

Increasing the Efficiency of Large Two-Stroke Diesel Engines for Lower GHG Emissions

9 - New Engine Developments - Diesel

Paper 357

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ABSTRACT

Future requirements on prime movers in shipping will focus, apart from the reduction of pollutant formation, on the reduction of GHG such as CO₂. This is apparent in the enforced and upcoming legislation, such as the EEDI initiative or the IMO strategy to cut CO₂ emissions from shipping by 50% within 2050. In addition ship owners & operators strive to reduce both initial and operating costs, notably fuel expenses. In the case of the traditional diesel cycle this basically boils down to lowering the BSFC at the same level of NO_x. Possibilities comprise amongst others improvements in the layout of the engine, notably the combustion development as well as changes in the turbocharging of the engines.

As in the automotive world where smaller engines are being offered, downsizing can be a path worth considering also in the marine industry. As the power requirements from the ships are to be seen as a given factor, the only way to make the engine smaller is to increase the power density. This is a trend that has been apparent in the past many years – it is likely to continue in the future as technological advances are being made.

Changes in the turbocharging of the engines can also help achieving the goals of lower GHG emissions. In this paper two technologies, 2-staged turbocharging and sequential turbocharging shall be examined. 2-staged turbocharging is by far not a new technology and has found its use in 4-stroke marine engines. In the 2-stroke world however it has not found broad application as the installation is complex and expensive – with the upcoming changes in legal requirements on the one side and costs evolution both for hardware and for fuel on the other the payback time for such installations might come into a reasonable range.

Sequential turbocharging on the other hand is a low-cost solution that requires relatively few hardware changes on the engine and the turbochargers. The design effort in order to create an inherently safe system is significant – this can however be transferred to different engine types once developed. Here one TC is cut out at low loads without having to switch off the engine which significantly increases the scavenge air pressure and allows for greatly reduced fuel consumption at the same level of NO_x. This technology too comes with its associated uncertainties and also only makes sense in some specific applications.

The three possibilities sketched above are to a very large extent ready for the market and for general application. They all show however the trend towards a higher complexity of the installations, even more so when Tier III solutions are included. This not only raises the stakes for engine, component and ship builders but also increases the requirements for the operators, namely the engineers on board. WinGD stays committed to continuing the development of technologies that can significantly reduce both CO₂ emissions as well as the total cost of ownership despite the challenges mentioned above. We are positive that this can be achieved without adding onto the burden of the crew and – last but not least – without sacrificing the high level of operational safety and reliability known from today's installations.

1 INTRODUCTION

The development of more efficient engines has been the main driver for technology development in the marine engine industry since many years, with the target to reduce the operating costs related to fuel consumption for ship operators. More and more fuel savings have been achieved in the past years and currently the highest efficiency levels for commercial internal combustion engines are reached. In the last decade this challenge has become even more complex due to the enforcement of tighter and tighter regulations on pollutant emissions, adding additional pressure to developers in the definition of the engine specifications.

The future ahead is most likely even more challenging. Several new legislation requirements have been approved recently and of importance for our purposes is the overall ambitious regulation framework depicting long term strategies towards the reduction of CO₂ emissions from shipping. In this context, we can mention the Energy Efficiency Design Index (EEDI) regulation which is already in force but foresees more stringent limitations or the decisions from IMO to cut CO₂ emissions from shipping by 50% within 2050.

As mentioned earlier, current marine engine technology is already mature and progressing towards higher efficiency on the basis of a continuous evolution; new rules are likely to accelerate the technological innovation cycles and possibly pave the way for the introduction of breakthrough technologies. Nevertheless, other factors have to be accounted for in the overall evaluations: Initial capital cost of equipment and systems, total operating costs, easy serviceability and, last but not least, technology maturity, even more so as shipping is a traditionally conservative industry.

Looking at the ship prime movers market, diesel low speed two stroke engines are the preferred choice by designers and owners in the merchant sector for their performance and reliability. For this engine segment, some relevant improvements in engine efficiency are still possible according to our analysis. Main improvements will regard, as always, combustion enhancement and turbocharging areas. Among the solutions we have analysed, three technologies appear to favourably fit the overall industry trends and requirements and are discussed in this paper:

Combustion developments

- Engine Downsizing

Turbocharging developments

- Two-stage turbocharging
- Sequential turbocharging

2 TECHNOLOGY REVIEW

2.1 Emissions regulations

Enforced NO_x emission rules for marine large two stroke engines are set in the MARPOL Annex VI [1] and prescribe the maximal allowable limits of pollutant emissions and their measurement duty cycle. The emission regulations which are currently in force are based on a territorial scheme: while a statutory limit applies globally, synthetically referred to in the following as IMO Tier II limit, a much lower one is enforced in specific NO_x emissions control areas, known as the IMO Tier III limit (Figure 1).

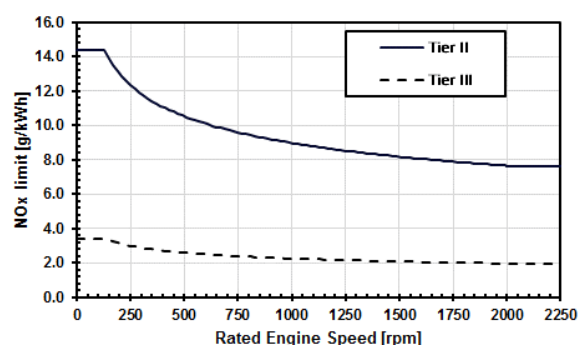


Figure 1: Allowable NO_x levels for IMO Tier II and Tier III legislation schemes.

As these regulations heavily affect the engine and plant specifications the major diesel large two-stroke engine developers have implemented solutions to cope accordingly.

IMO Tier II compliance has been achieved in the vast majority of the cases by engine combustion and turbocharging adaptations, e.g. with engine internal measures, while IMO Tier III standards are met with add-on measures and limited modifications of engine settings.

The engine specifications and development trends are heavily affected by the Tier II regulations. In particular, the pursuing of higher engine efficiency at same NO_x emissions levels reserves particular challenges to diesel developers. In the majority of the cases in fact, a modification with the aim of a reduction of fuel consumption normally leads to the simultaneous increase of the NO_x emissions. As a consequence, the developers have to find

the best possible compromise between fulfilling the emission limits and optimum fuel consumption. Most of the developments in this respect are targeted to the combustion phase and its control and added a compelling need of flexibility in the definition of settings in order to grant optimum performance in the typical operation range of a specific vessel. Accordingly, electronically controlled engines have become the industry standard, featuring fuel injection systems of the common rail type and hydraulically actuated exhaust valve drives that allow the application of different engine tunings. Turbochargers featuring pressure ratios reaching nearly 5:1 are commonly selected in order to exploit a higher Miller timing.

IMO Tier III standards instead are met at a price of higher capital costs and complexity of installation. SCR technology is today often selected for such scope. Due to its maturity level, the availability of liquid fuels, the existing infrastructure as well as the manageability of the technology it is to be assumed that SCR will be selected also in the coming years for many installations. A review of SCR technology for large 2 stroke engines is presented in [2].

Alternative ways to achieve IMO Tier III compliance are:

- Switch of combustion technology, selecting Winterthur Gas & Diesel's (WinGD) X-DF engines capable of operating with low pressure gas admission according to lean burn Otto combustion mode.
- Utilization of Exhaust Gas Recirculation (EGR).

To reduce the complexity for the reader, in the following of the paper we will only refer to the mainstream solution of engines equipped with SCR and we will focus on the optimizations in Tier II mode, taking for granted that the SCR can reduce the emissions to Tier III levels.

2.2 Rating field and derating

One of the biggest advantages of two stroke engines over their four stroke counterparts is their capability to operate in direct coupling with the propeller. This eliminates the need for a gearbox which reduces the complexity and cost of the installation. It also minimizes efficiency losses in gears, couplings or in conversions between different energy forms. In order to be coupled with different propellers and fit different ship designs, the engines must be prepared to operate at different points in terms of speed and power. The entirety of possible selection points for engine maximum continuous rating is referred to as rating field and the specific Contracted Maximum

Continuous Rating (CMCR) as engine rating (Figure 2).

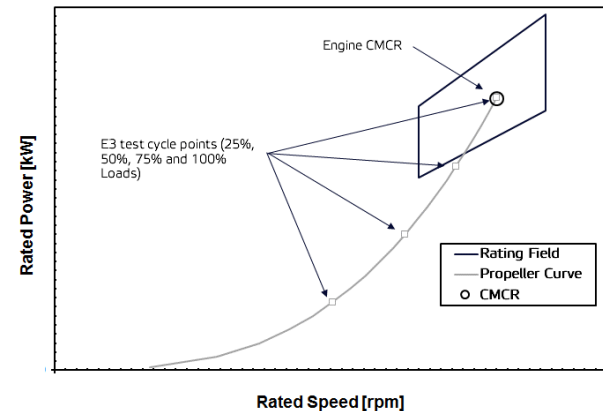


Figure 2: Exemplary depiction of an engine rating field within which a CMCR point can be selected. Based on this point the E3 test cycle modes are then defined.

Each rating is optimized to provide the best possible efficiency, leading to a situation where engines designed for a lower power output (derated engines) are providing a consistent fuel consumption gain compared to their peers optimized for higher power. This is due to the possibility to utilize a more efficient firing pressure over BMEP ratio. The situation is visualized in Figure 5, showing the trend of what we will refer to as IMO weighted BSFC over the rating field. IMO weighted BSFC is a synthetic index obtained through a weighted average of BSFC values at different load points (100%, 75%, 50%, 25%) according to the following formula:

$$BSFC_{IMO} = \frac{\sum_i m_{fuel} W_i}{\sum_i P_i W_i} \quad [1]$$

with

$BSFC_{IMO}$ = IMO weighted BSFC

m_{fuel} = Fuel flow at a specific load, in [g/h]

P_i = Engine Power Output, in [kW].

W_i = Weighting Index at a specific load

The weighting factors at different loads are the same as used to calculate the specific NOx emissions according to the IMO regulations. In order to discuss the fuel benefit gains in the remainder of this document, we will often refer to this index as it is capable of taking into account the whole load curve and is therefore quite

independent from the specific optimization strategy (e.g. the tuning).

The fuel consumption gains described deeply affected the engine selection of ship designers and derated engines are fitted in most of the modern merchant vessels. In principle, this means that the market has accepted to increase the capital expenditures (CAPEX) for the machineries by selecting bigger engines (bigger bore size or higher cylinder number) by paying them back with the reduction in operating costs achievable with reduced fuel bills.

Derated engines optimization adds an even higher degree of complication to the developer tasks for the mentioned need to keep the emissions of NOx below the Tier II limit. Lower engine speeds naturally lead to higher NOx due to the longer residence times of the burned gases at high temperatures, increasing the emissions formation rates. Additionally, also a reduction of BMEP normally leads to an increase of the specific emissions, as the combustion of fuel will be more concentrated close to top dead centre. Therefore, NOx emissions become the most limiting factor for the diesel engine optimization in derated applications.

3 ENGINE DOWNSIZING

3.1 Increase of firing pressure

The continuous development in engine structures and designs now makes it possible to significantly relax the limitations in firing pressures, maximum engine power and torque compared to the standards only a decade ago. Increasing the maximum allowed firing pressure is a very important leverage to improve the engine fuel efficiency, as it directly affects the engine thermodynamics. Such widely known trend can be highlighted already from the theoretical formulation of efficiency of the ideal diesel cycle, by rearranging its classical expressions. By calculating a series of ideal turbocharged diesel cycles, constraining the power output to be constant and regulating the maximum cylinder pressure by varying the scavenging air pressure at various compression ratios, one obtains the chart in Figure 3 (cited from [3]).

Whilst theoretical cycles may be of help in understanding some basic mechanisms reality tends to be much more complex which limits their applicability. Simulations can be instead a very powerful tool for these investigations as in our experience they have provided a high degree of precision in the prediction of performance of large two stroke engines and will be widely used in this work. For our purposes, a 1-D model of a possible

future two stroke engine has been developed within a commercial simulation platform (GT-Power) and utilized in the following.

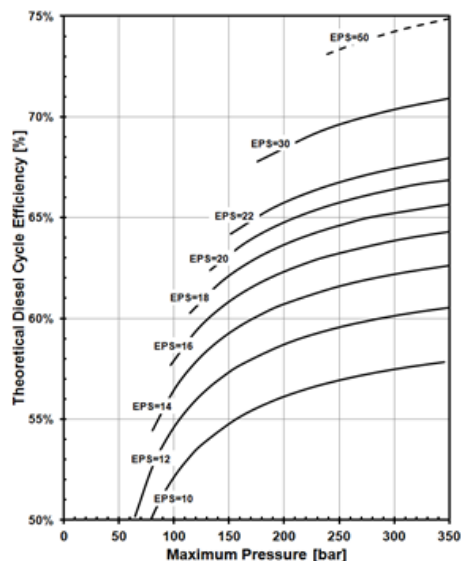


Figure 3: Maximal efficiency of the diesel cycle as a function of the compression ratio and maximum firing pressure applied [3].

In first instance, the model is utilized to show the effect of several hardware and settings modifications (more in particular: scavenging air pressure, start of injection, exhaust valve timings, compression ratio) to judge their effect on efficiency. Despite the wide range of parameter changes simulated the achievable BSFC gain is bounded by a pareto front when plotted against firing pressure. This clearly demonstrates the importance of the maximal firing pressure as one of the main thermodynamic restrictions to the increase of efficiency of large two stroke engines (Figure 4).

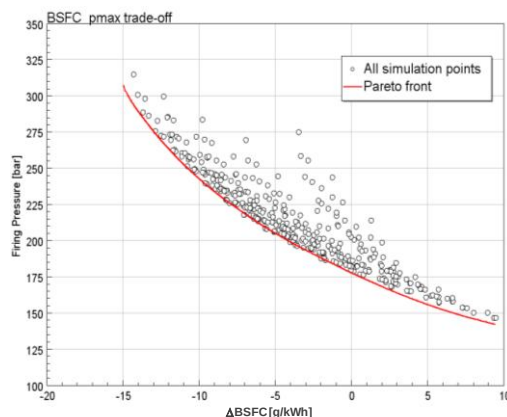


Figure 4: The reductions in BSFC achieved in a series of simulations with a wide range of parameter variations bounds against a Pareto front when plotted against maximal firing pressure.

3.2 Concept description

When higher firing pressures can be used, the developer challenge consists in constraining the NOx emissions within the compulsory limits. In this case, when no abatement technology is available, the diesel combustion shall be optimized to reduce the emissions, with the additional target to limit as much as possible the tendency towards duration increase, which in turn would reduce the overall efficiency benefit. Combustion optimization is achieved normally in a complex loop where theoretical trends are researched with the help of simulation tools and then verified with experimental tests. Typical parameters for optimization are the fuel injection settings and hardware (in particular the nozzle tip spray holes dimensions and orientation), the shape and dimensions of the combustion chamber and the in-cylinder flow field.

In the following a study case for a possible future product release is presented in which such combustion improvements are coupled with a high firing pressure design. At this purpose, a simulation model of a low speed two stroke engine having a bore size of 820mm and representative of the successful X82 engine series, has been created and validated. At each rating, various optimizations were assessed, always applying the latest developments of diesel combustion and the supposed upgraded level of firing pressure. Finally, for each engine rating, the optimum configuration was selected as the one delivering the highest predicted fuel efficiency with NOx compliance.

The various results obtained (hereafter referred to as new design or future product) were compared with the analogue calculations computed for the current product release. In many of the simulated ratings, the reduction in fuel consumption granted by the new design will amount (and sometimes even exceed) the level of 5g/kWh (BSFC IMO weighted result). This will constitute a substantial achievement in terms of CO2 reduction, as well as an economic benefit for the customers. Our preliminary evaluations comparing the current product and the future product (at the same rating) show in fact that the increase of CAPEX resulting from the increase of structure resistance to allow operation with higher firing pressure is paid back by the OPEX reduction in a very short time.

3.3 Engine selection

In addition to this important achievement, we would draw the reader's attention to another substantial improvement reported in the next charts. The two figures (Figure 5, Figure 6) depict the levels of BSFC reduction (IMO weighted, in

[g/kWh]) obtainable as a function of the rating compared to the ones for the full power and speed rating point.

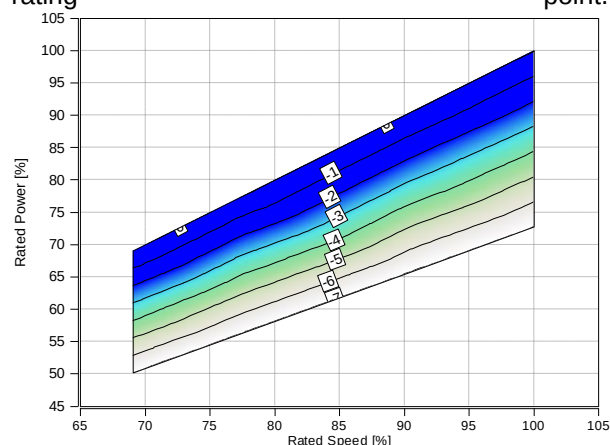


Figure 5: BSFC improvement (IMO weighted, in [g/kWh]) when derating an engine of the current engine release.

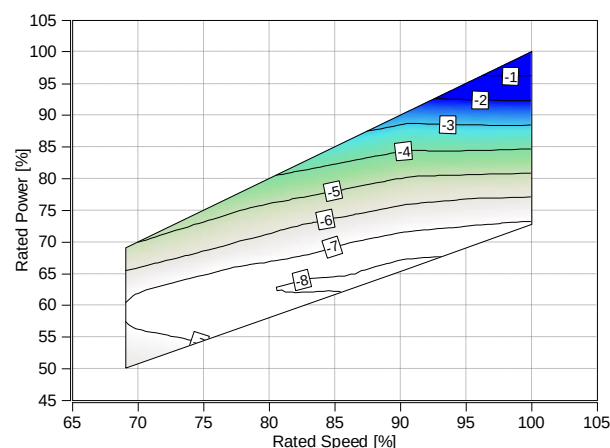


Figure 6: Expected BSFC improvements (IMO weighted, in [g/kWh]) when derating an engine for the proposed future product setup.

As one can notice, for the Future Product (Figure 6) the improvement with derating at low engine speed is quite reduced. This is largely due to the achievement of the limit of diesel combustion adaptation measures. Below such limit, the firing pressure must be reduced in order to keep the NOx under control and fuel consumption remains almost constant despite the derating. We expect this tendency to have several consequences for the engine selection at low engine speeds. In particular, the business case backing the selection of derated engines will become much less favourable for the owners, leading in many cases to the selection of products rated at higher specific power output, e.g. with less cylinders or smaller bore sizes. Mutuating the term from the automotive industry, we will refer to the advances above as engine downsizing, even if the

fundamental reasons for the reduction of the engine displacement are quite different.

From Figure 6 in fact, it appears that for a rating selection at lower engine speeds, an engine with less cylinders (i.e. a downsized engine) will perform at similar or slightly lower efficiency level than its peer with more cylinders, but at the same time the customer can benefit from other positive contributions to both CAPEX and OPEX. Capital expenditures will be significantly reduced due to the reduction of purchase price arising from the lower number of cylinders. In addition, another improvement will come from the reduction of the engine room space, due to the smaller engine dimensions and resulting in higher cargo capacity for the same ship. Also, lower service costs are to be expected with downsizing due to the reduced number of parts.

To clarify more, we would present the results of our economic evaluations for an actual application case. This case study is the calculation of the possible engine application for an oil tanker, with Contracted Maximum Continuous Rating (CMCR) of 25MW at 65rpm. Here two possible solutions are compared: the heavily derated 7X82 with new design and the corresponding 6 cylinders version. The calculated net present value for the selection of the second solution (e.g. the 6 cylinders version) against the first solution is presented in next Figure 7. In this figure, the x axis (e.g. NPV=0) represents equal cash flows between the two cases; a positive NPV instead represents an economic benefit for the selection of the 6 cylinders version. The obtained values are normalized to the expected engine price (CAPEX) and accounts for OPEX differences arising from Service and fuel expenditures. Interest rates have been supposed at 6%, while the calculation does not account for the additional positive cash flows that can be generated by the increase of the cargo capacity in the case of the smaller engine. Two hypothesis on the fuel cost are considered; one being an optimistic scenario with fuel at 300USD/ton and another one is instead rather pessimistic with fuel at 600USD/ton. The differences in fuel consumptions between the two solutions are taken from the previous Figure 5.

The result of the calculation displayed in Figure 7 shows that there is a large economic benefit for the customer in selecting the downsized 6 cylinders engine version. In the optimistic fuel price scenario, the overall economic saving after 15 years with this selection amounts to about 15% of the overall engine purchase price. In the case of higher fuel price, the selection of higher cylinder number generates positive cash flow only after 10 years.

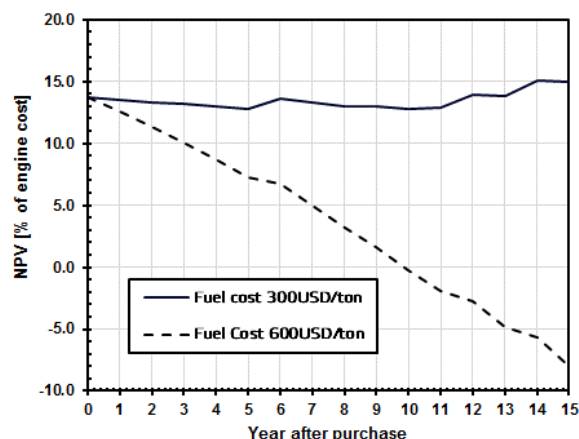


Figure 7: Net present value for the selection of a downsized engine compared to a basis case (NPV=0 axis) where instead the version with one cylinder more is selected. Application case: oil tanker, CMCR = 25MW at 65rpm, comparison of 6 vs 7cylinders future versions of the 820mm bore engine product.

The technology disclosed above displays a development trend for the next WinGD diesel products. The future engine releases are tracked to conform to such a development line, with specific applications on the particular product and combustion conditions. The first engines being developed according to the concept are the ones for the big bore segment, i.e. the X92-B and X82-D.

In conclusion, we expect the technology to bring an active and relevant benefit compared to today's levels and to be warmly welcomed by owners seeking for a sound financial technology investment.

4 TWO STAGE TURBOCHARGING

4.1 Technology review

In the last decade, the large engine industry has witnessed the introduction of two stage turbocharging technology, more specifically in application with four stroke engines. The application of higher boost pressures and supercharging efficiency featured by this technology, coupled with earlier intake valve closure (e.g. higher Miller timings) and increased firing pressure levels allowed consistent fuel consumption gains. Higher Miller timings in fact result in lower temperatures of the charge at top dead centre at same firing pressure, reducing the flame temperatures and resulting in lower NOx formation rates; it is therefore possible to increase the firing pressure while still keeping the NOx within the Tier 2 limits. On the other hand, due to the characteristics of four stroke cycles, the increased supercharging efficiency and supercharging pressure result in a more positive

work balance on the piston during the intake and exhaust strokes, i.e. during the charge replacement part of the cycle, consequent to the higher expansion of the exhaust gases.

Some studies and experiments for the application on large two stroke diesel engines have been already carried out by WinGD (Wärtsilä Switzerland at that time) [4], with the targets to reduce NO_x emissions or improve the fuel efficiency. More recently, Ryser has presented an innovative concept at the CIMAC 2016 congress [5] where two stage Turbocharging is coupled with an asymmetric port device and applied to a high firing pressure engine. Similarly to four stroke, a higher Miller timing is achieved (but realized with a late Exhaust valve closure) and coupled with high firing pressures to achieve efficiency gains at same level of NO_x. The increased supercharging efficiency is instead utilized to reduce the overall intake phase duration without major scavenging efficiency penalty. This was achieved by delaying the intake port opening timing with an asymmetric port device. In this way, a later exhaust valve opening is possible without resulting in exhaust gases blowback in the intake receiver and realizing a higher expansion degree of the combusted gases, which in turn benefits the thermodynamic cycle.

4.2 Thermodynamic Analysis

In the literature references, the overall efficiency gains reported or expected for the application of two stage turbocharging on marine two stroke engines are anyhow quite limited in comparison with the achievements declared for four stroke engines. Our first target is to understand the thermodynamic grounds for such a difference. On this purpose, we will utilize the kind of analysis proposed by Codan for four stroke engines [6] and apply it to study the effect of two stage turbocharging on the two-stroke cycle. For this purpose, we will focus our attention to the low pressure part of the cycle (Figure 8).

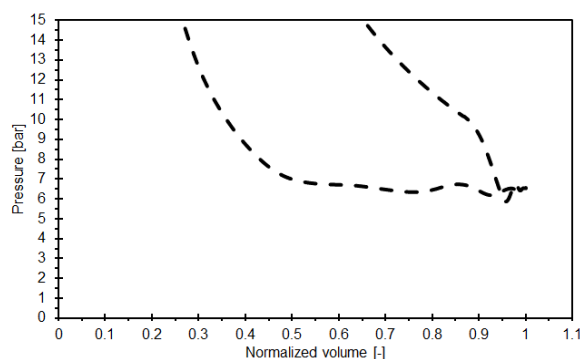


Figure 8: Low pressure part of the p-V diagram of a two stroke engine with high scavenging air pressure level.

Such cycle can be interpreted as the composition of four different contributions, as visually resumed in the following Figure 9.

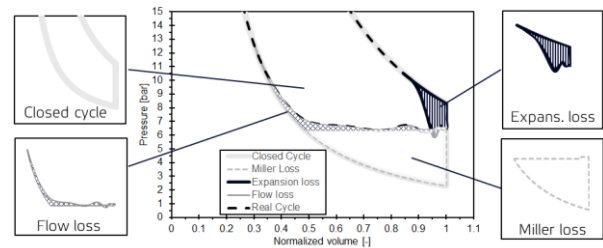


Figure 9: Visualization of the different contributions considered on the engine p-V diagram.

Those contributions are:

- A closed cycle: This is obtained by extrapolating the compression and expansion curves at closed valves until the bottom dead centre. This quasi-ideal situation corresponds to a case where charge replacement would happen instantaneously and the same quantity of air as in the real cycle is trapped into the cylinder. In other words, all the possible losses arising with the scavenging phase have now been eliminated

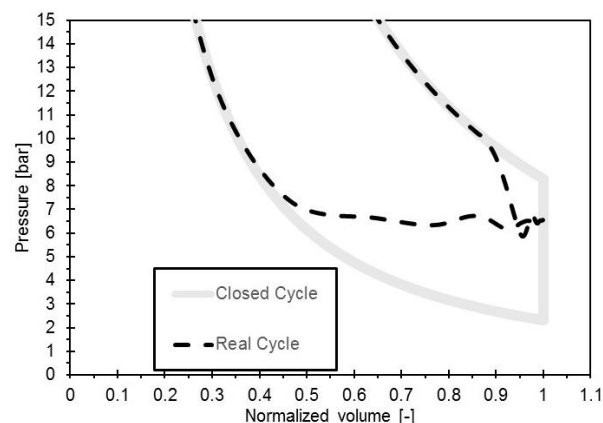


Figure 10: Construction of the closed cycle starting from the real cycle.

From this quasi-ideal situation, the thermodynamic losses arising with the scavenging phase are evaluated and subtracted:

- Miller loss: We now consider the presence of intake port and exhaust valve which are open only during the compression stroke to realize a non-instantaneous scavenging process. Port and valve are now supposed to operate with no pressure losses between the cylinder and respectively the intake and exhaust manifold, which are pressurized at the engine scavenging and exhaust pressure level. Intake duration lasts from BDC to the real engine intake port closure, while the ideal exhaust

valve is closed at the interception with the Closed Cycle previously defined. A new cycle is now described and being denoted below as Miller cycle. The difference between such cycle and the Closed Cycle is denominated Miller loss.

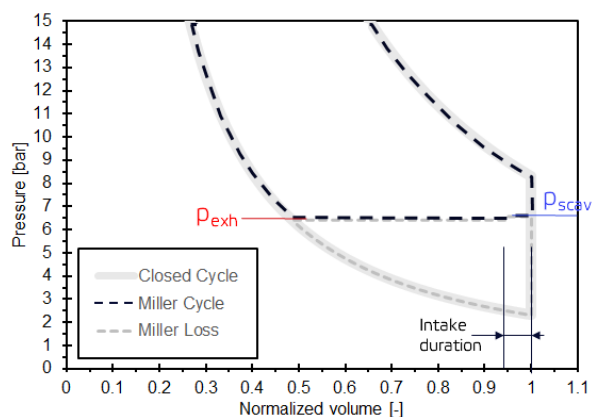


Figure 11: Illustration of the definition of Miller loss with the definition of relevant quantities for its construction.

- Expansion loss: As the actual intake port timings are symmetric, there is the actual need to open the exhaust valve before the BDC to reduce the pressure in the cylinder at the intake level and avoid exhaust gases to flow back into the air receiver. By considering real phenomena between the exhaust valve opening and the BDC and replacing this part of the cycle to the previously defined Miller cycle, one obtains the expansion loss.

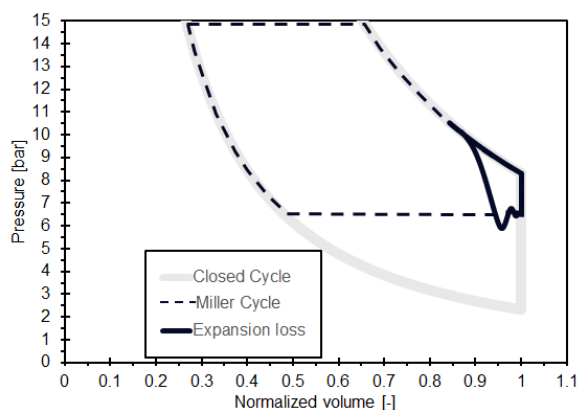


Figure 12: Illustration of the definition of Expansion loss.

- Flow loss: The real flow through valve and port creates pressure differentials between cylinder and respectively intake (intake port open) or exhaust (intake port closed) manifolds. This last contribution is to be evaluated only during the compression stroke, as the contributions during the expansion have been already accounted within the evaluation of the expansion loss. With this last modification, the real cycle is reproduced.

The above cycle analysis methodology has been employed to assess the effect of two stage turbocharging on two stroke engine. A 1D model of the system was built. The model was calibrated according to experimental data available from 2 stage turbocharging tests executed at the WinGD Diesel Technology Centre, which are for example shown in [4]. The model was then modified to simulate different levels of scavenging air pressure, regulating the Miller timing in order to keep constant the compression pressure. At the same time the start of injection was slightly regulated to keep the firing pressure constant; also the supercharging efficiency was set to be constant. The resulting thermodynamic cycles from the simulation were then analysed according to the methodology mentioned earlier and results are discussed below.

The following figure shows the development of each contribution in terms of overall cycle efficiency net effect compared to the reference and plotted as a function of scavenging air pressure.

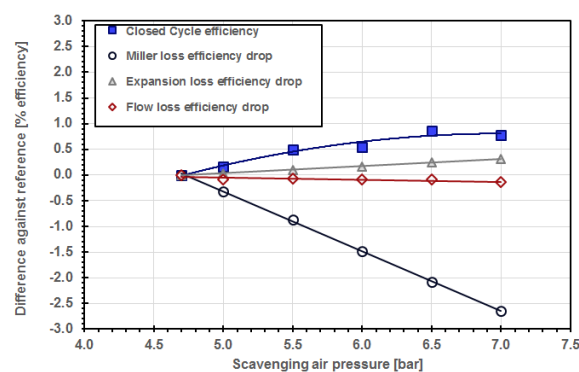


Figure 13: Evolution of the main losses with increasing scavenge air pressure.

As one can infer, the increase of scavenging air pressure only slightly improves the closed cycle efficiency (less than 1% by increasing the scavenge air pressure from 4.7 to 7bar). A slightly positive efficiency contribution is given also by the reduced expansion loss. Anyhow, those gains are more than balanced by the excessive efficiency reduction given by the Miller loss, consequent to the higher scavenging air pressure level. Comparing these results with the ones calculated by Codan for four stroke engines, we can find rather similar trends. The overall main difference lies in the absence, among the contributions for two stroke engines, of the pumping work (e.g. the work done on the piston during the intake and exhaust strokes), which in the case of four stroke was leading to a strong positive effect, increasing with the level of scavenging air pressure.

The global effect on efficiency of the indicated contributions on two stroke engines is the following.

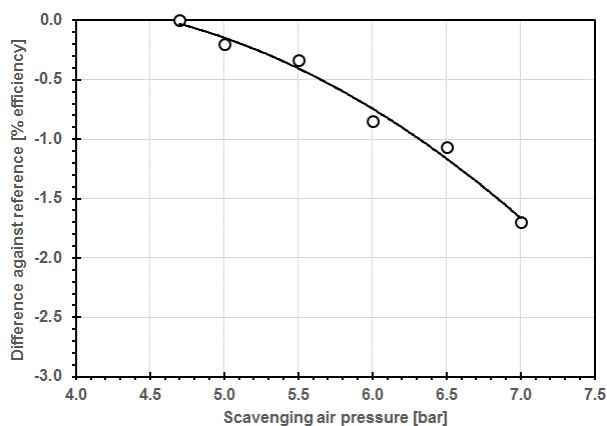


Figure 14: Sum of expected losses over scavenge air pressure.

As one can notice comparing the two figures (13 and 14), the dominant effect in the explanation of the trend is the increase of the Miller loss for high supercharging pressures. This is rather different from what would result on four-stroke engines, where the indicated efficiency was raising significantly with supercharging pressure and peaking at a boost pressure of about 6 [bar]. We would therefore conclude that the application of two stage turbocharging technology on a two-stroke engine has a more limited potential than on a four stroke engine due to the above explained thermodynamic mechanism.

4.3 Concept description

Even if the application of a two stage turbocharging system by itself is not beneficial for the efficiency improvement of a two-stroke engine, the increased Miller timing will result anyhow in lower compression temperatures as shown in the following figure.

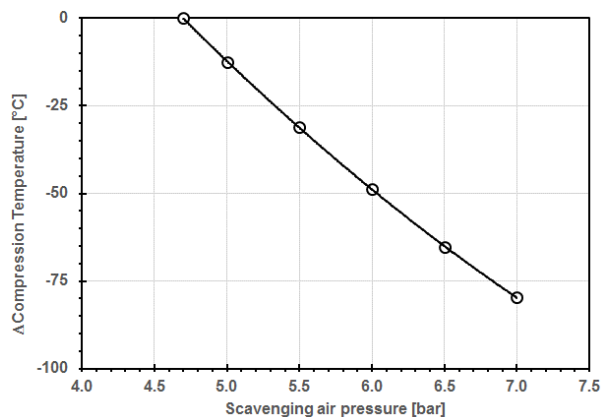


Figure 15: Decrease in compression temperature with increasing scavenging air pressure.

The above shown reduction of compression temperatures at higher scavenging air pressures leads to a reduction of the peak temperatures in the cylinder during the combustion, reducing the flame temperatures and consequently reduces the NOx formation. As already explained in this paper, it is possible to translate such NOx reduction into fuel consumption benefit by increasing the engine firing pressure.

The above mentioned simulation model of the engine with two stage turbocharger was therefore modified for this purpose and run to evaluate two different supercharging pressure levels at three different levels of engine BMEP. No other changes on the engine layout were done; only the settings were regulated. The firing pressure was regulated through injection begin to keep the NOx at a specified limit, the compression pressure was regulated with exhaust valve closure modification to achieve a similar ratio with firing pressure as in the baseline case (e.g. same firing ratio) and the exhaust valve opening was regulated for achieving a similar margin from blowback into the intake receiver. The results of our evaluation are shown below (Figure 16) in terms of IMO weighted BSFC gains compared to the current reference.

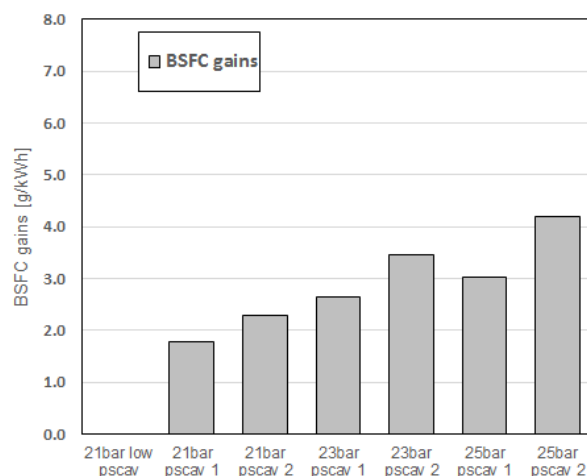


Figure 16: Expected BSFC gains (IMO weighted) at constant NOx for different combinations of the charging concept, the scavenge air pressure and BMEP.

As one can notice, the gains are quite limited compared to the increase of complexity of the layout. At the same time, the engine firing pressure was increased significantly in order to achieve such results.

As a last step, we considered also the idea proposed by Ryser about the regulation of the intake port opening timing to exploit the increased supercharging efficiency level. As in his proposal, we considered a sleeve system which is regulated to slide on the outer liner surface by means of a

hydraulic actuation. For details about the mechanism we would refer to the cited publication [5]. A model of the device was created in the simulation environment and applied to the engine model. Optimization of the device timings and speeds was performed to achieve the best fuel efficiency still compatible with a sufficient engine scavenging level. The baseline engine setup was selected to be the best one resulting from the previous optimization, e.g. the one featuring high scavenging air pressure and BMEP = 25bar. Final results of the optimization are showed below and report a possible additional fuel consumption gain of about 2 g/kWh with the utilization of an asymmetric port timing device. The overall results do not fundamentally deviate from the ones reported by Ryser in his work, so we would confirm his findings that there is a potential in the application of two stage turbocharging on large two stroke engines for improving the efficiency; such improvement potential is anyhow limited for basic thermodynamic reasons, which were also clarified.

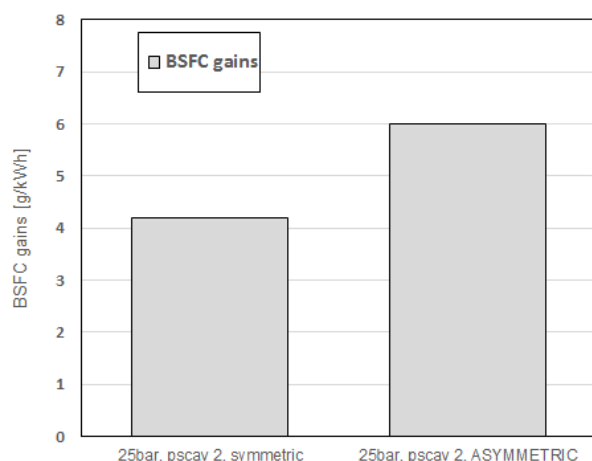


Figure 17: Improvement in expected BSFC (IMO weighted) if asymmetric port timing could be applied.

4.4 Challenges

An actual application of the technology is challenging for technical and business reasons. From the technical part, we would mention the following as the major barriers to the introduction:

- Engines would need a complete redesign, as the firing pressure levels requested for the proposed concept are clearly higher than for today's engines.
- A reduction of exhaust gas temperatures is expected and some mitigation measures might be needed to avoid sulphuric acids condensation in the exhaust funnel and in the boiler.

- Even if the application of two stage turbocharging on two stroke engines may be less complex in design and operation than for four stroke applications, it is nonetheless a challenging endeavour due to the huge product customization present in the low speed sector where each engine can be ordered anywhere within the rating field. In particular we may expect quite many complications for the turbocharger matching definition compared to today's procedures.
- The asymmetric port device, which accounts for a consistent part of the efficiency benefits, is not yet developed and its design presents some challenging points.

From the economic perspective, the conclusion appears to be mixed. The second stage turbocharger and cooler, with the additional systems and the redesign of the engine would substantially increase the costs of the product. Anyhow, this will be partly balanced by the increase of the power density which will result in the selection of engines with less cylinders or smaller bore size, reducing the cost of the basic engine. According to our preliminary estimations, the improvement of OPEX consequent from the increase in efficiency is expected to return the additional investment for the customer in some years. In this case, the future evolution of regulations towards a CO₂ reduction, such as the compulsory reduction of EEDI in the coming years, may trigger additional interest on the application of the technology.

5 SEQUENTIAL TURBOCHARGING

Whilst two stage turbocharging is an optimization of the installation aimed at increasing the scavenge air pressure, the efficiency and the power density, the sequential turbocharging described here is more focused on optimizations arising from the operational profile of the installation.

5.1 Concept Description

Under the same name of sequential turbocharging technology, few variants are possible. Traditionally, sequential turbocharging has been more widely used in marine industry in its variant of turbocharger cut-off, for applications on large 2 stroke engines already on the field where it is proposed as a retrofit upgrade. In such variant, one or more turbochargers are permanently blinded off and are thus taken out of operation. The remaining charger(s) then supply the engine with the required air amount, albeit at a higher scavenge air pressure. This finally results in a lower BSFC but comes at the cost of a limited operation range/speed of the engine in order to

prevent over-speeding of the remaining turbocharger(s). Such operating limit varies as function of the total amount of turbochargers and the number of the ones that are cut-off. In the case of cut-off of one charger in a configuration with two identical turbomachines, the operation is normally restricted to approximately 50%.

In the most classical sequential turbocharging concept instead, the number of turbochargers used over the operating load varies thanks to the flexible actuation of turbocharger cut-off valves, the load range itself though is not restricted. As this concept can be applied in principle on any engine featuring a design with multiple turbochargers, a multitude of possible configurations and actuation strategies exists; for simplicity an installation with only two turbochargers will be considered in the following to explain the working principle. At lower engine loads, one of the turbochargers is deactivated. The exhaust gases are therefore fed through only one of two turbines which leads to a much higher scavenge air pressure which in turn results in higher firing pressures. This again leads to a significantly lower BSFC (and thus CO₂) at the same amount of NO_x emissions. One drawback is that the amount of residual gases present in the cylinder tends to increase due to a deteriorating scavenging behaviour [7] resulting from the increase of the density of the charge air. At high engine loads instead, all the turbochargers are activated. The switching of operation from one to two turbochargers in marine application is normally quite constrained and in many cases the operators are instructed to perform such operation after having significantly reduced the engine load.

A further upgrade of this solution has been investigated and proposed jointly by WinGD and ABB. Starting from the classical concept, two main new features have been introduced:

- The flexibility in the definition of the turbochargers size ratio. The turbochargers selected no longer need to be identical but can consist of a “larger” and a “smaller” specifications;
- The cutting-in/out of turbocharger(s) is dynamically optimized, so it can be done during normal engine operation without the need for a load reduction.

The first change compared to a traditional layout with equal turbochargers allows an increase of the flexibility in the definition of the strategy, leading to an improvement of the possible benefits in terms of efficiency. As an easier example, we consider a configuration with two turbochargers where the total capacity is split in ratio 60-40% between the

two turbomachines. In this case, the achievable load with only one charger active is increased to approx. 60%, compared to about 50% achievable with equal sizing, thus enlarging the load range where fuel benefits can be realized. Additionally, three different operating modes can be envisaged:

- Running only the “small” charger
- Running only the “large” charger
- Running both chargers together

Figure 19 shows the four most promising actuation strategies obtainable by combining the three operating mode for the example configuration (2 turbochargers, 60%-40% sizing). Of course, in some installations, such high degree of complexity might not be needed; in such cases, it will rather be preferable to install the cut-off valves only on one charger and realize an easier actuation strategy. In addition to the operating mode, the developer will also have the possibility to optimize the turbochargers capacity ratio (e.g. the size of the various turbochargers). The limits in this definition depend on the fuel efficiency gains that can be achieved, on the necessity to grant sufficient scavenging of the engine and in the effect on the main thermomechanical loadings (which are application specific). From a turbocharger point of view the concept is much simpler as even different turbo frame sizes can be applied on the same engine without problems.

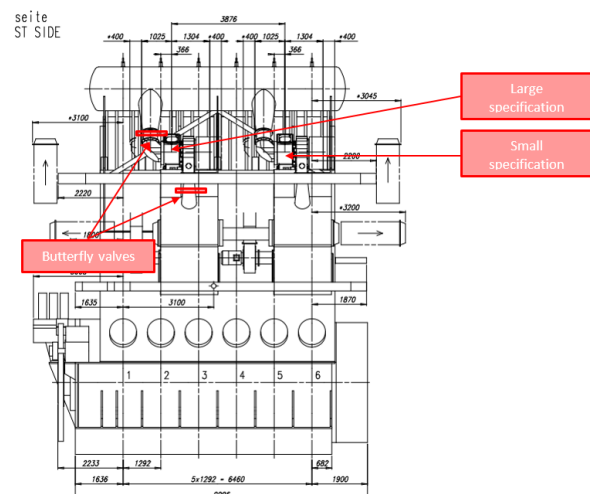


Figure 18: Sketch of the setup for sequential turbocharging depicting the large and the small specifications as well as the butterfly valves installed.

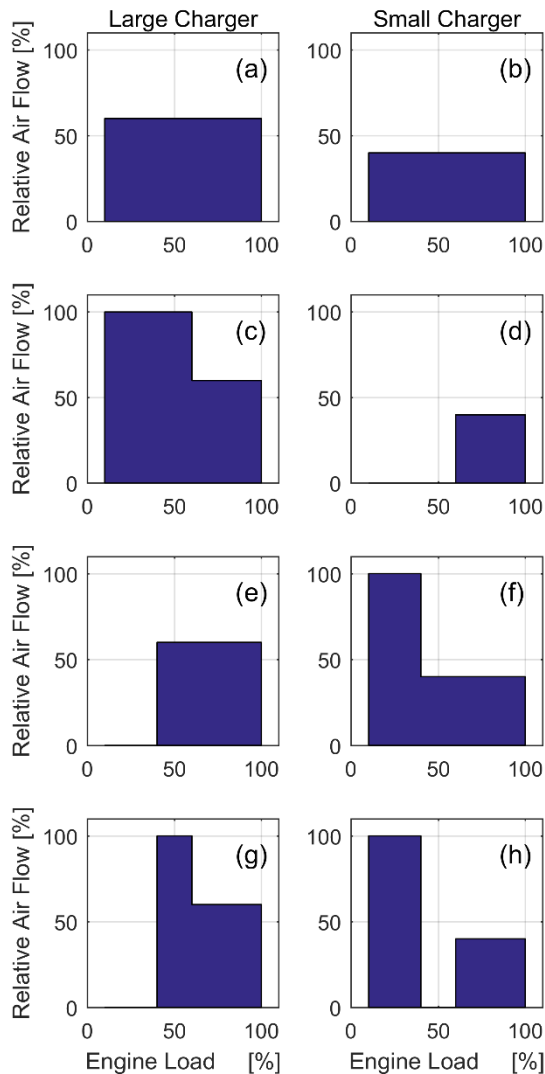


Figure 19: Four different actuation strategies possible with two differently sized chargers (e.g. 60%-40%). (a-b): Strategy 1 – All turbochargers always activated. (c-d): Strategy 2- only large charger operating up to 60% load, all chargers active above 60% load. (e-f): Strategy 3 – only small charger operating up to 40% load, all chargers active above 40% load. (g-h): Strategy 4 – only small charger operating up to 40% load, only large charger operating in the range 40-60% load, all chargers operating above 60% load.

5.2 Achievable Reductions in BSFC

The achievable reductions in BSFC can be quite significant (Figure 20). In the chosen example reductions by ~3% at the same levels of NO_x are well feasible. This translates to up to 5g/kWh fuel savings which, especially in view of the small additional installation cost (mainly valves) is substantial.

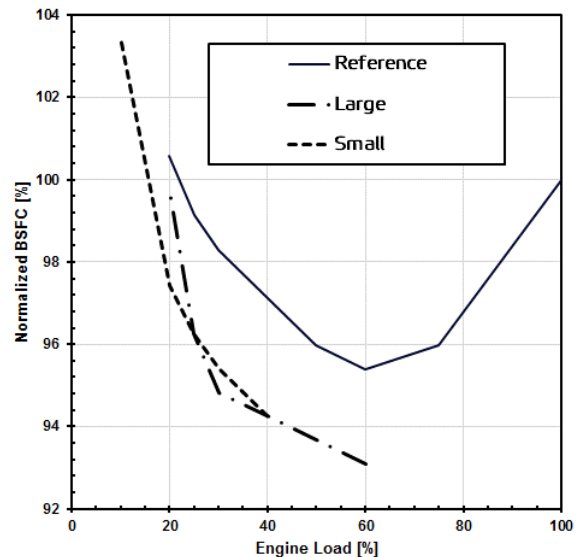


Figure 20: Normalized BSFC for different Sequential Turbocharging setups, all at same NO_x. Results for a turbochargers configuration with a 60-40% capacity ratio.

5.3 Risks

The higher scavenge air pressure achievable with sequential turbocharging leads to higher scavenge air density, which in general leads to a lower scavenging efficiency. This in turn leads to a higher amount of residual gases in the cylinder which potentially, due to both the acidic nature of the combustion gases as well as the particle content can lead to piston running issues. In former times this was seen as one of the major obstacles for the implementation of turbocharger cut-off. With the advent of high BN (base number) lubrication oils however, the associated risk of scuffing has been significantly decreased. Obviously, the use of higher BN oils might lead to somewhat higher operating costs associated with the lube oil consumption, these costs should however be small compared to the savings in fuel.

Other risks that need to be considered in the context of sequential turbocharging are obviously those connected to any kind of valve failure. To a large extent these can be alleviated by ensuring a “normally open” position of the valves so that in the fallback operation both chargers would be active at all loads (Figure 19, (a) & (b)). This is from an operational point of view nearly identical to running with identical turbochargers, e.g. the current standard.

5.4 Installation-specific Layout

The layout of an installation with sequential turbocharging needs to be optimized on a case-by-case basis, as the influence of various parameters on BSFC as well as other operational

parameters such as exhaust temperatures is rather large. The optimum solution depends both on the rating of the engine (e.g. firing pressure margin available), on the tuning selected, on the operational profile and on the turbocharger configuration available. Additionally, due to the large increase in firing pressures at lower speeds the influence on torsional vibrations in the shaftline needs to be carefully considered already in the planning stage of an installation.

5.5 Engine Behaviour when Cutting-In/Out

Switching on or off a turbocharger from operation whilst the engine is running in principle may be associated with concerns on the dynamic response of the installation. Items that need to be taken into consideration are e.g. the timing and the speed of the valve movements on turbine respectively compressor side, the speeding-up behaviour of the turbocharger and the associated dynamic scavenge air pressure change, the reaction of the engine control system to the sudden change in scavenge air pressure as well as the reaction of the components such as cylinder liner on the change in operating conditions.

As experienced during testing, the engine reacts rather slowly to the sudden change in hardware. In general the temperatures move smoothly from one steady-state condition to the other while the transition of operation is completed quickly.

The experiments were repeated at several different loads, all of which showed the same benign behaviour. When cutting-off one turbocharger, the remaining charger needs some time to speed up and to build up the new, higher scavenge air pressure (Figure 21). This process is largely completed within 60 seconds. The turbocharger cut-in procedure is much faster, the speeding-up from standstill is completed within 10...20 seconds (Figure 22).

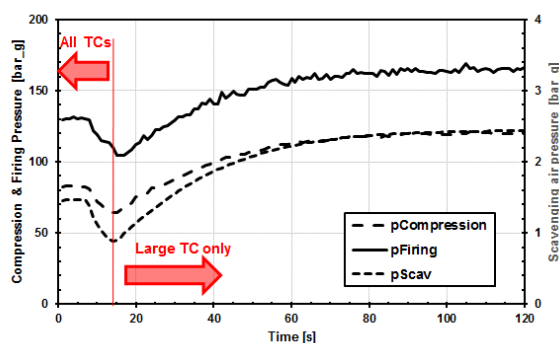


Figure 21: Evolution of Scavenge, Compression and Firing Pressures when taking one TC out of operation.

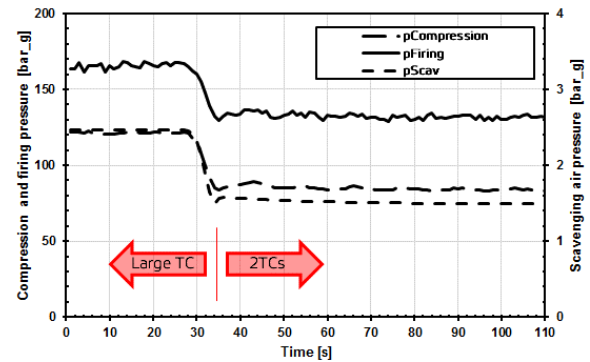


Figure 22: Evolution of Scavenge, Compression and Firing Pressures when putting both TCs into operation.

Overall the sequential turbocharger concept, when implemented in the manner presented above, has proven to be a relatively simple and robust way to increase the efficiency of the engine at part load and thus reduce overall fuel consumption and CO₂ emissions.

6 CONCLUSIONS

The three studies presented above all show the potential of further increasing the overall system efficiency, reducing the (cost of) fuel consumption and thus also the CO₂ emissions of a two stroke main engine.

The three cases selected are very different in terms of functioning principle, achievable gains and also complexity of the installation. Despite they have been studied separately, they are also not mutually exclusive, as in general one could envisage merging of the three technologies. The limiting factors here are the installation cost on the one hand and system complexity on the other.

When comparing an existing installation consisting of a diesel engine with single stage turbocharging and a possible future installation consisting of a diesel engine, an aftertreatment system and possibly a special charging concept, each with their associated operating peculiarities and subsystems, it becomes obvious that the complexity of operations is orders of magnitude higher.

The rising number of systems and their interaction however, many of which are due to stricter emissions regulations, poses higher requirements to the crew which needs to know how to operate all of these systems individually but also understand the interactions between the systems and the limitations that arise.

We believe that it is, from a technological point of view well possible to significantly reduce the

emissions footprint of a vessel, both with regards to e.g. NO_x as well as CO₂. This reduction however comes at a cost which today probably is underestimated.

WinGD is committed to continuing the development of technologies that can significantly reduce both CO₂ emissions as well as the total cost of ownership despite the challenges mentioned above. We are positive that this can be achieved without adding onto the burden of the crew and – last but not least – without sacrificing the high level of operational safety and reliability known from today's installations.

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8 REFERENCES AND BIBLIOGRAPHY

- [1] IMO (International Maritime Organization), Revised MARPOL Annex VI, www.imo.org
- [2] Kadau, D., Brutsche, M., Zürrer, S., Soppelsa, F. Compact marine high pressure SCR system technology development, *Cimac Congress 2019, Vancouver (Canada)*: paper No.405.
- [3] Rebecchi, P., Scrocco, P., Signorile, M.C., Nanda, S.K., Ott, M. 2015. Insight into the Combustion Control of WinGD 2-Stroke Dual-Fuel Dual-Combustion Engines, *Symposium for Combustion Control 2015, Aachen (Germany)*: 19:24
- [4] Ryser, R., Weisser, G., Wik, C. 2010. Application of 2-stage turbocharging to large diesel engines: Recent developments and new perspective, *15. Aufladetechnische Konferenz 2010, Dresden (Germany)*: paper No.102
- [5] Ryser, R., Mathey, C., Mutter, A. 2016. Power2 - Is 2-Stage Turbocharging Interesting for 2-Stroke Engines?, *Cimac Conference 2016, Shanghai (China)*: paper No.281
- [6] Codan, E., Huber, T. 2012. Potential of 2-stage Turbocharging, *17. Aufladetechnische Konferenz 2012, Dresden (Germany)*
- [7] Weisser, G., Scrocco, P., Ioannou, M., Hattar, C. 2013. Sequential Turbocharging on Large 2 Stroke Engines: A Promising Concept for Existing Installation as well as Future Products, *18. Aufladetechnische Konferenz 2013, Dresden (Germany)*: paper No.1

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