Approaches to Meeting Fluctuating Natural Gas Quality in Large Bore Engine Applications

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Abstract. State-of-the-art large bore gas engines for power generation are traditionally designed for long-term operation at high load. They run at high power density and high engine efficiency, which is guaranteed as long as the engine boundary conditions, e.g., fuel gas quality, stay within certain limits. A growing share of gases from alternative sources such as biomass, hydrogen from power-to-gas technologies or LNG will be found in the pipeline grids of the future. This is further promoted by the harmonization process for gas qualities in Europe. The European standard EN 16726 as well as EASEE Gas Directive set rather wide limits for relative density, calorific value and methane number. For both engine manufacturers and operators, new challenges arise that have to be met by improving operating strategies.

Operating a stationary large engine with gaseous fuels – spark or direct ignited (diesel pilot) – leads to the problem of auto-ignition. The detection of knocking combustion is an important measure for the active prevention of engine damage. Different regimes during the combustion process result in a diversification of the cylinder pressure signal as well as the acceleration sensor signal. This requires different strategies for the distinct identification of knocking combustion. The methods shown are mainly based on the use of the cylinder pressure signal and focus on knock detection, detection of knock onset and evaluation of knock intensity.

If knocking combustion is detected, the engine control system has to react appropriately in order to ensure stable and safe engine operation. Here one can distinguish between approaches that adjust soft parameters of state-of-the-art engines (ignition timing or mixture quality), and approaches that need additional hardware and their functionality to be realized (variable intake valve timing or variable compression ratio). It is shown how these approaches to knock control impact engine performance.

1 Introduction

1.1 Volatility of Gas Quality

Current large bore gas engines are highly efficient, very robust and nearly maintenance-free engines for power generation. In addition to these attributes, such combustion engines can be used to burn a wide range of gases of various origins. Over

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the last few years it has become very common for gas engine manufacturers to offer many different configurations derived from the same engine platform that run on a specific gaseous fuel-composition.

Figure 1 indicates the most important sources of gases and how they are converted into electricity and fed into the grid.

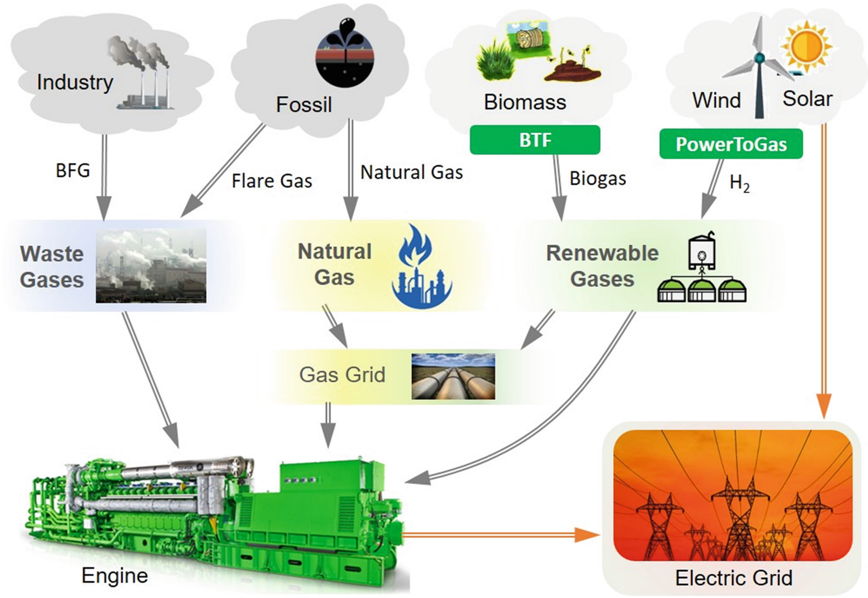

Fig. 1. Power generation with gases from various sources [1]

In general, gas engines have a comparatively low environmental impact, which has been steadily reduced in recent years due to progress in development. If renewable gases are burned, this benefit is even enhanced. As shown above, such gases are created from biomass via gasification processes, which are often referred to by the synonym biomass to fuel (BTF), or via power to gas, where hydrogen is produced from excess electric power from wind parks or solar plants. Renewable gases can either be directly fed into the engine or added to the conventional gas grid. The latter presents a great challenge for stable operation of a gas engine because this is the main driver of volatility in pipeline gas quality. The properties of the gas in the grid change depending on the amount of renewable gases fed into the pipeline system. The great unknown is the exact composition in the grid, hence which quality can be expected at each engine's inlet pipe.

Besides renewable gases, the most pervasive approach is natural gas from fossil resources. Special adaptations enable gas driven large bore engines to use flare gases and waste gases, e.g., blast furnace gas (BFG), from industrial processes.

For sustainable energy production, it has become more and more important to use gases from alternative, renewable resources. To this end, the European Union and the

European Committee for Standardization (CEN) in particular created the standard EN 16726 “Gas infrastructure – Quality of gas – Group H” in 2015. This standard defines requirements for gas quality with the goal of allowing free gas flow in the pipeline grid within the CEN member states [2, 3]. The restrictions from EN 16726 are shown in Fig. 2.

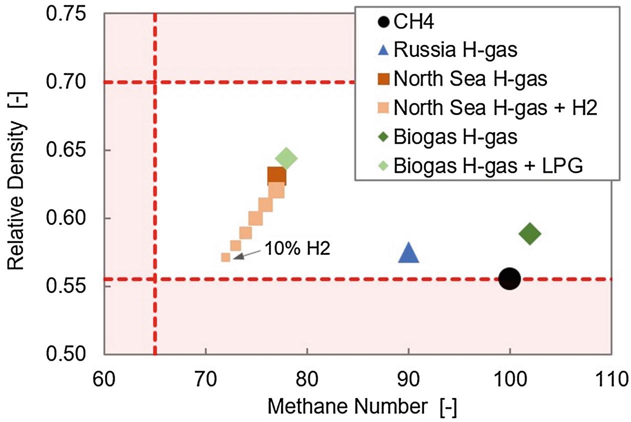

Fig. 2. Real pipeline gases and mixtures of them with limits according to EN 16726

In EN 16726 the member states were able to agree on limitations in methane number and relative density. The methane number has a lower limit of just 65. To calculate the relative density, the gas density is divided by the air density at reference conditions (0 °C and 101 kPa). The resulting value’s lower limit is 0.555 and the upper limit is 0.7. As illustrated in the figure, the examples stay within the limits indicated by the dashed red line.

It can be seen that an admixture of hydrogen (e.g., from Power to Gas) is not possible with each starting gas composition. Pure methane, biogas as well as H-gas from Russia are already close to the minimum relative density. Hydrogen can only be admixed to North Sea H-gas due to the latter’s high relative density, providing even up to 10% of volumetric share without exceeding the lower limit.

1.2 Consequences for Engine Operation

The most important gas property for combustion in a common gas engine is the methane number. It can be seen as the key parameter to quantifying the knocking tendency of a specific gas composition. A high methane number indicates good resistance of the gaseous fuels to knocking combustion and vice versa. Pure methane, for example, has a methane number of 100, as illustrated by the black dot in Fig. 2.

In addition to changes in environmental boundary conditions such as temperature, ambient pressure and humidity, another great challenge that has to be taken into account is volatile gas composition. Figure 3 provides an example of the variations in methane number in the natural gas grid in southeastern Austria observed over the last two years.

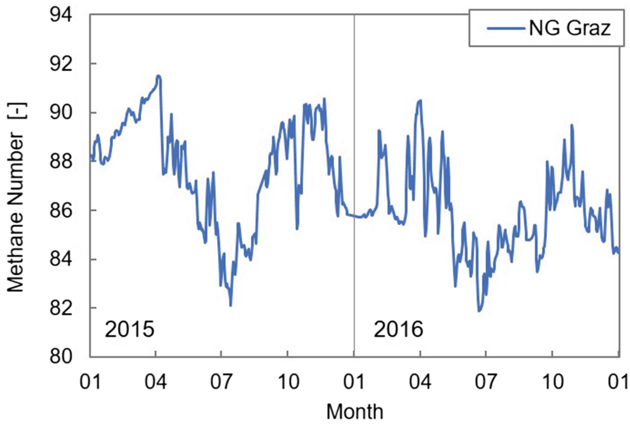

Fig. 3. Fluctuations in methane number of natural gas in Graz, Austria

The seasonal influence on gas quality is obvious; especially during spring and autumn, the methane amount in the gas grid seems to be quite high, whereas in summer, the methane number reaches its annual minimum. Fluctuations in this quantity have to be considered in order to guarantee stable and satisfactory engine performance during the entire year.

Diverse measures can be taken to respond to comparatively low methane numbers; what they have in common is that they should reduce the tendency for knocking combustion. Possible measures include [4]:

- Lowering air or mixture temperatures in the intake manifold
- Employing exhaust gas recirculation
- Cooling the cylinder charge via Miller timing
- Avoiding deposits to keep wall heat transfer constant
- Increasing charge motion to reduce time for pre-reactions
- Raising the engine speed level
- Increasing excess air ratio

Many of these measures attempt to reduce the temperature of the cylinder charge, which also has a positive effect on nitrogen oxide production. For a detailed description of selected measures, see Sect. 3.

2 Methods for Detecting Knocking Combustion

Knock detection aims to separate knocking combustion cycles from non-knocking combustion cycles using calibrated algorithms as well as to determine the precise knock onset.

If severe, knock can result in fatal engine damage within a couple of combustion cycles. [5] Thus to prevent engine damage and ensure knock-free engine operation at any time during commercial engine operation, a knock detection algorithm is mandatory. The knock detection algorithm provides the basis for counteractive measures such as control of engine parameters and/or advanced hardware functionalities. Researchers also use knock detection algorithms to determine the knock limit under certain boundary conditions.

Following, the measured cylinder pressure trace is the basic physical parameter for knock detection because it is superimposed by high frequency pressure oscillations when knock occurs.

Since combustion regimes for spark ignited combustion and diesel pilot ignited combustion differ, they require different knock detection algorithms.

2.1 Spark Ignited Combustion (Gas Engine Concept)

With spark ignited combustion using gaseous fuels, knock detection based on the in-cylinder pressure trace involves identifying the sudden transition to knocking combustion, whereas the in-cylinder pressure trace is superimposed by high frequency pressure oscillations as shown in Fig. 4.

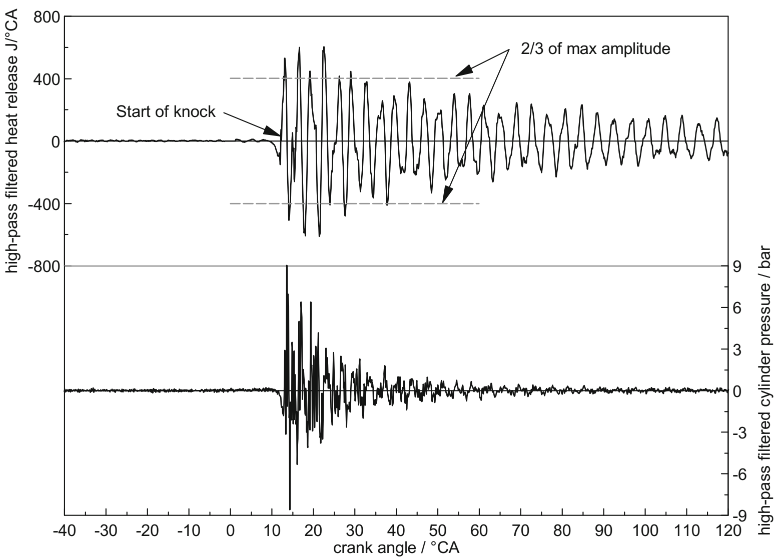

Fig. 4. High-pass filtered heat release and cylinder pressure [9]

The knock detection algorithm is basically a combination of several criteria. To increase sensitivity by differentiation, the apparent heat release rate is calculated from the pressure curve using the first law of thermodynamics. The high frequency part is extracted using a high-pass filter, after which a critical threshold value is determined that is two thirds of the maximum amplitude. Second, the amplitude that first exceeds this threshold value is determined. The zero value before this amplitude is the start of knocking, see Fig. 4.

To ascertain that this is in fact the start of knock, the high frequency pressure oscillations in the area of several degrees crank angle before and after the start of knock are integrated and set in relation to each other. This ratio must also exceed a threshold value. In addition to the start of knock, knock intensity is another parameter used to characterize knock behavior and can be determined from the high-frequency part of the cylinder pressure curve [6–9].

2.2 Diesel Pilot Ignition (Dual Fuel Concept)

The challenge in setting up a knock detection system with diesel pilot ignited combustion systems is the appearance of ringing combustion, or simply ringing. Ringing is a well-known phenomenon induced by diesel pilot combustion. Both ringing and knocking are the result of a sudden conversion of large parts of fresh cylinder charge that leads to pressure oscillations.

Three parameters influence the severity of ringing:

- Rail pressure (p_Rail)
- Start of injection (SOI)
- Amount of diesel injected (φ_Diesel)

Increasing the rail pressure, advancing the SOI and increasing the amount of injected diesel increase the tendency for ringing to occur.

Investigations have also shown that ringing intensity greatly fluctuates in consecutive in-cylinder pressure traces. An example is given in Fig. 5. Cycle 1 exhibits strong ringing whereas cycle 2 has light ringing and cycle 3 has a ringing intensity somewhere in between.

Under extreme conditions, ringing and knocking can naturally occur within the same combustion cycles. As seen in Fig. 5, ringing combustion is present throughout the entire combustion cycle, whereas knocking combustion sets in after TDC – glow ignition is not taken into account.

Therefore, knock detection with diesel/natural gas dual fuel combustion must be capable of distinguishing between ringing and knocking combustion. This can be achieved by calculating the energy of the superimposed oscillations in a time range of the combustion cycle when only ringing combustion occurs and in a time slot during which knocking combustion occurs.

The superimposed oscillations are extracted from the in-cylinder pressure trace using a bandpass filter (p_{bp}). Based on this data, the signal energy (SE) of the signal in the particular time slots during the combustion cycle are calculated according to:

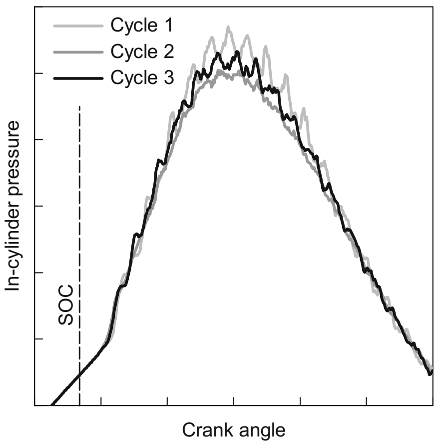

Fig. 5. Consecutive in-cylinder pressure traces with various ringing intensities [9]

$$SE = \int_{\varphi_1}^{\varphi_2} p_{bp}^2 \, d\varphi$$

Both phases are compared to each other and an index is calculated.

To increase the sensitivity of this method, the peak pressure of the bandpass filtered in-cylinder pressure trace is also evaluated in both of these combustion phases according to:

$$p_{max,} = max(p_{bp}|_{\varphi_1}^{\varphi_2})$$

In the same manner as with the signal energy, a second index is calculated with the peak pressure. Both indices are multiplied and the product is a collective knock indicator – the so-called dual fuel knock indicator (DFKI), which is then compared to a threshold value.

In fact, this threshold already shows good agreement, but it can be improved by introducing a second threshold. According to the mathematical definition of the previously described knock detection algorithm, dividing by a figure close to zero may lead to an exceeding of the first threshold value. The remedy for this mathematical misinterpretation was the determination of a minimum amount of energy of the phase where knock is expected – which is the second threshold value. Investigations of a broad variety of non-knocking and knocking combustion cycles in diesel/natural dual fuel operation have shown that this approach is independent of various operation conditions and the definition of just two thresholds is required.

Figure 6 shows the results of a previously conducted subjective separation of the measurement matrix into non-knocking and knocking combustion cycles (gray

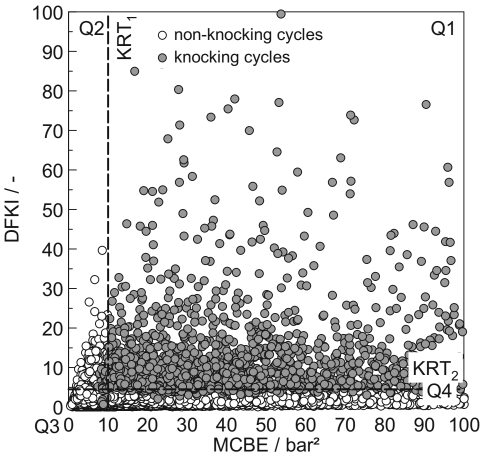

Fig. 6. Overall results of knock detection with diesel/natural gas dual fuel operation

dots = knocking combustion cycles, white dots = non-knocking combustion cycles). It also provides the results of the knock detection algorithm based on the x-axis and y-axis values; the dashed lines indicate the thresholds (knock related threshold, or KRT). These thresholds result from the differentiation between the knocking and non-knocking combustion cycles. The two thresholds divide the measurement matrix into four quadrants (Q1 to Q4).

For a more detailed description of knock detection with diesel/natural gas dual fuel combustion see [9].

Figure 7 provides an example of a series of consecutively measured measurement points where the EAR was varied in equal steps. Each EAR step is a separate

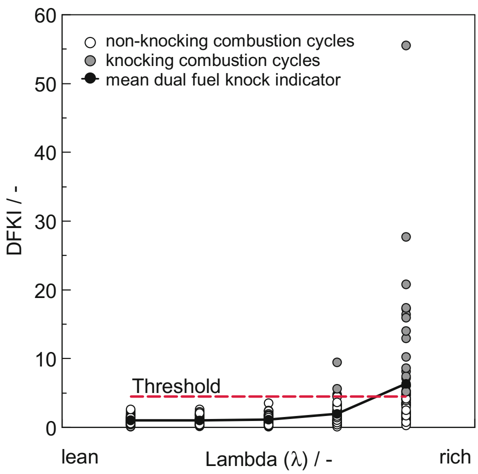

Fig. 7. EAR variation

measurement point that consists of 60 consecutively measured combustion cycles and is indicated by the dots.

This example was chosen because all measurement points exhibited decent ringing whereby knock detection is exclusively dependent on the first threshold value (DFKI). A gray dot once again indicates a knocking combustion cycle whereas a white dot indicates a non-knocking combustion cycle. When the charge is enriched, knocking combustion occurs. The first three measurement points from left to right contain no knocking combustion cycles. The fourth measurement point already contains a few knocking combustion cycles and is considered to indicate light knocking. Enriching the cylinder charge even further increases both knock frequency and knock intensity. With more than 30% knock frequency (31 out of 60 consecutively measured combustion cycles) and a knock intensity of 8 bar, this measurement point is considered to exhibit severe knocking.

3 Approaches to Preventing Knocking in Large Bore Engines

Four different approaches were chosen to react to knocking that occurs during engine operation and ensure stable and safe combustion. These approaches are described in the following chapter and all follow the same principle of cooling down the temperature of the in-cylinder charge, which directly results in a lower temperature in the unburned zone (two thermodynamic zones: burned zone and unburned zone). This can be understood as the common goal of all strategies to prevent knocking.

The investigated approaches can be divided into two categories. The first includes strategies applicable to nearly all common large bore engines and are often already in use. The second category consists of strategies that require additional devices or more advanced hardware, which requires changes or even adaptations in engine design.

All four approaches were investigated on a state-of-the-art multi-cylinder engine (MCE) application based on simulation in GT-Power. The simulations were conducted on a spark ignited lean burn gas engine with a scavenged prechamber. Such engines are equipped with a turbocharger unit followed by a mixture cooler. In our case, we chose a two-stage turbocharged configuration so that the model engine was able to reach an overall power level of more than 4 MW. The engine load is controlled by a standard throttle valve as well as an additional valve that opens a compressor bypass path.

The engine's behavior with reference to its knock tendency was derived from an extensive database of measurement results from a single cylinder research engine at the LEC. The single cylinder was operated under real-life conditions and ran on the same configuration later used for the MCE simulations. From these investigations, a knock limit was determined for this specific engine configuration. Providing examples of six different measurement points, Fig. 8 indicates how the knock criterion defines whether it is a knocking or a non-knocking operation point. As can be seen in the figure, there is good agreement of the points with the linear approach to the knock limit over the entire range of methane numbers.

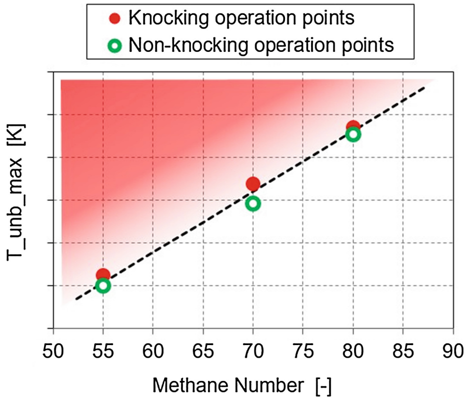

Fig. 8. Definition of knock limit (= dashed line)

Next, the correlation for the maximum temperature of the unburned zone (T_unb_max) was transferred to the MCE model. This made it possible to quantify the knock margin by taking the difference from T_unb_max (from a particular operating point) to the knock limit in Kelvin, subsequently referred to as ΔT_knock. A constant value of 10 K was selected in order to guarantee a standard knock margin for all the anti-knock strategies described in the next chapters.

3.1 Control of Engine Parameters

Adjustment of ignition timing

It was first considered how lowering the methane number affects the combustion process itself without any other intervention, cf. the gray line in Fig. 9. Since an increased number of higher hydrocarbons speeds up combustion, its phasing advances with lower MNs. Consequently, the maximum temperature of the unburned zone rises and ΔT_knock is reduced. While MN goes down, knocking combustion will start; this can be seen at MN 69, where the gray line meets the knock limit. It should be noted that the two operating points below the knock limit (against the red background) cannot actually be reached and only represent theoretical behavior.

In practice, every engine retards combustion itself by adjusting the start of ignition to withstand upcoming knocking. This is accomplished very fast and quite effectively by means of engine control. The blue curve in Fig. 9 illustrates this common strategy. Starting from MN 85 on the right, the knock margin decreases until it reaches the chosen distance of 10 K. To keep this constant, it is necessary to adjust ignition to late timings. For example, when the MN is 70, ignition has to be retarded by 2 °CA. As a result, it is not surprising that engine efficiency becomes worse, although it must be remembered that the gray line is theoretical and cannot be reached under any circumstances. Nevertheless, the gray line will serve as the *baseline* for assessing the engine efficiency potential, called ΔETA_eff, of all further approaches.

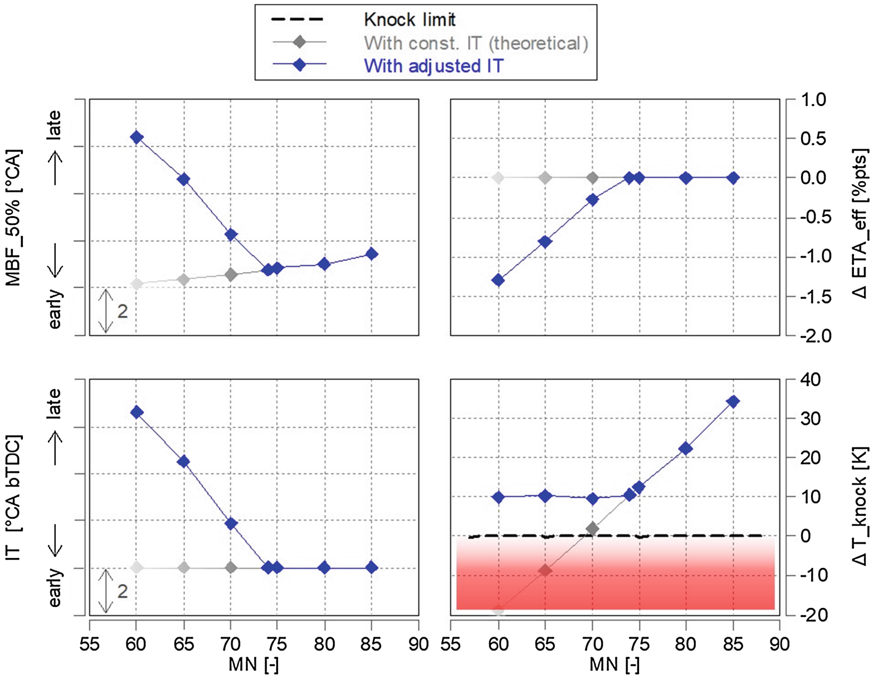

Fig. 9. Reacting to MN change by adjusting ignition timing (strategy 1)

Adjustment of mixture temperature

Another approach that is also standard practice is to cool down the charge's temperature directly by lowering the manifold air temperature (MAT). This measure is not as fast as adjustment of ignition timing because cooling down the entire air flow of a MCE with the attached mixture cooler takes time. Especially with stationary large bore engines, the size of the mixture cooler unit as a quantity for capacity is not restricted by the packaging; thus this strategy can be used on these types of engines.

Figure 10 shows the simulation results using this strategy. Engine control begins to intervene at a MN of 74. As methane number continues to decrease, the charge temperature goes down along the cyan-colored line. To ensure knock-free combustion at MN 67, the MAT has to be decreased by roughly 15 K, which is quite a large amount considering that MAT usually has a value in the range of 45 to 60 °C.

By applying *strategy 2*, combustion phasing remains fairly unaffected, hence the efficiency continues to become better than the *baseline* with lower MNs. Yet in very humid conditions, this particular strategy could lead to condensation problems along the intake air-path.

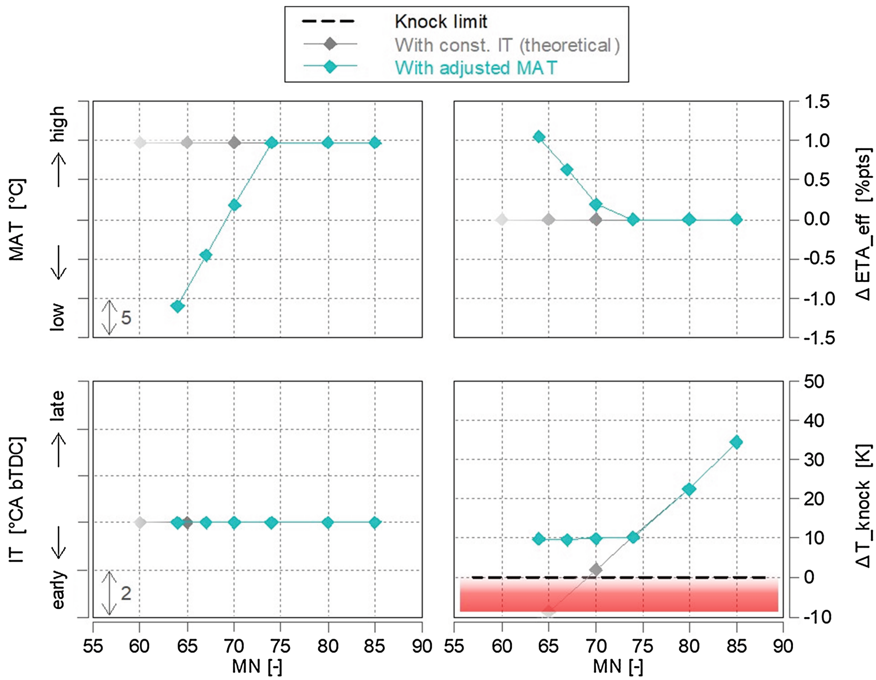

Fig. 10. Reacting to MN change by adjusting manifold air temperature (strategy 2)

3.2 Enhanced Hardware Functionalities

Variable valve timing

With this approach, the engine must be capable of controlling the timing of the valves. In the specific case implemented into the simulation model described in this paper, variability only refers to the intake valves. To keep it simple, only the closing of the intake valve is changed, i.e. the valve's opening ramp remains the same with all different closing angles (IVC) while the opening duration is varied. Such behavior can be obtained using a variable valvetrain, which has already been described in previous publications [10, 11].

The simulations summarized in Fig. 11 were carried out to demonstrate the impact of variable valve timing. At the starting point of MN 85, the efficiency of the IVC controlled model (green curve) as well as the distance to the knock limit is the same as the *baseline*. The shape of the valve lift curves is different, which is the reason for a retarded IVC in relation to standard valve timing at MN 85 [12]. IVC is controlled so that engine load is kept constant. In the *baseline* case, this is achieved using the compressor bypass valve so that the ignition timing can remain unchanged.

If the MN decreases so that combustion advances, IVC occurs later until MN 76, at which point the maximum duration of intake valve opening is reached. From that point on, the controller starts to advance the IVC; the ignition has to be retarded in parallel to

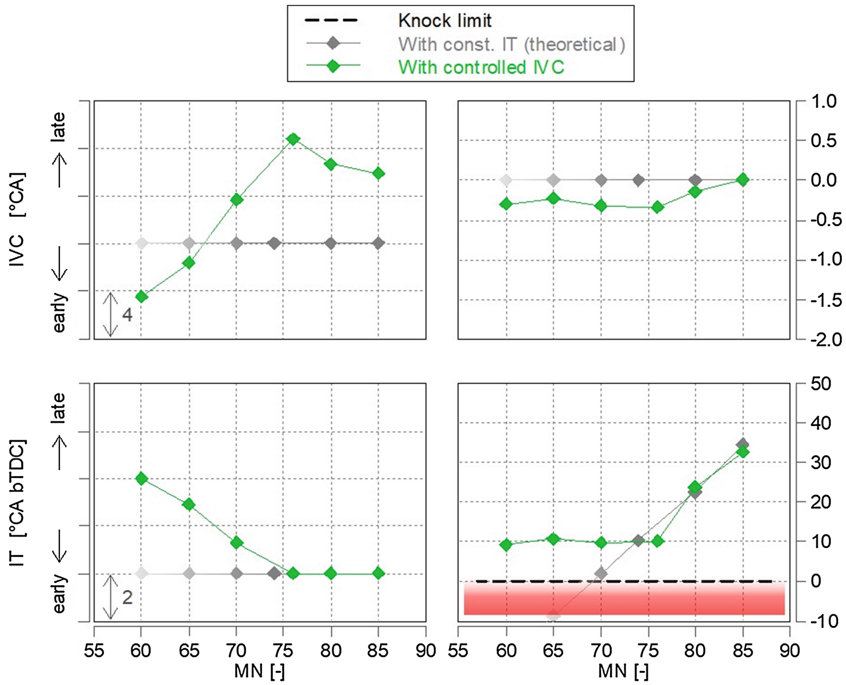

Fig. 11. Reacting to MN change by adjusting intake valve closing angle (strategy 3)

ensure that the engine load remains constant. With the earlier IVC, the temperature of the unburned zone decreases so that ΔT_knock can be kept constant at 10 K. The efficiency stays at nearly constant throughout the entire range below MN 76.

Variable compression

A more complex change in engine design is recommended when variable compression is used. There are several theoretical approaches to how to achieve a variable compression ratio [13] and the first passenger vehicle equipped with this new technology will be launched on the market in 2018 [14].

Our focus was on developing a strategy to prevent knocking, not to increase efficiency. As previously mentioned, we used a MCE simulation model for our investigations and changed the compression ratio (CR) within this model, disregarding how it might be put into practice on a large bore engine.

As explained above for *strategies 1* and *2*, an adjustment must be made only if the MN is lower than 74. In this case, the CR is steadily reduced while ignition timing remains unchanged. This results in worse conditions at ignition timing in terms of pressure and temperature, which in turn leads to a shift in combustion, which lasts longer. This fact is shown in MFB_50% curve in the top right diagram of Fig. 12. Together with the reduced compression ratio the latter is the reason for the drop in effective efficiency, indicated by ΔETA_eff, which steadily rises as the MN decreases.

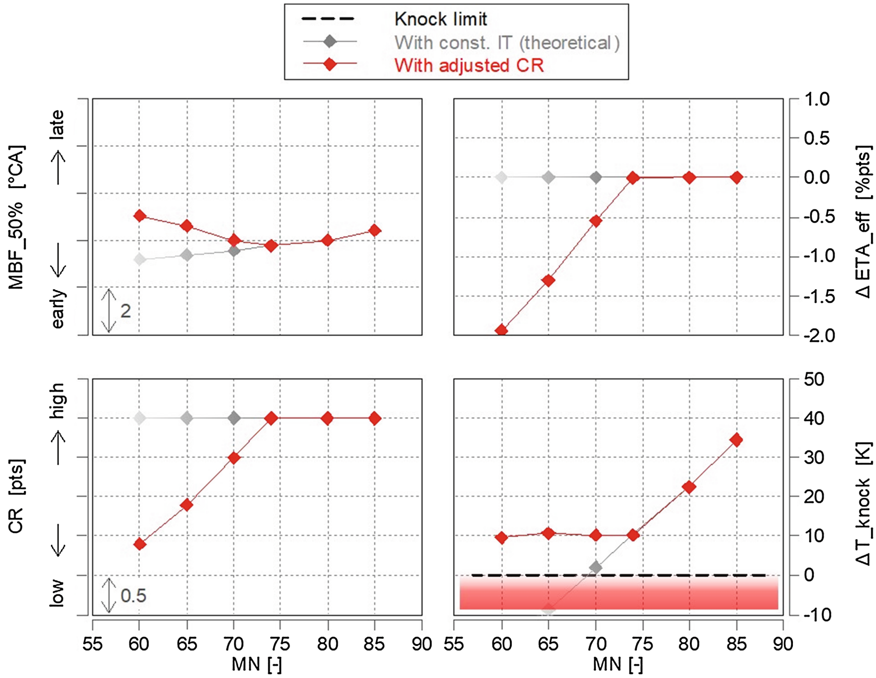

Fig. 12. Reacting to MN change by adjusting compression ratio (strategy 4)

4 Summary

Knock is a combustion anomaly that – in severe cases – can lead to engine failure. Since large bore gas engines have to deal with a broad variety of gas qualities, knock is likely to occur. To prevent this harmful form of combustion, robust knock detection is mandatory. To this end, algorithms are used that detect knocking combustion cycles. It was pointed out that spark ignited combustion and diesel pilot ignited combustion both have different combustion regimes and thus require different strategies for detecting knock.

When knock is detected, countermeasures are initiated. The strategies described in this paper provide an overview of potential approaches to preventing knocking yet make no claim to completeness. It is certain that adjustment of ignition timing (strategy 1) is the most straightforward way, especially in contrast to the enhanced hardware functionalities that already have to be considered during the engine design phase.

Although the focus of the strategies was on preventing knocking, we would like to take a look at the expected efficiencies shown in Fig. 13. Two operating points are compared for each strategy, one at the highest MN of 85 and the other at the lower limit of MN 65 defined in EN 16726 (see Sect. 1.1).

As can be seen in the figure, if the ignition timing is adjusted to operate the engine at enough distance to knocking (ΔT_knock = 10 K), efficiency (ETA_eff) drops by approximately 0.4%pts. Only *strategy 4*, where the compression ratio is reduced by

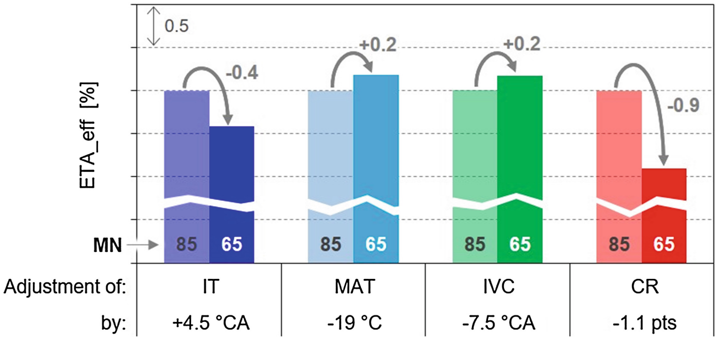

Fig. 13. Direct comparison of the four strategies

almost 1.1 pts, is worse regarding ETA_eff. The two strategies in the middle of Fig. 13 perform the same. But it has to be considered that the mixture temperature must be cooled down by 19 °C, which is quite a large amount, so that this strategy – *strategy 2* – cannot be applied to every engine concept. Consequently, the best efficiency can be achieved using intake valve variability.

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Glossary

bp	Index for a bandpass filtered value
BFG	Blast furnace gas
BTF	Biomass to fuel
CR	Geometric compression ratio
DFKI	Dual fuel knock indicator
EAR	Excess air ratio
ETA_eff	Engine break efficiency
IT	Ignition timing
IVC	Intake valve closing angle
KRT	Knock related threshold
MAT	Manifold air temperature

MBF_50%	Crank angle at which 50% of fuel mass is burned
MN	Methane number
NG	Natural gas
P_RAIL	Rail pressure
SE	Signal energy
SOC	Start of combustion
SOI	Start of injection
TDC	Top dead center
T_knock	Temperature of the unburned zone where knocking occurs
T_unb	Temperature of the unburned zone
φ_Diesel	Energetic amount of diesel

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