Technical information

ABB Turbocharging
Turbocharging medium speed diesel engines with extreme Miller timing
Abstract
Miller timing is one of the few measures that can be applied in an internal combustion engine to simultaneously reduce NO\textsubscript{x} emissions and fuel consumption – a fact engine builders are acknowledging by introducing it on almost all types of engine.

ABB’s new-generation turbochargers allow operation at very high pressure ratios and with very high turbocharger efficiencies. Engine builders can use the new potential these turbochargers offer either to improve the Miller process or to further increase engine output.

An overview of the theory of the Miller process is followed by a look at how some important parameters of the gas exchange and turbocharging system influence engine operation. Finally, the turbocharging system of a medium speed diesel engine with extreme Miller timing up to 60 °CA before BDC is presented.

Key Words: Miller; emissions; two stage turbocharging
The basic principle underlying the Miller process is that the effective compression stroke can be made shorter than the expansion stroke by suitable shifting of the inlet valve timing. When both the engine output and boost pressure are kept constant, this will reduce the cylinder filling and the pressure and temperature in the cylinders will be lower.

The original purpose of the Miller process was to increase the power density of engines without exceeding their mechanical and thermal limits [8, 11]. In the 1990s, attention turned to how it could be used to reduce the temperature in the cylinders for a constant engine output, and to using this positive effect to minimize NO\textsubscript{x} formation. In the case of gas engines, an additional benefit is that the operating range can be increased since there is less tendency for the engines to “knock”.

The Miller process is one of the few options that engine builders have for simultaneously reducing emissions and improving engine efficiency. Since all engine builders strive to meet engine emission limits without any loss of efficiency, practically every modern engine is operated today with at least moderate Miller timing.

The drawback is that ever-higher boost pressures are necessary for a constant engine output, i.e. increasing demands are made on the turbocharging system. This is clearly shown by the boost pressure versus bmep diagram in Fig. 1. The lower the engine’s charging efficiency ($\lambda_t$), the higher the boost pressure has to be for the engine to achieve its bmep.

The first part of this study presents some of the basic principles of the Miller process and shows how the pressure ratio and turbocharging efficiency affect engines operated with Miller timing.

In a further section, results of experimental work undertaken as part of the CLEAN project at Flensburg University of Applied Sciences’ Institute of Marine Technology are presented. Since the work carried out by ABB Turbo Systems within this project focused on the theoretical optimization of the turbocharging system and on its experimental implementation on the test engine, this section will:

- discuss the possibility of supercharging the engine with the Miller timings that were tested ($\varphi_{IC} = \text{up to 60}^\circ\text{CA before BDC}$);
- show the limits of single stage turbocharging; and
- present and analyze, based on the results of the measurements, controlled two stage turbocharging as a solution that allowed the test engine to be operated with the extreme Miller timing of $\varphi_{IC} = \text{60}^\circ\text{CA before BDC}$.

All simulations were performed with the SiSy program system [2], which makes use of the Woschni/Heider model [6] to calculate the NO\textsubscript{x} formation.
In an ideal cycle, late and early closing of the inlet valve are equivalent. It can be seen from the p-v diagram (Fig. 2) that both ideal processes begin with the effective compression at point 1', i.e. their compression phase is shorter than the expansion phase – a characteristic of the Atkinson process. The only difference between "early" and "late" closing is the part between 1' and bottom dead center. In the first case, i.e. "early closing", sub-process 1'-1-1' takes place in the cylinder with closed inlet valves; in the second case, "late closing", (sub-process 1'-1''-1') with open inlet valves. Both sub-processes exhibit zero area in the ideal case. Thus, all the other parts are identical. The idea that the Miller process reduces the charge temperature due to expansion of the charge air in the cylinder is especially worth noting. This can be clearly seen in the case of "early closing", but would seem to be missing in the case of "late closing". In fact, it is the virtual expansion from the charge pressure $p_{Rec}'$ to the initial pressure of the compression $p_{ac}$ that is relevant for the reduction of the process temperatures. As long as the same compression curve is achieved from the same charge pressure and the changes of state continue to be isentropic, then the changes of state necessary until the point 1' (via 1'-1-1' or via 1'-1''-1') will be irrelevant.

It should be noted in connection with the processes compared in Fig. 2 that they show the same pressures but different temperatures. As a result, the masses and outputs must be different.

### 2.1 Efficiency of the Miller process

If only ideal processes are considered, then the overall efficiency of the Miller process is worse than that of the process with conventional valve timing. The reason is that the part of the positive “gas exchange loop” lying between the compression curve and the BDC (see Fig. 2) is cut off. This loss, referred to in the following as “Miller loss”, can lead to a reduction in engine efficiency of up to 0.5%. In a real process it can be expected that when the pressure behaves in the same way in the high-pressure section, more output at higher efficiency will be achieved since the gain from the lower heat losses will more than compensate for the Miller loss. If the output and all other process parameters are kept constant,
the pressure level will be generally lower, thus improving the efficiency of the high-pressure process. The lowering of the peak pressure has, as a rule, the effect of freeing up some of the engine’s mechanical potential for a further improvement in efficiency. This improvement can take the following forms:

– An increase in air/fuel ratio $\lambda_V$
– An increase in compression ratio $\varepsilon$
– An increase in the combustion pressure rise due to earlier injection

In the investigated case the last of these proved most effective, as the engine already exhibited in the reference case a high $\varepsilon$ and no combustion pressure rise. However, depending on the boundary conditions, the two other possibilities, either alone or combined, could also be interesting.

To be able to evaluate the influence of the Miller process on engine operation (i.e. independently of the inlet valve cam profile), the term “Miller Effect” has been introduced. This effect is described by $(1 - \lambda_i)$ and defined as follows:

Miller Effect [%] = $(1 - \lambda_i) \cdot 100$

By simulating operation of the test engine described in sect. 4 with different Miller timings and assuming that the cylinder output, $\lambda_V$, $\varepsilon$ and peak cylinder pressure remain constant, it was possible to determine the influence of the charging efficiency and thus of the Miller Effect on the overall engine efficiency (Fig. 3). The efficiency has two components: a high-pressure, i.e. closed cycle, component and a gas exchange component:

$\Delta \eta_{\text{engine}} = \Delta \eta_{\text{closed cycle}} + \Delta \eta_{\text{gas exchange}}$

It is seen that the efficiency gain in the closed cycle increases linearly with the Miller Effect. The efficiency of the gas exchange component remains approximately constant up to a Miller Effect of 15 – 20 %. A further increase in Miller Effect causes a sharp increase in both the throttle losses and the Miller loss. An optimum for the overall efficiency occurs at a Miller Effect of about 30 %.

2.2 “Early” or “late” closing?
As already mentioned, in an ideal cycle late and early closing of the inlet valves are equivalent. In real cycles, however, there are differences:

– With early closing the inlet valves must start to close very early; therefore, the difference between the cylinder pressure and the charge pressure is larger due to throttling; in the case of late closing the intake and discharge of a portion of the charge air also involves some throttling losses. Which of these two effects is the more dominant will depend on the valve geometry and cam profile.
– With late closing it is also necessary to consider the heat transfer: the charge air which is forced back has already been heated up in the cylinder and this heat is stored in the inlet channel until the inlet valve opens again in the next cycle. This partly reduces the theoretical cooling of the cylinder charge, compared with early closing.
These effects can be observed in the p-v- and T-v diagrams (Fig. 4). The pressure curve has a direct influence on the gas-exchange work, while the temperature curve influences the closed cycle work due to the dependence of the heat losses and the thermodynamic properties on the actual gas temperature. The possibility of further indirect influences on the combustion also cannot be excluded, since the movement of the charge in the cylinder can change in accordance with the valve closing point.

The global effects can be derived from an analysis of the results in Fig. 5. Since the engine has a camshaft with steep flanks, the inlet closing time can be varied without changing the maximum valve lift. In this case the advantage of engine operation with “Miller early” can be clearly seen compared with operation with “Miller late”. The second curve applies to scaled inlet cams with reduced maximum valve lift, which simultaneously results in a reduction of the inlet valve area. The determining parameter:

\[ C_{VE} = c_K \frac{A_K}{A_{VE,max}} \]

\( c_K \) = mean piston speed, \( A_K \) = piston area, \( A_{VE} \) = effective valve area has increased by approx. 40%. In high speed engines it can occur that owing to the mechanical limitations for valve acceleration, the maximum valve lift with “Miller early” must be reduced. In such cases, better results can be achieved with “Miller late”.

Previous considerations apply to the full load point. For engine operation with constant speed it can be expected that all load points exhibit a similar behavior. For operation with variable engine speed, e.g. in propeller operation, the engine behavior is better with “Miller early” at part load, since the charging efficiency, in contrast to “Miller late”, is higher at reduced speed.

This effect can also be seen qualitatively in Fig. 5, since \( C_{VE} \) is proportional to the engine speed.

For operation with fixed valve timing, the part load behavior is normally better with “Miller early” than with conventional timings. “Miller late” is not suitable for operation with variable engine speed.
In operation with natural aspiration, i.e. with operating points of very low load, in which the turbocharging system supplies no pressure, a reduction of the air/fuel ratio compared with conventional engines must generally be expected. With extreme Miller timing and reverse scavenging the air/fuel ratio can fall in such a way that the engine reacts with heavy emission of smoke when starting and on the first application of load. In this case, operation could be improved with “Miller late”, since the residual gas is mixed with more fresh air and is then partly exhausted from the cylinder.

It is known that the Miller process can supply very interesting results with variable valve timing. Systems with full variability have the highest potential, but are demanding. The Miller process is simpler to apply to systems in which the entire inlet valve opening phase can be displaced. These are not optimal, however, since $\Phi_{IC}$ and valve overlapping are varied simultaneously.

These effects can be summarised as follows:

- “Miller early”, full load: this provides the best possibility for controlling the peak pressure, since the displacement of the cam in the “early” direction simultaneously produces a reduction in charging efficiency and charge pressure.
- “Miller early”, part load: increase in air/fuel ratio in part load operation is limited, since the displacement in the “late” direction increases the charging efficiency, but reduces the charge pressure by reduction of the scavenging.
- “Miller late”, full load: the peak pressure is difficult to control, since the Miller Effect is performed with reduced scavenging, which results in an increase in the charge pressure.
- “Miller late”, part load: the reduction in the Miller Effect provides a marked increase in the air/fuel ratio here, since charging efficiency and charge pressure increase simultaneously. This positive effect could be reduced, however, by the pressure-reduction measures, which must necessarily be introduced at full load.

Fig 5: $A_{VE}$–$\Phi_{IC}$ variation.
2.3 Scavenging in engines with Miller timing

If an engine is to be designed for operation with extreme Miller and fixed timing, the problems of valve overlap must also be considered. The scavenging of an engine can be described by three parameters: the delivery ratio $\lambda_R$, the charging efficiency $\lambda_l$ and the scavenging factor:

$$\beta_1 = \frac{\lambda_R}{\lambda_l}$$

An increase in $\lambda_R$ generally leads to a reduction in gas exchange work, which has a negative effect on engine efficiency. If the scavenging is reduced so far, however, that the proportion of residual gas has increased considerably in the cylinder, deterioration of the engine efficiency occurs, being derived from the high pressure cycle. There is then an optimum efficiency for every engine at the rated point, which corresponds to very moderate scavenging. Scavenged engines are almost always designed with more scavenging, since this improves the thermal loading and part load behaviour.

Variation of the inlet closing point $\varphi_{ic}$ from the conventional to Miller timing with constant valve overlap results in equal reductions of $\lambda_R$ and $\lambda_l$, so that the scavenging factor tends to rise. This can be explained by the following considerations:

- With increasing Miller Effect and constant air/fuel ratio, the air density before the cylinder is increased and the volume to be filled is simultaneously reduced ($\lambda_l$ falls).
- The scavenge volume with unchanged valve overlap rises approximately proportional to the density before the cylinder and the scavenging factor accordingly increases.

If the scavenging factor is to be kept constant, a reduction in valve overlap is recommended, this resulting in a further improvement in engine efficiency (see Fig. 6).

The reduction in valve overlap has a positive influence on engine behaviour in the lowest load range, where, with extreme Miller timing, the available fresh air is significantly reduced.

- The intersection of the lines of charge pressure and exhaust gas pressure after the cylinders is displaced to areas of lower load and the range in which the exhaust gas pressure after the cylinder is higher than the charge pressure becomes smaller.
- The negative effects of the reversed scavenging are reduced by the smaller valve areas during scavenging.

This is shown in exemplary fashion in Fig 7. Less valve overlap significantly improves the air/fuel ratio in the part load range. Smoke problems when starting and on application of load to the engine can thereby be avoided.

![Fig. 6: $\Delta \varphi_{vo}$ vs $\eta_a$ variation (Ref.: $\Delta \varphi_{vo} = 90^\circ$CA, $\eta_a = 0.65$).](image-url)
2.4 Turbocharging efficiency

The results shown in Fig. 5 were calculated with a constant turbocharging efficiency of $\eta_a = 0.65$. As a result the pressure difference over the cylinder and therefore the theoretical value of gas exchange work remains constant.

This corresponds to a turbocharger efficiency of 69 %, which is an extremely good value for a small 4-stroke engine turbocharger, even under normal pressure conditions. With increasing Miller Effect the required pressure ratio rises and it becomes increasingly difficult to maintain the turbocharger efficiency at this level. Since, however, the turbocharging efficiency has a significant influence on engine efficiency (see Fig. 8) efforts will also be focused in future on achieving high pressure ratios with high turbocharging efficiencies. The first curve ($\eta_a = 0.65$) in Fig. 8 shows the optimum engine efficiency with a Miller Effect of approx. 30 % and a compressor pressure ratio of around 5. It is extremely difficult to achieve this with the required turbocharger efficiency.

With higher turbocharging efficiencies it is possible to achieve a further improvement in engine efficiency. The engine efficiency optimum is displaced depending on the turbocharging efficiency level to Miller Effect ranges of over 30 %. The current standpoint, however, is that these results are only possible with turbocharging systems in which higher efficiencies can be achieved with intercooling.

Fig 7: $\lambda_v$ at part load.

Fig 8: $\eta_a$-\$\pi_v$ variation.
It was shown in the previous section that for implementation of the Miller process in modern engines very high pressure ratios and efficiencies are necessary. The possibilities for single or two stage turbocharging with and without control are considered below.

### 3.1 Single stage turbocharging

The latest generations of ABB turbochargers for 4-stroke applications permit pressure ratios greater than 5 in continuous operation with aluminum compressors.

The product range covers two turbocharger families:
- TPS: turbochargers with radial turbine for the power range 500 – 3200 kW [1]. The TPS..-F versions are a further development of the proven TPS..-D/E series. The range of operation of TPS turbochargers and the cross-section of a TPS..-F turbocharger are shown in Figs. 9a and 9b.
- TPL: turbochargers with axial turbine for the power range from 1000 kW [10]. The TPL..-C versions are a further development of the proven TPL..-A/B series. The range of operation of TPL turbochargers and the cross-section of a TPL..-C turbocharger are shown in Figs. 10a and 10b.
Fig. 11 shows the development of the achievable pressure ratio of ABB turbochargers over the course of the years. The curve shows a levelling off up to the early nineties, since the components were developed in mature products (VTR and RR) with their corresponding limitations.

With the introduction of the new generations (TPS and TPL, 1996) it was possible to open up further development potential with new concepts. Shortly afterwards it was seen that a compressor design with a simple “trim” concept was not suitable for optimum coverage of the required flow rate range. The introduction of different design areas according to the mass flow rate, as well as the use of stabilizers [7] and the latest tools for compressor development led to a further improvement in pressure ratio.

This progress made it possible to turbocharge modern engines with the highest mean effective pressure and with a moderate Miller Effect. In engines not designed for the highest mean effective pressures this results in an even greater potential, which can be used for the introduction of extreme Miller timing.

If, however, an engine is to be turbocharged for example at mep = 28 bar with 30% Miller Effect, a pressure ratio of at least 6 is necessary. No suitable turbocharger is yet available for this. Far higher pressure ratios are technically possible with single stage turbocharging, but a technological leap would be necessary with a change to better materials. This,

Fig. 11: Achievable pressure ratio with single stage aluminum compressor (50,000 h, base load) [9].
however, would significantly increase development and production costs. Essential turbocharger requirements such as high turbocharger efficiency, a wide compressor map and high reliability and flexibility, also mean that there is a very high technical risk involved. For these reasons it is also necessary to consider two stage in addition to single stage turbocharging.

3.2 Two stage turbocharging
With two stage turbocharging, the air and gas side enthalpy heads are divided between the two turbochargers. This division alone enables higher turbocharger efficiencies to be achieved in the individual stages owing to the lower loading of the turbocharger components. The turbocharging efficiency achievable can be even further increased, however, by additional measures:

- Intercooling improves the process of air compression, since the process of isothermal compression is more closely approached. The theoretical gain (see Fig. 12) increases with the pressure ratio.
- At the turbine-end the sum of the individual isentropic enthalpy heads of the two stages is always greater than the enthalpy head of the overall stage, since the losses in the high pressure stage at least increase the inlet temperature of the low pressure stage. Optimal connection of the turbines, however, would allow the outlet losses of the high pressure stage, which comprise the greatest source of loss of a turbine, to also be largely converted into pressure. The achievable gain in efficiency of the two stage turbine can be 2 to 8% points compared with the efficiency of the single stage.

Generally considered, turbocharging efficiencies up to around 80% would be achievable with two stage turbocharging with optimized efficiency.

The cost, weight and physical volume of a turbocharging system of this kind, however, would be extremely high. And the system complexity would increase. On top of this, the high full load efficiency of the turbocharging system would make sophisticated control intervention necessary for part load operation of the engine to be practicable.

3.3 Controlled two stage turbocharging
Solutions to the problems outlined above are offered by controlled two stage turbocharging. A layout for optimum turbocharging efficiency leads to an even distribution of the pressure ratios:

\[
\pi_{\text{HP}} = \pi_{\text{LP}} = \sqrt{\pi_{\text{System}}}
\]

The result is that very large turbocharger stages are required. If the pressure ratio is increased in the low pressure stage, the result is as follows:

- The low pressure compressor is smaller: it supplies the same flow at a higher pressure.
- The low pressure turbine is smaller: it must produce more energy from the same exhaust gas flow.
- The high pressure compressor remains the same or is slightly smaller; the operating point moves parallel to the surge limit. If low pressure ratios are necessary, smaller compressor wheels with high specific swallowing capacity can be used.
- The high pressure turbine is larger: it must produce less energy for the same inlet conditions.
Obviously, for a compact solution the high pressure turbine is critical. If, however, some of the exhaust-gas mass flow can be made to bypass this turbine at the system’s design point, the turbocharging system will be compact and an ingenious means of control is provided, effectively solving the part load problems described.

Bypassing of the high pressure turbine is controlled by a throttle device, which can be progressively closed in the part load range. The boost pressure can therefore be increased as required. In full-load operation, losses in efficiency must be expected, but these will be much lower than with a conventional waste gate, since the exhaust gas flow diverted is lost over just a small part of the head.

The different sizes of the turbochargers for various turbocharging systems are shown in Fig. 13.

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**Fig. 13: Turbocharger sizes for various turbocharging systems.**
4 Test engine

The experimental work was performed as part of the CLEAN partial project “NO₂-reduction in large diesel engines by application of the Miller process” [3] and the subsequent project “Particle and NO₂ reduction in large diesel engines by combined application of charge pressure controlled high pressure injection and turbocharging according to the Miller process”. ABB Turbo Systems, in close collaboration with the other project partners ISF and FMC, carried out a thermodynamic analysis of the influence of the Miller process and its effects on engine operating values and NOₓ emissions, and also fitted the engine with the appropriate turbochargers. In this way it was also possible to confirm experimentally the engine operation and NOₓ reduction calculated from the thermodynamic analysis.

The test plant was realized using the research engine ISF4524 of the Institute of Marine Technology at FH-Flensburg [4]. The most important engine data are given in Table 1.

<table>
<thead>
<tr>
<th>Table 1: Engine ISF4524</th>
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<tr>
<td>Bore</td>
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<td>Stroke</td>
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<tr>
<td>Stroke/bore ratio</td>
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<tr>
<td>Compression ratio</td>
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<tr>
<td>Cylinder capacity</td>
</tr>
<tr>
<td>Number of cylinders</td>
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<tr>
<td>Turbocharging system</td>
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<tr>
<td>Inlet closing</td>
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<td>Injection system</td>
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</table>

Fig 14: Controlled two stage turbocharging on the test engine and schematic arrangement.
The engine has six cylinders, but it was decided to run it on three to keep costs down. ABB installed on this engine its smallest turbochargers from the TPS series: A TPS 48-D for the reference measurement without Miller timing, a TPS 48-E for single-stage turbocharging, and a combination of TPS 48-D and TPS 48-E for the two stage turbocharging (Fig. 14). Some specially made components were necessary to optimize the controlled two stage turbocharging. The control valve itself (GloTech60) was provided by Woodward.

Since, in each of these configurations, and especially in the last one, the turbochargers were operated well below their capacity limits no conclusions can be drawn as to the real potential in terms of turbocharging efficiency and compactness. The results therefore serve only to confirm the different configurations’ thermodynamic potential from the point of view of engine operation. To optimize the turbocharging it would be necessary to use smaller turbochargers or to run the engine on six cylinders.

**Test results**

At the start of the project a variation in $\varphi_{IC}$ was calculated for the engine for the full load point with various turbocharging systems (see Fig. 15). This produced the target values and forecasts shown in Table 2, which also shows the test results.

![Figure 15: Influence of $\varphi_{IC}$ variation on the operating parameters of the research engine at full load.](image)
In general the forecasts were exceeded both for fuel consumption and the NOₓ values. It was not possible to test the planned version with inlet closing 30 °CA bBDC owing to non-availability of the parts. Instead a version with 45 °CA bBDC was tested experimentally for both single and two stage turbocharging. The turbocharging systems were not optimized for this purpose and therefore the engine’s charge pressures were too low in case II.a and too high in case II.b (see Table 2).

The values specified apply to engine operation with a feed pressure in the injection system of 300 bar. By varying this feed pressure it is possible to move along a be-NOₓ trade-off line of the engine. An increase in feed pressure in the injection system leads to a reduction in fuel consumption and an increase in NOₓ emissions, and vice-versa [5]. In the extreme case: III. Miller/ϕIC = 60 °CA bBDC, a further significant reduction in NOₓ emissions, aimed at emissions optimization, can be obtained without losing the entire efficiency advantage.

The configurations in Table 2 are generally only suitable for generator operation. The engine was also operated, however, according to the propeller law. The good experience so far with the engine as well as close agreement between the calculations and measurements (Fig. 16) indicate that the propeller operation is reproduced well by the simulation.

Both the measurements and the calculations (see Fig. 17) confirm that reliable propeller operation is not possible without control intervention. It is anticipated that by re-specifying the turbocharger for λmin = 1.8 at part load and regulating the charge pressure by means of a waste gate at full load, propeller operation will also be possible with the extreme Miller timing considered (Fig. 17).

Although the turbocharging efficiency is relatively low owing to the considerably over-dimensioned turbocharger, the results look interesting. Improvements can only be expected with a version of the turbocharging group for the “full engine”.

Fig. 16: Comparison between simulation and measurement (propeller law, ϕIC = 60 °CA bBDC, PVD = 300 bar).
Table 2: Fuel consumption and NO\textsubscript{x} emissions reduction with different inlet close timing compared with the basic engine version (feed pressure = 300 bar).

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<thead>
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<th>Forecast</th>
<th>Measured</th>
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<tr>
<td><strong>I. Standard/\psi_{IC} = 20^\circ CA aBDC</strong></td>
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<td>Engine operation with TPS 48-D/single stage turbocharging</td>
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<td>$P_{\text{max}}$ (100 %-load) = 3.4 bar</td>
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<tr>
<td><strong>II. Miller/\psi_{IC} = 30^\circ CA bBDC</strong></td>
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<tr>
<td>Engine operation with TPS 48-E/single stage turbocharging</td>
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<tr>
<td>$\Delta \dot{m}_g$ [g/kWh]</td>
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<td>-</td>
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<td>$\Delta \dot{N}O_{x}$ [g/kWh]</td>
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<td>Charge pressure at full load</td>
<td>$P_{\text{rec}}$ [bar]</td>
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<td><strong>II.a Miller/\psi_{IC} = 45^\circ CA bBDC</strong></td>
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<td>Engine operation with TPS 48-E/single stage turbocharging</td>
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<td>$\Delta \dot{m}_g$ [g/kWh]</td>
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<td>+ 4.8</td>
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<tr>
<td>$\Delta \dot{N}O_{x}$ [g/kWh]</td>
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<td>- 3.8</td>
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<tr>
<td>Charge pressure at full load</td>
<td>$P_{\text{max}}$ [bar]</td>
<td>4.39</td>
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<td><strong>II.b Miller/\psi_{IC} = 45^\circ CA bBDC</strong></td>
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<td>Engine operation with TPS 48/two stage turbocharging</td>
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<td>TPS 48-D (LP stage) + TPS 48-E (HP stage)</td>
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<td>$\Delta \dot{m}_g$ [g/kWh]</td>
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<td>- 11.6</td>
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<tr>
<td>$\Delta \dot{N}O_{x}$ [g/kWh]</td>
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<td>- 2.2</td>
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<tr>
<td>Charge pressure at full load</td>
<td>$P_{\text{rec}}$ [bar]</td>
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<td><strong>III. Miller/\psi_{IC} = 60^\circ CA bBDC</strong></td>
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<td>Engine operation with TPS 48/two stage turbocharging</td>
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<td>TPS 48-D (LP stage) + TPS 48-E (HP stage)</td>
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<td>$\Delta \dot{m}_g$ [g/kWh]</td>
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<td>Charge pressure at full load</td>
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Table 2: Fuel consumption and NO\textsubscript{x} emissions reduction with different inlet close timing compared with the basic engine version (feed pressure = 300 bar).
5 Summary and outlook

It was shown that the Miller process has very high potential with regard to improvement of emissions and engine efficiency. This potential can even be increased by introducing Miller timings that produce far more Miller Effect than is usual today. However, this requires very high boost pressures.

ABB Turbo Systems has developed two new turbocharger generations that permit pressure ratios of over 5 in a version with favorably priced aluminum compressors. At the same time multi-stage solutions are being examined for even higher pressure ratios.

It has been possible to confirm these concepts experimentally in a research project lasting several years. At the same time, the experience and knowledge gained generate new ideas and increase confidence in the simulation tool, which continues to yield information of ever-better quality.

Controlled two stage turbocharging has proved to be a highly promising solution for the turbocharging of large diesel engines, particularly for operation at variable speed and substantially increased charge pressures.

Modern turbochargers, however, are not designed for this operation. An important task still to be undertaken is the preparation of design solutions for optimized multi stage turbocharging, therefore making it economically attractive.

There are also applications, as in the case of gas engines, where the proposed control provides no benefit, since these engines employ different control concepts. The definition of suitable turbocharging systems for all high pressure applications is another demanding task requiring attention in the future.

![Fig. 17: Controlled two stage turbocharging – engine operating parameters (propeller law, \( \varphi_c = 60^\circ\text{CA} \ bBDC \), feed pressure = 300 bar).](image-url)
References/acknowledgments


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