Reducing emissions using 2-stage turbo charging

By cooling the combustion process using a Miller cycle, NOx emissions can be efficiently reduced. This demands high boost pressures, which can be achieved with a 2-stage turbo charging (TC) system. Thus, these two technologies form an optimum combination.

In order for the engine to cope with start-up problems caused by the cooler combustion chamber having earlier set inlet valve closure timings (IVC), a variable inlet valve closing (VIC) system will also be needed. Simulations made with a commercial 1D-simulation code have proven the potential that 2-stage turbo charging, together with extreme Miller timings, offers in respect of fuel consumption savings, as well as NOx emission reductions.

As a result, a series of engine tests using a Wärtsilä 4-stroke medium-speed diesel engine was planned to verify the potential for this technology. The outcome of the tests, as well as the design changes needed to the engine, is reported in this article.

The testing reported in this article has been performed with a prototype system. Several years of further development work needs to be invested, both on the engine as well as on the turbocharger side, in order to get 2-stage TC systems ready for production use.

Why a Miller cycle?

With a Miller cycle one understands changed inlet valve closure timings, which are normally made before BDC. The cycle was originally intended to increase IMEP of petrol engines, and is based on a shortening of the compression stroke and a lowering of the charge temperature inside the cylinder. [1]

The working principle of the low-pressure loop with a Miller cycle, compared to a normal diesel cycle, is shown in Figure 1.

One major difference can be seen in the decreased compression, due to the additional expansion of the cylinder load after the IVC and before BDC. Because of this, the total compression work is less than with the standard cycle. Since the total expansion work remains the same, there is a positive effect on the engine’s...
efficiency, which results in lower overall CO₂ emissions. This is also ensured by increased air receiver pressure, which guarantees that the pressure inside the cylinder remains the same at BDC and thus also at the end of compression.

The other major difference is seen in the overall temperature level inside the cylinder. The additional expansion of the cylinder load after the IVC before BDC, ensures the attainment of a considerably lower temperature inside the cylinder at the start of combustion. This difference is maintained and even increased during the high-pressure cycle, as shown in Figure 2. Positive effects of the lower temperatures inside the cylinder include:

- lower NOx emissions
- lower exhaust/component temperatures
- lower heat transfer from the cylinder contents to the cylinder boundaries, i.e. better efficiency.

**DESIGN CHANGES NEEDED WITH A 2-STAGE TC**

Several checks had to be made in order to see if the test engine would withstand the considerably higher charge air and exhaust gas pressures, compared to the standard engine. Stress and strength calculations have been made for the following components:

- Engine block, since the air receiver is integrated into the engine block.
- Charge air system with charge air cooler housing, diffuser and other piping between the compressor and cylinders.
- Exhaust side piping with a SPEX pipe construction.
- The charge air cooler.

According to the calculation results, all components could be expected to last and remain well within the limits set for each component. The biggest area of stress was on the diffuser side walls (see Figure 3). One weak spot in the engine block was found, however, caused by the wall thickness being too thin for the air receiver. Due to this, reinforcements were installed outside of the engine block.

Furthermore, the inlet valve springs had to be changed to stiffer ones since theoretical calculations indicated that the pressure difference over the valves would become too high at pressures downstream of the compressor of 8.7 bar. This would lead to uncontrolled opening of the inlet valves with standard valve springs.
However, the biggest change to the engine has clearly been in the construction of the turbocharger shelf. The system was designed so that the high-pressure (HP) turbocharger is located on the engine, and the low-pressure (LP) charger is located on a separate bracket behind the engine.

The intercooler is integrated in the same separate bracket as the LP TC, whilst the after-cooler is in the original position on the engine. For safety reasons, a burst protection has been designed around the whole turbocharger system. This consists of a steel bar frame, with 28 mm thick plywood inside and a sheet metal plate outside.

The very early IVC has the drawback of very low temperatures in the cylinder at the start of combustion (low effective compression ratio) at engine start-up and part loads, resulting in starting problems and high smoke emissions at low loads. The most efficient means of coping with the increased Miller timings is to use a variable inlet valve closing (VIC) system, whereby the inlet valve closing could be adjusted to later timings at start-up and part loads, for aiding the engine’s operation.

The design of a fully flexible VIC system, though, is not economically viable for a small size engine, such as the Wärtsilä 20, as was to be used in the 2-stage TC tests. A viable solution for solving these problems in small engines would be to use an auxiliary blower, as was proven in the 1-stage TC pre-tests at HUT, and/or installing a heating system for the cooling water temperature. Another option would be just to design a simple on/off VIC system for the lower-end bore sizes, which would be sufficient to improve engine start-up.

Other changes done to the test installation included:
- Installation of a pre-heating system for the low-temperature water to the charge air cooler (CAC) for securing start-up and low-load smoke behaviour at extreme Miller timings
- Installation of a separate low-temperature water circuit for the IC
- Exhaust stack modifications
- Installation of an oil supply and drain for the LP module, as well as a de-aeration vessel and flow meter for the blow-by flow.

A principal sketch of the system is shown in Figure 4. The whole engine 2-stage TC test set-up is shown in Figure 5.

2-STAGE TC TEST RESULTS
The experimental testing of the advanced Miller timings and 2-stage TC were performed at the Wärtsilä Engine Laboratory in Vaasa during the period October 2006 – February 2007. The main objective of these experiments was to gain knowledge about engine performance with advanced Miller and high boost pressure.
All tests were performed on a Wärtsilä 20 engine at both constant and variable speeds. The engine specifications were as follows:
- Bore/stroke: 200/280 mm
- BMEP: 27.3 bar
- Maximum power: 200 kW/cylinder
- Engine speed: 1000 rpm constant and variable speed
- Maximum cylinder pressure: 200 bar.

See the actual test set-up in Figure 6.

The TC specifications used were as follows for the HP and LP chargers respectively:
- ABB TPS48 HP TC
- ABB TPS52 LP TC

A Miller timing of 81°CA before BDC was chosen as the most extreme timing to be tested. Since a camshaft having such extreme timings has a very short opening timing, the profile was shortened as much as deemed possible, and additional Miller timing would demand a decrease in the valve lift (see Figure 7).

Based on some pre-tests made with a 1-stage TC up to compressor pressure ratios (PIC) of 6.2, the PIC need for the 2-stage TC tests was estimated at about 9 for an IVC 66°CA earlier than the standard (corresponding to the dotted line to the left in Figure 8). The corresponding estimation of NOX reduction was about 50% with the same IVC (see Figure 8).

The initial PIC level used in the tests was about 9.1:1 with a split between the LP and HP compressor stages as follows:
- LP = 4.05:1
- HP = 2.30:1
This resulted in a split LP/HP of ~64/36% (see Figure 9).

In the initial tests an extreme Miller timing, IVC 81° before BDC, was used. The chosen initial overall charge air pressure was found to be too high, causing excessive firing pressures and air flow. It was therefore reduced from 9.1 to 8.3:1 for the remaining tests with this Miller timing. The split between LP and HP was also moved towards a slightly higher ratio for the HP stage.

As expected, the starting and acceleration capability with such an early IVC timing was very poor, in spite of the relatively high geometrical compression ratio (16:1) and in spite of cooling water pre-heating. The water pre-heating temperature was therefore further raised to 50 °C, and in addition a heating fan was installed at the suction air intake. This improved the engine starting capability, but the fuel admission still had to be strictly limited during acceleration in order to avoid heavy misfiring.

Also steady-state performance and loading capability were very poor at low load. Only at loads above 25% was the combustion good enough to give acceptable engine performance and load acceptance. Figures 10 and 11 show some performance parameters as a function of engine load, with initial engine and TC specifications.

As expected, at high load NOX emissions were strongly reduced, while the thermal load and engine efficiency remained quite unchanged. The NOX level was reduced by more than 40% at full load, and by even more than 50% at 75% load, during these initial tests. The reason for the strong increase in NOX emissions at low load in this case is mainly the ignition delay caused by the low compression end temperature, followed by a very fast heat release which results in high cylinder pressures. A variable inlet closing (VIC) system would probably improve the situation.
The fast combustion resulting in high cylinder pressures at low load is also reflected in high exhaust gas temperatures (see Figure 11). A positive issue with the fast pre-mixed combustion at low load, however, was a reduction in smoke emissions, which were in fact lower than in the reference case at loads <40%. But at 50% load, smoke emissions increased by some 45% at constant speed, and by about 15% at variable speed operation, thus confirming the assumptions made beforehand.

During these initial tests, the charge air and firing pressures were still too high, and the overall turbo charging system efficiency still too low compared to the target levels, partly because of a slight mismatch of the turbochargers, and partly because of the high intake air temperature in the HP stage turbo (after the IC). Also, the charge air receiver temperature was higher than the targeted level of 55 °C.

A further reduction of the charge air pressure, and an improvement of the cooling system in order to achieve lower HP suction air and charge air receiver temperatures, would reduce the NOx emissions further. By these means it appears to be possible to reach a NOx reduction in the region of 50%, compared to the standard engine concept.

Furthermore, the BSFC will also benefit from these measures, and further improvement of the turbocharger matches will reduce the BSFC even further. Figure 12 shows the measured efficiency levels of the turbocharger systems during this initial test series. However, a level in excess of 72% at full load and up to 75% at part load is within reach.

Figure 13 shows an example of the pressures in the inlet port, cylinder and exhaust port during the gas exchange period with extreme Miller timing. The pressure in the cylinder at BDC drops 4.3 bar below the charge air pressure mainly due to the internal expansion, which explains the need for a high boost pressure, as well as the resulting low NOx emissions and thermal load.

With this big influence of extreme Miller timing on in-cylinder conditions, it is obvious that special means are needed to ensure ignition and good engine performance at idling and low load. One such solution would be, for example, a VIC system that makes it possible to set the IVC later (close to BDC) at start-
up and part load operation. With this, the effective compression ratio of the engine would increase. Filling of the cylinder with air would also improve considerably, resulting in higher firing pressures and lower thermal loads.

An example of the influence of a VIC system on some engine performance parameters is shown in Figures 14 and 15. This example reflects a rather moderate Miller timing, IVC = 33° before BDC, but it is obvious that the benefit would be larger with an even stronger Miller timing.

Another advantage with a VIC system is that the TC compressor can be better optimized, because the operating line in the compressor map is moved to the right (away from surge line) when the system is operated in "late mode" (low load), but moved back into higher efficiency when the system is in "early mode" (high load).

An air bypass system is often used to improve the engine part load performance, but with a VIC system such a bypass is not needed. During the initial tests, the scavenging period was kept at a standard length, but it is clear that this could be reduced with a variable cam system. This would reduce the BSFC considerably at high load, without causing excessive thermal load at low load.

Thus, a combination of extreme Miller timing, 2-stage TC technology, and a shorter scavenging period makes it possible to reach BSFC savings in the magnitude of 2-3% (corresponding to the same reduction in CO₂ emissions) together with NOₓ reductions of 50%. In combination with a VIC system, the part load performance could also be optimized and the drawbacks minimized. Test results with this combination are seen in Figures 16 and 17.

CONCLUSION

All assumptions based on the simulations were confirmed by the 2-stage TC tests. The estimated difficulties with engine start-up and smoke levels at part load, were also confirmed. The possibilities to cope with low loads are limited unless VIC technology is developed. The heating of cooling water of the aftercooler is not sufficient, but needs to be used in combination with an external blower for the supply of hot air.

NOₓ reductions in the region of 50% are achievable with extreme Miller
timings in combination with 2-stage TC technology. These reductions were in excess even of those shown in the simulations, where only 37% NOx reductions were estimated with extreme Miller timings.

Also full load BSFC and thermal load improve somewhat, due to the increased TC efficiencies and boost pressures respectively.

The main drawbacks are the difficult engine start-up and poor low-load running as a result of the very low in-cylinder temperatures. The most efficient solution for this is to use a fully flexible VIC system in large bore engines. For small bore engines, a simple VIC system or other alternatives such as an external blower, as well as a cooling water temperature heating system is recommended where the investment of a fully flexible VIC system is not economically viable.

**NOMENCLATURE**
BDC  Bottom dead centre  
BSFC  Brake specific fuel consumption  
CAC  Charge air cooler  
CO2  Carbon dioxide  
IVC  Inlet valve closure (0 mm valve lift)  
HP  High pressure  
IC  Intercooler  
LP  Low pressure  
NOx  Nitrogen oxides  
TC  Turbocharging/turbocharger  
VIC  Variable inlet valve closure

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**REFERENCES**