In this configuration the exhaust gases are recirculated without being expanded in the main turbine.

It is further assumed that with EGR the gas mass flow rate through the engine remains the same as in a standard configuration without EGR.

a) Engine operation in IMO III mode

The required EGR rate can be different over the load range. As a consequence it is possible that the optimal boost pressure needed to achieve good combustion will not be achieved with a fixed nozzle ring. A turbocharger provided with variable nozzle geometry is able to adjust the boost pressure, thus reducing operating costs and emissions.

b) Switching between IMO II and IMO III mode

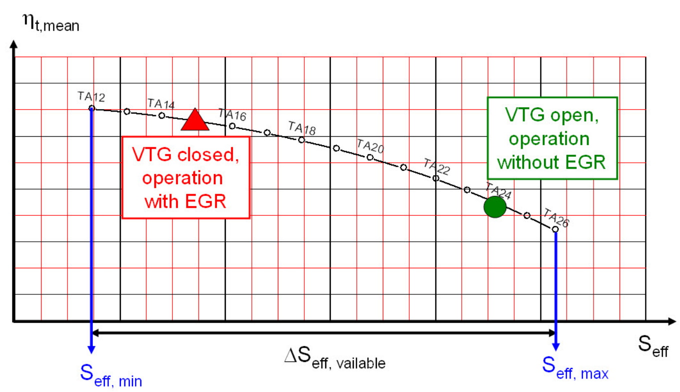
One of the main challenges for the turbo charging system running with EGR is to operate efficiently in both ECA and non-ECA modes, i.e. with the EGR turned on and off.

In the case of a high pressure EGR system the EGR rate (amount of recirculated exhaust gas) has a direct impact on the required specification of the main turbocharger(s). With an EGR rate of 25%, the air flow through the turbocharger(s) is reduced to 75% of a reference case without EGR. In other words, an engine running with four turbochargers with no EGR would require only three similar turbochargers when running with an EGR rate of 25%. Simulation calculations performed by ABB Turbo Systems Ltd have shown, that it will probably not be possible to run an engine satisfactorily in the two operating modes at an EGR rate of 25% or higher without additional measures being taken on the turbocharging side.

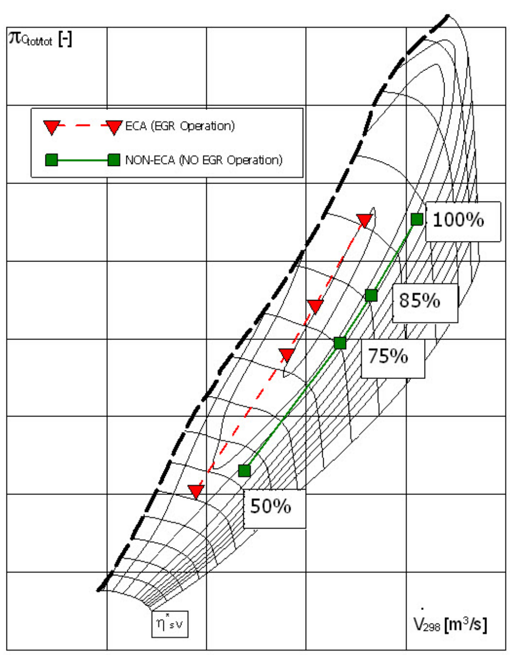
One possible technical solution is to apply sequential turbo-charging. With this kind of turbocharging system an additional turbocharger is switched on at the same time the EGR system is turned off. An advantage of this system is that it can be adapted well to any EGR rate. A drawback is the relatively high complexity.

For moderate EGR rates up to approx. 25% the VTG-technology represents a simpler and more cost efficient alternative. The engine is run with a small turbine area (VTG “closed”) when the EGR is turned on and with a large turbine area (VTG “open”) when the EGR is turned off (see Fig. 15 and Fig. 16).

For EGR rates above approx. 25% the VTG variability needs to be supported by other measures. The operating range is limited on the turbine side by the available efficiency and blade vibrations (see Fig. 15). On the compressor side the operating range is limited by the surge margin (EGR “on”) and available efficiency (EGR “off” / see Fig. 16).



15 Compressor inlet temperature variation, nozzle ring area is adjusted to keep the boost pressure constant.



16 VTG to optimize the engine operation at different load.

Conclusions and outlook

The field experience collected during the last 10 years has shown that VTG technology can reliably be applied to different types of ABB turbochargers. Various methods have been used to qualify and optimise the technology for axial and radial versions.

Until recently the use of VTG technology was directed mainly at gas engines, where fuel consumption can be reduced directly by applying VTG instead of an exhaust waste gate. Lately, new applications in the field of electronically controlled 2-stroke low speed diesel engines have been opening up. Here, VTG technology can make it possible to reduce fuel oil consumption in the medium and low load range. The first ABB turbocharger with VTG technology for a low-speed application has been running on a 2-stroke engine since 2007.

With the introduction of the IMO emission regulations according to Tier III the possibility of switching between two engine running modes will have to be considered as a means of achieving the lowest possible CO2 emissions, fuel oil consumption and running costs on the open sea. In this context VTG technology offers a possible means of control with the capability to support running mode switching and help optimise the performance of future engines.

Considering the potential of the new application areas, ABB has decided to offer the VTG technology as an option also for the new A100-L turbocharger generation. This turbocharger has been developed exclusively for 2-stroke engines and will fulfil the future technical requirements in this engine segment. For the A100-L ABB Turbo Systems Ltd will for the first time not only provide the mechanical parts but also a completely integrated VTG system, including actuator and drive. The control will remain an integral part of the main engine control to ensure optimum overall performance. A first turbocharger of this generation with VTG, an A175-L32T, is already running in commercial operation.

Contact us

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