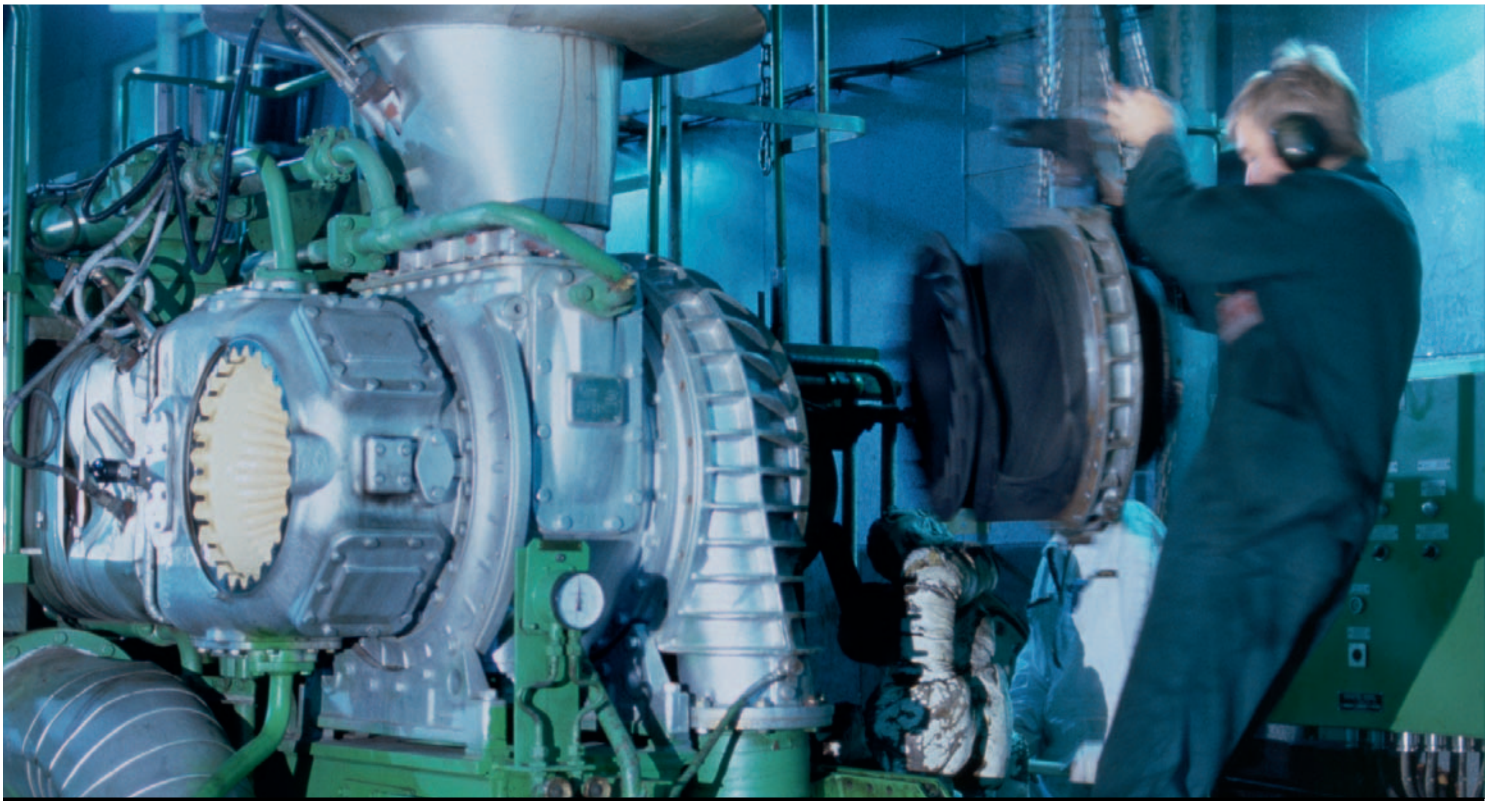

Technical information

ABB Turbocharging
Operating turbochargers – Collection of articles by Johan Schieman published in Turbo Magazine 1992 – 1996

Power and productivity
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Contents

BBC turbochargers in operation

Turbochargers retain their efficiency over their service life providing they are operated properly and kept clean by in-service washing.

The job of a diesel engine turbocharger is to supply compressed air to the engine. The air, heated by the compression, passes through a cooler which reduces its temperature and increases its density.

The air mass is compressed in the engine cylinder to a high pressure. Fuel is injected and burnt. It is normal for a small portion of the cylinder lubricating oil to also be burnt during combustion. The exhaust gases that are produced pass to the turbocharger's turbine, which drives the compressor.

How does the turbocharger become polluted?

Particles in the air are deposited on the compressor and diffusor blades, in time forming a layer which reduces the compressor's efficiency (Fig. 1). These particles have several origins. The air taken by the compressor via filter from the ship's engine room is often mixed with oil vapours. Also, the engine room ventilation system sucks in air from outside, and this can be laden with dust, e.g. in ports where ores are handled. Another factor is the salt in sea air.

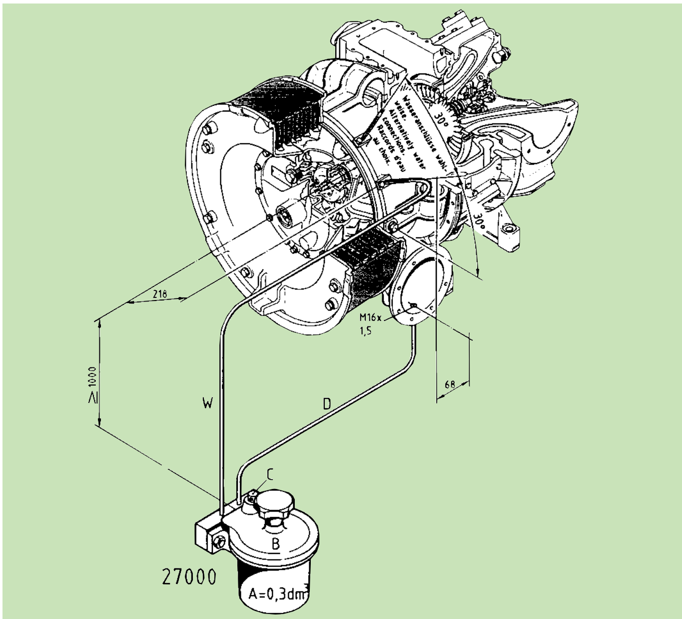


Fig. 2: Waterwashing arrangement for cleaning a compressor.

What to do?

To counteract the drop in efficiency a certain amount of water is injected periodically (once a week to once daily, as necessary) during full-load operation (Fig. 2). This measure considerably lengthens the intervals between turbocharger cleanings requiring dismantling (Figs. 3 and 4).

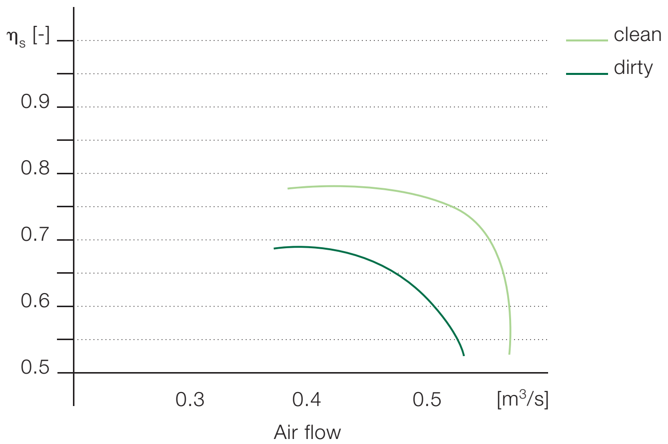


Fig. 1: Reduction of compressor efficiency due to fouling.

In time, the air flowing through the cooler also causes a layer of dirt to form on the fins and tubes, with the result that the pressure loss across the cooler increases and the air supplied to the engine becomes hotter. This affects how the compressor behaves, in that the operating curve in the compressor characteristics moves closer to the point where surging would occur.

Depending on the quality of the fuel (MDO, HFO or mix) burnt in the engine and on the constituents of the lubricating oil, residues may be formed which pass together with the exhaust gases into the turbine.

The composition of these residues decides whether or not a layer of dirt is deposited on the nozzle rings and turbine blades. Dirt layers can be moved by periodical cleaning (once a week to once daily, as required), allowing longer intervals between cleanings requiring dismantling of the turbocharger.

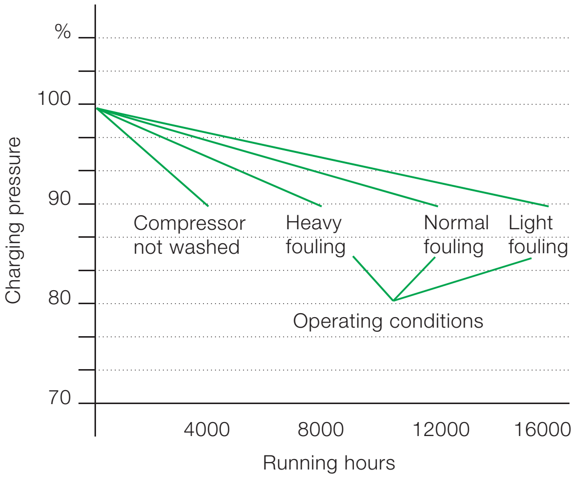


Fig. 3: Example: Reduction in charging pressure with time.

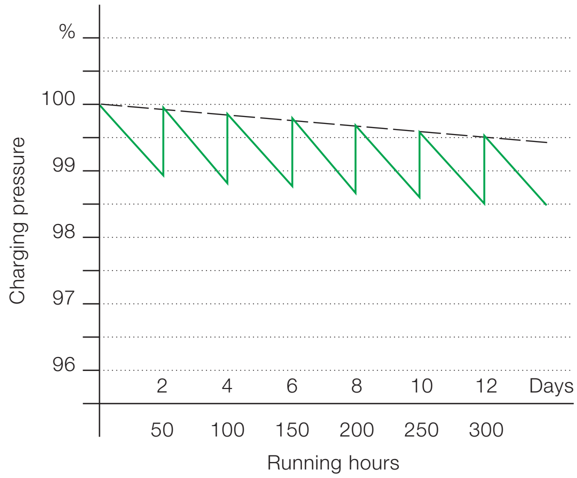


Fig. 4: Compressor cleaning in service. Example: cleaning every second day.

Proven methods of cleaning are waterwashing of the turbine during part-load operation and cleaning by blasting with ground nutshells, etc. The combustion residues can contain hard and chemically corrosive components which can initiate erosion and corrosion of the turbine as well as the nozzle and cover rings. By selecting special materials or applying coatings to relevant parts, the service life of these components can be lengthened substantially.

The importance of cleanness

The processes described above underscore the need to keep the turbine clean to ensure trouble-free operation. Pollution reduces the through-flow area and turbine efficiency. Consequently, compressor surging can also occur in the case of two-stroke engines.

Erosion and corrosion also enlarge the clearance between the moving blades and the cover ring, causing a drop in turbine efficiency as well as turbocharger speed. Providing the plant is properly looked after, i.e. the compressor and turbine are cleaned periodically during operation and the silencer filter is kept clean, the efficiency will diminish only very slowly. A normal figure today for the interval between two cleanings requiring dismantling is 8,000 to 16,000 hours, depending on the operating conditions and unit size.

A turbocharger which has been dismantled, cleaned and had its worn parts replaced, will have an efficiency “as good as new”. Some 80 service stations are available worldwide to help owners maintain high efficiencies for their turbochargers.

Turbochargers pollution and engine operation

The consequences of a drop in turbocharger efficiency for the engine are:
– Higher exhaust gas temperatures
– Lower charging pressure
– Smaller air mass flow

The usual complaint by ship’s engineers concerns high exhaust gas temperatures, although these are really a result of lower pressure and smaller air mass flow. As has been said, turbocharger efficiency decreases only very slowly with time. Despite this, turbocharger suppliers often hear complaints of high exhaust gas temperatures, wrongly reported as being caused by the turbocharger.

Often, high exhaust gas temperatures are caused by changes in the properties of the engine’s fuel injection system over its lifetime. With fuel injection, pressure build-up is not so fast and combustion is slower due to wear and leaks in the fuel pumps.

ABB Turbo Systems has investigated these factors in some detail. It was seen that the engine itself can cause higher exhaust gas temperatures even when turbocharger efficiency is good. At part load, this influence is far less pronounced. The first reaction is to run the machine with reduced output.

An often-heard request in connection with complaints of high exhaust gas temperatures is for the turbocharger specifications to be adapted to allow more air to flow. Before any decision is taken to modify the turbocharger, it pays to first analyze why the temperature is so high. Under certain circumstances this can save a considerable amount of money.

The described causes of higher exhaust gas temperatures usually appear in combinations of one or the other. In time, the turbocharger supplies less pressure and air, and the engine combustion starts later and lasts longer.

For older plants, it is often better to limit the output to the maximum that is required and adapt the turbocharger specifications to the new situation or replace the old unit with a new, modern one with higher efficiency.

This will allow a considerable improvement in the “performance” of an installation which otherwise would not be capable of “new” operating values.

How ambient conditions affect turbocharger performance

In addition to proper operation and washing, ambient conditions also play a role in turbocharger efficiency.

In an earlier article in Turbo Magazine II/1992 we explained how the diesel engine and turbocharger interact, that the compressor supplies air to the engine through an air cooler, and that the engine feeds the turbine with exhaust gases to drive the compressor. Providing the turbocharger is kept adequately clean during operation, its performance will stay at a high level for a long period of time. However, another factor – the ambient conditions – also plays a role and could cause an unexpected change in performance.

How do ambient conditions affect performance?

The properties of the air taken in by the compressor are:
- barometric pressure
- air intake temperature
- humidity

While the first and second of these have an immediate effect on turbocharger performance, there are also some secondary influences of an external nature which could cause the turbocharger performance to change. These can be, for example:
- the temperature of the cooling water used in the air cooler, or
- increased torque requirements at lower engine speeds, e.g. due to fouling of the ship's hull.

The conditions that count

Barometric pressure

The barometric pressure depends on the height above sea level, and decreases as this height increases. However, even at sea level considerable variation is possible. (The lowest pressure known – 0.915 bar – was measured in January 1993 in the North Sea; the highest values lie at about 1.04 bar). At any given engine load the pressure ratio across the turbine increases as the barometric pressure goes down. The result is a higher turbocharger speed (Fig. 1).

Many turbochargers are in operation on ship's propulsion engines. For a given engine output the turbocharger speed may differ slightly because it depends on the barometric pressure. The pressure ratio of the compressor increases with the turbocharger speed. The resultant charging pressure, having built up at the lower barometric pressure, remains slightly lower than the charging pressure at a higher barometric pressure. Above sea level the barometric pressure is lower and for a given engine load the charger speed will increase accordingly (Fig. 2).

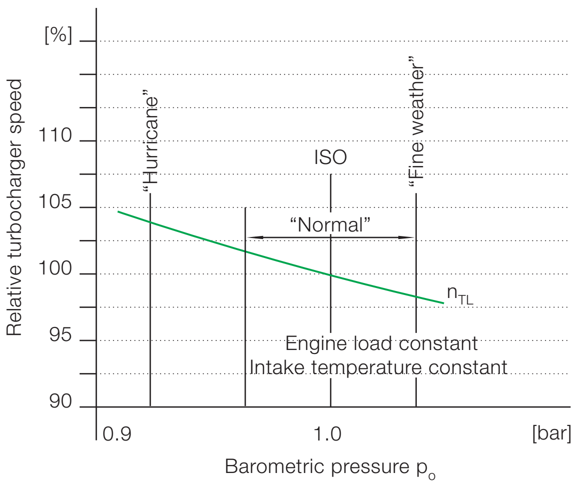


Fig. 1

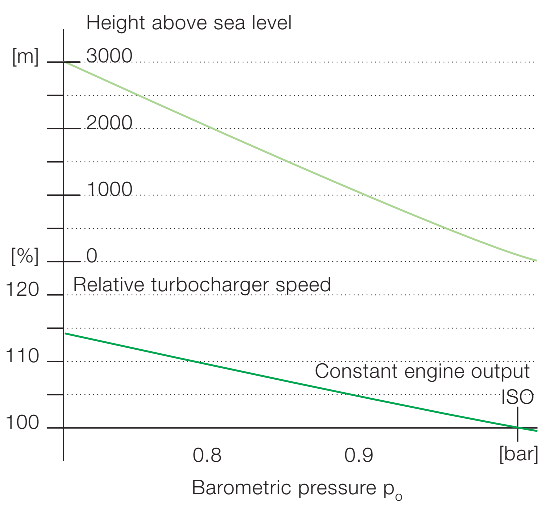


Fig. 2

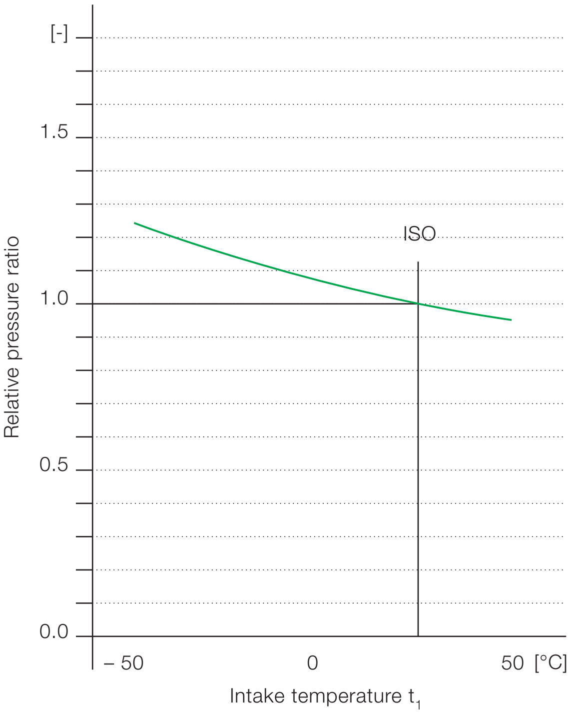


Fig. 3

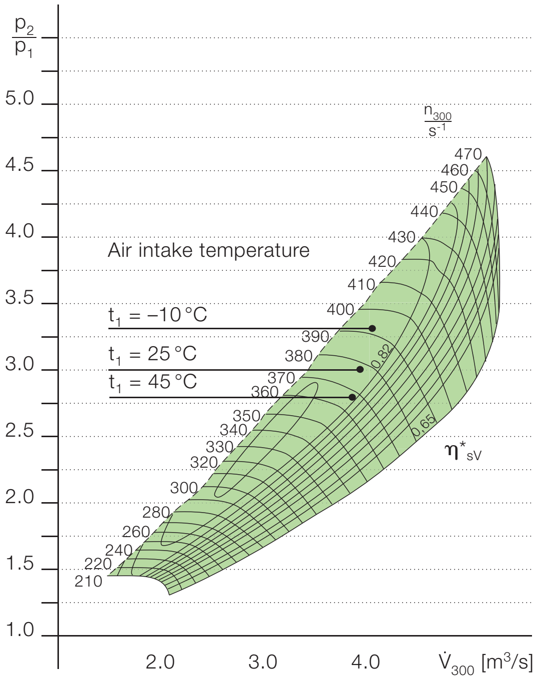


Fig. 4

Turbochargers on diesel locomotives in mountainous regions have to take account of these conditions. When the maximum permissible operational speed has been reached, the engine output has to be reduced to prevent the charger from overspeeding. It may also be necessary to use other measures (e.g. an exhaust gas waste gate) to prevent overspeed. Usually, a turbocharger will be specified for International Standards Organization conditions, i.e. 1 bar ambient pressure. If deviations are considerable, the engine may have to be derated.

Air intake temperature

The energy needed to compress the air is directly proportional to the intake temperature. Therefore, at a given turbocharger speed, the pressure ratio of the compressor decreases with increasing intake temperature, and vice versa.

Fig. 3 shows that the influence is considerable. If we sail with a ship from a moderate to a tropical region (from 25 to 45 °C intake temperature) the absolute charging pressure decreases by 6 % (from 1.0 to 0.94 in Fig. 3).

The consequences of this for a given engine load are:
- less air throughput
- lower air-fuel ratio
- higher exhaust gas temperatures

If we now sail to Arctic waters, where the intake temperature may be –10 °C, the absolute charging pressure increases by 10 %. We now have to be careful that the maximum firing pressure in the engine is not exceeded.

The stability of the compressor may be impaired as these conditions move the working point on the compressor map nearer to the surge line (Fig. 4).

Fishing vessels operating in Arctic waters are confronted with this problem now and again. A good remedy is to prevent outside air from entering the engine room near the charger air intake.

Other applications in tropical climates may be taken care of by carefully specifying the turbocharger. In most cases the specification is based on ISO air intake conditions of 1 bar and 25 °C. Depending on the location and application, ISO-recommended derating may be applied to prevent thermal and/or mechanical overload of the engine and turbocharger.

Air humidity

The performance of the turbocharger is not affected by the humidity. However, compressed humid air carries a considerable amount of water as it passes through the air cooler and this may condense into the air receiver. This may be overcome by either frequent or continuous venting, though with a subsequent loss of some air pressure.

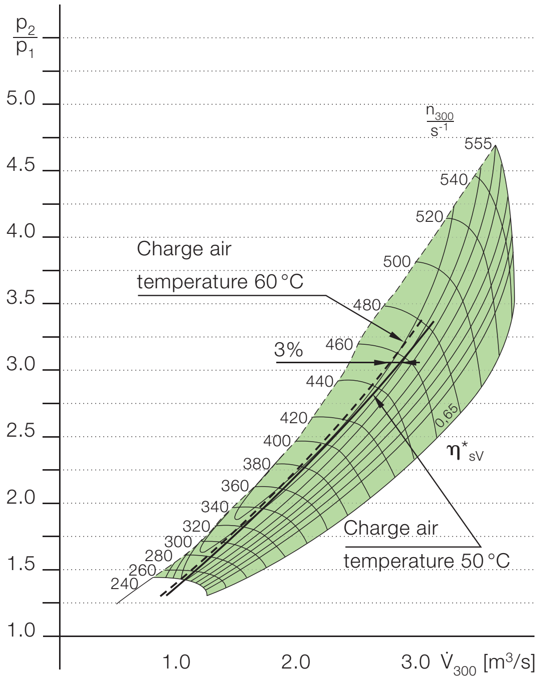


Fig. 5

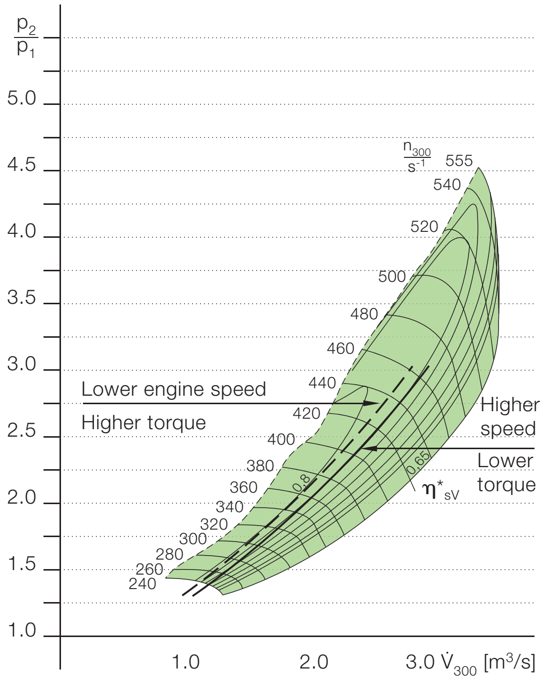


Fig. 6

The impact of secondary influences

Temperatures of cooling water used by air cooler

In the case of turbocharged 4-stroke engines the temperature of the charge air has a significant influence on the air througput. If, for example under ISO conditions, the charge air temperature is 50 °C and, due to a high cooling water inlet temperature, this temperature increases by 10 °C to 60 °C, the air throughput will decrease by 3 % (corresponding to absolute temperatures of 323 K and 333 K, respectively). This moves the working point on the compressor map nearer to the surge line (Fig. 5). Occasionally, this phenomenon is found to be the reason for surging.

Torque requirements

For 4-stroke engines for marine propulsion the condition may occur whereby a high torque is required at relatively low engine speed. As the air throughput is mainly determined by the engine speed, the operating line on the compressor map moves to the surge line (Fig. 6). Such behavior could be caused by fouling of the ship's hull. In severe cases, this may even lead to surging.

It goes without saying that these possibilities are all considered in the turbocharger specification. However, the influence of high compressor efficiency near the surge line and the conflicting need to have a wide safety margin separating the working point from the surge line call for a compromise.

Modern compressors with backswept vanes, e.g. those used in BBC® VTR..4 turbochargers, feature excellent stability in terms of surging. As a result, the compressor's high efficiency is available over the whole load range of the engine.

Turbochargers, efficiency and the diesel engine

The turbocharger has a decisive influence on the diesel engine's performance, while its efficiency plays an important role in how the diesel engine and turbocharger interact.

To understand these relationships better, we have to distinguish between the turbocharger efficiency and the turbocharging system efficiency, and establish in what way they affect the performance of the turbocharged diesel engines.

The turbocharged diesel engine

Fig. 1 depicts the various components of a diesel engine installation. The air flowing to the engine first passes through the silencer filter, which causes a small pressure drop. The air is then compressed from pressure p_1 to p_{21}.

The velocity of the air leaving the compressor is reduced in a so-called diffuser before it enters the air cooler. This process causes some loss of pressure. Afterwards, the air passes through the air cooler. The air sustains another pressure drop.

A pressure ratio p_{21}/p_1 is generated by the compressor. The engine senses only p_R/p_o. The reduction in the pressure ratio from p_{21}/p_1 to p_R/p_o is due to the air section of the turbocharging system.

Similar conditions exist on the exhaust side. The connection between the exhaust valve and the turbine inlet must be shaped to prevent pressure losses. The backpressure after the turbine, p_4, has to be kept as low as possible in order to keep the pressure ratio over the turbine, p_3/p_4, high.

This situation shows us that:
- The turbocharged engine senses the turbocharging according to the pressure ratio p_R/p_o, p_R/p_3 and p_3/p_o.
- The turbocharger has to cope with the pressure ratios p_{21}/p_1 and p_3/p_4.

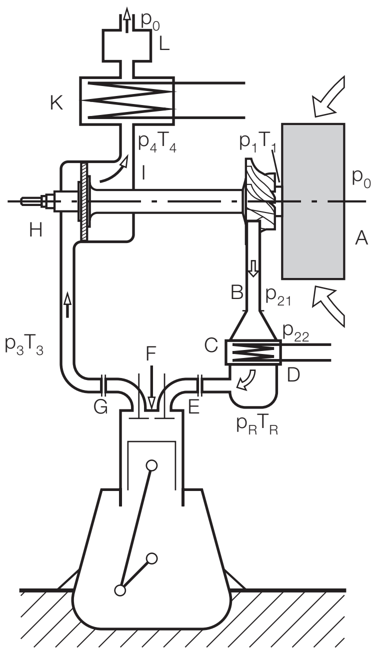


Fig. 1

p_o	Barometric pressure
p_1	Pressure, compressor inlet
T_1	Temperature, compressor inlet
p_{21}	Pressure, compressor outlet
p_{22}	Pressure, cooler inlet
p_R	Pressure, air receiver
T_R	Temperature, air receiver
p_3	Exhaust gas pressure
T_3	Exhaust gas temperature
p_4	Exhaust gas pressure after turbine
T_4	Exhaust gas temperature after turbine
A	Air intake, silencer filter
V	Compressor of turbocharger
B-C	Compressor outlet duct
C-D	Air cooler
D-E	Air receiver, connection to inlet valve
F	Fuel injection nozzle
G-H	Exhaust gas duct to turbine
T	Turbocharger turbine
I	Exhaust gas manifold after turbine
K*	Exhaust gas boiler
L	Exhaust silencer, spark arrester

*for bigger engines

We can express the efficiency required from the turbocharger as

$$\eta_{TC} = \frac{T_1 \cdot R \cdot \frac{\kappa_A}{\kappa_A - 1} \cdot \left(\pi_c^{\frac{\kappa_A - 1}{\kappa_A}} - 1 \right) \cdot m_c}{T_3 \cdot R \cdot \frac{\kappa_g}{\kappa_g - 1} \cdot \left(1 - \frac{1}{\pi_T^{\frac{\kappa_g - 1}{\kappa_g}}} \right) \cdot m_T} \qquad (1)$$

and the system efficiency as

$$\eta_{system} = \frac{T_1 \cdot R \cdot \frac{\kappa_A}{\kappa_A - 1} \cdot \left(\left(\frac{p_R}{p_o}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1 \right) \cdot m_c}{T_3 \cdot R \cdot \frac{\kappa_g}{\kappa_g - 1} \cdot \left(1 - \frac{1}{\left(\frac{p_3}{p_o}\right)^{\frac{\kappa_g - 1}{\kappa_g}}} \right) \cdot m_T} \qquad (2)$$

The system efficiency is lower than the turbocharger efficiency.

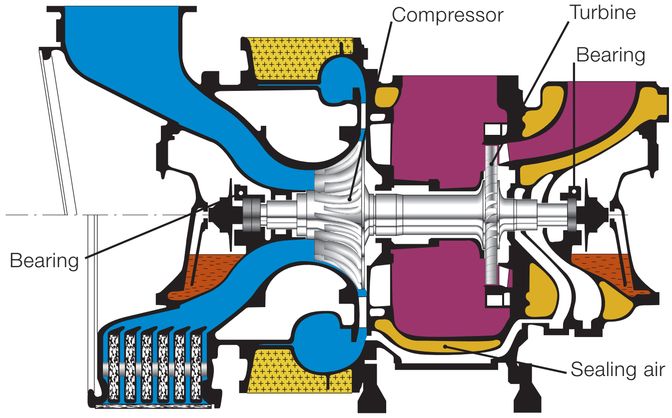


Fig. 2

The symbols represent:

η_{TC}	Overall efficiency turbocharger
T_1	Compressor intake temperature
R	Gas constant
κ_A	Ratio of specific heats of air
π_c	Compressor pressure ratio
m_c	Compressor air flow
T_3	Temperature before turbine
κ_g	Ratio of specific heats for exhaust gas
π_T	Pressure ratio over turbine
m_T	Turbine gas flow
p_R	Charge air pressure
p_o	Barometric pressure
p_3	Exhaust gas pressure before turbine
η_{system}	Turbocharging system efficiency

Turbocharger efficiency and engine performance

Fig. 2 shows a cross-section of the VTR turbocharger with its main components:

- Compressor
- Turbine
- Bearings

These parts have an efficiency of their own:

η_C	= Compressor efficiency
η_T	= Turbine efficiency
η_{mec}	= Mechanical efficiency (bearing friction)

To prevent exhaust gases from entering the bearing chamber on the turbine side and to provide some cooling for the turbine disc, sealing air is taken from the compressed air flow. Hence:
η_{vol} = Volumetric efficiency (sealing air)

There are cooled and uncooled gas inlet casings. The cooled gas inlet casing reduces the exhaust gas temperature somewhat before the gas reaches the nozzle ring of the turbine. Therefore:
kW = Cooling effect (uncooled casing, kW = 1)

Without going into the details of η_C and η_T, it can be stated that:

$\eta_{TC} = \eta_C \cdot \eta_T \cdot \eta_{mec} \cdot \eta_{vol} \cdot kW$ (3)

The engine experiences η_{TC} (1) = η_{TC} (3)

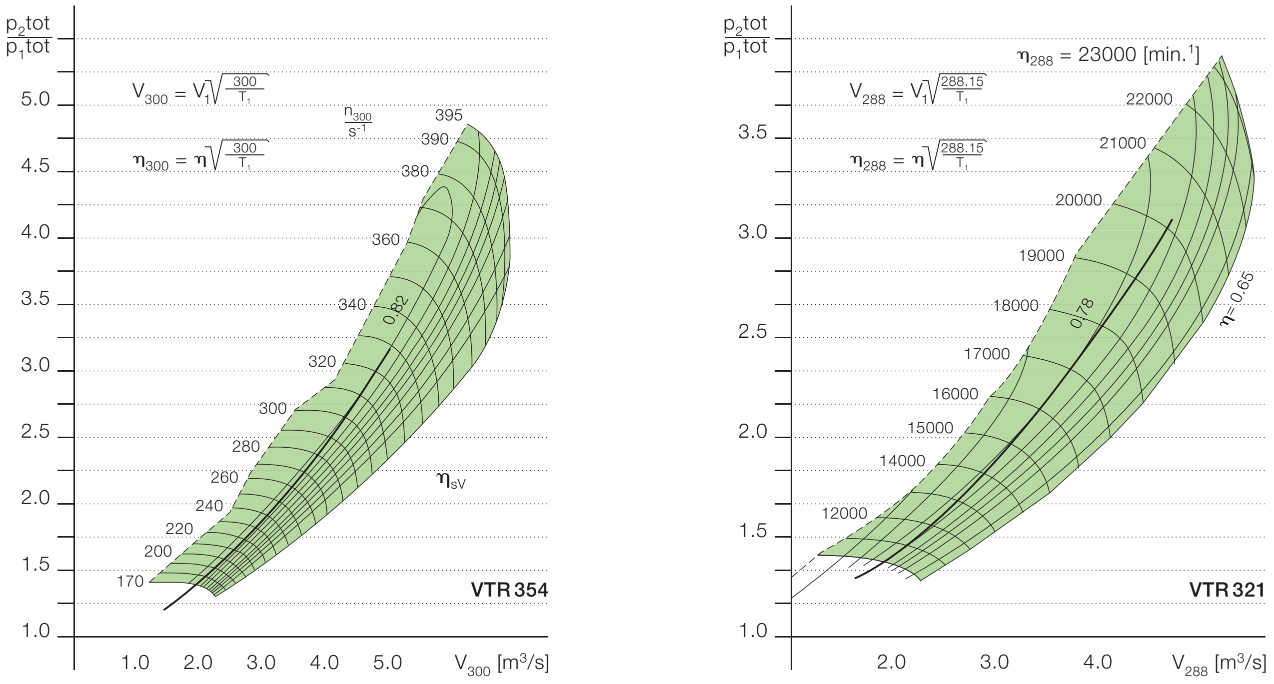


Fig. 3

Fig. 3 compares a compressor map of the VTR 321 with one for the VTR 354, given approximately the same air flow and pressure ratio. Fig. 4 compares the turbine maps. Fig. 5 shows the difference in performance of a 4-stroke engine when a better charger is used.

The indicator diagram of a 4-stroke engine is shown in Fig. 6. From formula (1) we can see that if η_{TC} has a higher value and all other values remain more or less constant, the turbine pressure ratio will have a lower value. This means that the pressure ratio over the engine, p_R/p_3, increases. It allows a larger turbine area.

Fig. 4

Comparison of engine test results obtained with the VTR 321 and VTR 354

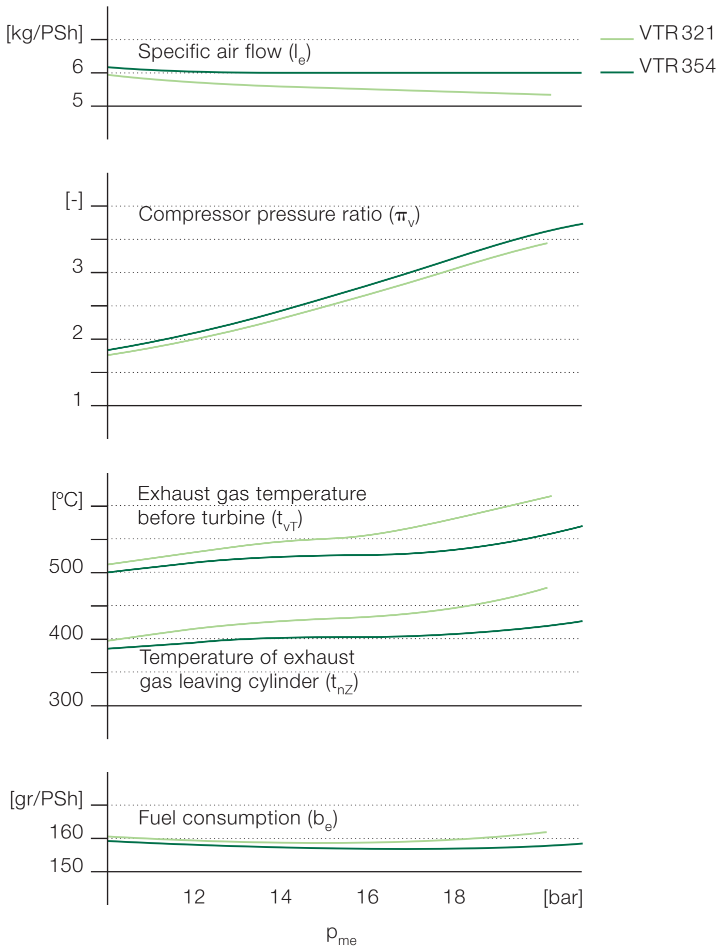


Fig. 5

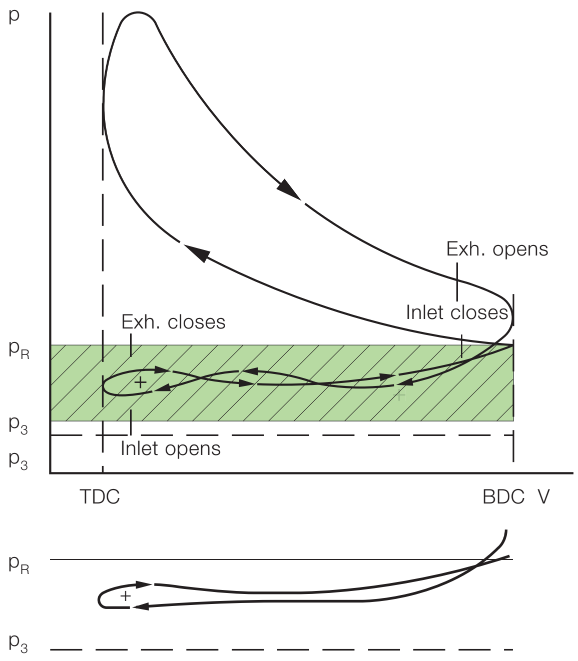


Fig. 6

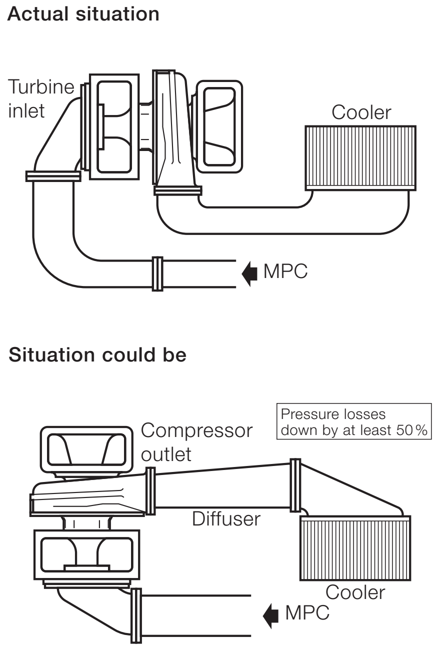


Fig. 7

More positive work can be achieved during the gas exchange period. This has, as its reward, a reduction in fuel consumption. It does of course assume, though, that the timing of the inlet and exhaust valves has been correctly adjusted.

Turbocharging system efficiency

A typical example is shown in Fig. 7. By comparing the two arrangements in terms of the flow conditions for air and gas, we quickly see that the configuration with the fewer bends in the ducts provides most benefits. A diffuser between the compressor outlet and the cooler inlet considerably reduces the pressure loss that would be caused by two 90 ° bends in the duct.

A straight pipe allowing a high-velocity flow of gas to the turbine avoids the pressure loss due to a 90 ° bend. Fig. 8 shows the increasing differences in the turbocharger and system efficiencies as the pressure losses increase on the air-side.

Obviously, the ducts for the air flow must be of very good aerodynamic design. And it is essential to keep the air inlet filter and the air cooler of an installation clean so as to keep the flow resistance low. High pressure losses can cause the compressor to surge.

An increasing backpressure after the turbine, for instance due to fouling of the exhaust boiler, will cause the charging pressure to drop, Fig. 9. This can cause the compressor to surge.

Turbocharging systems

A comparison of different turbocharging systems will be given in a later issue of Turbo Magazine.

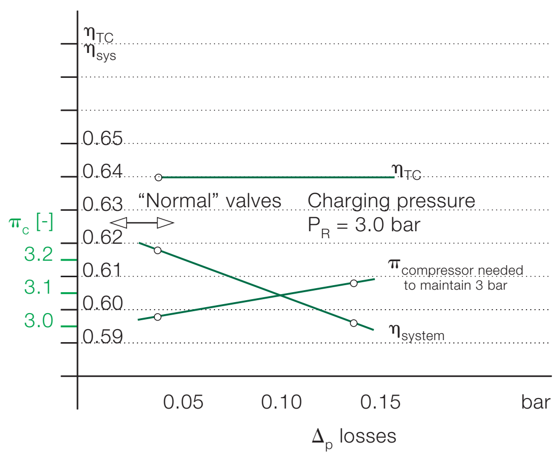


Fig. 8

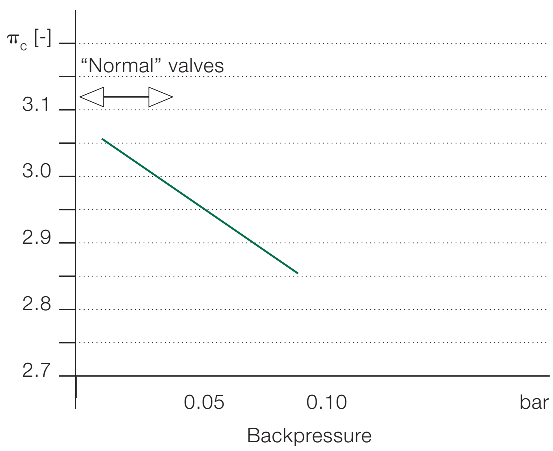


Fig. 9

Turbochargers, load cycles and overload

The way fishing vessels are operated determines the duty cycle of their diesel engine and turbocharger. Some breakdowns can be avoided by understanding how these machines react to the ships' working schedules.

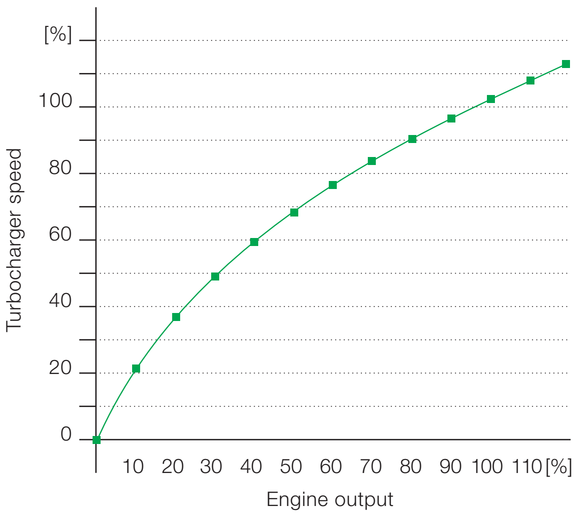


Fig. 1

The working schedule for fishing vessels operating in the North Sea usually begins on Monday, when they leave for the fishing grounds, and ends the following Friday, when they return to port at full speed. The engine power output and speed therefore have to satisfy two requirements: one is the need for the ship to arrive as quickly as possible at the fishing grounds, and later to return to port, with the ship "steaming" freely; the other involves the actual fishing operations, when the nets are out and being dragged through the water, possibly over the sea bottom.

Often, government regulations restrict the length of time a fishing vessel may stay at sea, i.e. it has a limited number of "seadays". These seadays include the time needed to reach the fishing grounds. It goes without saying that the ship will be run at its maximum possible speed in order to reach the fishing grounds and return to harbour as fast as it can. During the fishing, when the nets are out, the engine is run at the maximum possible torque, and every time the nets are pulled in the load and speed are reduced. This is a procedure that repeats itself at least every hour.

How the machines respond

How do the engine and turbocharger react to this type of operation? In Fig. 1, the turbocharger speed is plotted against the engine output. The 100 percent engine output is the limit which should never be exceeded. This is ensured by blocking the fuel pump rack on the engine. Usually, the blocking shim is secured by a lead seal.

The turbocharger speed at full engine load is determined during the acceptance testing of the engine. Its operational reliability for applications on board fishing boats is also checked. As shown above, this application involves many hours of full-speed running to and from the fishing grounds as well as large numbers of wide variations in engine load and speed, with the turbocharger speed varying accordingly. An example of the time-related load is shown in Fig. 2, the number of load cycles over a period of time in Fig. 3. The values have been compiled from measurements on board fishing vessels.

Load cycles show the variation in engine load from low to high and high to low. This can cover "idling/full load/idling", "half load/full load/half load", "part load/higher part load/part load" or "idling/part load/idling" for the engine. The turbocharger speed varies as shown in Fig. 3.

Given the very heavy duty expected of the engine and its turbocharger, it is clear that if the load level is increased beyond the output for which the engine has been built the lifetime of some parts of the turbocharger may be reduced, resulting in a breakdown.

The large number of load cycles, especially at an increased load level, may cause a reduction in the lifetime of turbocharger parts. Turbocharger overspeed is a possibility when the ship is "steaming" freely and the output increases rapidly with the engine speed.

Time-related engine load | Time-related load

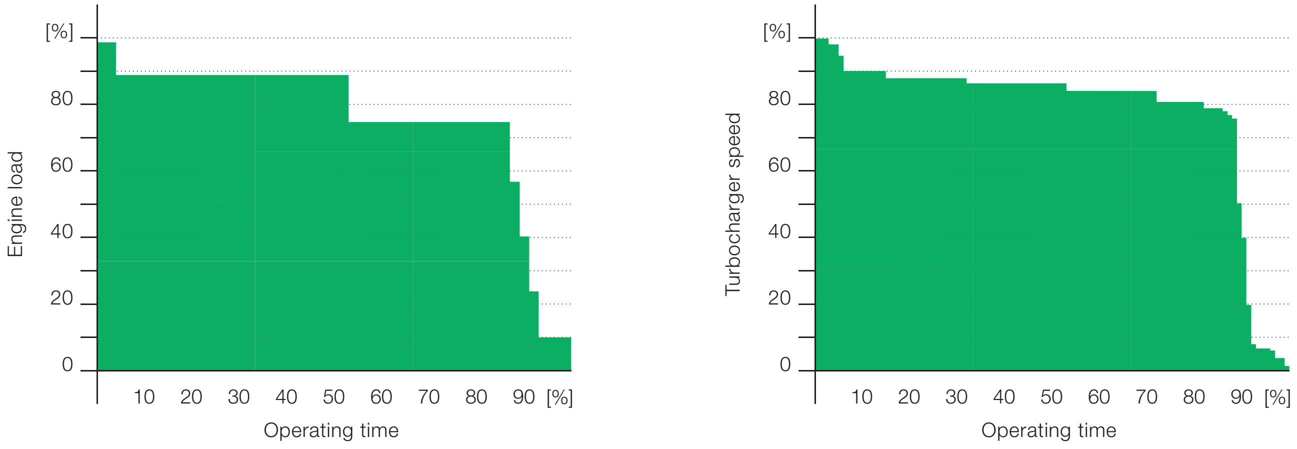


Fig. 2: Fishing vessel operation. Measured engine load and related turbocharger speed.

"Readjusting" for disaster

ABB Turbo Systems has been confronted several times with serious turbocharger breakdowns caused by "readjustment" of the engine fuel pump racks with the objective of running the engine at a higher output than the one it was intended for. This is a very dangerous thing to do, and the consequences can be disastrous. Although a lot of effort has been put into containment of the turbocharger (by designing it so that if a breakdown does occur, any broken parts remain within the casings), there have been situations where broken parts did fly out of the charger. This was brought about by the extreme overload conditions in which the installation had been operated for a prolonged period of time.

Consult the engine builder first

It should be clear from this that the engine should never be "readjusted" for a higher output without first consulting the engine builder. The engine builder will make sure that the turbocharger is adapted to match the new engine output requirements. In this way, severe breakdowns can be largely prevented and the risk to personnel minimized.

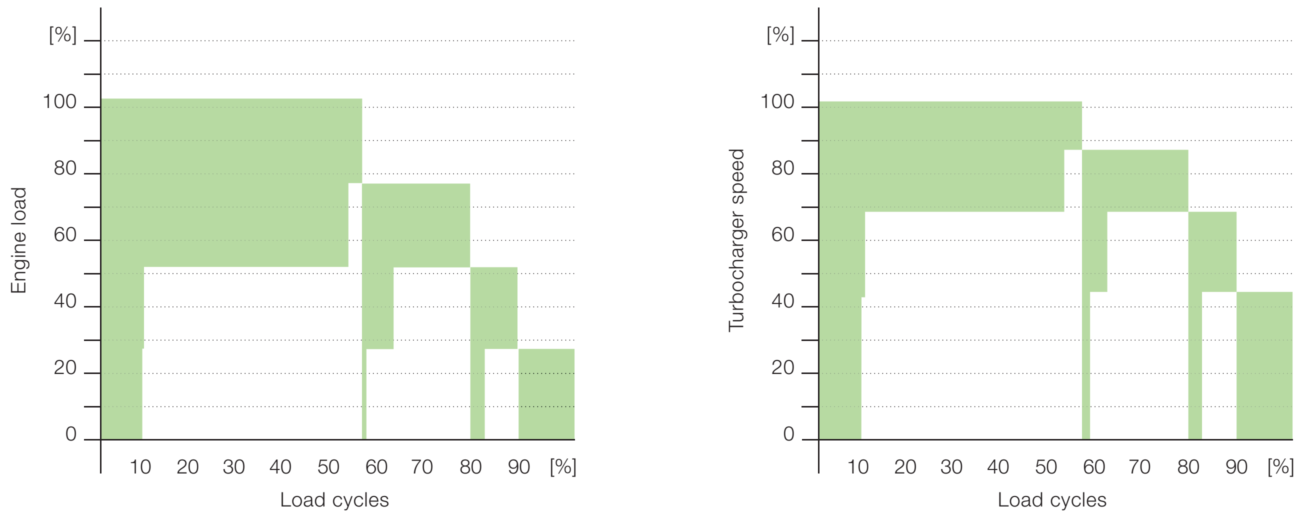


Fig. 3: Fishing vessel operation. Measured load cycles. 100 % corresponds to 53,000 load cycles in 10,000 hours.

Turbocharging systems for diesel engines

In another article for our readers with a technical leaning, Johan Schieman describes how the turbocharging system influences diesel engine performance and discusses the pros and cons of different turbocharging systems.

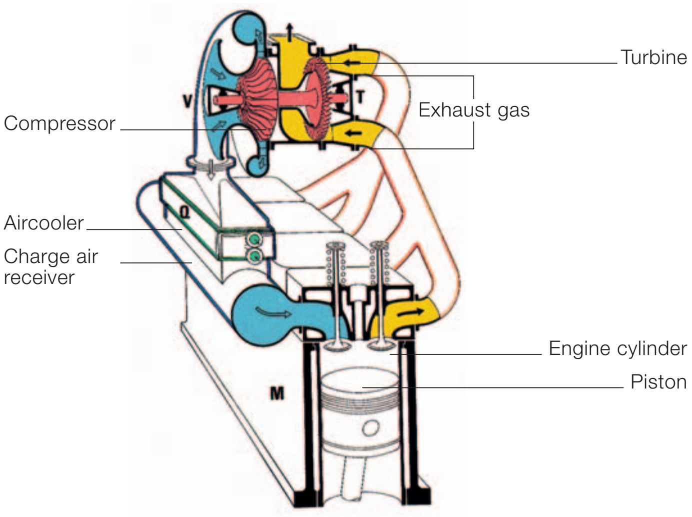


Fig.1: Cutaway drawing of a turbocharged diesel engine.

Turbocharging systems

ABB turbochargers are in use on diesel engines rated at 500 kW and more. In the following, we shall consider only those engines on which ABB's turbochargers can be used. Fig. 1 shows a turbocharged diesel engine. The exhaust gases drive the turbocharger's turbine, which in turn drives the air compressor, both of which are mounted on the same shaft. The compressed air is passed through an air cooler to the engine's air receiver.

The name given to the turbocharging system comes from the arrangement of the exhaust ducts between the engine cylinders and the turbocharger gas inlets. Fig. 2 shows the different arrangements. These are in use on 4-stroke diesel engines. Pulse and constant pressure systems are also possible for 2-stroke engines. Pulse systems use narrow pipes with small volumes between the cylinders and the turbine gas inlets.

The available turbocharger efficiency plays a key role in the choice of turbocharging system. As part of the search for better efficiency for the turbocharging system, research was also carried out into so-called two-stage turbocharging.

Through the development of turbochargers for higher pressure ratios and higher efficiencies, it has been possible over the years to meet practically all the requirements. With today's compressor designs, pressure ratios of approximately 5 are possible. It can be seen from Fig. 3 how the turbocharger efficiencies have developed over the years.

Pulse systems versus constant pressure systems

The following may help the interested reader to understand the gas exchange process of the 4-stroke engine better. Fig. 4 shows the valve timing of the inlet and exhaust with the overlap for scavenging the top dead center cylinder space with fresh clean air. A pressure difference is necessary to allow the scavenging air to flow from the air receiver through the top dead center cylinder space into the exhaust duct. This pressure difference is determined by the turbocharging system and the turbocharger efficiency. When the turbocharger efficiency is low, the exhaust gas pressure is only slightly lower than the charge air pressure.

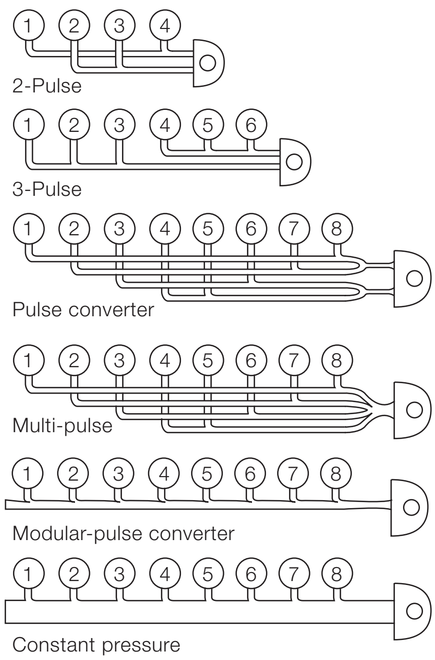


Fig. 2: Arrangements of exhaust ducts.

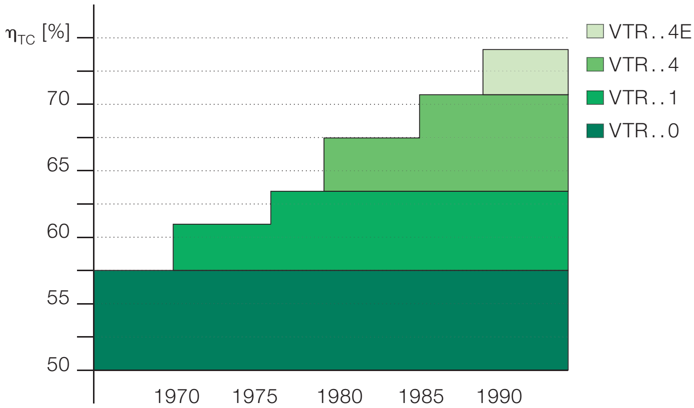


Fig. 3: Development of the turbocharger efficiencies over the years.

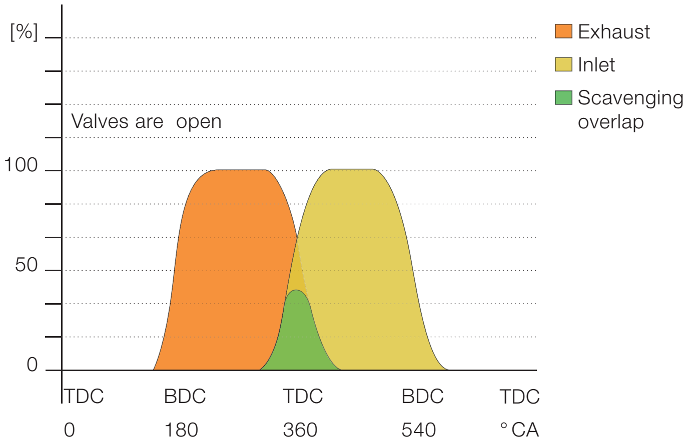


Fig. 4: Valve timing of inlet and exhaust.

From Fig. 5 it is seen that if the turbocharger efficiency is low the pulse system offers advantages by allowing an optimal clean cylinder charge of fresh air. Due to the large pressure variations in the exhaust pipe after the cylinder during the exhaust period, the difference between the charge air pressure and the exhaust pipe pressure is also quite large during the inlet-exhaust overlap (Fig. 4). This allows ample scavenging. The constant pressure system permits hardly any pressure difference over the cylinder, and scavenging of the top dead center cylinder space is very limited. As turbocharger efficiencies have become higher, this disadvantage has become less important.

The engine firing order is chosen so as to ensure that stresses in the crankshaft as a result of torsional vibration are as low as possible. Using a pulse system and the given firing order, combinations of cylinder exhausts on one pipe to one turbocharger gas inlet must be chosen in such a way that the exhaust pulse of one cylinder cannot disturb the scavenging period for another cylinder on the same pipe.

Three-cylinder combinations with a firing interval of 240° crank angle (CA) are in use on 6-, 9-, 12- and 18-cylinder engines, while for 8- and 16-cylinder engines the firing intervals used are 360°CA or 270/450° CA. Pulse systems have also been built for 5- and 7-cylinder engines.

The constant pressure system has the same large exhaust gas receiver irrespective of how many cylinders the engine has. The same applies to single pipe exhaust systems, where the exhaust receiver volume is reduced in comparison with constant pressure systems.

The efficiency with which the turbocharger turbine can operate depends on how it is supplied with the exhaust gas. A continuous gas mass flow (full flow) to the turbine supports optimal utilization of the energy in the exhaust gas. 3-pulse cylinder combinations, constant pressure and single pipe systems assure this mode of gas supply, whereas 2-pulse cylinder combinations provide an intermittent supply of gas to the turbine.

In Fig. 6 the exhaust gas pressure, charging pressure and cylinder pressure are plotted over °CA for the 3-cylinder combination. Fig. 7 shows the same quantities for the constant pressure system. Fig. 8 illustrates, for the 2-cylinder combination, the periods when there is no pressure in the exhaust gas duct, i.e. no gas supply to the respective gas inlet of the turbine. The turbine, driven by gas from the other gas inlets, drags the blades through stagnant gas, which obviously reduces the efficiency of the system.

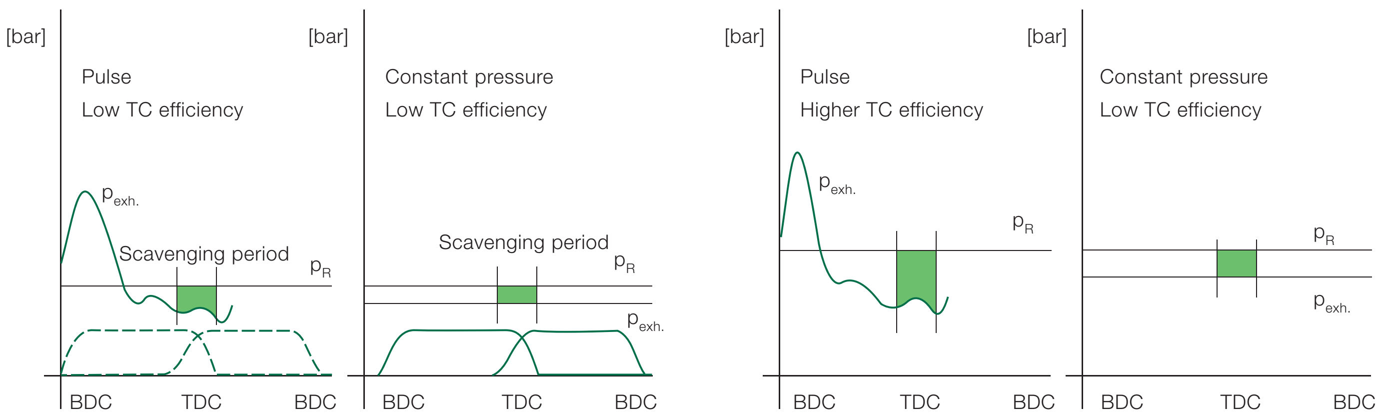


Fig. 5: Influence of turbocharger efficiency on the scavenging process for pulse and constant pressure systems.

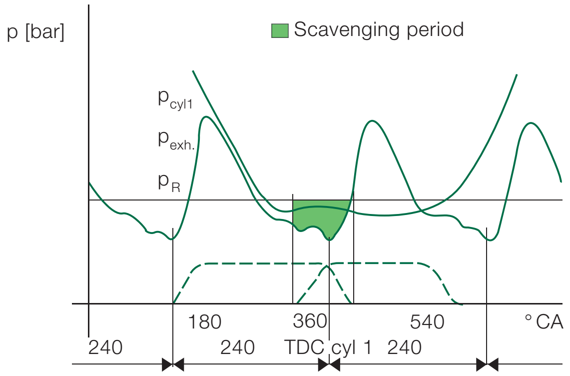


Fig. 6: Pressures of a 3-pulse system plotted over degrees crank angle (° CA).

Solutions

To overcome the problems mentioned, various systems have been introduced to achieve a full flow gas supply to the turbine for those cylinder numbers that cannot be divided by 3, e.g.:

- Pulse converters
- Multi-pulse converters
- Single pipe exhaust gas ducts with a relatively small gas flow area
- Constant pressure exhaust gas receivers

In the full-load operation range of the engines, these solutions have brought improved performance compared with the two-pulse arrangements. However, in some applications the part-load performance deteriorates in comparison with "straight" pulse operation. This applies particularly to ships' propulsion systems and the traction sector (diesel electric locomotives), where the engine output is proportional to the engine speed to the power of 3 ($Pe = n^3$).

How the system functions

We shall now use the example of an 8-cylinder engine with pulse converters to explain how such a system achieves full flow exhaust gas to the turbine. Fig. 9 shows the exhaust strokes of all the cylinders over °CA of cylinder 1, the crank rows with firing order, and the exhaust duct arrangement for four gas inlets. By combining the ducts 1 – 8 and 2 – 7 with a pulse converter on one turbine gas inlet, as shown in Fig. 10, exhaust gas is supplied with an interval of 180 °CA on one gas inlet instead of with an interval of 360 °CA on two gas inlets. Full flow is achieved. However, it is seen from the diagram that the high pressure pulses of 2 – 7 are exactly there where 1 and 8 have their scavenging period. Fortunately, it takes some time for a disturbing pulse from no. 2 cylinder to reach the turbine and for its reflection to reach the scavenging cylinder (no. 1), with the result that the disturbance arrives at the end of the scavenging period. By correctly dimensioning the pulse converter areas, the intensity of the reflected pulse can be reduced to a value lower than the charging pressure. In this way, the scavenging process is hardly impaired when pulse converters are used.

The "straight" pulse system uses narrow pipes having a small volume. This enhances the build-up of intensive pulses and turbocharger response to demand for higher engine load. By combining two ducts via a pulse converter, the duct volume connected to one gas inlet increases and the build-up of the exhaust pulse becomes less intensive. The maximum pulse pressures are lower than with a "straight" pulse system. Thus, the turbocharger responds more slowly to an increase in engine load. The multi-pulse converter has an even larger pipe volume and is even slower to react. The constant pressure system, with no pulses at all, is slowest here. Single-pipe systems are comparable with the latter two systems.

For ship's propulsion systems with a fixed pitch propeller, the pulse system offers considerable advantages in terms of engine part-load performance over systems with large exhaust duct volumes. Fig. 11 shows the reason for the favourable part-load performance. Looking at the three-pulse system, it can be assumed that there is a constant pressure region with superposed pulses. At full load the pulses contain only a small part of the energy supplied to the turbine. At part load, however, the pulses contain a relatively large proportion of the energy, this proportion increasing as the loads become lower. Similar considerations are valid for two-pulse arrangements. At part load, the pulse system can therefore supply more air at a higher charging pressure than systems with larger exhaust duct volumes. Fig. 12 shows the relative differences in achievable part-load charging pressures.

The question can be asked, given the pulse systems' clear advantages in terms of performance, why many engines have a single pipe or a constant pressure turbocharging system? Let us consider the pros and cons of the latter systems.

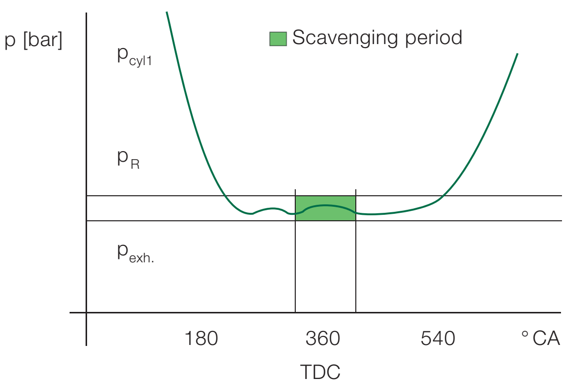


Fig. 7: Pressures of constant pressure system plotted over °CA.

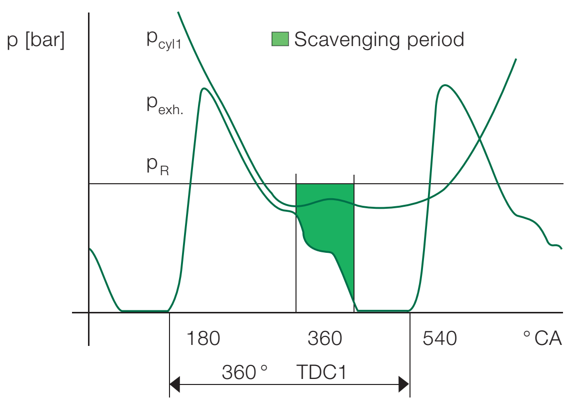


Fig. 8: Pressures of 2-pulse system plotted over °CA.

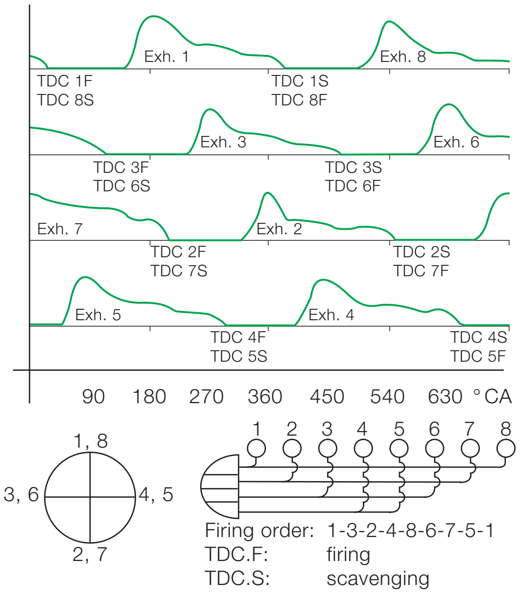


Fig. 9: Exhaust gas pressure sequence of an 8-cylinder engine with 2-pulse system. 4 exhaust ducts, 4 turbine gas inlets.

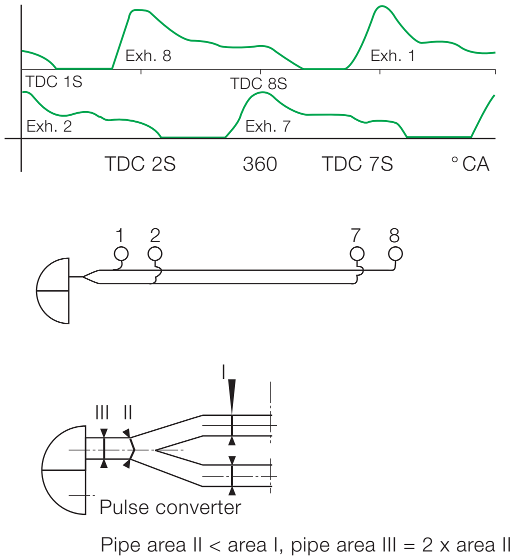


Fig. 10: Pulse converter arrangement for two 2-pulse exhaust ducts.

Advantages:
Simplicity
Suitability for any number of cylinders
Easy standardization of many identical parts
Constant engine speed application, few problems
One turbine gas inlet, possibly next smallest charger size

Disadvantages:
Large expansion compensators
Slow load take-up response
Low part-load charging pressure for propeller law/traction

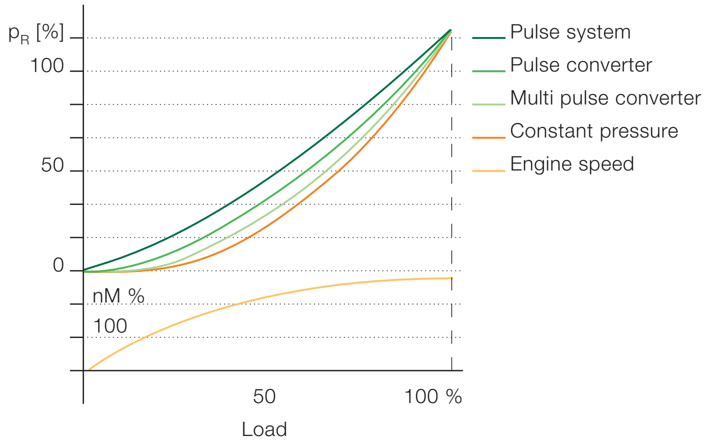


Fig. 12: Relative difference in achievable part load charging pressure p_R.

The low part-load charging pressure leads to a low air/fuel ratio for the combustion and high thermal loading of the parts of the combustion chamber.

Several methods exist with which the low part-load charging pressure can be remedied.

- The charger is specified for an adequate part-load charging pressure. An air waste-gate for the full-load range can be used to limit the otherwise too high charging pressure.
- The charger is specified as before, but an exhaust gas waste-gate is used.
- A bypass duct guides air from before the air cooler into the turbine inlet to increase the charging pressure at part load, allowing the turbocharger compressor to be specified for higher efficiency at full load.

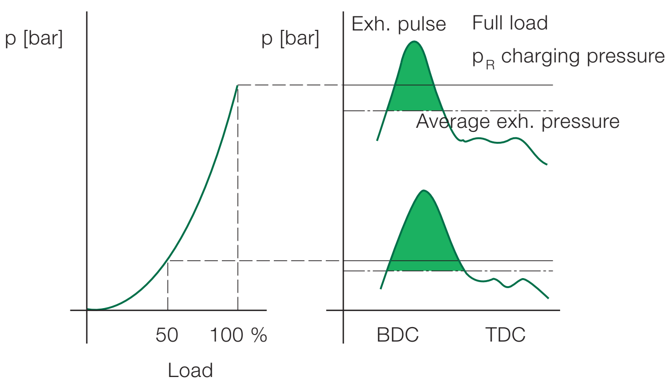


Fig. 11: Pulse system. Good part load performance because of relatively high pulse energy at low load.

At first sight the constant pressure/single pipe systems appear to be more favourable in terms of ease of manufacture. However, the auxiliary measures which have to be taken to meet requirements such as fast take-up of load, satisfactory part-load performance for fixed pitch propeller drives, etc., add to the costs.

Fig. 13 compares the performance of constant pressure and 3-pulse systems. Note the difference in λ_v, a measure of the air/fuel ratio for combustion.

Trends and preferences for some applications

We can distinguish between three main fields of application for ABB turbochargers:

- Ships' propulsion systems
- Generator drives (power stations, ships' auxiliary engines)
- Locomotive drives (traction)

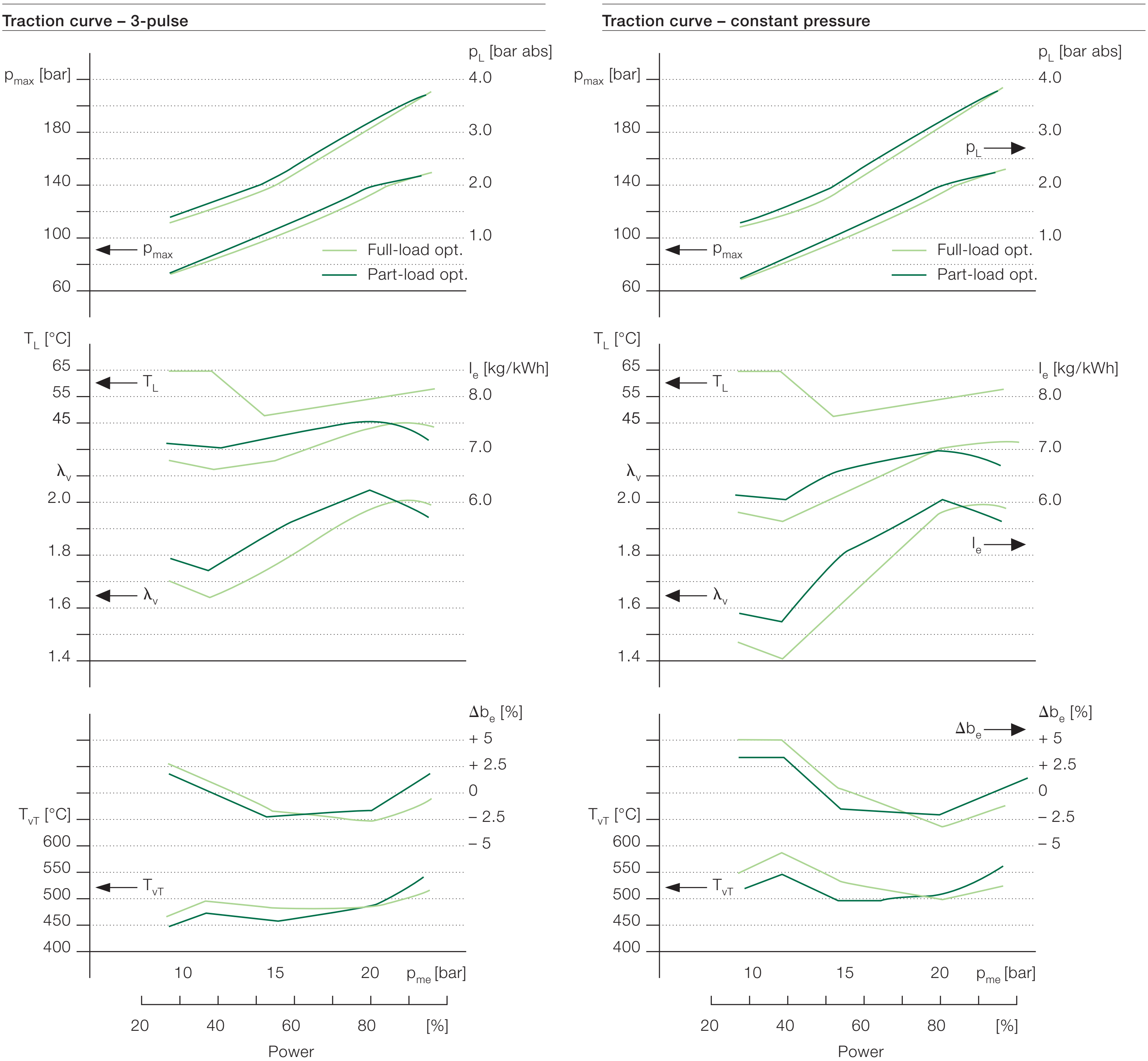


Fig. 13: Comparison of the performance of 3-pulse and constant pressure systems. Note the part load difference λ_v.

Occasionally, there are also applications where full torque is required at variable speed.

More recently, we have seen an increase in applications with pulse systems. There are even signs of 2-cylinder combinations staging a comeback – a trend also applying to the highly turbocharged engines. This is due to the fact that although the turbocharging system itself is not very efficient, the efficiency of the turbochargers now available compensates in part for the low system value. The negative influence on fuel consumption is partly compensated for by modern fuel injection apparatus, assuring the best possible specific fuel consumption. Two-pulse systems are found on modern 4-cylinder auxiliary engines for ships.

As regards ships' propulsion systems and traction applications, three-pulse arrangements are frequently used on 6-, 9- and 12-cylinder engines. This is also the case for generator drives with fast load take-up requirements.

Single pipe or constant pressure turbocharging is frequently used on big engines for power generation, especially when these are used for base-load operation. This type of turbocharging may also be found on locomotives, e. g. with 16-cylinder engines and possibly equipped with ancillary apparatus such as a waste-gate or bypass.

So, from the point of view of the turbocharger, we can conclude that:

– The three-pulse turbocharging system is the best in terms of system efficiency and achievable specific fuel consumption, and does not require ancillary apparatus.
– The high efficiencies offered by ABB turbochargers make the use of two-pulse systems possible up to the higher engine outputs. The turbocharger efficiency largely compensates for the low system efficiency.
– Single pipe and constant pressure systems are generally used on big engines ("V" engines). Ancillary apparatus is fitted for certain applications.

Turbocharger compressors – the phenomenon of surging

The many different applications of turbocharged engines make turbocharger performance and reliability key issues, which turns the spotlight on the compressor's performance. Johan Schieman explains how a radial compressor works and why surging occurs, and looks at some problems he has experienced.

Fig. 1: Partly disassembled VTR 354-11 turbocharger. 1 = Diffusor, 2 = Air inlet casing, 3 = Silencer filter, 4 = Compressor wheel (inducer, impeller), 5 = Air outlet casing.

The working of a centrifugal compressor

The main parts of the turbocharger compressor are the compressor wheel (inducer and impeller), diffusor, air inlet and air outlet casing. These parts can be seen in Fig. 1, which shows a turbocharger partly dismantled. It is usual for a silencer-filter or a suction branch to be fitted to the air inlet casing.

Fig. 2 shows a schematic of a compressor wheel. It will be assumed that the wheel stands still. The space between the blades is filled with air at ambient pressure and temperature. We shall consider a small air mass volume at a radius r.

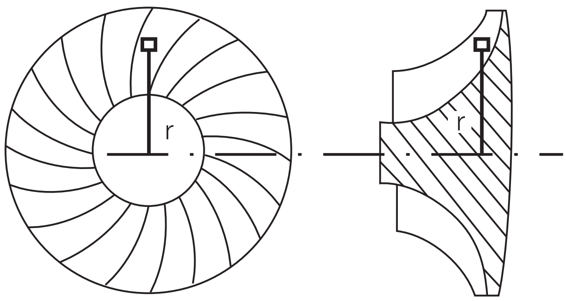
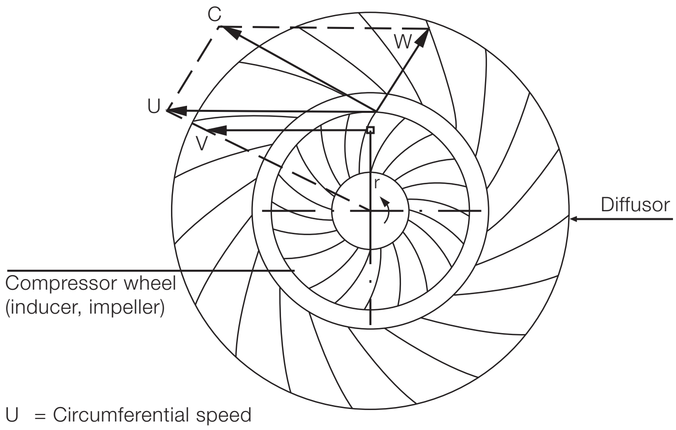


Fig. 2: Compressor wheel. Airmass volume at radius r.



U = Circumferential speed
V = Speed of airmass particle
C = Absolute speed of air leaving the compressor wheel,
W = Speed of air relative to blades

Fig. 3: Rotating compressor wheel.

$$\eta_{TC} = \frac{\dot{m}_C \times T_1 \times R \times \kappa/(\kappa-1) \times (\pi_C\text{^}((\kappa-1)/\kappa)-1)}{\dot{m}_T \times T_3 \times R \times \kappa^*/(\kappa^*-1) \times (1-\pi_T\text{^}((1-\kappa^*)/\kappa^*))}$$

η_{TC}	Turbocharger efficiency
$\dot{m}_C$	Air mass flow
T_1	Air intake temperature
R	Gas constant
π_C	Compressor pressure ratio
κ	Ratio of specific heats for air
$\dot{m}_T$	Exhaust gas flow
T_3	Exhaust gas temperature before turbine
π_T	Turbine pressure ratio
κ^*	Ratio of specific heats for exhaust gas

Fig. 5

We start the compressor wheel rotating (Fig. 3) with a circumferential speed U. At radius r, the speed is V. The small air mass volume is subjected to a radial acceleration, V^2/r, which causes it to move radially outwards. All the air experiences this influence and air begins to flow into the air inlet casing, through the compressor wheel and into the diffusor and air outlet casing. The air leaves the compressor wheel at its circumference with an absolute velocity C.

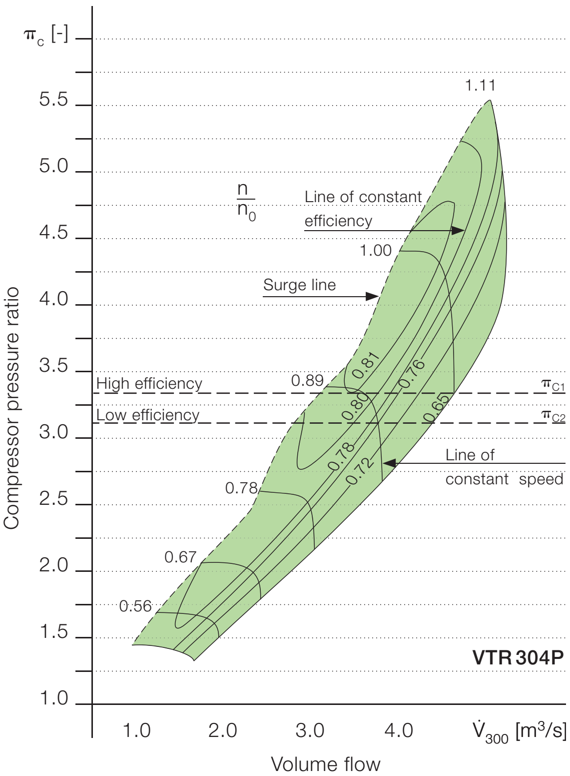


Fig. 4: Compressor characteristic.

For a given design operation range the blades of the diffusor must be arranged with the correct angle of incidence and a profile that reduces to a minimum the losses caused by collision. This will give the highest compressor efficiency. The general performance is positively influenced by low air friction losses on the various flow surfaces of the compressor wheel and the diffusor.

Fig. 4 shows the characteristic of a modern compressor. At a given speed and with increasing volume, the achievable pressure ratio is lower due to the lower efficiency. We find the explanation of this in Fig. 6. Theoretically, a compressor with backswept vanes and no losses due to incidence or to friction will exhibit a decreasing pressure ratio with increasing volume (line a). But there is friction, and it increases with the volume (line b). To the left and right of design point A on the constant speed line c, the angle of incidence of the flow into the diffusor is not optimal. The hatched areas show the magnitude of the losses.

The surging phenomenon

The compressor runs at constant speed and supplies air to the air receiver of the engine, where a required pressure must be maintained. Fig. 7 shows the line of constant speed and the operating line of the engine. The intersection of the two lines is the working point A. If a slight increase in air volume occurs, more pressure is required on the operating line and the pressure becomes lower on the constant speed line. The volume has to decrease again to the point of equilibrium A. If, at the same charger speed, a slight reduction in airflow occurs, the pressure will increase although less pressure is required on the working line. Equilibrium is then once more at point A. The working point A is stable on the part of the constant speed line inclined downwards with increasing volume.

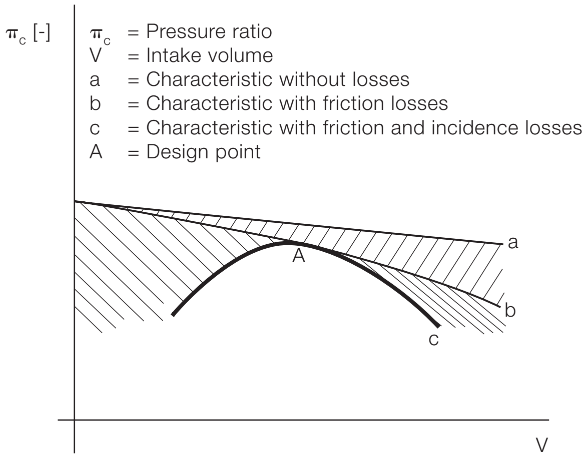


Fig. 6: Compressor characteristic with backswept vanes.

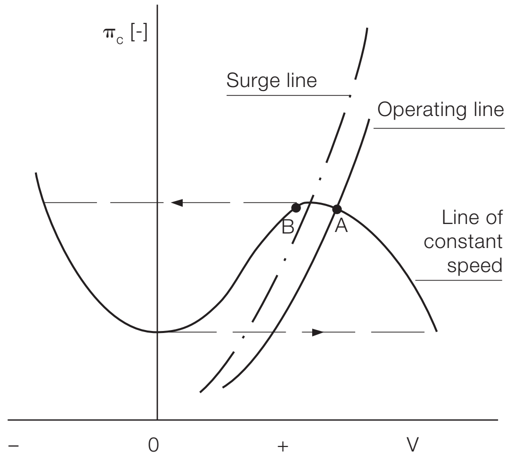


Fig. 7: Surging cycle.

If a slight decrease in volume occurs at point B (at the same pressure as A), then the pressure on the constant speed line decreases. The compressor cannot maintain the required pressure, the volume continues to decrease and the compressor surges. Point B is not stable on that part of the constant speed line that is inclined upwards (with increasing volume).

Theoretically, instability of the compressor starts where the line of constant speed is level. Since we combine a turbomachine, which supplies a continuous airflow, with an engine taking air intermittently, there will always be a pressure fluctuation which influences the point where instability starts. It goes without saying that this overview of the functioning of a centrifugal compressor has been greatly simplified.

A case where surging occurred

Before we look at this case, we shall first consider how a turbocharger specification is determined. Generally, it is based on so-called ISO (International Standards Organisation) conditions, namely ambient pressure of 1 bar and ambient temperature of 25 °C. Margins must be incorporated in the specification to take certain deviations into account.

The deviations can be due to:
- The engine application, e.g. for a ship's propulsion with fixed pitch propeller, for a generator drive or dredger pump drive (full torque, variable speed).
- The ambient conditions, e.g. variations in the atmospheric pressure or in the intake temperature.

Where there is considerable deviation from the ISO conditions, it may be necessary to adjust the output of the engine to the prevailing conditions. For stationary engines, a turbocharger specification may be adapted to the site conditions.

Given the knowledge available nowadays, the specifications virtually exclude the possibility of modern compressors surging. Why then, does surging occur and especially after several years of operation? In some of the cases that we have encountered the reasons have had something to do with the way the turbochargers have been maintained. On a 2-stroke engine with constant pressure turbocharging, two or more turbochargers supply air in parallel to a common air receiver. The compressors and turbines are kept clean by waterwashing during operation. Sometimes, "dry cleaning" is used for the turbines. The intervals between overhauls that include dismantling of the turbochargers are long. And on ships running a tight time schedule, not all the turbochargers can be overhauled at the same time. The state of wear and cleanliness changes with time. When the differences become considerable, the turbocharger efficiency will also change. The charger in the worst condition may start surging. The turbochargers on the engine have practically the same intake and exhaust gas temperature as well as the same pressure ratios over the compressors and over the turbines (Fig. 5).

The combined performance of the turbochargers yields an average efficiency. The turbocharger with the lower efficiency must achieve the pressure ratio prescribed by the other turbocharger(s) and will react by reducing its air mass flow. This may result in surging. The influence of turbine wear can be seen in Fig. 8. The turbocharger with the worn turbine works in parallel with turbochargers in a better condition. It must achieve the pressure ratio prescribed by the others even though it tends to run at a somewhat lower speed. A considerable difference in the state of turbine wear may also result in surging, starting from the "worst" charger.

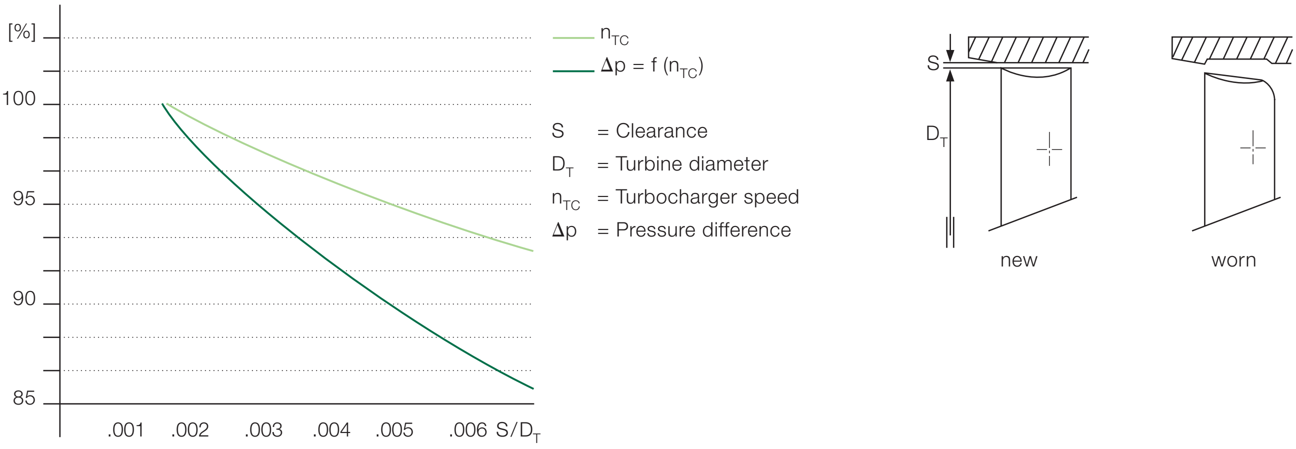


Fig. 8: Influence of wear on turbine blades and cover ring.

When surging begins, it is usual to first reduce the charge air pressure by reducing the engine output slightly or blowing off some air from the air receiver. Such situations, which arise after many years of operation, have been met with in practice. The time schedules of the ships being all-important, it may be difficult to plan maintenance so that more than one turbocharger can be overhauled at the same time. In such cases, the charger specification can be adapted so as to cope better with the prevailing situation.

Modern turbochargers having compressors with backswept vanes perform very satisfactorily on 4-stroke engines and generally exhibit good stability. The reasons for surging occurring are usually due to a reduction in airflow, caused by obstruction of the air flowpath. This may be due to:

- A dirty filter in the silencer-filter
- Increased backpressure after the turbine (dirty exhaust boiler, dirty exhaust silencer)
- A dirty aircooler (airside)
- A high aircooler cooling water temperature
- A dirty, contaminated nozzle ring/turbine
- The ship's hull being dirty, causing the engine to run at full torque/reduced speed

A schematic of the flow path for the charge air and the exhaust gas is given in Fig. 9. The various flow obstructions generally develop simultaneously, although they will not be equally severe. Margins for deviations from the layout data are incorporated in the turbocharger specification. Thus, only extreme cases of airflow obstruction are likely to make the compressor begin to surge.

Of course, there are many more reasons (e.g. concerning the engine and its adjustment) for the turbocharger running outside of its design range and for surging starting. However, it would go beyond the scope of this article to consider them. A reasonable maintenance schedule aimed at keeping the engine and charger(s) in good condition is the practical way to prevent unexpected anomalies from occurring.

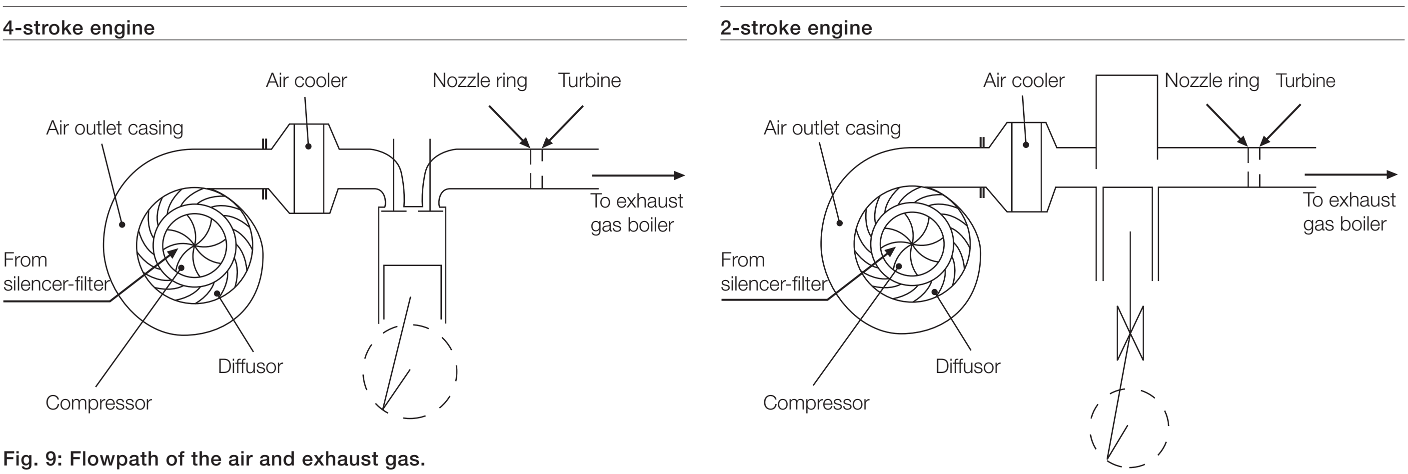


Fig. 9: Flowpath of the air and exhaust gas.

Turbocharger turbines: aspects of axial and radial turbines

Exhaust gas turbines of two completely different designs – axial flow and radial flow – are used to drive turbocharger compressors. The author considers the pros and cons of these turbine types for various turbocharger applications.

Fig. 1: The windmill – ancestor of the axial turbine. This type has been in use since the 13th century.

The axial turbine

We have seen from a look at the compressors (TM 1/1995) that air can be compressed by spinning a wheel with appropriate blading in a suitable housing. In a turbine, the opposite happens. Compressed air, or in the case of the turbocharger turbine exhaust gas, is blown against suitably shaped blading to make the turbine rotor spin and thereby generate power with which to drive the compressor.

These ideas are not new. Fig. 1 shows a very basic example of the axial turbine, a windmill. This particular type was in use as early as the 13th century. For turbocharger turbines, the air, or rather the gas flow has to be used much more efficiently. The turbine is therefore equipped with a large number of blades and the gas is guided to these blades through carefully directed nozzles. The cutaway drawing of a turbocharger in Fig. 2 shows the arrangement of the nozzles and the turbine.

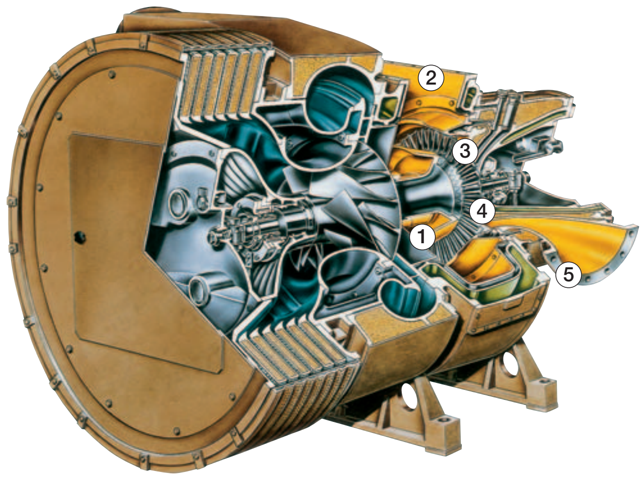
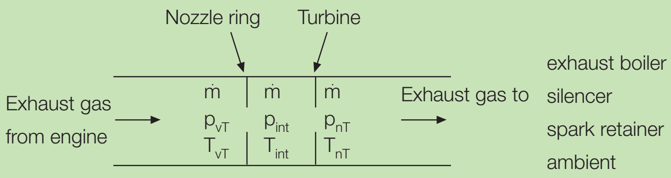


Fig. 2: VTR..4E turbocharger: Arrangement of nozzle ring and turbine. 1 = Diffusor after turbine, 2 = Gas outlet casing, 3 = Nozzle ring, 4 = Turbine, 5 = Gas inlet casing.



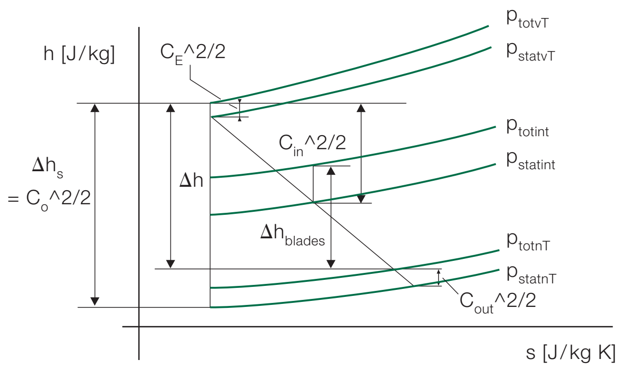
ṁ	Gas mass flow, kg/s
p_{vT}	Pressure before turbine, bar
T_{vT}	Temperature before turbine, K (°C + 273)
p_{int}	Pressure between nozzle ring and turbine, bar
T_{int}	Temperature between nozzle ring and turbine, K
p_{nT}	Pressure after turbine, bar
T_{nT}	Temperature after turbine, K

Fig. 3: Flowpath of the exhaust gas through the nozzle ring and turbine.

The flow path of the gas through the turbine is shown schematically in Fig. 3. Exhaust gas from the engine enters the gas inlet casing and flows to the nozzles. The nozzles direct the gas jets at the turbine blades, where the energy is transferred from the gas to the turbine. As the gas passes through the nozzle ring, its velocity increases considerably, but no power is extracted here. The pressure between the nozzle ring and the turbine blades, p_{int}, decreases due to the gas expanding. Further expansion over the turbine blades, down to the back pressure after the turbine, allows power to be extracted from the exhaust gas.

In the field of thermodynamics, it is usual to show these processes in so-called enthalpy-entropy diagrams. The enthalpy is the specific energy content per unit of gas mass (Joule/kg) while the entropy is a variable of state, expressed in Joule/kg °K.

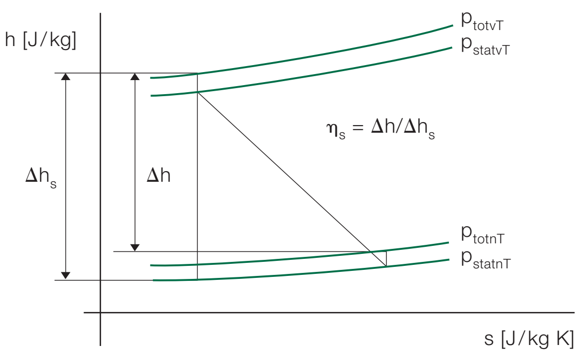
The turbocharger has a single-stage turbine which allows the combination of the nozzle ring and turbine to be considered as one flow area, thereby eliminating the intermediate pressure and temperature, which are very difficult to measure due to the high gas velocities between the nozzle ring and turbine blades.



h, s, Δh_s, Δh, see Fig. 4

C_o	Gas velocity corresponding to isentropic enthalpy drop Δh_s, m/s
C_E	Gas velocity at nozzle ring inlet, m/s
C_{in}	Gas velocity at turbine blade inlet (equal to velocity at nozzle ring outlet), m/s
C_{out}	Gas velocity at turbine outlet, m/s
Δh_{blades}	Enthalpy drop over turbine blades
$\Delta h_{blades}/\Delta h_s$	Degree of reaction

Fig. 5: Enthalpy-entropy diagram showing absolute velocities.



Δh_s	Isentropic enthalpy drop, J/kg
Δh	Usable enthalpy drop, J/kg
η_s	Isentropic efficiency

Fig. 4: Enthalpy-entropy diagram for the nozzle ring and turbine assembly.

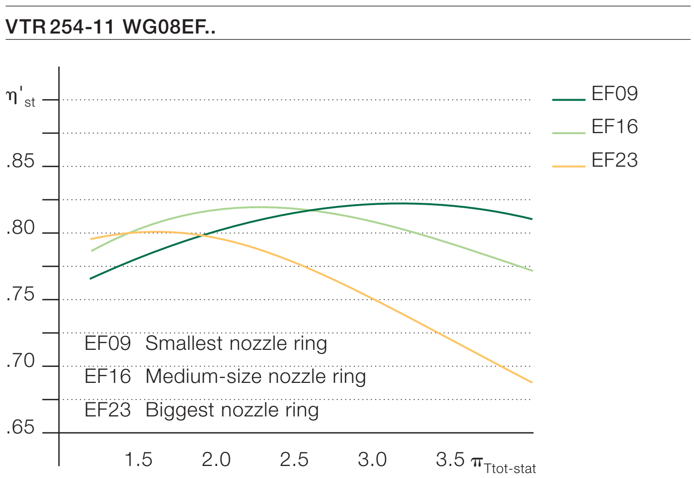


Fig. 6: Isentropic turbine efficiency versus pressure ratio $\pi_{TtotvT\text{-}statnT}$.

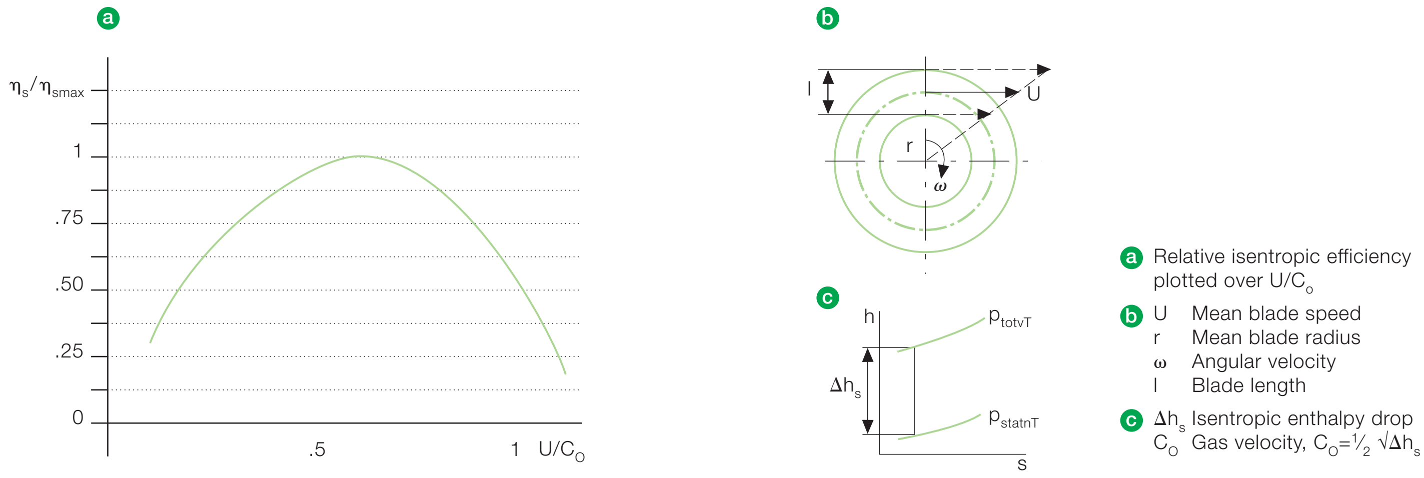


Fig. 7: Relative isentropic efficiency versus U/C_O

Looking at Fig. 4, it is seen that for a given pressure ratio p_{totvT}/p_{statnT}, Δh_s depicts the specific energy difference (enthalpy drop) without losses to the surroundings. No increase in entropy takes place. However, the technical processes we apply always incur losses. The processes are said to be irreversible. The entropy has to increase. Δh depicts the enthalpy drop that can be used. The ratio $\Delta h/\Delta h_s$ is referred to as the isentropic efficiency.

The gases have a certain velocity on arriving at the nozzle ring, where the gas expands, increasing the velocity. The gas expands further over the turbine blades, which run a certain speed, and exits the blading at a relatively low velocity. The gas velocity-enthalpy relationship at the different points can be seen in Fig. 5.

The enthalpy drop between the nozzle ring inlet and the turbine exhaust can be divided into several parts – the gas velocity at the nozzle ring inlet, the turbine blade inlet (equal to the nozzle ring outlet velocity) and the turbine blade outlet. The gas velocity at the turbine blade inlet is converted into usable output with the best possible efficiency that can be achieved. Obviously, after the gas has left the turbine blades it cannot perform any more work.

If the entire enthalpy drop (expansion pressure ratio) were used to increase the gas speed in the nozzle ring, we would speak of an impulse turbine. But if only a part is used and the remaining enthalpy (pressure ratio p_{int}/p_{nT}) expands over the turbine blades, we speak of a reaction turbine. The degree of reaction is the ratio of enthalpy drop over the turbine blades to the total enthalpy drop.

From Fig. 3 we have seen that there is an intermediate pressure, p_{int}, between the nozzle ring and the turbine blades. Different turbine blade heights can be applied for a given turbocharger frame size. Nozzle rings with a gas flow area ranging from small to large may be chosen for every blade height. A small nozzle ring area combined with a turbine of fixed area will cause the intermediate pressure to be low. The degree of reaction will be low. A large nozzle ring area for the same turbine will result in p_{int} being higher, as will the degree of reaction.

The degree of reaction influences the turbine efficiency. In Fig. 6, the turbine isentropic efficiencies have been plotted over the pressure ratios π_T (p_{totvT}/p_{statnT}). The figure also shows clearly that, depending on the engine application requirements, a combination of nozzle ring and turbine can be chosen for higher turbine efficiency at full load or part-load operation of the engine.

Turbochargers are frequently fitted to engines with a pulse turbocharging system. The pressure ratio over the turbine is then highly variable. To determine the influence of the turbine efficiency on this type of operation, it is convenient to plot the efficiency over the so-called blade speed ratio, U/C_o. U is the mean turbine blade speed and C_o is the gas velocity corresponding to the isentropic enthalpy drop over the nozzle ring and turbine assembly. Fig. 7 shows the main details.

Fig. 8 depicts the exhaust gas pressure during the gas exchange period of a 4-stroke engine. The mean turbine blade speed, U, is constant, while C_o varies with the gas pressure and temperature. U/C_o changes accordingly, and with it the efficiency. Fortunately, at low pressures, when the efficiency is low, only a small enthalpy drop and a reduced gas mass are involved, so that not much energy is converted with low efficiency.

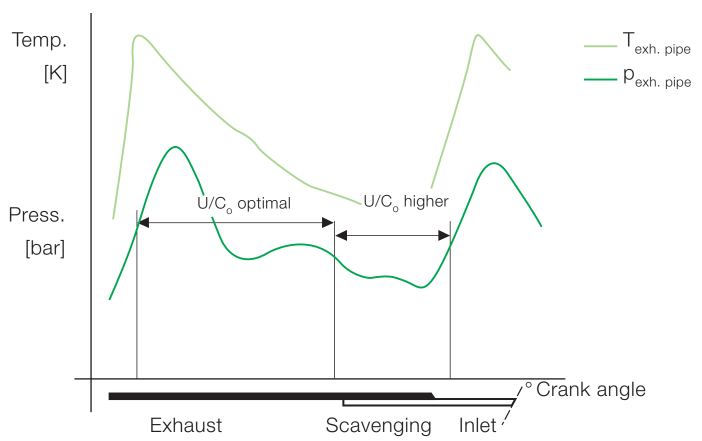


Fig. 8: Period of gas exchange in a 4-stroke engine with 3-pulse turbocharging.

Fig. 9: Turbine blades with profile twisted from root to tip to take account of the increase in blade speed with the blade length.

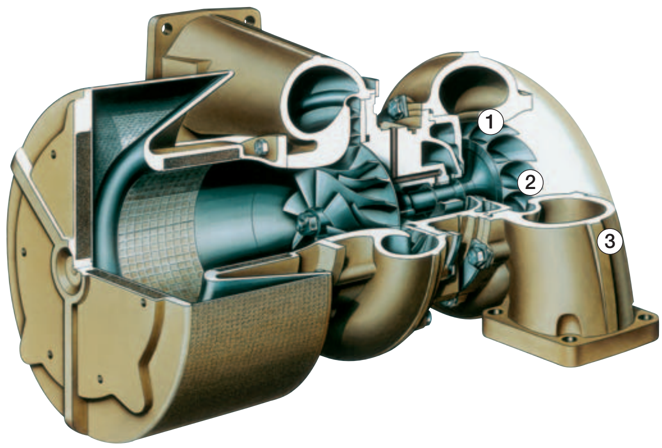


Fig. 11: RR..1 turbocharger. 1 = Volute outlet around the turbine, 2 = Turbine, 3 = Gas inlet casing.

The nozzle ring blades have a fixed outlet angle. The shape of a turbine blade is shown in Fig. 9. Its profile is twisted from root to tip to take into account the fact that the circumferential speed U increases with the radius. It can be seen in Fig. 10 that the twisted turbine blade is necessary for optimal flow conditions over the blade length.

To make best possible use of the energy contained in the exhaust gases, correctly shaped gas-inlet casings must be provided to keep pressure losses due to the flow to a minimum. For this reason, channels are used in which the gas velocity gradually accelerates from the inlet to the nozzle ring (see Fig. 2). In addition, the gas passage after the turbine has also been configured to limit pressure losses. The diffusor has a shape which allows the low static pressure after the turbine to be utilized, thereby increasing the available pressure ratio p_{tot}/p_{stat} over the turbine assembly and the corresponding enthalpy drop. This layout is used in VTR..4E turbochargers, where very high overall efficiencies are required.

Turbochargers with axial turbines are used on medium-speed and slow-speed engines. They are perfectly capable of accepting the exhaust gas from engines running on heavy fuel oil. The turbine retains its high efficiency over a very long period of time, the more so when reasonable maintenance is provided. The axial turbine is able to supply an adequate output with good efficiency to drive the compressor from low pressure ratios upwards, thus assuring good part-load performance of the engine. The latter is especially important for fixed-pitch propeller drives. Turbochargers with axial turbines are found on ships, in diesel power stations and on diesel locomotives, dredgers, etc.

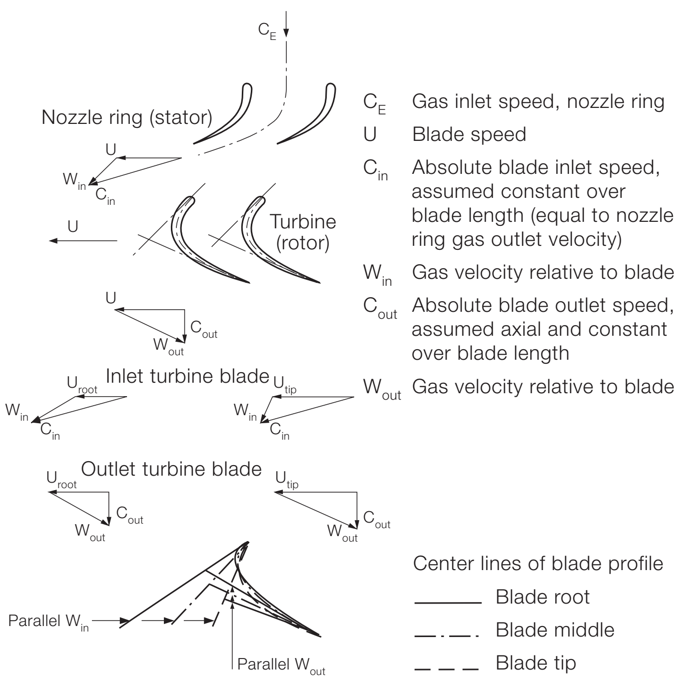


Fig. 10: Velocity diagrams for blade profiles at the blade middle, blade root and blade tip. Comparison of profile center lines.

The radial turbine

A turbocharger fitted with a radial turbine of the so-called mixed flow type is shown in Fig. 11. This particular design, a type RR..1, has no nozzle ring. The gas inlet casing is shaped in such a way that the required exhaust gas velocity at the turbine inlet is obtained by correct dimensioning of the circular outlet of the volute around the turbine. For different applications different gas inlet casings are needed. This design is used especially for large series of turbochargers having the same turbine specification. A construction with nozzle ring is advantageous for easy adaptation.

In the enthalpy-entropy diagram, the thermodynamic process of the radial turbine can be shown in the same way as for the axial turbine (Fig. 5). Unlike the axial turbine, the radial turbine has an inlet with a diameter which is larger than that of the outlet (Fig. 13). The energy transfer to the rotor can be written as shown in Fig. 12.

$$\Delta h = \tfrac{1}{2}\cdot((U_{in}{}^{\wedge}2 - U_{out}{}^{\wedge}2) - (W_{in}{}^{\wedge}2 - W_{out}{}^{\wedge}2) + (C_{in}{}^{\wedge}2 - C_{out}{}^{\wedge}2))$$

Fig. 12

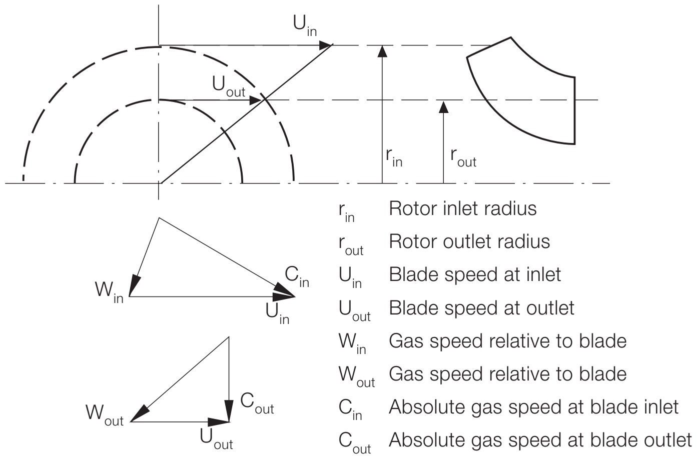


Fig. 13: Schematic of turbine rotor showing inlet and outlet radius and speed.

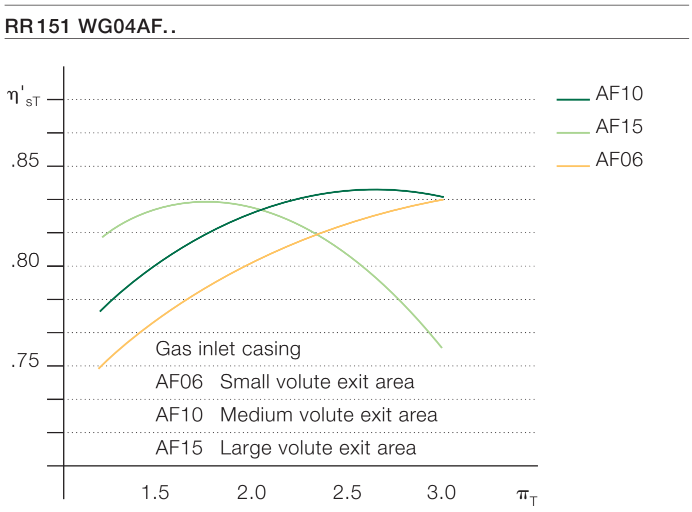


Fig. 14: Isentropic turbine efficiency plotted over expansion ratio p_{totvT}/p_{statnT}.

The difference between the circumferential speeds adds considerably to the turbine work. If care is taken to see that the gas is accelerated while expanding in the rotor passages (relative velocity $W_{out} > W_{in}$), this will also make a positive contribution to the turbine work. It is clear that the absolute velocity C_{out} cannot do any work and should be as low as possible. Considering this situation, it would appear that for small (automotive) turbochargers the radial turbine is a must. An axial turbine with small dimensions is not feasible in practice. The small-size radial turbine does not suffer from efficiency loss due to clearance leakages as much as a small axial turbine would.

Turbochargers with radial turbines can be produced more economically. This is the reason for the overlap of turbocharger sizes where both axial and radial turbines are available. Their application then depends on the engine requirements.

The efficiency of a typical RR..1 turbine is plotted over its pressure ratio in Fig. 14. At low pressure ratios there is a wide gap between the efficiency with a large and with a small volute exit area. The large volute exit area causes the degree of reaction to be relatively high, so that the influence of centrifugal force working in a direction opposite to the gas flow has less influence than when the volute exit area is small. At high pressure ratios the influence of the centrifugal force on the gas particles is not so significant due to the much higher energy content of the gas. The small volute exit area exhibits high efficiency for a relatively low degree of reaction, the large volute exit area exhibiting low efficiency.

Under adverse conditions – gases with a low energy content, high turbocharger speed (e.g. a 4-stroke engine at full speed with no load) – the turbine may tend to act as a centrifugal compressor due to the centrifugal force acting on the gas particles.

Turbocharging of automotive engines is already under way, so that small turbochargers with radial turbines can be said to have found their place in the engine industry. Their use on high-speed industrial engines is also widely accepted. However, for bigger engines running on poor-quality fuels the axial turbine is generally the preferred type. ABB supplies turbochargers with radial turbines for engines rated at up to about 2000 kW.

The above overview is just that – an overview. Details of the performance and operation of the turbocharger turbines would go beyond the scope of this article. If you have a question that the article does not answer, please send it to us. We will do our best to answer it.

Turbocharging over the years

Turbocharging has played a vital role in the development of the diesel engine. This article takes us back to its beginnings and charts its progress to the present day.

The idea of supplying air under pressure to a diesel engine was voiced by none other than Dr. Rudolf Diesel as early as 1896. The use of a turbocharger for this purpose, however, was the result of work by a Swiss, Alfred Büchi, who patented his so-called "pulse system" in 1925. This system feeds the exhaust gases of the engine through narrow pipes to the turbocharger turbine, thus driving the compressor. The pressure variation in the small-volume pipes allows overlapping of the inlet and exhaust, permitting scavenging of the compression space of the engine cylinders with clean air. Cylinders that do not disturb each other's scavenging process can be connected to one pipe (turbine gas inlet) in accordance with the firing order of the engine (see Fig. 1). This pulse system was the foundation for the future success of turbocharging.

In December 1928, Alfred Büchi gave a lecture at the Royal Institute of Engineers at The Hague in the Netherlands. From this lecture, we learn that already then it was known that the thermal load of a diesel engine does not essentially increase when turbocharged (Fig. 2). An improvement in fuel consumption due to the better mechanical efficiency could also be shown. And it was seen from a comparison of four turbocharged 10-cylinder single-acting 4-stroke engines and four turbocharged 6-cylinder double-acting 4-stroke engines used for a ship's propulsion installation rated at 36,000 bhp that the

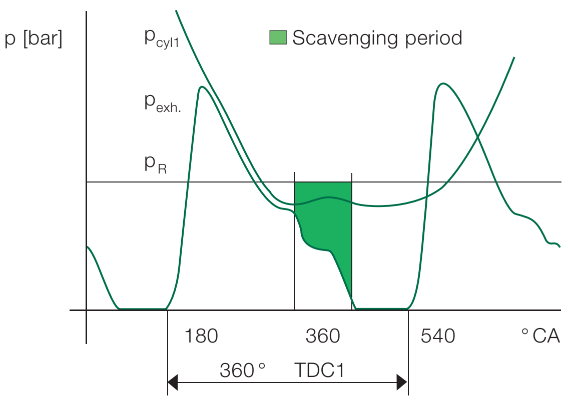


Fig. 1: Diagram showing the pressure p_{cyl} in the cylinder, p_{exh} in the exhaust pipe and p_r in the air receiver of an 8-cylinder engine. The scavenging period, where the inlet and exhaust are simultaneously open, is not disturbed by the exhaust pulses of other cylinders.

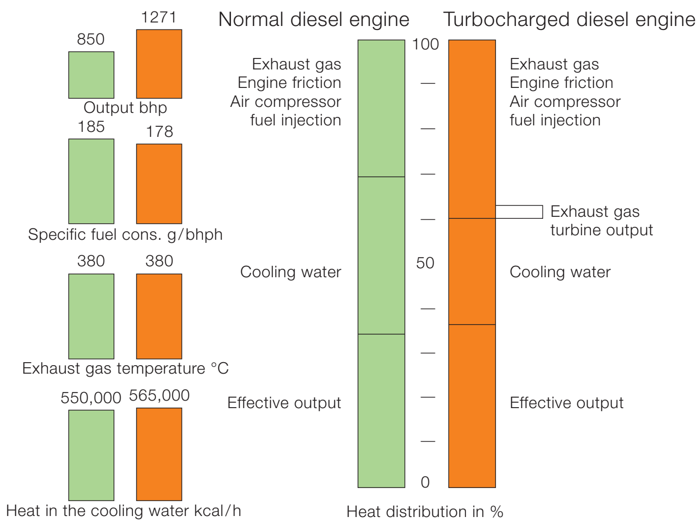


Fig. 2: Some engine test data from the lecture given by Alfred Büchi in 1928.

double-acting engines offered several advantages. Neither single-acting nor double-acting 2-stroke engines could reasonably compete. Thought was given very early on to turbocharging the engine-driven scavenging pumps of the 2-stroke engine. However, since the turbocharger efficiency necessary for this was not available, it took many more years for this goal to be reached.

With the market introduction of the VTR..0 turbocharger, it became possible to turbocharge 2-stroke engines with engine-driven scavenging air pumps. 2-stroke engine builders began to make use of this possibility in a big way after 1945. However, to eliminate the scavenging pumps and reduce fuel consumption, the turbocharging system has to be a pulse system. This feature was introduced successfully in 1951 on a B&W marine propulsion engine.

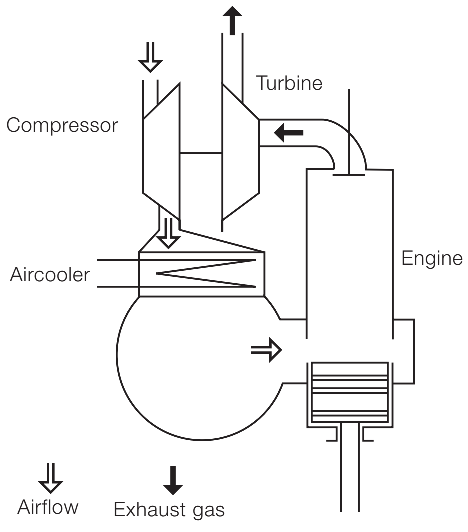
Use of the VTR-type turbochargers on 2-stroke and 4-stroke engines increased rapidly. Through continuous development, improvements were made to the achievable compressor pressure ratios, diesel engine designs (for higher permissible firing pressures) and fuel injection systems. These improvements and broad utilization of the thermodynamic properties of the diesel cycle have led to the current state of the art of turbocharging.

2-stroke engines

It is interesting to look at some of the development stages through which turbocharging of 2-stroke engines has passed. For the sake of simplicity, longitudinally scavenged engines (i.e. with air inlet ports and exhaust valves) will be considered.

The air supply of a 2-stroke engine without turbochargers is provided by scavenging pumps. The VTR..0 turbochargers were used in two lines of engine development, one retaining the scavenging pumps and one omitting them but using the pulse system to obtain sufficient energy to drive the turbocharger(s) over the entire load and speed range of the engine. The typical arrangements of the turbochargers are shown in Fig. 3. Fig. 4 compares the timing of the inlet and exhaust for the constant pressure scavenging pump system with that of the pulse system without scavenging pumps. Fig. 5 shows a Götaverken engine with an 850 mm bore, a stroke of 1700 mm and 11 cylinders, each cylinder having two double-acting scavenging pumps with a 480 mm bore, turbocharged with three VTR 630 units. A Burmeister & Wain engine of type 10 K98FF with four VTR 750 turbochargers in pulse operation is shown in Fig. 6. The photographs were taken around 1970.

Arrangement for 2-stroke engine with pulse turbocharging (B&W 1970)



Arrangement for 2-stroke engine with constant pressure turbocharging and scavenging pumps in series with the turbocharger compressor (Götaverken 1970)

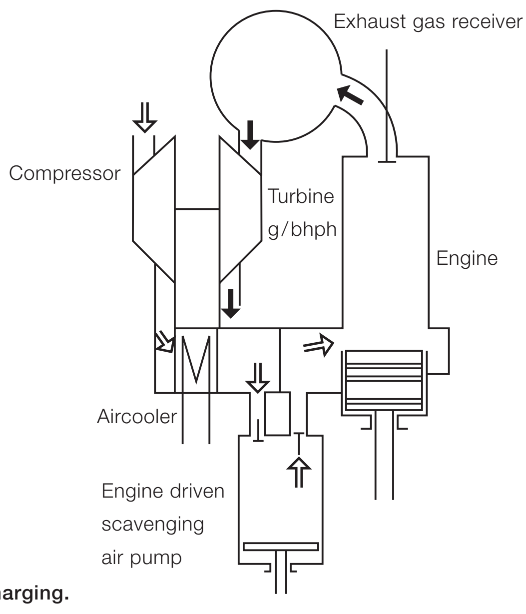


Fig. 3: Diagrams of 2-stroke engines featuring pulse and constant pressure turbocharging.

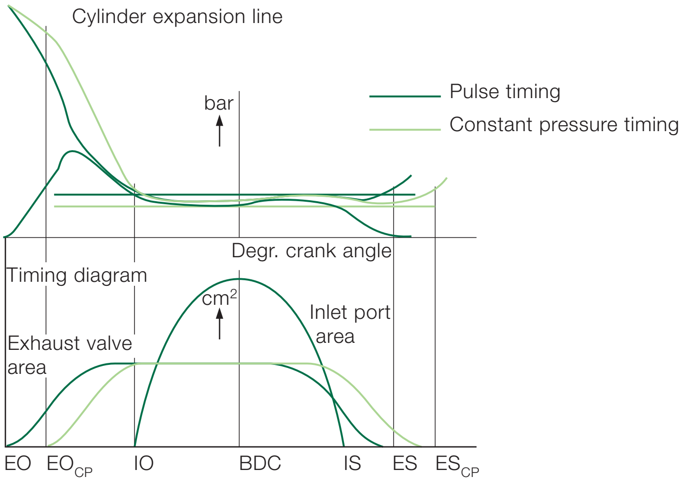


Fig. 4: Timing of inlet and exhaust for pulse and constant pressure turbocharging.

The advantage of the pulse system was that no mechanical system was necessary at any point in the engine load range to guarantee the air supply. This was also due to the very high mechanical efficiency of the VTR turbochargers. The exhaust gas energy needed was obtained by opening the exhaust valve early (Fig. 4). However, this reduced the effective expansion stroke of the engine.

The engine with scavenging pumps and featuring constant pressure turbocharging can make do with a much later opening point for the exhaust valve, but takes part of the energy for the air supply from the crankshaft. The engine has to drive its own scavenging pumps. Thus, the fuel consumption of these types of engine was not essentially different, amounting to 150 – 155 g/bhph (204 – 211 g/kWh).

From the above, it is obvious that the combination of constant pressure turbocharging and no scavenging pumps would improve the fuel consumption. This became possible with the introduction of turbochargers with improved efficiency. The VTR..1 was introduced in 1971. Changeover from pulse to constant pressure turbocharging on the B&W engines came in due course, and the expected reduction in fuel consumption with it (Fig. 7). An electrical auxiliary blower had to be used to start the engine and for low part-loads.

The higher pressure ratios and higher efficiencies with the VTR..4 turbochargers allowed further increases in output and a reduction in the specific fuel consumption. A better understanding of the thermodynamics of the diesel cycle and the properties of the air and gas flows has resulted in only the longitudinally scavenged 2-stroke diesel engines being able to hold their own in the marketplace. VTR..4D turbochargers (combining high pressure ratio and high efficiency) meet the present-day requirements of the engine manufacturers (actual status: pressure ratio 3.7:1, mean effective pressure 18 – 19 bar).

Fig. 5: Götaverken engine of type VGS11U, rated 26,400 bhp, 19,400 kW at 119 rev/min. Three VTR 630 turbochargers are installed.

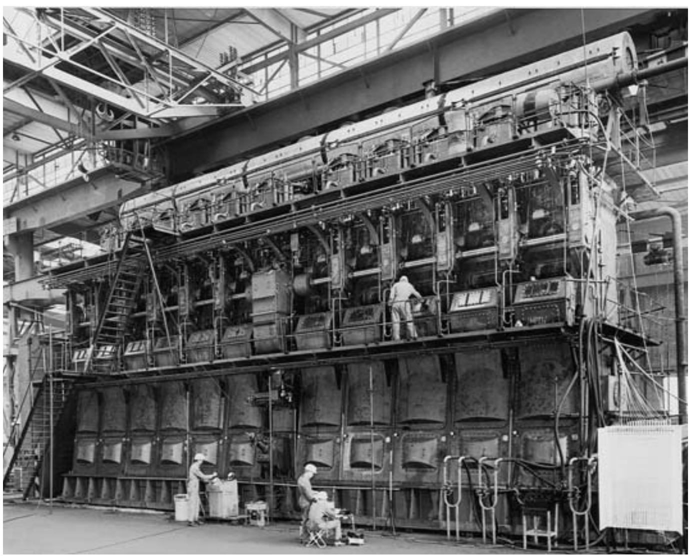
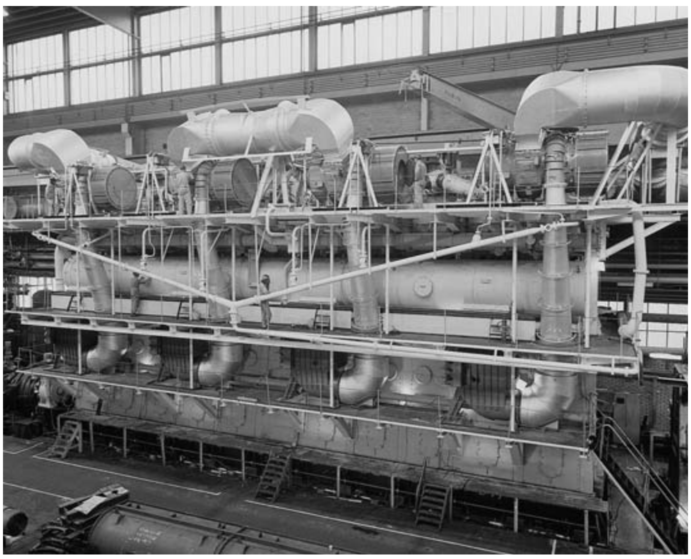


Fig. 6: Eriksberg B&W engine of type 10K98FF, rated 38,000 bhp, 27,950 kW at 103 rev/min, fitted with four VTR 750 units.



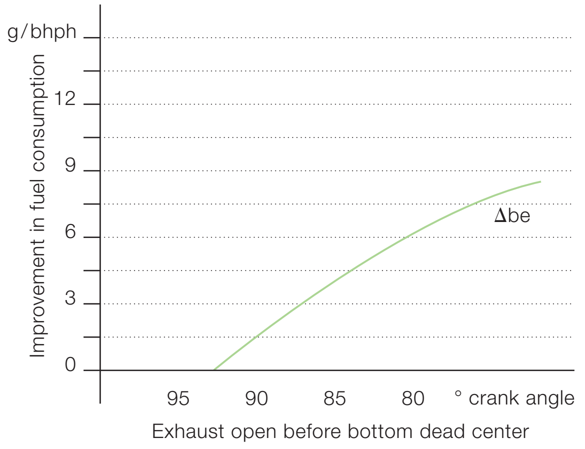


Fig. 7: Improvement in fuel consumption due to later opening of the exhaust valve.

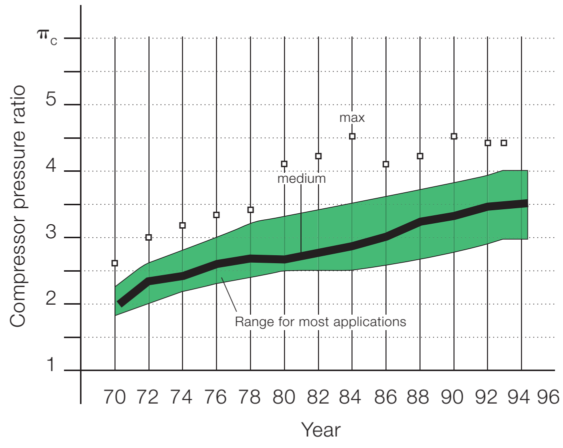


Fig. 8: Increase in compressor pressure ratio over the years.

The next turbocharger generation, the TPL series, is in the process of being introduced and will allow further improvements in engine output and fuel consumption.

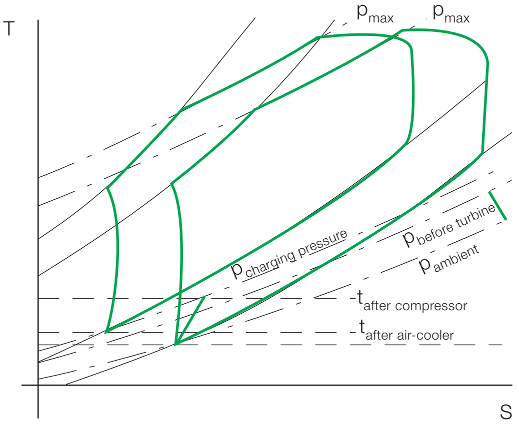
4-stroke engines

The turbocharging of 4-stroke engines has, of course, also been strongly influenced by the availability of adequate turbochargers. While the 4-stroke engine is basically self-aspirating, it is not so dependent on high turbocharger efficiency. What is it then, that makes turbocharging so attractive?

The turbocharger allows:
- an increase in engine output for a given speed which is approximately proportional to the absolute charging pressure supplied by the turbocharger,
- a reduction in required space for the installation of a specific engine output,
- more favorable fuel consumption.

The increase in compressor pressure ratio over the years is shown in Fig. 8. Twenty years were needed to increase it from about 1.3 to 2 in 1970. Initially, a given 4-stroke engine would have a compression pressure of, say, 35 bar, and a maximum firing pressure of 50 bar. To achieve a charging pressure of 0.3 bar gauge, the compression ratio of the engine would be reduced from approximately 13.4 to 11.1. The compression and maximum firing pressure would then remain the same, but the output would increase by some 30 percent. This would eventually have a negative influence on starting the engine.

Encouraged by the results achieved, the engine manufacturers designed for higher pressures. The introduction of air-coolers to reduce the temperature of the compressed air entering the engine cylinders helped to increase the engine output at more moderate charging pressure levels (e.g. a mean effective pressure of 10 bar, a charging pressure of 0.6 bar gauge with air-cooler and 0.85 bar without air-cooler, for the same charge of air (kg) in the cylinders; without an air-cooler the exhaust gas temperature would be some 50 °C higher).



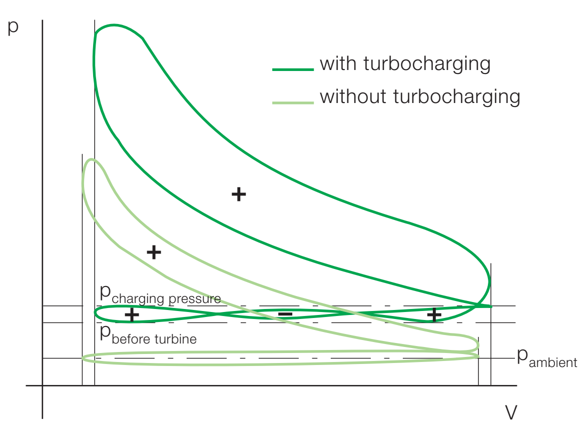


Fig. 9: Comparison of the cylinder diagrams of a 4-stroke engine with and without turbocharging.

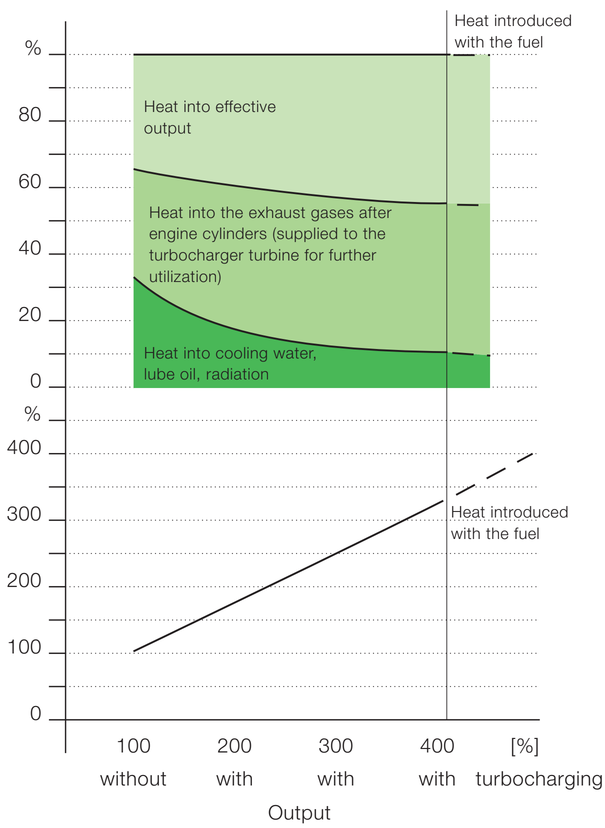


Fig. 10: Distribution of the heat introduced with the fuel for an increasing level of turbocharging.

Fig. 9 shows the difference in the cylinder diagram of a 4-stroke engine with and without turbocharging. It is easily seen that the turbocharged engine has considerable advantages. In the T-S diagram the turbocharged engine shows a smaller increase in entropy (S) than the non-turbocharged engine. This results, thermodynamically, in better efficiency. In this diagram, the slope of the lines of constant pressure becomes steeper with increasing pressure. From the p-V diagram, it is seen that the area representing positive work increases considerably. The turbocharged engine shows positive work in the area representing the gas exchange. For the non-turbocharged engine, only negative work is shown. The area representing positive work includes the output required to overcome the friction of the engine. This friction is largely dependent on the speed of the engine and to a lesser extent the pressures. This is the reason for the mechanical efficiency of an engine steadily increasing with the degree of turbocharging. Together, these factors mean that, essentially, a turbocharged engine will exhibit decreasing specific fuel consumption with an increasing degree of turbocharging. In addition, deeper knowledge of the thermodynamic properties of the diesel process and a better understanding of the fuel injection systems have helped to substantially reduce the specific fuel consumption of diesel engines.

With the steady increase in the mean effective pressures, the exhaust gas temperatures have remained approximately constant due to the higher charging pressures. This and the cooling of the air increases the density of the air entering the cylinders. In this way, the ratio of air-to-fuel (kg/sec/kg/sec) could be kept more or less constant. The distribution of the heat supplied with the fuel, however, changes considerably

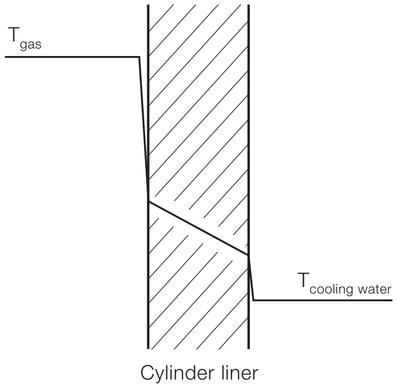


Fig. 11: Heat transfer from the hot-gas side to the cooling water through the wall of the cylinder liner.

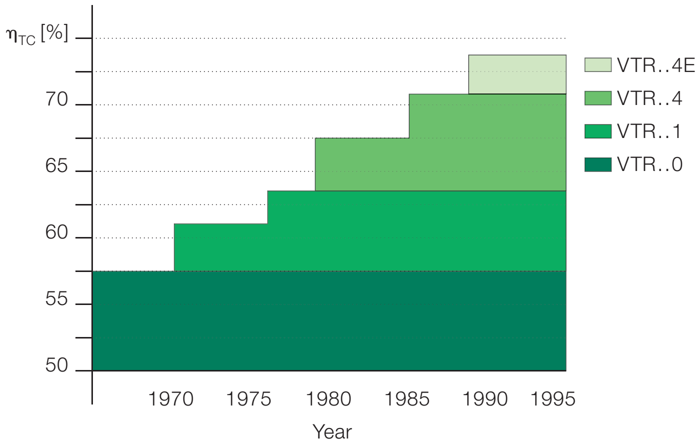


Fig. 12: Increase in turbocharger efficiency as a result of continuous development over the years.

with the increasing level of turbocharging. This is illustrated in Fig. 10. The heat in the lubricating oil represents roughly the output needed to overcome friction, i.e. the mechanical efficiency of the engine. The heat in the cooling water is roughly equivalent to the heat load of the combustion chamber parts. With turbocharging, we succeed in keeping the heat flow to the cooling water, in absolute terms, approximately constant, this being largely independent of the level of turbocharging. This is also one of the most important successes of turbocharging. Alfred Büchi mentioned this in his lecture of 1928, and it still holds true today. The very simplified explanation is shown in Fig. 11. The heat transmission coefficient from the gas to the wall depends on the pressure ($\sqrt{p_{gas}}$). The heat transmission coefficients through the wall and from the wall to the cooling water have hardly changed. The process temperatures still being at the same level as without turbocharging (fuel-air ratio) and the surface areas still the same, the heat flow to the cooling water could not change very much.

Further advantages of turbocharging

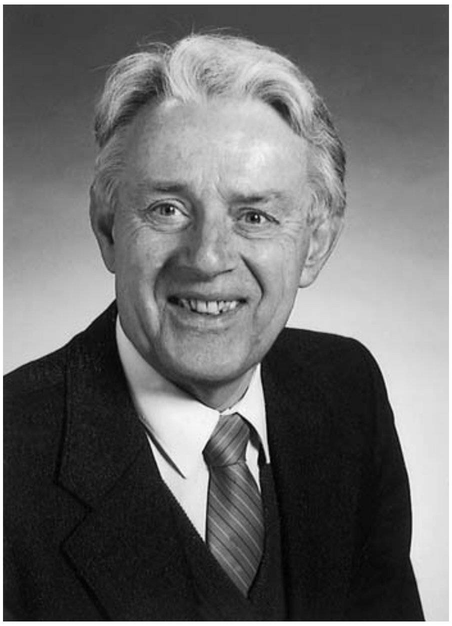
The state of the art for modern engines is an output 4 times (400 percent) as high as the non-turbocharged engine for the same speed and dimensions. From Fig. 10, it can be seen that the heat introduced with the fuel at 400 percent output has increased to only 320 percent. This means that the specific fuel oil consumption (sfoc) of modern engines amounts to only 80 percent of the sfoc of the non-turbocharged engine. Thus, turbocharged engines are a must in terms of environmental protection. Fuel that is not burnt neither causes air pollution nor produces CO_2. Also, the modern turbocharger combining high pressure ratios with high efficiency is an important factor when the engine process has to be adjusted to achieve a low NO_x emission level through low process temperatures (high air-to-fuel ratio, high charging pressure). Put in general terms, the materials and energy required to produce a turbocharged engine for a given output and speed are considerably less than for a non-turbocharged engine for the same output and speed.

To complete this overview, Fig. 12 shows how the turbocharger efficiencies have developed up until the present day. As high efficiencies allow a large turbine area and this in turn reduces the work necessary to expel the exhaust gas from the 4-stroke engine cylinders, the full-load fuel consumption is influenced favourably. The 2-stroke engine may further reduce the timing for the inlet and exhaust, or allow a power turbine to be installed to convert excess exhaust gas energy into useful output.

Looking to the future, ABB is preparing to introduce its new TPS and TPL turbocharger designs combining compactness with high pressure ratios and high efficiencies. Since non-turbocharged engines will hardly be built anymore, the future looks bright for our turbochargers.

About the author

From 1992 to 1996 Turbo Magazine carried a series of articles by Johan Schieman on the subject of operating turbochargers. The series came to an end with Johan's death in April 1996. This collection of his articles is both a response to the many requests received from readers and a modest memorial to the author.

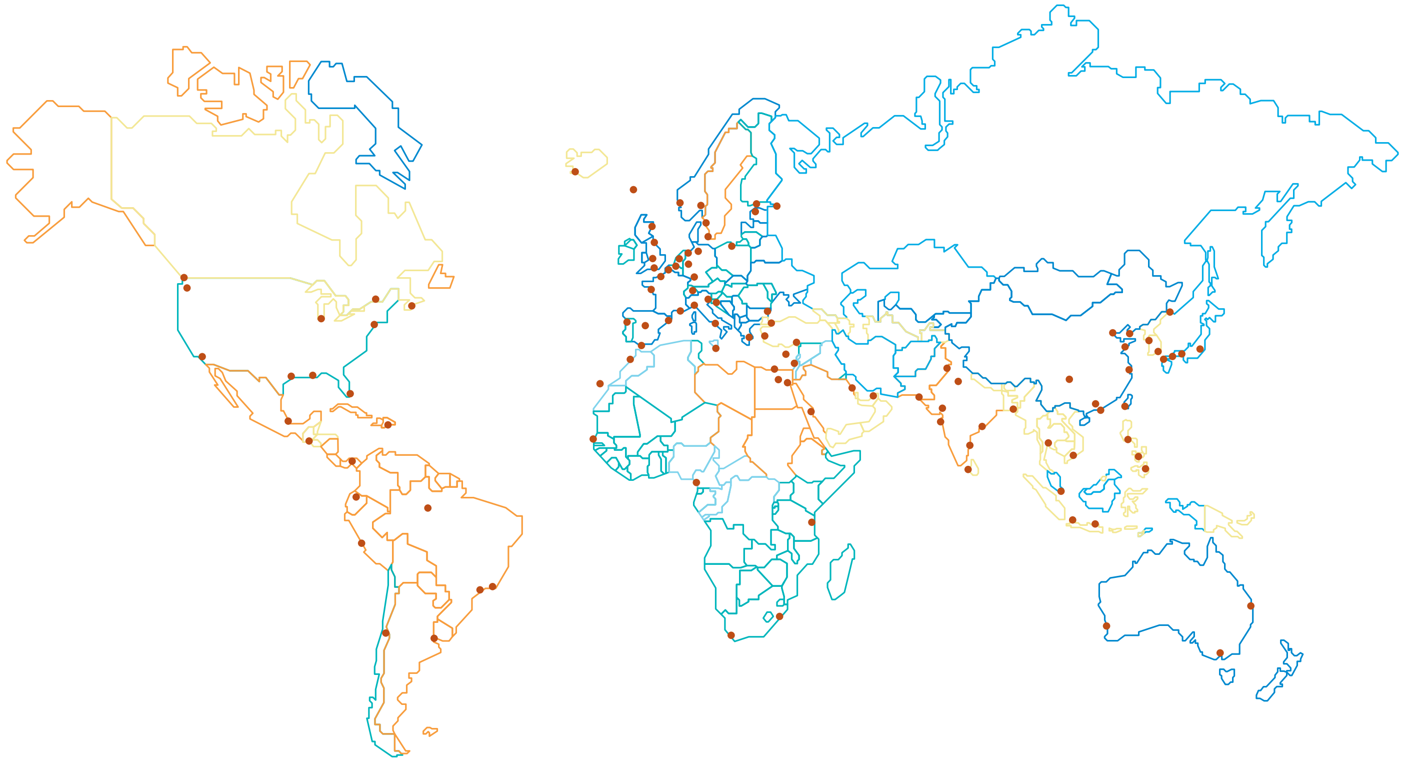


Johan Schieman's insights into the workings of turbochargers made the articles an instant and eminent success. Highly instructive, they were much appreciated, especially by ship's operators, who kept additional copies on board their vessels as reference for their engine crew. Schools used the articles for teaching and commercial trade journals asked for permission to reprint them.

In 1956 Johan Schieman joined ABB (then Brown Boveri) as a sales and application engineer at our Rotterdam offices. His many visits to Baden made him a familiar face at headquarters, and in 1967 he was persuaded to stay "for three years". He remained in Baden for the rest of his life. During 38 years with ABB Turbochargers Johan became a reputed specialist in the turbocharging of 2-stroke diesel engines and an experienced China hand to boot. From 1985 until 1987 he was the company's liaison to licensees and engine builders in China, based in Hong Kong. In the following years he visited China two or three times a year. His interest in foreign cultures went as far as to study the Chinese language, along with other tongues like Spanish and Swedish. The latter was the result of having worked in Scandinavia, where he was instrumental in the introduction of ABB turbocharging technology for 2-stroke diesel engines at Götaverken and B&W.

Johan Schieman had that rare talent of being able to motivate others to take a genuine interest in his work. Through his ability to communicate, the sincerity of his conviction and the perseverance with which he pursued his goals, he became the mentor of many younger application engineers in our company and abroad.

ABB Turbocharging Service network



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