Abstract: Gas and dual fuel engines applying the Otto cycle are commercially available solutions undercutting the NOx limits set out by the IMO Tier 3 regulation. Furthermore, it has been demonstrated that the concept of the strong Miller process enabled by two-stage turbocharging and variable valve timing is an attractive solution to considerably improve the trade-off between fuel consumption and NOx emissions for diesel engines. For gas engines, the concept can be utilised to bring the trade-off between fuel consumption and power density to a new level. The potential has been confirmed on real gas and diesel engines with excellent results. A comprehensive simulation study has been carried out in order to transfer the concept of strong Miller cycle, two-stage turbocharging and variable valve timing to dual fuel engines.

The study identified great potential for improving fuel consumption in both diesel and gas mode when integrating ABB’s two-stage turbocharging Power2 and variable valve train VCM® into a seamless concept. Furthermore the flexibility of VCM resolves some of the compromises needed when designing an engine capable of running according to the Diesel and the Otto cycle. In addition, FPP operation as well as improved manoeuvrability and load pickup for gas and dual fuel engines may be realised, as the extensive thermodynamic simulation work indicates. The publication presents the assumptions and boundary conditions that were used, the underlying modelling approach and the potential found, while also discussing expected challenges.
INTRODUCTION

Pure gas and dual fuel (DF) engines are widely discussed alternatives for marine propulsion, because they are able to comply with the NOx emission limits set forth by IMO Tier III. However, the performance of today’s established marine engine designs is limited mainly by the phenomenon of knocking combustion. In the case of pure gas engines, this limitation sets the achievable efficiency and power density. Additionally, in the case of DF engines, operation in diesel mode also suffers from lower efficiency.

Based on a cycle simulation study, this paper will introduce a concept for DF engines, which allows for increasing the power density while reducing the fuel consumption in both the diesel and the gas operation mode. Additionally, the concept contains a control scheme for more spontaneous load response. The simulations results predict a reduction of fuel consumption of up to 20 g/kWh, in the relevant load range of roughly 15 g/kWh compared to today’s commercially available reference engine. At the same time, the power density was increased to bmep = 26 bar for both diesel and gas mode operation.

In order to realize the proposed concept, the following technology building blocks are needed:

- Highly efficient two-stage turbocharging in order to provide a suitable boost pressure required by applying strong Miller cycle. With Power2®, ABB has a solution which perfectly fits the requirements
- Variable inlet valve closing for air/fuel ratio control and switching between fuel optimized diesel and gas operation mode (e.g. ABB’s variable valve train solution VCM®)
- Pilot fuel injection system capable of flexible SOI setting
- Optimized but fixed compression ratio
- Mechanical structure for bmep = 26 bar and cylinder firing pressure up to 220 bar

Nowadays, it is a well-known fact that introducing strong Miller cycle and two-stage turbocharging contributes to improving engine efficiency and allows for a higher power density. [1], [2], [3], [4] and [5]. With Power2, ABB provides a two-stage turbocharging solution in place that supports OEMs in implementing such concepts. A simulation based case study will demonstrate this effect for marine gas and DF engines in the present paper.

Moreover, it will be shown that applying variable valve train technique, for instance the ABB variable valve train solution VCM, has the potential to resolve the design compromises of DF engines. Both diesel and gas mode operations will be able to work at their individual best thermodynamic potential in the range of nominal engine power. By further exploiting the capabilities of the variable valve train, e.g. skip firing, part load efficiency in the gas mode is further improved.

The proposed concept for the gas mode will also be applicable to pure gas engines since VCM enables the individual optimization of the gas and diesel mode operation.

Starting with established DF engine technology, the material is organized in the following manner. First, currently used engine control strategies are introduced. Then, fields for engine development are indicated leading to opportunities for power density increase and fuel saving. Finally, cycle simulations have been employed to investigate the new concept and to estimate and discuss its potential but also its challenges.

AVAILABLE TECHNOLOGY AND CURRENT DEVELOPMENT PATHS

Established DF Engines

Today’s commercially available marine DF engines have a bmep in the range of 20 to 22 bar. They feature a single stage turbocharging system combined with moderate Miller cycle and fixed valve timing. The compression ratio $\varepsilon$ is rather low (in the range of 11 to 13) in order to prevent engine knock in the gas mode. Often the bore size is somewhat enlarged to match the output of their diesel counterparts. Commonly, DF engines are operated at constant engine speed. However, engines for FPP operation have also been announced lately, [6], [7].

Gas Engine Control Strategies

The power of port injected gas engines is controlled by the gas admission valve in the inlet ports. In order to operate the engine at a desired NOx emission level within the applicable operating window between the limits of misfiring and knocking combustion, the air excess ratio $\lambda_V$ is adjusted by a corresponding control device. For established DF engines, an Exhaust Waste Gate (EWG) is used for the above purpose. Figure 1 shows other possible
control elements, such as compressor recirculation, throttle valve and Variable Turbine Geometry (VTG). All the control devices shown serve the purpose of controlling the air excess ratio $\lambda_V$ (or engine power in the case of premix gas engines), however with a different impact on engine performance, as will be explained below.

![Figure 1 – Engine topology with control elements](image)

In the case without valve timing variability, the charging efficiency $\lambda_l$ results from the steady state engine operating point, depending mainly on engine mass flow rate and speed $n_{Eng}$. According to Eq. 2, the air excess ratio $\lambda_V$ is given by the prevailing charging air pressure $p_{Rec}$. If the receiver pressure, and consequently $\lambda_V$, is re-adjusted using a compressor recirculation or throttle valve, e.g. in the case of lower ambient temperature, the engine does not perceive any changes, because intake and exhaust receiver pressure levels stay approximately the same. On the other hand, if with the same turbine specification, the receiver pressure is adjusted by blowing off more gas from the exhaust receiver with an EWG, engine back pressure is reduced. This leads to reduced gas exchange losses and as a result engine efficiency is improved.

$$\lambda_V = \frac{p_{Rec} \cdot \dot{\lambda} \cdot H_u \cdot \eta_{Mot}}{L_{min} \cdot bmep \cdot T_{Rec} \cdot R_{Rec} \cdot L_{min}} - \frac{1}{L_{min}} \quad \text{Eq. 1}$$

$$\lambda_V \propto p_{Rec} \cdot \dot{\lambda}_l \quad \text{Eq. 2}$$

Key Points of Future Engine Development

For the future development of DF engines the following four points are expected to be of major importance:

1. Improvement of efficiency in both gas and diesel mode
2. Extended power density
3. Improvement of load step response or engine acceleration without decrease of steady state engine efficiency
4. Enabling direct drive propulsion by mitigating the tendency to knocking combustion

The next section will sketch an engine and turbocharging concept that aims at achieving the above mentioned aspects.

OPPORTUNITIES FOR DF AND PURE GAS ENGINES

Control of unwanted knocking combustion is a key factor to increase power density and efficiency of engines operating according to the Otto cycle.

The occurrence of combustion knock can have many sources, for instance low methane number of the fuel gas, insufficient homogeneity of the in-cylinder charge, cylinder-to-cylinder or cycle-to-cycle variability of the charge composition, hot spots in the combustion chamber, deposits, evaporating lube oil, unfavorable pressure-time-history in the bulk of the unburned gas of the cylinder charge, etc.

Prediction of knocking combustion is generally very difficult and needs to be properly addressed, e.g. when designing air/gas mixing and distribution to the individual cylinders or when designing the cooling system of the cylinder head, to name only a few of the involved design tasks.

From a turbocharging point of view, the impact of the pressure-time history in the unburned gas in the cylinder is of major interest, since it directly influences the requirements of the turbocharging system. Therefore, the following section will mainly focus on the self-ignition starting in the bulk of the unburned gas.

Improving Efficiency in Gas and Diesel Mode

As mentioned above, the occurrence of unwanted knocking combustion in gas engines depends on in-cylinder temperature and pressure history. Residence time of the unburned gas at high values of temperature and pressure leads to increased knock tendency, [8], [9]. For a given cylinder head, cam design, and receiver temperature level, this leads to limited design values of compression ratio and bmep. The higher the required design power density, the lower the allowable maximum compression ratio $\varepsilon$ will be, [10].
For a given air excess ratio the reduction of the cylinder charging efficiency \( \lambda_l \) (increase of Miller effect) results in reduced in-cylinder cycle temperatures which enables an increase of the compression ratio or/and the power density. However, for maintaining a constant air excess ratio \( \lambda_v \), the charge air pressure needs to be raised to a higher level (\( P_{rec} \) varies inversely with the charging efficiency \( \lambda_l \) as shown in Eq. 2). Only moderate levels of Miller effect can be realized because of the limited pressure levels that can be achieved with single stage turbocharging. As mentioned above, present engines use the available charge pressure to achieve power densities of roughly bmep = 20 bar at a compression ratio of about \( \varepsilon = 11 \) to 13. Because of the constraints imposed by the gas mode operation, engine efficiency suffers especially in diesel mode, for which a compression ratio around \( \varepsilon = 16 \) would be favorable.

Figure 2 shows that an increase in compression ratio gives rise to significantly improved closed cycle efficiency. The plot has been derived at constant power density, air excess ratio and cycle start temperature for an ideal limited pressure cycle (Seiliger process) based on ideal gas properties. The blue lines show variations from the isochoric to the isobaric process for various levels of maximum cycle pressure. Consequently every blue line represents a constant ratio of \( P_{max} / imep \). The plot illustrates the unused efficiency potential of the diesel mode when operated at a compression ratio \( \varepsilon = 12 \). An increase to a common value of about \( \varepsilon = 16 \) allows for an increase in closed cycle efficiency of more than \( \Delta \eta = 5\% \). The increase in the compression ratio up to a value of 16 could be made feasible by increasing the Miller effect, improving thus engine efficiency in both modes. Of course, this also requires a corresponding substantial increase of charge air pressure. The required levels of charge air pressure cannot be supplied at a power density level of about bmep = 24 bar and above by single stage turbocharging, consequently two-stage turbocharging becomes indispensable.

Meanwhile pure gas engines are leaving behind their diesel counterpart in terms of engine efficiency by consequent application of the Miller cycle and two-stage turbocharging. With respect to power density, gas engines are approaching the level of established diesel engines, [2], [3]. Also the compression ratio is being shifted towards the values common for diesel engines.

Two-stage turbocharging efficiencies are far above the single stage level mainly because of its intercooled compression process. The higher the required total pressure ratio the more efficient the intercooled two-stage compression process becomes. High turbocharging efficiency results in increased engine pressure drop and thus strongly improves the gas exchange work.
losses, the lowered process temperature due to the increased Miller effect substantially increases the closed cycle efficiency.

**Figure 3 – Gain in gas exchange efficiency with EWG and VCM compared to control with throttle valve (or compressor recirculation)**

**Improving Power Density and Enabling FPP Engine Operation**

As mentioned above, knocking combustion limits gas engines at higher power densities since increased in-cylinder pressure is an important trigger of engine knock. Moreover, residence time plays a very critical role too besides temperature, pressure, air excess ratio and gas properties as will be shown in a subsequent section about modeling the knock phenomenon. The longer a gas mixture is exposed to a certain level of pressure and temperature the more probable the occurrence of knocking becomes. For this reason, engine operation at reduced engine speed and increased torque (bmep), e.g. FPP operation, is an especially demanding task for gas engines as the residence time of the charge mixture within the cylinder becomes longer. This challenge can be met by applying a correspondingly strong Miller timing. By the resulting reduction of the cycle process temperature the risk of engine knock is mitigated, thus allowing for improved power density and high torque at reduced speed.

From the perspective of a simulation-based study, the subsequent section will introduce a concept to improve efficiency and power density of Otto-cycle engines (pure gas as well as DF engines), to come to par with their diesel counterparts. ABB is able to support the development of future gas and DF engines with high charge pressure and high efficiency with its assets in the area of two-stage turbocharging and the new variable valve train system VCM.

**SIMULATION-BASED APPROACH TO NEW DF AND GAS ENGINE CONCEPTS**

Starting point of the simulation-based study is an engine model calibrated with measured data obtained from a single stage turbocharged DF engine with moderate Miller cycle and fixed camshaft featuring a standard main diesel injection system, a CR system for pilot fuel injection and a port injection system for gas admission. This simulation model has been extended and modified with the following features:

- two-stage turbocharging system
- increased compression ratio $\varepsilon$
- strong Miller effect
- variable inlet valve train

The increase of the cylinder compression ratio $\varepsilon$ will consequently raise the diesel mode engine efficiency. In order to prevent knocking combustion in the gas mode however, the Miller effect has to be increased and/or the combustion phasing needs to be retarded. Moreover, the right choice of the parameters such as compression ratio, Miller effect (charging efficiency), and combustion phasing is not only restricted by the combustion knock limit, but also by safeguarding conditions for safe ignition of the pilot diesel spray. In addition, other design limits such as the maximum allowable cylinder pressure and turbine inlet temperature have to be considered as well.

The prediction of combustion knock and the ignition of the pilot spray (in the gas operating mode) by simulation are challenging tasks. The next subsection will show in more detail how the boundaries of knocking combustion and pilot spray ignition have been estimated. Subsequently the setup of engine and turbocharging system considered in the simulation study will be introduced.

In the case of pure gas engines, typically a scavenged pre-chamber is used for ignition of the charge. It is assumed that such a system imposes less constraint in terms of pressure and temperature for safe and reliable ignition. Therefore, the next section focuses on the ignition using a pilot spray.

**Key Model Assumptions**

a) Combustion

In gas mode, the heat release rate has been simulated according to the Wiebe model. While the combustion shape parameters have been kept constant over all engine load points in the gas
In the diesel mode, the combustion length and shape parameters are recalculated for each operating point according to Woschni and Anisits, including a supplementary modification of the shape parameter from ABB’s experience. While the assumption of constant combustion length and shape is common practice for conventional gas engines, it is expected that the combustion parameters are influenced by changes in charging efficiency $\lambda$. Unfortunately no reliable combustion model is available yet. Ongoing research [11] will help to work out more reliable simulation results.

For the simulations in gas mode, the amount of injected pilot fuel is neglected. Heat is only added to the process by gas combustion. No additional mass is added to the cylinders after the gas exchange.

b) Pilot Spray Ignition Delay $\tau_{ID}$

In the gas mode of a DF engine, the lean gas mixture charge is set off by the auto ignition of a small amount of directly injected diesel fuel (micro pilot spray). The duration of the ignition delay of the pilot diesel spray influences combustion quality. A very short ignition delay results in poor dispersion of the ignition centers. On the other hand, a too long ignition delay eventually leads to an increased number of misfiring cycles. In every case injection timing, and thus the resulting ignition delay, influence the variation coefficient of indicated mean effective cylinder pressure [12]. Increased cycle-to-cycle deviations require a higher margin towards knocking combustion and maximum cylinder pressure. Consequently this gives rise to a lower engine efficiency.

In the present study, ignition timing is estimated by an ignition delay model which is based on pressure and temperature trajectory after the injection of the fuel to provide for a better comparability of the simulation results.

A simulation tool for the combustion of lean burn gas engines with ignition by liquid micro pilot fuel injection has been elaborated upon in a recent research project [13]. Included is also a model for pilot spray ignition delay calculation. Unfortunately, the tool had not been released when carrying out this study. However, measurement data from [13] available at that time has been used to check and calibrate the less detailed model described in the following paragraph.

For the calculation of the ignition delay $\tau_{ID}$ of common diesel engines, the correlation described in [14] is often applied, see Eq. 3. Calculations using the latter model resulted in a reasonably good fit with the measurements described above, only by adjusting the model parameter $C_0$, as depicted in Figure 4 (model parameters $C_1$ to $C_3$ are applied according to [14]). Actually, the original model of Sitkei did not include the correction factor $C_0$. The latter has been introduced by ABB in order to match experimental results with different fuel properties.

The calibration of the described model with the data mentioned above resulted in a reasonable agreement between the calculation and measurement. Evaluation of roughly 200 measurement points resulted in a mean value of the difference between calculation and measurement of approximately 0 °CA with a standard deviation of less than 2 °CA (for the nominal rotational speed of $n_{Eng} = 1500$ rpm, 1°CA is equivalent to 0.11 ms).

$$
\tau_{ID} = C_0 \cdot \left[ \tau_0 + \left( \frac{C_1}{p^{0.7}} + \frac{C_2}{p^{1.3}} \right) e^{C_3} \right]
$$

Eq. 3

![Figure 4 - Ignition delay evaluated from measurements compared to calculation](image-url)

The measurements cover only a small range of in-cylinder pressure and temperature (rather low power density and low charging efficiency $\lambda$). The range of special interest at reduced temperature (680-780 K) and increased pressure (75-100 bar) represents an extrapolation beyond the calibration range of the model; therefore, the accuracy of the predicted values cannot be determined. But as the model is a physical and not an empirical one, an extrapolation seems to be feasible. According to different measurement evaluations for high power density and high Miller diesel engine, the same model has proven to also fit well at high pressures and low temperatures with respect to the main diesel injection. The measurement points of the pilot-injected gas engine and the pure diesel engine are shown in Figure 5. The grey markers show the start of injection timing SOI, the blue ones the start of combustion SOC. Lines of constant ignition delay – according to the calculation model – are shown.
as black solid lines. The window of temperature and pressure relevant for the shown investigation results is indicated by the blue shaded area (nominal engine load).

![Figure 5 – Ignition delay map](image)

**Figure 5 – Ignition delay map**

c) Knocking

For a given air excess ratio \( \lambda \), the occurrence of combustion knock depends on temperature and pressure before and during the combustion process. By applying the Miller cycle, the cycle temperature can be substantially decreased, reducing thus the risk of knocking. On the other hand, by reducing the Miller effect by means of a variable inlet valve timing, such as for improved transient engine response, knock tendency is correspondingly increased. For a fair comparison of the potential of different control concepts of air excess ratio \( \lambda \) including the variable valve train, an estimation of knock tendency has to be considered. Therefore, engine knock has been estimated based on a widely used phenomenological knock model, [15], [16]. This type of model allows for predicting the tendency of knocking combustion based on the pressure-temperature-time history in the bulk of the unburned gas of the cylinder charge. However, it does not give evidence about pre-ignition or knocking based on other triggers such as in-cylinder charge inhomogeneity, cylinder-to-cylinder or cycle-to-cycle variability of the charge composition, hot spots in the combustion chamber, deposits or evaporating lube oil. These issues need careful engineering on the part of the engine builder.

According to the model used, spontaneous ignition of the gas takes place within the unburned zone after a certain period of time \( \tau_{\text{Knock}} \). According to Eq. 4, this time is a function of temperature, pressure, air excess ratio and mixture composition (considered in model parameters \( X_1 \) to \( X_3 \)). With a continuously changing state of the cylinder charge, this equation needs to be integrated over time. Spontaneous ignition is assumed to occur when the knock integral shown in Eq. 5 exceeds a certain threshold, usually calibrated with measurement data (the theory foresees a value of unity). In the calculations of the present study a constant value of \( I_{\text{Knock}} \) has been set as limit. This value has been chosen based on engine measurements obtained from operation at high bmep and Miller effect, where knocking combustion could be excluded. Of course, this assumption will only be valid for a certain range of methane number.

Since many factors within the cylinder may contribute to auto ignition or knocking combustion, the model applied is not expected to provide exact information about the occurrence of combustion knock, but serves as an instrument for better of engine operating points with different Miller timings and/or compression ratios \( \varepsilon \).

\[
\tau_{\text{Knock}} = X_1 \cdot P \cdot e^{X_2 / T} \quad \text{Eq. 4}
\]

\[
I_{\text{Knock}} = \int_{t_{\text{end of combustion}}}^{t_{\text{rec}}} \frac{1}{\tau_{\text{Knock}}} \, dt \quad \text{Eq. 5}
\]

d) UHC emission

The emission of unburned hydrocarbons UHC is not only an important loss in terms of engine efficiency. It is also relevant regarding the issue of global warming due to the very high greenhouse effect of methane (about 20 times higher than CO₂).

A very compact overview on the UHC emission is presented in [17]. According to that, UHC emission is especially high for large bore (slow rotating) engines. Because of the increased knock risk at low speed (high residence time), a very lean combustion is required with an adverse effect on UHC emission. Furthermore, the combustion chamber of DF engines cannot be optimized for gas mode and compromises towards the diesel mode may result in increased methane emission.

In the engine simulations presented, a constant value of 4% unburned fuel (\( \eta_u = 0.96 \)) was assumed, irrespective of the load point due to a missing modeling approach. This value will probably overestimate UHC emission at high bmep and vice-versa at engine part load. Furthermore, as the compression ratio increases, the relative volume of the crevices increases, which potentially leaves some fuel unused and increases UHC emissions. In this respect, an OEM might respond in turn by redesigning the top land area or increasing the stroke/bore ratio.
e) VCM, Additional Torque Demand

In the simulations, no additional torque demand arising from the VCM system has been considered. The additional VCM torque is too small for reliable measurements on a test rig. Simulations of the cam supported electro-hydraulic system predict very low values which back up the findings on the test rig. On the other hand, it is proved that VCM allows for much faster closing of the valves than conventional valve train systems. Therefore, a steeper inlet valve closure slope is assumed in the simulation. Commonly, in the case of very early Miller timing, a reduction of the maximum valve lift has to be considered to avoid excessive forces within the valve train. The reduction of the maximum valve lift is much less pronounced due to the faster valve closing ability with VCM. Therefore, lower gas exchange losses could be assumed with VCM. It is expected that the improvement of the gas exchange process will help to cancel out the rather small additional torque demand by the VCM.

f) Gas Supply System

With strong Miller and increased power density elevate the receiver pressure in gas mode up to 7.5 bar. This requires also a correspondingly elevated pressure level in the gas supply. For our investigation, the gas supply pressure was assumed to be provided at the required level and was not considered in system efficiency calculations. However, if the required gas supply pressure is not available on site, additional power consumption due to fuel gas compression has to be taken into account. For instance, if gaseous fuel is supplied at ISO ambient conditions, an increase of the engine gas supply pressure, e.g. from 7 to 10 bar, raises the required compression power from roughly 1.5% to 2% of the engine power.

**Simulation of Engine and Turbocharging System**

Engine cycle simulations have been carried out using the ABB in-house simulation software SISY, [18]. Subsequently, more in-depth simulation work has been done with the newly developed simulation environment ACTUS (also in-house software). The tool offers highly sophisticated features for the simulation of modern turbocharged engines.

As stated in Table 1, a simulation based investigation has been performed that compares an engine setup with single stage turbocharging and EWG air excess ratio control (reference case EWG) to a future engine setup with two-stage turbocharging. The latter case allows for increased power density enabled by a strong Miller effect. The increased bmep allows for a reduction in cylinder bore diameter. As shown in the section about available technology and current development paths, the bore diameter of DF and gas engines is commonly enlarged by a certain amount to maintain the engine power despite the reduced power density at the same level as a diesel engine of the same size category. Since a reduction of the bore leads to a reduced bearing force, the maximum cylinder pressure limit can be increased in the latter case (for the cases of future engine design a further increase of the firing pressure is considered beyond the scaling due to the bore size reduction). Due to the strong Miller timing, even a large step in compression ratio of $\Delta \varepsilon = +4$ can be afforded. Two different gas mode air excess ratio control systems have been compared for the cases with two-stage turbocharging. In the case EWG-2s, the air excess ratio is controlled by an exhaust waste gate bypassing both turbine stages. In the VCM case, the intake charge mixture is controlled just by a variable inlet valve train. In the present study, only a variation of the inlet valve closure timing is considered.

**Table 1 – Simulation cases**

<table>
<thead>
<tr>
<th>Simulation Case</th>
<th>TC system</th>
<th>bmep</th>
<th>bore</th>
<th>$\varepsilon$</th>
<th>$\lambda_\text{V}$ control</th>
</tr>
</thead>
<tbody>
<tr>
<td>EWG</td>
<td>single stage</td>
<td>Ref</td>
<td>Ref</td>
<td>Ref</td>
<td>EWG</td>
</tr>
<tr>
<td>EWG-2s</td>
<td>two-stage</td>
<td>+30%</td>
<td>-6%</td>
<td>+4</td>
<td>EWG HP+LP</td>
</tr>
<tr>
<td>VCM</td>
<td>two-stage</td>
<td>+30%</td>
<td>-6%</td>
<td>+4</td>
<td>VCM</td>
</tr>
</tbody>
</table>

Bypassing of the high pressure turbine would be normally preferred as a control strategy in two-stage turbocharging since it results in improved engine gas exchange. Engine efficiency appears to be somewhat higher with HP bypass solutions. Anyhow, bypassing the high pressure turbine in the gas mode considerably increases the low pressure compressor ratio in the case of DF application. Consequently, compared to the diesel mode where the HP bypass is preferably closed, the operating lines of gas and diesel modes end up being considerably separated in the compressor map. This leads to comparably low compressor efficiency in diesel mode (near choke line) or low surge margin in gas mode. For this reason, in the present investigation a system turbine bypass (two-stage turbine system exhaust waste gate) has been considered as a better choice for DF operation. In spite of this, the HP turbine bypass strategy can be more beneficial depending on the application and also on the width of the compressor map.
The fully variable inlet valve closure timing in the VCM case provides a high degree of freedom for the engine performance optimization, especially in the knock-limited gas mode. However, the feasible range of design parameters is restricted to certain limits to guarantee engine and turbocharger operation in the applicable range. For example, the knocking integral value must stay below a defined value in the gas mode engine operation, to assure knocking free operation while the ignition delay of the pilot spray must not exceed a certain threshold value in order to guarantee ignition at the desired combustion start. The complete list of limiting boundary conditions is stated below in Table 2.

Table 2 – Limiting conditions

<table>
<thead>
<tr>
<th>Gas Mode Limits</th>
<th>Diesel Mode Limits</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. knocking integral value</td>
<td>Max. turbine inlet temperature</td>
</tr>
<tr>
<td>Max. pilot spray ignition delay</td>
<td>Max. ignition delay</td>
</tr>
<tr>
<td>Air excess ratio constant</td>
<td>Min. air excess ratio</td>
</tr>
<tr>
<td>Max. cylinder pressure</td>
<td>Max. cylinder pressure</td>
</tr>
</tbody>
</table>

**SIMULATION RESULTS**

This section is divided into two parts. In the first, the results of an optimization calculation regarding the nominal engine operating point are shown. Based on this optimization, case studies have been carried out which are presented in the second part of this section.

**Design Parameter Optimization**

Increased compression ratio $\varepsilon$ and Miller effect are expected to increase engine efficiency, whereas the increase of the compression ratio is expected to be especially beneficial for the diesel cycle. Optimization calculations have been carried out at full engine load for several values of bmep considering both gas and diesel mode engine operation in order to arrive at optimum engine design parameters.

The compression ratio $\varepsilon$ is limited by geometrical constraints, depending on the value of stroke to bore ratio chosen by the designer. Furthermore, the efficiency benefits at high values of compression ratio will be restrained by increasing friction and heat losses (unfavorable ratio of surface and volume of the combustion chamber).

**a) Ideal Turbocharging Components Performance**

In the case of idealized turbocharging components, calculations have been carried out assuming constant turbocharger component efficiencies. Compressor power consumption is calculated by means of a constant stage loading coefficient. The component efficiencies are chosen so as to achieve a high level of turbocharging efficiency typical for two-stage turbocharging.

The engine compression ratio $\varepsilon$, the inlet valve closure timing IVC and combustion phasing of diesel and gas modes have been adjusted to achieve maximum engine efficiency while satisfying all the stated boundary conditions. The results in Figure 6 show that the engine performance benefits considerably from an increase in the compression ratio. The plot shows the increase of thermal engine efficiency for several values of bmep as a function of the compression ratio $\varepsilon$. The increase refers to the efficiency achieved at the lowest compression ratio and power density. According to the plot an efficiency increase of more than +2% points is still possible compared with today’s established engine technology.

**Figure 6 – Average of diesel and gas mode engine efficiency gains for different levels of bmep as a function of the compression ratio $\varepsilon$.**

Since both the Diesel and the Otto process are limited by the maximum cylinder pressure and knocking combustion respectively, an increase in compression ratio needs to go hand in hand with an increase in Miller effect. Due to the given constraints, the optimal increase of the compression ratio is reduced towards higher engine power density, Figure 7.
Figure 7 – Optimum compression ratio and achievable engine efficiency rise

Figure 8 and Figure 9 show the maps of Miller effect and combustion phasing for diesel and gas mode respectively. The effects of increasing bmep (from 20 bar up to 30 bars) are shown in the maps for different levels of compression ratio. The increase in power density leads to a counter-clockwise trajectory in the plot. For example, excessive ignition delay of the pilot is the limiting boundary in the gas mode at very low compression ratios. Consequently, an increase in bmep allows for advanced combustion timing. More Miller timing would also prolong ignition delay. At a certain point, the knocking boundary is reached and this has to be countered by an increase in Miller effect. A retarding of SOC becomes necessary if at even higher compression ratios the maximum cylinder pressure is reached. The stated constraints cannot be fulfilled anymore in the regions of extreme values of bmep and compression ratio. For this reason, e.g. for ε = + 8, not all levels of bmep are plotted. The black circles indicate the optimum values as shown in Figure 7. It is shown that maximum average efficiency of diesel and gas modes can be gained by applying Miller timing corresponding to a charging efficiency of roughly \( \lambda_i = 0.5 - 0.55 \) in diesel mode and \( \lambda_i = 0.45 - 0.50 \) in gas mode. However, for a reasonably high engine power density such high levels of Miller effect can only be provided by two-stage turbocharging.

Figure 8 – Diesel mode: charging efficiency vs. start of combustion timing

Figure 9 – Gas mode: charging efficiency vs. start of combustion timing

b) Real TC Components Performance

The same investigation as described in the subsection before has been carried out with real TC component characteristics. The component sizes were adjusted so as to achieve a proper location of the operating point within the respective compressor and turbine maps. The study results do not deviate significantly from the ones achieved with ideal TC components. However, the width of real compressor maps limits the allowable difference of charging efficiency \( \lambda_i \) between the diesel and gas modes. Furthermore, turbocharging efficiency lower than specified in the ideal case was obtained outside the nominal operation range of the chosen TC components. This leads to a much more limited set of solutions.

Based on the optimization results shown, the engine parameters for a more detailed case study have been chosen as indicated in Figure 10. Compared to established DF engines an increase of power density by \( \Delta \text{bmep} = 6 \) bar with a
simultaneous rise in compression ratio by $\Delta \varepsilon = 4$ points has been considered for further study cases.

![Graph](image)

**Figure 10 - Average of diesel and gas mode engine efficiency gains for different levels of bmep as a function of the compression ratio $\varepsilon$**

**Case Studies**

Starting from the results of the design parameter optimization, according to Table 1 (page 8) two case studies have been carried out for,

a) DEP engine operation  
b) FPP engine operation

For FPP engine operation with EWG, the engine control margin at part load becomes too small due to the increased brake mean effective pressure at reduced engine speed. Therefore, in both cases of EWG air excess ratio control, the engine simulation model needs to be equipped with an inlet valve shift mechanism enabling at least two fixed closure timings.

a) DEP Engine Operation

Figure 11 and Figure 12 show the reduction of fuel consumption achieved in the cases EWG-2s and VCM compared to the reference case EWG in gas, as well as in diesel mode as function of engine load. As expected from the optimization results in the previous subsection, the potential gain of engine efficiency with the recommended engine design parameters combined with increased Miller effect enabled by two-stage turbocharging is very high. In both operating modes a reduction of fuel consumption in the range of 10 to 15 g/kWh is predicted by simulation at full engine load. While in diesel mode the VCM case shows better performance towards full load, in gas mode the EWG-2s displays a more pronounced fuel reduction potential at engine part load.

![Graph](image)

**Figure 11 – Diesel mode fuel saving**

![Graph](image)

**Figure 12 – Gas mode fuel saving**

The disadvantage of the VCM case compared to the EWG-2s is due to the air excess ratio control strategy based on the fully variable Miller timing. The required cylinder throttling towards part load by further advancing of the inlet valve closure event leads to a very high reduction of the in-cylinder temperature level, in addition to the reduced part load in-cylinder pressure. As a result, the ignition of the pilot diesel spray is impeded. The retarded combustion thus produced leads to the depicted difference in fuel consumptions.

Among other valve control strategies, the issue of excessive Miller timing for DEP part load operation could be mitigated by the application of skip firing. [19]. By cutting out a certain number of cylinders, the mean effective pressure of the remaining firing cylinders would be increased. In order to maintain the desired air excess ratio $\lambda_V$, the cylinder filling has to be increased, i.e. in the case of VCM the Miller effect has to be reduced by retarding the closure of the inlet valves. As a result, both in-cylinder temperature and pressure are increased and pilot spray ignition is enhanced. The combustion can be advanced until reaching the knock limit criteria. The corresponding expected
benefit (simplified simulation assumptions) in terms of engine fuel consumption is shown in Figure 13. The dotted line marks the envelope of the fuel consumption achieved by skipping up to 4 out of 10 cylinders. The additional fuel saving at part load is very high and would clearly outplay the EWG-2s case in gas mode. Without variable valve timing the potential of skip firing is much more limited for EWG control. In both cases, the positive impact of skip firing on UHC emission – due to increased bmep of the fired cylinders – has not been considered and constant fuel conversion efficiency has been assumed.

b) FPP Engine Operation

For FPP engine operation, the cases EWG and VCM have been simulated with the same design parameters as in the DEP section shown above and consequently exhibit the same performance parameters at nominal load. However, the reference case performance is different because for an applicable low load engine behavior, a part load optimized turbine specification and an inlet valve closure timing shift below the 50% load point is applied.

Figure 14 and Figure 15 show the specific fuel saving relative to the reference simulation results as a function of engine load. It is evident that the simulations predict a substantial reduction of fuel consumption in the whole engine load range. Whereas for DEP operation with VCM (without considering skip firing) fuel consumption is higher than with EWG control because of the retarded combustion, the VCM case benefits from the increased part load mean effective pressure of FPP operation. Consequently, the ignition of the pilot spray is facilitated and consequently combustion phasing is limited by knocking.

c) Loss Analysis

Figure 16 is presented for a better understanding of the significant increase in the engine efficiency. It depicts the relevant loss mechanisms relative to the reference simulation in the case of FPP at full load. As can be seen from the plot, changes in the closed cycle process contributes more than 3% points in the diesel mode and more than 2% points in the gas mode to an increase in engine efficiency. The increase of closed cycle efficiency originates from the increase in the compression ratio, the elevated level of maximum cylinder pressure and the reduction of the process temperature (Miller effect, $\Delta\eta_{\text{Miller,cycle}} = 0.5$ to 1% points). Furthermore, gas exchange efficiency is improved by the substantial increase of turbocharging efficiency and optimized valve timing ($\Delta\eta_{\text{GE}} = 0.5$ to 1% points). While the mean effective friction pressure in the diesel mode is assumed to be increasing with bmep, the fmep in the gas mode is kept constant independent of engine bmep. Consequently, the cases EWG-2s and VCM for gas mode at elevated bmep perform at an increased mechanical efficiency compared to the reference gas case.
With only slight differences, the statements above are also valid for the DEP full load case.

**Figure 16 – Loss analysis compared to the reference case**

**CHALLENGES**

The above proposed concept for DF engines utilizing strong Miller, high compression ratio, two-stage turbocharging and variable valve timing discloses new potential for efficiency and power density improvement of gas and DF engines. However, engine operation parameter values are outside of known ranges. It lies in the nature of such extrapolations that certain challenges might arise.

**Homogeneous Mixture**

A homogeneous mixture of fuel gas and air is crucial in order to achieve low NOx emission as well as reducing the risk of knocking combustion. Therefore in port injected engines the admission of the fuel gas becomes a challenging task, especially at increased Miller effect with very early closure of the inlet valves. Continuous admission of fuel gas is supposed to achieve best mixing results. On the other hand, it is constrained due to the issue of UHC emission. Too early gas admission leads to increased accumulation of rich fuel gas in cylinder crevices which is a main source of unburned hydrocarbons. Too late admission however increases the amount of fuel gas in the inlet port after closure of the inlet valves which is then possibly scavenged into the engine exhaust in the subsequent intake cycle.

**UHC Emission**

Research into the calculation of UHC emission in gas engines is still ongoing. Even a reasonable estimation is difficult for changing engine design parameters such as the compression ratio and Miller effect. Therefore, the presented results only consider a constant percentage of unburned fuel. Of course a more sophisticated model regarding UHC emission would help to produce more reliable simulation results of engine efficiency.

**Diesel Pilot Spray Ignition**

In the gas mode, relatively strong Miller cycle was utilized in the simulations to prevent engine knock. However, while creating conditions that mitigate the tendency to knocking, the quality of ignition of the pilot spray suffers. Thus, a balance has to be found between safe ignition of the pilot spray and sufficiently high margin towards knocking combustion. Also, in the light of the latest research results, safeguarding reliable ignition at low temperatures is a key factor. While it was possible to initiate combustion with pilot spray in a single stroke combustion chamber even at a compression temperature of 720 K, a single-cylinder research engine could not be started at the same ignition conditions [11].

**Ignition with a Scavenged Prechamber**

In the case of pure gas engines, a scavenged pre-chamber is typically used for ignition of the charge. It is assumed that such a system will be less susceptible to cycle-to-cycle variations at conditions with strong Miller than a pilot spray ignition system.

**Lubrication Oil Ignition**

For Otto cycle engines with pilot spray ignition abnormal combustion behavior due to ignition of lubrication oil has been described [19]. The feasible window of pressure and temperature where the combustion can still be deterministically initiated by the ignition of the pilot spray becomes narrower towards increasing engine power density.

**Combustion Chamber Design**

The combustion chamber design of DF engines is a compromise between the specific needs of diesel and homogeneous lean gas combustion. A further increase towards higher compression ratios adds further complexity to optimum combustion chamber design. A corresponding increase of the bore/stroke ratio could be a viable alternative.

**ABB CONTRIBUTION**

**Power2: ABB Two-Stage Turbocharging Solutions**

The present investigation showed that increased power density of DF and gas engines, even at increased compression ratios and thus engine
efficiencies, can be achieved by consequent application of the Miller cycle. Together with the high air excess ratio the required charge air pressure exceeds the range achievable with single stage turbocharging and as a consequence, two-stage turbocharging has to be introduced. At ABB, two-stage turbocharging systems have been developed in recent years and a first generation is being successfully operated. With the products at hand, very high pressure ratios as well as very attractive turbocharging efficiencies in the order of 75% can be realized. While the first family of two-stage turbocharging systems, Power2®, has been successfully introduced in the market, a second generation is under development in order to meet the future needs of ABB customers, [21].

Valve Control Management

High Miller effect diesel engines suffer from deteriorated combustion quality at engine part load. Both gas and diesel engines experience poor engine load acceptance and engine start becomes more difficult. Variable valve train is known to improve these problems opening the possibility to further optimize engine performance, [22], [23]

ABB is developing a cam supported electro-hydraulic valve train system in cooperation with Schaeffler Technologies GmbH & Co. KG, a well-known specialist for automotive engine components. It is based on Schaeffler’s UniAir/MultiAir® technology and meets the requirements of four-stroke engines with power outputs above 400 kW, [24], [25]. The resulting product, VCM (Valve Control Management), enables variations in the inlet and/or exhaust valves’ timing and lift by means of electro-hydraulic valve actuation. The prototype shown in Figure 17 has been successfully tested on a fired test engine, [23].

During testing, the power consumption of the electro-hydraulic unit has been shown to be very low. As a result the benefit of running the engine in thermodynamically more favorable conditions is not reduced.

While the focus was set on functionality and reliability for the prototype module testing without aiming at system cost and compactness, case studies of VCM modules for specific customer needs have been worked out. Such a study result considering a variable inlet valve actuation is shown in Figure 18.

Figure 17: VCM prototype module, with individual variability of exhaust and intake valves

Figure 18 – Generic case study of a VCM module for variable intake valve actuation

CONCLUSIONS

Based on an extensive simulation study, it was shown in the paper that efficiency and power density of pure gas and DF engines can be substantially improved by introducing two-stage turbocharging and variable valve timing. Increased power density would provide for reducing the enlarged bore diameter of today’s gas and DF engines to the level of diesel engines of the same frame size.

The following technology building blocks are needed to realize the above proposed concept:

- Highly efficient two-stage turbocharging in order to provide a suitable boost pressure required by applying strong Miller cycle. With Power2®, ABB has a solution which fits the requirements perfectly
- Variable inlet valve closing for air/fuel ratio control and switching between fuels optimized diesel and gas operation mode
(e.g., ABB’s variable valve train solution VCM®).

- Pilot fuel injection system capable of flexible SOI setting
- Optimized but fixed compression ratio
- Mechanical structure for a gas engine with bmep = 26 bar and cylinder firing pressure up to 220 bar

Engine control with an exhaust waste gate solution could be shown with improvements comparable to those achieved with VCM. However, solutions with VCM show several advantages:

- Increased transient response, in particular in the diesel mode, [22]
- Higher margin with regard to knock limit
- Facilitated engine start in the gas mode
- Fuel efficient control of air excess ratio without control device being exposed to hot exhaust gases (HFO operation)

With the concept introduced in this paper, pure gas and especially DF engines close the gap between today’s established diesel engines and marine DF or gas engines in terms of efficiency and power density.

Today, pure gas and DF engines already operate under the NOx and SOx limits in ECAs. Therefore, they represent an interesting alternative to diesel engines equipped with EGR or exhaust gas after-treatment as IMO Tier 3 abatement technologies.

With Power2 and VCM, ABB offers the key technology for the implementation of the concept outlined in this paper.

It is recommended to work in two directions to further assess the technical potential of the concept introduced:

- Investigate experimentally and, if necessary improve, ignition and combustion evolution in pilot-ignited Otto cycles when applying strong Miller valve timing
- Implement and test the control of a two-stage turbocharged gas engine under FPP operation

<table>
<thead>
<tr>
<th>NOMENCLATURE</th>
<th>Definition</th>
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<tbody>
<tr>
<td>BDC</td>
<td>Bottom Dead Center</td>
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<tr>
<td>CA</td>
<td>Crank Angle</td>
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<tr>
<td>EWG</td>
<td>Exhaust Waste Gate</td>
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<tr>
<td>EWG-2s</td>
<td>Exhaust Waste Gate, bypassing both turbine stages</td>
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<td>FPP</td>
<td>Fixed Pitch Propeller</td>
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<tr>
<td>Hu</td>
<td>Lower heating value of the fuel</td>
</tr>
<tr>
<td>Lmin</td>
<td>Stoichiometric air/fuel ratio</td>
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<td>SOC</td>
<td>Start of combustion</td>
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<tr>
<td>SOI</td>
<td>Start of injection</td>
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<td>T_ac</td>
<td>Compression start pressure</td>
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<td>TDC</td>
<td>Top Dead Center</td>
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<tr>
<td>UHC</td>
<td>Unburned Hydro Carbons</td>
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<td>VCM</td>
<td>Valve Control Management, variable valve train system by ABB</td>
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<tr>
<td>VTG</td>
<td>Variable Turbine Geometry</td>
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<tr>
<td>V, VD</td>
<td>Volume, cylinder displacement</td>
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<tr>
<td>bmep</td>
<td>Brake mean effective pressure</td>
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<tr>
<td>imep</td>
<td>Indicated mean effective pressure</td>
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<tr>
<td>p_cyl</td>
<td>In-cylinder pressure</td>
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<tr>
<td>pc</td>
<td>Compression end pressure at TDC</td>
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<td>pRec</td>
<td>Air receiver pressure</td>
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<tr>
<td>pfiring</td>
<td>Maximum cylinder pressure</td>
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<tr>
<td>Δp_fire</td>
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<tr>
<td>ε</td>
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<td>Delivery ratio</td>
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<td>Total air excess ratio</td>
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<td>τID</td>
<td>Ignition delay</td>
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ACKNOWLEDGEMENTS

The authors would like to thank Ms. Inna Tishchenko for her valuable work regarding the implementation and testing of optimization algorithms for DF engine cycle simulations.

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