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Investigation of Possibilities of λ = 1 Full Load Operation for Gasoline Engines in the Light of Future Emission Regulation

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Abstract

To date, huge amounts of money have been invested in the development of internal combustion engines to reach the current level of technology. High specific power and good thermal efficiency have been achieved, thanks to which, internal combustion engines are now widely used. However, the driving force behind the developments is no longer the high performance, but the compliance with strict emission standards. Future emissions regulation, namely Euro 7, will be challenging for engine and vehicle manufacturers. One possible technical solution may be to use a stoichiometric air-fuel mixture on the entire engine map to meet the requirements of the Euro 7 emission standard. This article analyzes the change in Euro regulations in the light of Euro 7, as well as the theoretical background of the $\lambda = 1$ operation. Several technical possibilities to achieve the stoichiometric ratio, such as e.g. water injection or variable compression ratio are presented.

Keywords

internal combustion engine, Euro 7, emission, stochiometric, air-fuel ratio, full load, three-way catalyst

1 Introduction

In the complete combustion of fuels that consist of only carbon and hydrogen, the components of the exhaust gas will include nitrogen (N_2), carbon dioxide (CO_2), and water vapor (H_2O). However, due to the insufficient time available for the combustion process during a real process and to the heterogeneous distribution of temperature in the combustion chamber, the combustion will be incomplete. A real combustion process also produces pollutants such as carbon monoxide (CO), unburned hydrocarbons (HC), nitrogen oxides (NO_X), and particulate matter. CO_2 is not considered a pollutant because it is not harmful to health, but it is harmful to the environment (Bagány, 2011; Eckert and Rakowski, 2012; Kalmár and Stukovszky, 1998; Vas, 2005). These statements are valid as far as conventional transportation fuels are concerned. The picture is different when it comes to alternative fuels (Török and Zöldy, 2010; Zöldy and Török, 2015).

The relative concentrations of the exhaust gas components are shown in Fig. 1. The exhaust gas contains 71% nitrogen (N_2), 13% water vapor (H_2O), 14% carbon dioxide (CO_2), and 1% other components (noble gases, oxygen and/or hydrogen). The remaining 1% are pollutants, which consist of 0.1% nitrogen oxides (NO_X), 0.2% hydrocarbons (HC), 0.005% particulate matters and 0.7% carbon monoxide (CO). The most important secondary components are carbon dioxide (CO_2), hydrocarbons (HC)

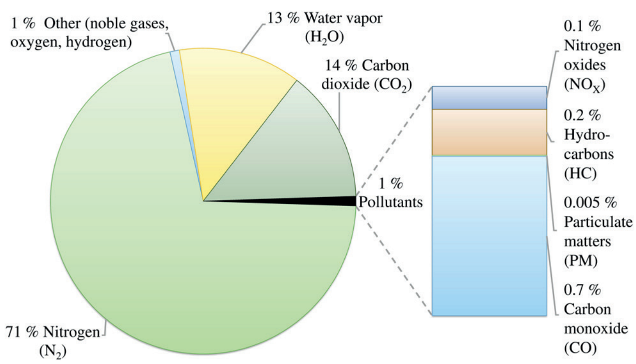

Fig. 1 The concentration of exhaust gas components of a gasoline engine (Reif et al., 2015b)

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and nitrogen oxides (NO_X). For a gasoline engine that runs with a homogeneous mixture, it is sufficient to simply equip the vehicle with a three-way catalytic converter. When the catalytic converter reaches the operating temperature, more than 99 % of the pollutants can be converted into harmless substances. These are carbon dioxide (CO_2), water vapor (H_2O) and nitrogen (N_2). If the gasoline engine is running close to full load with a rich mixture or at partial load with lean operation, the three-way catalytic converter cannot produce a sufficient conversion rate (Reif et al., 2015a; Reif et al., 2015b).

2 Relevant legislation and limits for gasoline engine's emission

There are different emission standards around the world, which are summarized in Fig. 2. Different regulations apply to the USA, the EU and Japan as well (Reif et al., 2015a).

Among the air pollutants emitted by road vehicle engines, the components currently regulated include CO, NO_X, THC, NH_3, Particulate Mass (PM) and Particulate Number (PN). Both the EU (European Union) and the UN-ECE (United Nations Economic Commission for Europe) provide technical specifications to test air pollutants separately for several vehicle types, such as passenger cars, truck engines, non-road engines, and two- or three-wheel vehicles. In the present article, passenger car engines and passenger car testing are investigated. The evolution of emission limits for passenger cars can be observed in Table 1 and in Fig. 3 (Baumgarten et al., 2018; Bielaczyc and Woodburn, 2019; Eckert and Rakowski, 2012; Giakoumis, 2017).

The first emission standards for road passenger cars were created in the 1960s for gasoline engines. The first EU regulation of emissions (Council Directive 70/220/EEC, 1970) was made in 1970, which required testing of gasoline engines' HC and CO emissions and set limit values for these exhaust components. An analogous regulation in the UN was Regulation No. 15 (UN-ECE, 1977). Initially, only HC and CO emissions were regulated, so manufacturers prescribed a lean mixture operation for the passenger car's gasoline engine, resulting in increased NO_X emissions. NO_X emissions only became regulated in 1977. Diesel engine emissions in Europe later became regulated in 1988 (Bagány, 2011; Baumgarten et al., 2018; Bielaczyc and Woodburn, 2019; Eckert and Rakowski, 2012; Giakoumis, 2017; Kalmár and Stukovszky, 1998; Vas, 2005). The "Euro" regulations were established with Euro 1 (Council Directive 91/441/EEC, 1991) in 1992. Manufacturers were able to meet the requirements by using exhaust after-treatment catalysts and unleaded gasoline. However, due to the heterogeneous mixture in diesel engines, particle formation is unavoidable. In compression-ignition engines, the particulate emission issue has been solved by the introduction of Diesel Particulate Filters (DPFs). This is not a problem for spark-ignition engines with intake manifold injection. Today, the issue of particulate emissions has come to the forefront of spark-ignition engine design with direct injection. From Euro 5 onwards, direct injection gasoline engines have already had a limit on the emitted mass of particles, thus the use of Gasoline Particulate Filters (GPFs) has become necessary. For diesel engines, Selective Catalytic Reduction (SCR) with a combined particulate filter keep both PM/PN and NO_X emissions sufficiently low. Fig. 4 shows the historical European and UN emission test regulations of passenger cars, tested components, typical exhaust after-treatment techniques and the test cycles used for both compression ignition and positive ignition engines (Bagány, 2011; Baumgarten et al., 2018;

Table 1 EURO specifications [mg/km] (Engeljehringer, 2018)

EURO	1	2	3	4	5	6
Spark ignition engine						
CO	2720	2200	2300	1000	1000	1000
HC + NO_X	970	500	–	–	–	–
HC	–	–	200	100	100	100
NO_X	–	–	150	80	60	60
HC	–	–	–	–	68	68
PM	–	–	–	–	4.5	4.5
PN [#/km]	–	–	–	–	–	6×10^{11}
Compression ignition engine						
CO	2720	1000	640	500	500	500
HC + NO_X	970	700	560	300	230	170
NO_X	–	–	500	250	180	80
PM	140	80	50	25	4.5	4.5
PN [#/km]	–	–	–	–	6×10^{11}	6×10^{11}

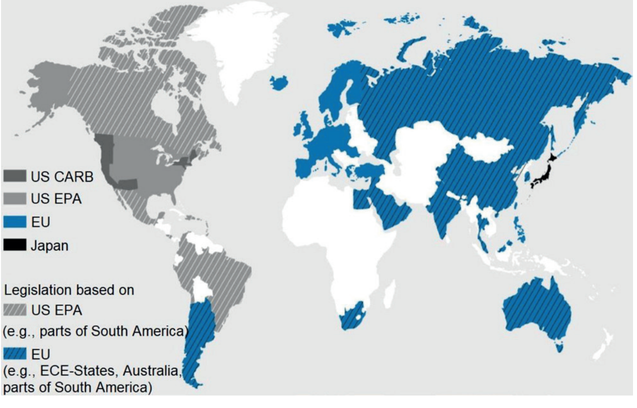

Fig. 2 Overview of the various emission legislations (Reif et al., 2015a)

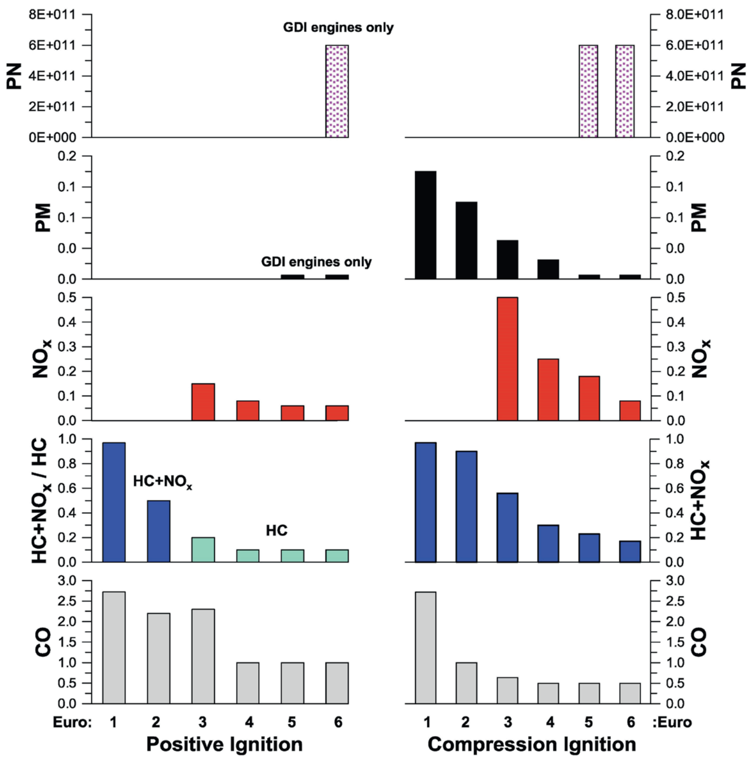

Fig. 3 Euro emission limits for M1 and N1 vehicles in the EU (Giakoumis, 2017)

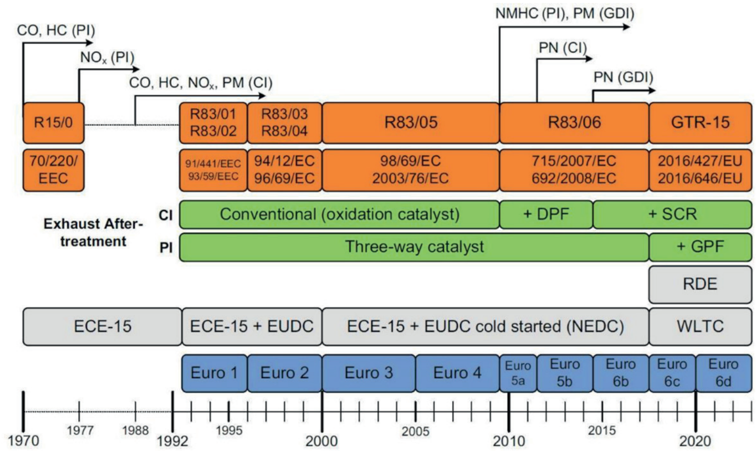

Fig. 4 Historical overview of emission test requirements for passenger cars (Giakoumis, 2017)

Bielaczyc and Woodburn, 2019; Eckert and Rakowski, 2012; Giakoumis, 2017; Kalmár and Stukovszky, 1998; Vas, 2005).

The initial driving cycle used to test emissions in the European Union was the test cycle from ECE-R15. Euro 1 (Council Directive 91/441/EEC, 1991) ECE+EUDC cycles were used next, followed in 2000 by the New European Driving Cycle (NEDC) and in 2017, with the Euro 6c standard (Commission Regulation 2017/1151, 2017), the Worldwide Harmonized Light Vehicles Test Cycle (WLTC) is used. The ECE+EUDC cycle had a 40-second idle time before sampling began (Bagány, 2011; Baumgarten et al., 2018; Bielaczyc and Woodburn, 2019; Giakoumis, 2017).

From Euro 3 (from 2000) (Directive 98/69/EC, 1998) the emission sampling started immediately with the starting of the engine, meaning the emissions during the 40 sec heating time were also sampled and measured. This was the NEDC test cycle and it presented a challenge for manufacturers in the automotive industry. Transient engine warm-up results in significant emissions as the catalyst has not achieved the operating temperature during this period. Hence, reducing emissions and new testing protocols (WLTC and Real Driving Emission) are the biggest challenges for the automotive industry, which have completely changed the approach to vehicle approval. The increasingly stringent restrictions must be met under ever-widening boundary conditions, and the regulated parameters (pollutant components) are also expanding. Road measurements and monitoring of nitrogen oxides (NO_X) and carbon-monoxide (CO) emissions have been carried out using Portable Emission Measurement Systems (PEMS) since the launch of Euro 6c in 2016 (Commission Regulation 2017/1151, 2017). RDE is an emission test method that measures the vehicle's tailpipe emissions under real driving conditions with a PEMS device under varying traffic conditions. Fig. 5 shows the different effects on vehicle emissions under real driving conditions (Bagány, 2011; Baumgarten et al., 2018; Bielaczyc and Woodburn, 2019; Giakoumis, 2017).

The first RDE package was published in Commission Regulation (EU) 2016/427 (2016) and the second and third packages in Commission Regulation (EU) 2016/646 (2016), 2017/1151 (2017) and 2017/1154 (2017). The fourth RDE package (Commission Regulation 2018/1832, 2018) also introduced testing of in-service vehicles (ISC, In-Service Conformity). When measured under real conditions, it was found that the emission values of some components – especially NO_X, but also particulate matter and other components such as CO_2 – can be much higher than during laboratory measurements. In the case of RDE, the emission limits must be met over a much wider engine operating range. Both the WLTC and RDE processes require a cold start of the engine, but WLTC is not the main challenge in terms of emissions due to the stringent requirements of the RDE (Bagány, 2011; Baumgarten et al., 2018; Bielaczyc and Woodburn, 2019; Giakoumis, 2017).

The "Euro 7" (AECC, 2020) will be put into effect in the period of 2023-2025. It will probably require a stoichiometric mixture ($\lambda = 1$) over the entire engine map to ensure that the three-way catalyst operates efficiently in all conditions, including the RDE conditions. The expected limit values of the future emission standard, namely Euro 7, are shown in Table 2. It is also expected that Euro 7 will prescribe the same test methods with stricter limits for all engine types and fuels. This would mean that the same limits would apply to a gasoline and a diesel-engine, so the regulations would be fuel and technology independent. Presumably, the number of particles will need to be examined over a larger size range (from 10 nm versus 23 nm). Another possible change could be the introduction of limit values for NO_2 and even for N_2O, NH_3 and aldehydes (Bielaczyc and Woodburn, 2019; Bontemps et al., 2019).

3 Possibilities and challenges of stoichiometric operation

In this section, the fundamentals of stoichiometric operation and the investigation of the effect on pollutants are described. Then we deal with full load operation possibilities and summarize the most important opportunities in achieving a stoichiometric mixture.

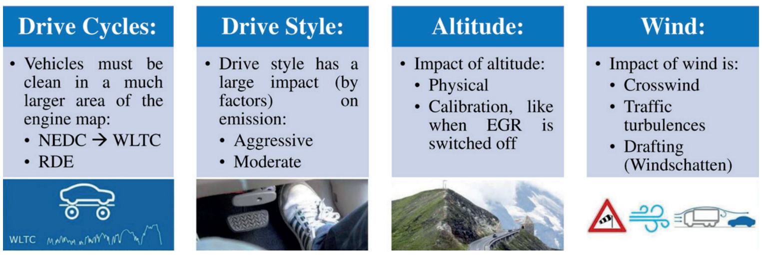

Fig. 5 Factors influencing vehicle emissions under RDE (Engeljehringer, 2018)

Table 2 Expected limit values of Euro 7 compared to Euro 6d (Hofegger, 2017)

	Euro 6d	Euro 7	**Change**
Test cycle	WLTP	WLTP	–
THC [mg/km]	100	50	**50%**
NMHC [mg/km]	68	35	**51.4%**
NO_X [mg/km]	60	35	**58.3%**
NO_2 [mg/km]	–	20	–
CO [mg/km]	1000	500	**50%**
PM [mg/km]	4.5	3.0	**66.6%**
PN [#/km]	6×10^{11}	6×10^{11}	**0%**
NH_3 [mg/km]	–	10	–

3.1 Air-fuel ratio and its effect on pollutants

The theoretical (ideal) air-fuel ratio prescribes the mass of air required for the complete combustion of the fuel. For gasoline, the theoretical air-fuel ratio is approximately 14.7. The actual air-fuel ratio is the ratio between mass of air and the mass of fuel currently available in the engine cylinder. The ratio of the actual air-fuel ratio to the theoretical air-fuel ratio is called the equivalence air-fuel ratio or lambda (λ) (Bagány, 2011; Dezsényi et al., 1999; Kalmár and Stukovszky, 1998; Vas, 2005):

$$\lambda = K / K_0, \tag{1}$$

where:

- λ – equivalence air-fuel ratio [–]
- K – actual air-fuel ratio [–]
- K_0 – theoretical air-fuel ratio (stoichiometric) [–].

If $\lambda < 1$, the mixture is deemed rich and the fuel cannot burn completely due to the oxygen deficiency. A rich mixture is required for cold starting as well as for accelerations. In this case, the extra fuel is not intended to participate in the combustion, rather its evaporation enthalpy cools the combustion chamber and the exhaust system. If $\lambda > 1$, the mixture is termed lean and an abundance of oxygen is available for combustion. To ignite the mixture, it is necessary to be in the range of $0.8 < \lambda < 1.1$, while ignition is uncertain for $\lambda \leq 0.6$ and $\lambda \geq 1.2$ for conventional gasoline engines. Maximum engine power is achieved with a slightly rich mixture, around $\lambda \approx 0.85$, while the best fuel consumption with a slightly lean mixture, around $\lambda \approx 1.1$. The decrease of equivalence air-fuel ratio is limited by the increase in CO formation while the increase in fuel consumption is due to deterioration in efficiency. The increase of equivalence air-fuel ratio is limited by the instability of combustion and the increase in specific fuel consumption. The formation of pollutants as a function of equivalence air-fuel ratio is shown in Fig. 6 (Bagány, 2011; Baumgarten et al., 2018; Eckert and Rakowski, 2012; Kalmár and Stukovszky, 1998; Meggyes and Boschán, 1993; Rakowski et al., 2012; Vas, 2005).

Complete combustion is not possible due to the inhomogeneity of the mixture and the constantly changing temperature, pressure, wall humidification and the short time to complete the processes, so carbon monoxide and unburnt hydrocarbons can always be found in the exhaust gas. The formation of CO, HC, and NO_X also changes with equivalence air-fuel ratio and temperature. While CO and HC increase with the enrichment of the mixture ($\lambda < 1$), the formation of NO_X is the highest at the highest combustion

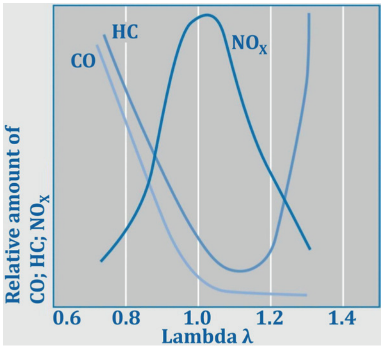

Fig. 6 Emissions of a gasoline engine as a function of the equivalence air-fuel ratio (Reif et al., 2015b)

temperature, around 1.1 equivalence air-fuel ratio. With a lean mixture, at $\lambda \geq 1.2$, the combustion temperature also decreases, so NO_X emissions also decrease, but HC increases. It can also be clearly seen in the figure that the CO and HC emissions are the most favorable for the slightly lean mixture, in the range of equivalence air-fuel ratio factors λ = 1.05–1.1, while the NO_X emissions are the highest in this area. Fig. 7 shows the range of NO_X formation as a function of temperature and local air conditions in a conventional and an alternative (low temperature) combustion process (Bagány, 2011; Eckert and Rakowski, 2012; Meggyes and Boschán, 1993; Merker et al., 2012; Rakowski et al., 2012; Vas, 2005).

In the classical stoichiometric ($\lambda = 1$) combustion process, we are in the NO_X formation zone. However, with a lower temperature combustion, the condition range of soot and NO_X formation can be avoided. An effective way to reduce NO_X formation is Exhaust Gas Recirculation (EGR), which also has a positive effect on the knocking tendency. EGR reduces the maximum combustion temperature, therefore it also reduces the NO_X formation (Dezsényi et al., 1999; Kalmár and Stukovszky, 1998; Merker et al., 2012; Vas, 2005).

Particulate matter can be solid or liquid pollutants emitted by the engine but the majority is soot. Fig. 8 shows the soot formation field as a function of equivalence ratio

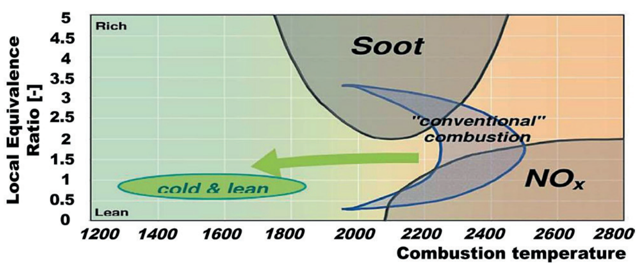

Fig. 7 Emissions as a function of local equivalence air-fuel ratio and temperature (Merker et al., 2012)

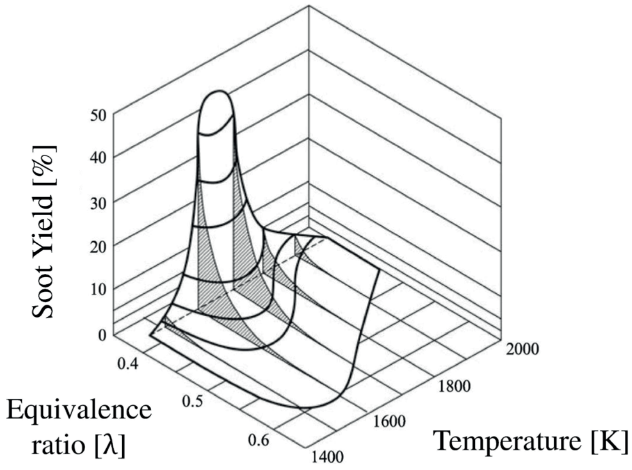

Fig. 8 Soot yield as a function of local equivalence air-fuel ratio and temperature (Eckert and Rakowski, 2012)

and temperature. For soot formation, the temperature window of 1500 < T < 1900 is critical for premixed flames. In this range, an extreme increase in soot is seen below 0.6 lambda (Eckert and Rakowski, 2012; Vas, 2005).

3.2 Stoichiometric operation at full load

Emissions legislation places ever-increasing expectations on the automotive industry, so new technical solutions are needed to meet them. Recently, the avoidance of a rich mixture ($\lambda = 1$) has grasped the attention of the development due to the tightening emission standards. With the loss of component protection from fuel enrichment, it appears that moving forward, the exhaust gas temperature must be limited in order to protect the components (e.g. turbocharger, cylinder head, catalyst). Thus, the maximum performance is limited by the heat resistance of those components in contact with exhaust gas. In constant compression ratio engines, combustion is delayed at high loads to avoid knocking, resulting in higher exhaust gas temperatures. Enrichment is used to reduce the temperature, but the three-way catalyst works most efficiently under stoichiometric conditions ($\lambda = 1$), so the enrichment results in an increase in HC, CO, and fuel consumption (Baumgarten et al., 2018; Collée et al., 2017; Fraidl et al., 2018; Neubauer et al., 2019).

The three-way catalyst converts the three main pollutants (CO, NO_X, HC) over the operating temperature range of 450–600 °C. It oxidizes CO and unburned HC to produce water and carbon dioxide. A portion of the oxygen required for oxidation is available from the exhaust gas and the other part from the reduction of nitrogen oxides. If the engine is running with a rich mixture for an extended period, there will not be enough oxygen to oxidize CO and HC, and their emissions will increase. The equivalence air-fuel ratio should be kept around 1 to optimize the conversion efficiency of the three-way catalyst. Then 99% of NO_X, 95% of CO and 70% of HC can be converted. This narrow range around $\lambda = 1$ is illustrated in Fig. 9 (Bagány, 2011; Meggyes and Boschán, 1993; Vas, 2005).

In the case of a gasoline engine with a stoichiometric mixture at high loads, the temperature in the cylinder increases, which causes an increase in nitrogen oxide emissions. Accompanying the rise in emissions is the tendency of knocking and the possibility of damage to the catalyst. To avoid this, gasoline engines are operated with rich mixture ($\lambda = 0.7$), leading to increased CO and HC emissions (Giakoumis, 2017).

The load-speed characteristic of a gasoline engine is illustrated in Fig. 10, showing two load points of the engine with their equivalence air-fuel ratios and the efficiency of the three-way catalyst. At partial load, with lambda 1 (gray), the CO removal efficiency of the catalyst is close to 100%, while at full load, the engine operates with 0.8 lambda (red), which reduces the efficiency of the catalyst to 40% and subsequently the CO emissions increase significantly (Baumgarten et al., 2018).

An increase of CO emissions at full load can be avoided with a stoichiometric mixture, at which the specific engine power will decrease. This foregoes the thermal protection of the component offered by fuel enrichment. There are two options for eliminating power loss (Fig. 11), which are (Baumgarten et al., 2018):

- The use of more heat-resistant materials e.g. allowing an exhaust gas temperature of 1050 °C for the turbocharger turbine (red arrow), or

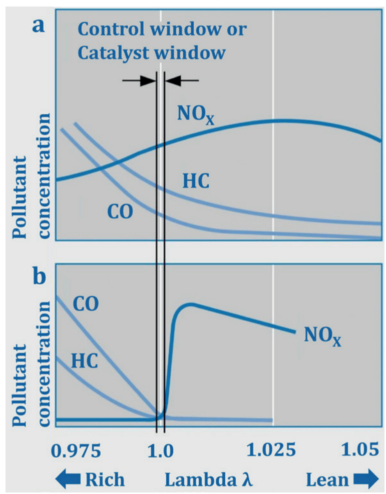

Fig. 9 Exhaust gas components before (a) and after (b) the three-way catalyst (Reif et al., 2015b)

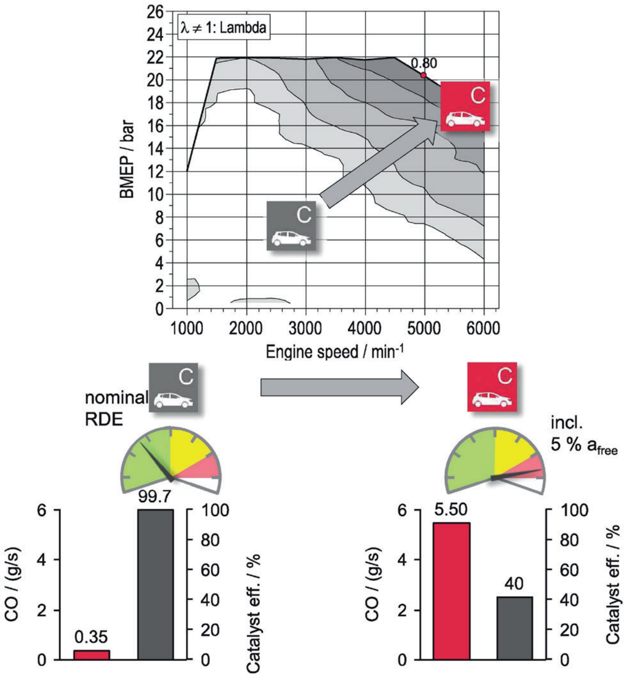

Fig. 10 Effect of the engine operating point on CO emissions and the three-way catalyst conversion efficiency (Baumgarten et al., 2018)

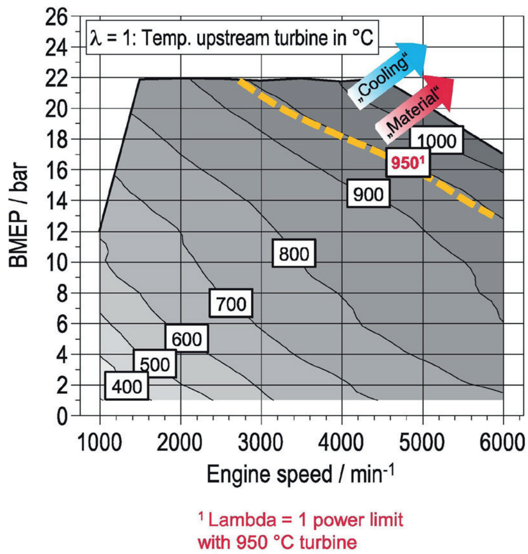

Fig. 11 Exhaust gas temperature in lambda 1 engine map (Baumgarten et al., 2018)

- Limiting the exhaust gas temperature to 980 °C by introducing additional cooling technologies (blue arrow).

For two turbocharged gasoline engines with different specific powers (110 kW/l and 90 kW/l), AVL List GmbH. investigated the possibility of a stoichiometric mixture while the original power of the engine was provided. For this purpose, several technical solutions have been devised (Baumgarten et al., 2018):

- 1050 °C turbocharging technology,
- Water injection,
- Variable compression ratio, and
- Electrically assisted turbocharger (eTC).

A summary of these technologies and their impacts is shown in Fig. 12. The base engine "A" requires a high fuel enrichment (approx. $\lambda = 0.7$) to achieve a specific output of 110 kW/l, while the basic engine "B" only achieves 90 kW/l with a lower fuel enrichment (approx. $\lambda = 0.8$). For both engines, the figure shows the power that can be achieved using different technologies, some of them allow $\lambda = 1$ operation. From the basic configuration (with full load enrichment), the following columns show, from left to right, the specific powers that can be achieved with (Baumgarten et al., 2018):

1. The maximum exhaust temperature is 980 °C (Lambda = 1),
2. Maximum temperature increased to 1050 °C by changing the turbine material,
3. Direct water injection,
4. Variable compression ratio (2-stage),
5. Larger turbine combined with electrically assisted turbocharger (eTC).

For both engines, the specific engine power is significantly reduced if an exhaust gas temperature of 980 °C is used without the introduction of additional technologies. The use of turbine material with an exhaust gas temperature limit of 1050 °C has significant potential, but this technology alone is not sufficient to fully compensate the power loss. Direct water injection eliminates power losses. The variable compression ratio (2-stage) eliminates ("B" engine) or moderates the power losses ("A" engine) and provides additional benefits in fuel consumption. Significantly increased use of turbine with electrically assisted turbocharger (eTC) compensates the power loss of both engines (Baumgarten et al., 2018).

4 Stratified mixture

In stratified mixture formation, a globally lean mixture is used that is inflammable with conventional spark ignition. At the moment of ignition, a cloud of flammable mixture is formed near the spark plug which is used to propagate ignition of the lean mixture farther from the plug. In the case of

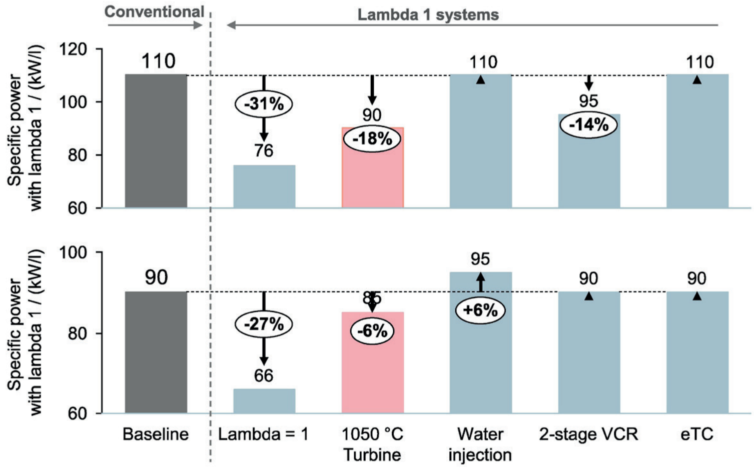

Fig. 12 The effect of different lambda =1 technology packages on the performance of two different base engines (Baumgarten et al., 2018)

a lean mixture ($\lambda > 1.2$), the speed of flame propagation and combustion is lower. The combustion temperature is also lower, leading to a decrease in NO_X emissions and increase in HC emission. Due to the globally lean mixture, three-way catalysts cannot be used to reduce NO_X. Instead, a storage catalyst is used for the reduction, which must be regenerated at short intervals by enrichment. This has a negative effect on fuel consumption. The direct injection spark ignition engine cannot be operated in the entire engine map in stratified charge mode. It must be used at low loads and low speed ranges. Three methods used for stratified charge mode are shown in Fig. 13 (Bagány, 2011; Kalmár and Stukovszky, 1998; Merker et al., 2006; Rakowski et al., 2012; Schäfer and Basshuysen, 1995; Schäfer et al., 2017).

In a wall guided process, the piston design guides the mixture cloud from the nozzle to the spark plug. In an air guided process, the air rotates around a horizontal axis, which can be ensured by the geometry of the intake port. In a spray guided process, the injector produces a flammable mixture with the injected fuel spray at the spark plug. Both the lean and the homogeneous mixture have advantages and disadvantages. The evaluation of the use of a homogeneous, stoichiometric mixture and a lean mixture is summarized in Table 3. In the evaluation, "+" means favorable, "0" means neutral, and "–" means

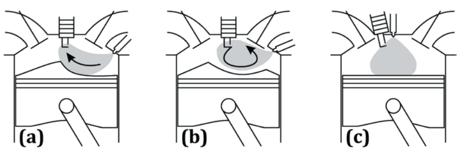

Fig. 13 Wall guided (a), air guided (b) and spray guided (c) stratified charge methods (Merker et al., 2006)

Table 3 Evaluation of various mixtures

Evaluation criteria	Homogeneous stoichiometric mixture	Stratified lean mixture
Specific fuel consumption	0	+
Complexity of the aftertreatment system	+	–
Carbon monoxide emissions	0	+
Hydrocarbon emissions	0	–
Nitrogen oxide emissions	0	+
Usability in the entire engine map	+	––
Result	2	–1

unfavorable (Bagány, 2011; Kalmár and Stukovszky, 1998; Merker et al., 2006; Rakowski et al., 2012; Schäfer and Basshuysen, 1995; Schäfer et al., 2017).

Explained further, the lean mixture can reduce specific fuel consumption and is also beneficial in terms of NO_X and CO emissions. However, lean mixture cannot be applied to the entire engine map. When using a lean mixture, the three-way catalyst operates at a lower efficiency compared to the stoichiometric operation, so the lambda 1 operation is the favorable process in this comparison as well (Bagány, 2011; Kalmár and Stukovszky, 1998; Merker et al., 2006; Rakowski et al., 2012; Schäfer and Basshuysen, 1995; Schäfer et al., 2017).

5 Several technical solutions for $\lambda = 1$

Future emission regulations will require operation with stoichiometric mixture ($\lambda = 1$) over the entire engine map. To ensure stoichiometric operation without loss of performance, new technologies or a combination of these will be required. In the case of gasoline engines, there are some technologies, which have already been utilized to achieve appropriate efficiency and fuel consumption: turbocharged gasoline engine with direct injection (TGDI) or three-way catalytic converter and particulate filter. Efficient methods to extend the $\lambda = 1$ range include (Fraidl et al., 2018; Neubauer et al., 2019):

- Exhaust gas cooling with integrated exhaust manifold into the cylinder head;
- Cooled exhaust gas recirculation (EGR);
- Miller or Atkinson cycles;
- Water injection;
- Variable compression ratio;
- HCCI combustion.

In the following section, selected solutions are presented in more detail.

5.1 Water injection

Water injection is a technology that can achieve $\lambda = 1$ operation and CO_2 reduction at the same time. The injected water, with its high enthalpy of evaporation (see Fig. 14), cools the charge in the cylinder, thereby reducing the exhaust gas temperature and the likelihood of knocking.

These two benefits are connected since the reduced proclivity of knocking permits a higher compression ratio and thus CO_2 can be reduced (Baumgarten et al., 2018; Hermann et al., 2019; Neubauer et al., 2019; Seeley and Fischer, 2019).

The evaporation enthalpy of water is 2257 kJ/kg, while the evaporation enthalpy of premium quality gasoline is 420 kJ/kg. This means that the evaporation enthalpy of water is more than five times greater, than that of gasoline. However, the use of water also has disadvantages, such as wall humidification, it can freeze in cold operation climates and it has a higher surface tension than gasoline (Neumann et al., 2019).

One way of grouping water injection concepts is by the position of injection. Water can be injected into the intake manifold (plenum), into the intake port or directly into the cylinder. Direct water injection can be accomplished by injecting pure water or injecting a water/fuel emulsion. The arrangement of the injectors is illustrated in Figs. 15 and 16 (Neumann et al., 2019).

In case of intake manifold injection, the injection time is theoretically optional, but injection with closed intake valves results in a layer of water on the wall of the intake manifold and port and on the intake valve(s). This leads to larger droplet size and slower evaporation. Because wall wetting cools the components instead of the mixture, it degrades the efficiency of water injection. Injection into the intake manifold normally cools the intake air, while injection into the intake port and into the combustion chamber (direct injection) removes heat from the combustion chamber (Hermann et al., 2019; Neumann et al., 2019).

Water injected directly into the combustion chamber requires a higher injection pressure than water injection into the intake manifold. In the case of direct water injection, the complexity and cost of the system are also high. With direct water injection, the water injection time is independent of the gasoline injection, but with direct emulsion injection, this cannot be said. The emulsion is injected at high pressure (> 200 bar), resulting smaller droplet size, better mixing, and evaporation. Simulation results, shown in Fig. 17, compare premixed direct injection and pure water direct injection based on the

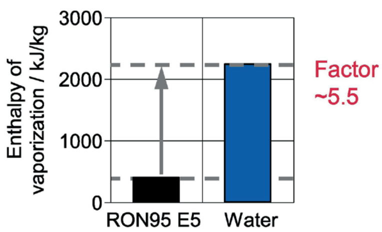

Fig. 14 Evaporation enthalpy of water and RON95 E5 gasoline (Baumgarten et al., 2018)

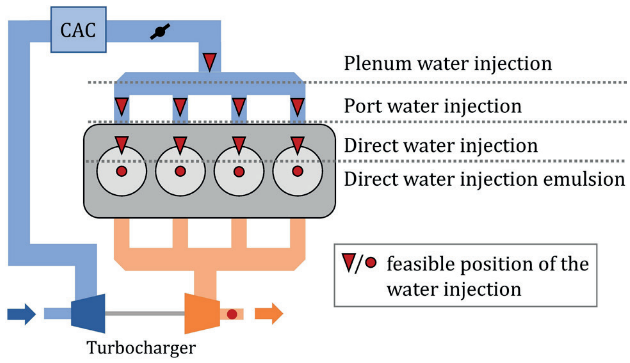

Fig. 15 Water injection concepts (Neumann et al., 2019)

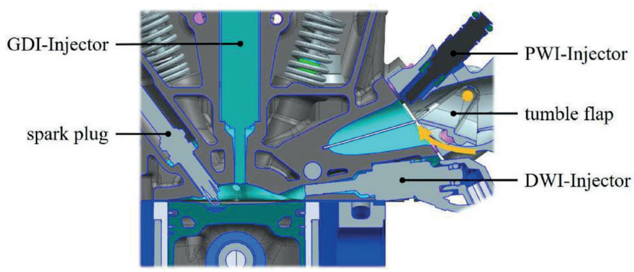

Fig. 16 Arrangement of injectors (Neumann et al., 2019)

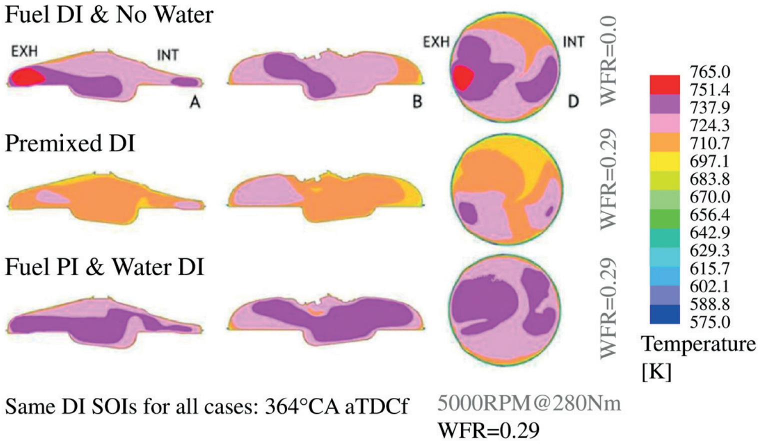

Fig. 17 Compression End Temperatures for water injection concepts (Hermann et al., 2019)

compression end temperatures in the combustion chamber. In all cases, water injection resulted in a decrease in temperature, but the highest temperature decrease was obtained through premixed direct injection (Hermann et al., 2019; Neumann et al., 2019).

Fig. 18 shows the effect of water injection concepts on emissions. Three water injection concepts were investigated: port water injection (white), single- (blue) and multi-direct water injection (red). For each water injection variant, an increase in water volume flow results in a continuous decrease in NO_X concentration since thermal NO_X formation decreases with temperature. Water injection also slightly reduces CO emissions. Though, HC, and particulate emission increase significantly with water volume flow especially with multi-direct water injections (Neumann et al., 2019).

A summary table, Table 4, shows a multi-aspect evaluation of the effect of different water injection technologies on engine operation. Based on the evaluation, it can be concluded that port water injection is the most advantageous. Furthermore, direct water injection also achieved

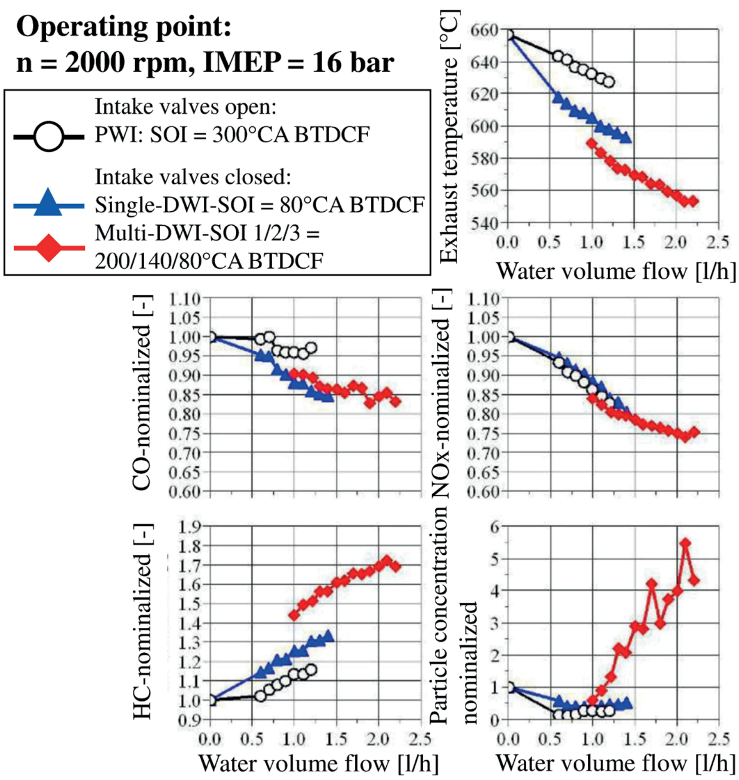

Fig. 18 The effect of water injection concepts on emissions (Neumann et al., 2019)

Table 4 Evaluation of water injection concepts (Hermann et al., 2019)

Water injection	Plenum	Port	Direct (emulsion)	Direct (water)
Distribution to cylinders	––	++	+	++
Mixture formation	––	0	++	+
Evaporation	––	0	++	+
Thermal efficiency	––	0	++	++
Water consumption	––	0	++	+
Transient operation	++	++	––	++
Integration effort	++	++	––	––
Costs	++	++	––	––
Sum	−3	+8	+3	+5

a high score, so this method is also beneficial. The use of intake manifold (plenum) injection is not recommended (Hermann et al., 2019).

5.2 Variable compression ratio (VCR)

One of the goals of the gasoline engine development is to increase the thermal efficiency. One way that this can be achieved is by altering the compression ratio. The compression ratio is shown in Fig. 19 as shown on a p-V diagram (Shinichi et al., 2017).

The compression ratio (CR) is defined as the ratio of the total volume of the cylinder ($V_C + V_S$) and the compressed volume (V_C). The thermal (theoretical) efficiency of the engine improves when the compression ratio increases because the engine performs more work: the same piston stroke has a larger positive working area (Shinichi et al., 2017).

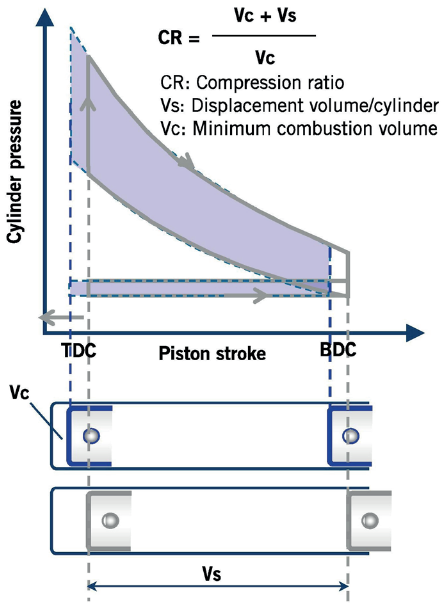

Fig. 19 Compression ratio on p-V diagram (Shinichi et al., 2017)

Knocking limits the efficiency improvements that can be realized through compression ratio at full load. Combustion temperature and pressure are lower at partial load, so a higher compression ratio is permitted in this operating state. If the compression ratio can be changed during operation, the optimum compression ratio can be set for all operating conditions in terms of efficiency and knocking. An example of a variable compression ratio is shown in Fig. 20 (Shinichi et al., 2017).

Several solutions have realized a variable compression ratio, e.g. the Nissan Infiniti engine. The basic structure and operation are shown in Fig. 21. The reduction of the compression ratio is achieved by lowering the top dead center (TDC) below. The stroke value can be adjusted by 1.2 mm. The minimum compression ratio of the engine is $\varepsilon = 8$, which is used at high load, and the highest compression ratio is $\varepsilon = 14$, which is used at partial load. This difference in the compression ratio is achieved by creating a 1.2 mm stroke difference (see Fig. 21). The compression ratio between the highest and lowest values can be adjusted continuously to meet the power demands. Compared to

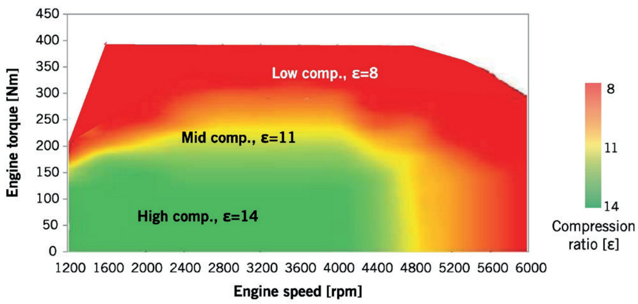

Fig. 20 Ideal compression ratio values in the engine map (Shinichi et al., 2017)

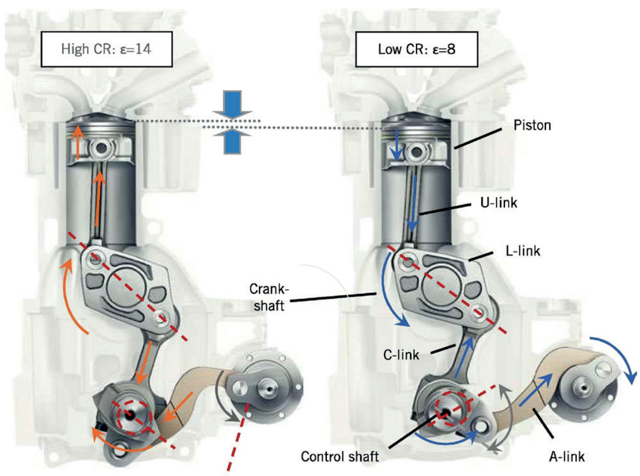

Fig. 21 Basic structure and operation of Nissan Infiniti (Shinichi et al., 2017)

conventional engines with a constant compression ratio, the high compression ratio of the system makes it possible to increase the efficiency of the engine and improve fuel consumption (Nagyszokolyai, 2016; Shinichi et al., 2017).

The FEV Group has developed an alternative method that involves a connecting rod with the ability to change the compression ratio during operation in two stages. With a rotating eccentric tappet, the length of the connecting rod and thus the compression ratio can be adjusted (Fig. 22). The position of the eccentric is moved by 2 hydraulic pistons. The compression ratio is changed within 0.2–0.6 seconds. Compared to other VCR systems, the solution of FEV Group does not require extensive design changes due to its modular and compact form making it a cost-effective solution. Disadvantageously, the VCR system significantly increases the alternating mass forces which creates other issues. Reduction in CO_2 emissions is mainly achieved at the partial load resulting from a reduction in fuel consumption (FEV, 2020; Nagyszokalyai and Őri, 2012).

5.3 Miller- and Atkinson-cycle

Increase of efficiency in throttled operation at partial load is also possible with using late (Atkinson-cycle) or early (Miller-cycle) closing of inlet valves. In the Atkinson cycle, the intake valves close later, after bottom dead center (BDC). At this point, the piston is moving upwards, returning some of the fresh air back to the port(s). To include the charge mass of the standard control time in the cylinder, the gasoline engine is de-throttled further, thereby increasing the efficiency. At high speeds, a relatively late intake valve closure is advantageous to achieve a high volumetric efficiency. Due to the long opening times of the inlet valves, dynamic gas effects can also be exploited in the Atkinson process, especially in naturally aspirated engines. In the case of Atkinson cycle, the secondary air charge motion is less intense. The (volumetric) compression ratio is reduced, but the (volumetric) expansion ratio remains the same. It should be understood that the entire displacement of the internal combustion engine is not used. This reduces the tendency of knocking, and has additional beneficial effects on thermal efficiency, fuel consumption, and NO_X formation. This procedure is used only for naturally aspirated engines because the degree of delivery cannot be reduced with supercharged engines because of the boost pressure (Barth et al., 2013; Isenstadt et al., 2016; Wittek, 2006).

Fig. 22 FEV 2-stageVCR system (FEV, 2020)

In contrast, the Miller method describes an early closing of the intake valves. The Miller cycle is the namesake of Ralph Miller and was described in his 1947 patent. The intention was to fit the Atkinson cycle to conventional engines. The intake valve closes before BDC, and with the piston still moving downward (intake stroke), the charge expands and cools in the cylinder. Compared to the standard timing, the following compression takes place at a lower pressure and temperature level. A decrease in the end compression temperature and end pressure reduces the tendency of knocking. The lower compression ratio also results less NO_X emissions (Audi, 2016; Isenstadt et al., 2016; Neubauer et al., 2019; Wittek, 2006).

To generate the same torque and to include the same mass of fresh air in the cylinder, the working process must be de-throttled, which significantly increases the efficiency. A turbocharger can also be used to compensate for the lower engine torque. The Miller process is such that the compression ratio is reduced and the expansion ratio is constant. For charged engines, the Miller cycle is preferable to the Atkinson cycle because a portion of the charge is returned to the intake manifold by the piston. The Atkinson cycle is therefore recommended for natural aspirated engines (Audi, 2016; Isenstadt et al., 2016; Neubauer et al., 2019; Wittek, 2006).

Since both methods decrease the temperature of the charge during compression, they can also be used to reduce the tendency of knocking, especially with supercharged gasoline engines at full load. In addition, they can be used to increase specific performance by exploiting the separate compression and expansion ratio. The disadvantage of these methods is that the total volume of the cylinders is not used for maximum output. For this reason, these processes cannot be used for a racing car to reduce the exhaust gas temperature as maximum output is always needed (Barth et al., 2013; Wittek, 2006).

5.4 Combustion development

Development of the combustion system and combustion process is necessary to reduce emissions and fuel consumption. Development of combustion systems include (Ogink, 2015; Seeley and Fischer, 2019):

- Intake and exhaust port design;
- Injection optimization (profile, strategy) and spray targeting;
- Piston and combustion chamber optimization;
- Flame propagation.

The intake manifold, intake port, piston design, and engine speed also affect the secondary air charge motion. Two main secondary charge motions are "swirl" and "tumble", which are depicted in Fig. 23 (Rakowski et al., 2012).

Swirl is a rotational motion around a vertical axis, which is largely characteristic of 4-valve engines and exists until ignition. Tumble is a rotating motion around a horizontal axis. The tumble flow is formed in the cylinder during the intake stroke, it collapses (break-down) during compression. The turbulent vortices mix the flame front, so the combustion process becomes faster. Turbulence as a device to increase flame front velocity is employed at high engine speeds, as less time is available for mixture formation and combustion. The velocity of flame propagation is strongly dependent on the flow conditions. For laminar flow (low Reynold's number) the flame velocity is typically between 0.3–1 m/s, while for turbulent flow (high Reynold's number) it can be up to 30 m/s. Tumble motion leads to rapid combustion and reduces fuel consumption. Intense tumble also has a positive effect on knocking (Ogink, 2015; Rakowski et al., 2012).

An important part of combustion system development is the design of the intake port(s). Combustion process

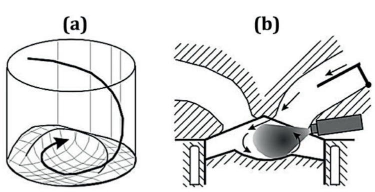

Fig. 23 Basic forms of secondary air charge motion: (a) "swirl"; and (b) "tumble" (Rakowski et al., 2012)

optimization of direct injection gasoline engines can also be performed using Computational Fluid Dynamics (CFD). The results of steady-state CFD analysis of different intake ports are shown in Fig. 24. The mass flow rates, and tumble ratios of the different intake ports are shown. For a given valve lift, there is a nearly linear relationship between the two parameters (Ogink, 2015; Seeley and Fischer, 2019).

High charge mass flow is achievable with high valve lift and it causes high tumble flow intensity. The target of the development is to design an intake port that provides sufficient mass flow to achieve the desired power level at the highest possible tumble intensity. Two intake port designs (fill port and tumble port) can be seen in Fig. 25 (Ogink, 2015; Seeley and Fischer, 2019).

Higher mass flow is available with the fill port. The flow enters the combustion chamber almost vertically. In the case of the tumble port, a much straighter design can be observed. The incoming air enters the combustion chamber mainly on the upper side of the valves and then rotates around a horizontal axis (Ogink, 2015).

A properly designed combustion system is characterized by high turbulent kinetic energy (TKE) before and during combustion, resulting in a stable and rapid combustion process. Fig. 26 shows the transient CFD results for the tumble and the fill ports, where the intensity of the tumble flow as a function of the crank angle degree (CAD) (Ogink, 2015).

During the intake and compression stroke, the tumble level of the fill port is much lower than the tumble

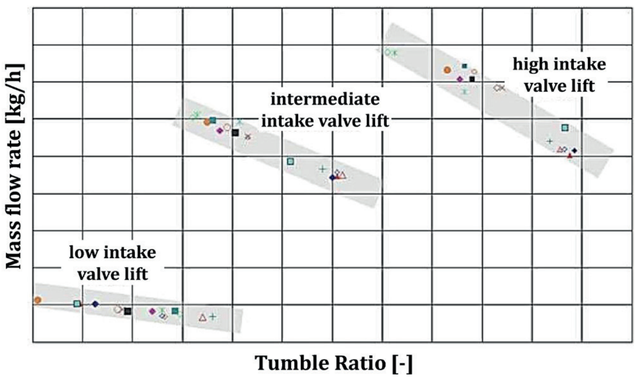

Fig. 24 Intake ports mass flow and tumble ratio relationships (Ogink, 2015)

Fig. 25 Intake ports designs: (a) Fill port; and (b) Tumble port (Ogink, 2015)

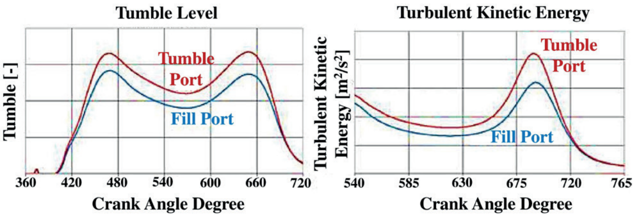

Fig. 26 Tumble level and TKE for Fill port and Tumble port (Ogink, 2015)

port's level (left diagram). Moreover, the average level of Turbulent Kinetic Energy (TKE) at the end of compression is much higher for the tumble port (right diagram). The results of experimental studies show lower fuel consumption, better mixture formation, lower emissions (HC, CO, and soot) and great combustion properties from the tumble port (Ogink, 2015).

The shape of the intake port can be characterized by 3 parameters, which are the height of the valve seat, the morphing amplitude, and the port angle. The effects of these input parameters on tumble intensity and charge mass are shown in Fig. 27. It was found that the larger the morphing amplitude parameter is, the larger the corresponding tumble and the less the charge mass will be. Additionally, the shorter the height of the valve seat is, the better the overall effect will be. Furthermore, a 90° port angle is not necessary but is a highly influential condition for high performance.

The CFD simulation results (velocity fields) for different intake port designs are shown in Fig. 28. The figure shows the velocity fields at BDC in the mid plane of intake valve. The maximum charge mass can be achieved with the green

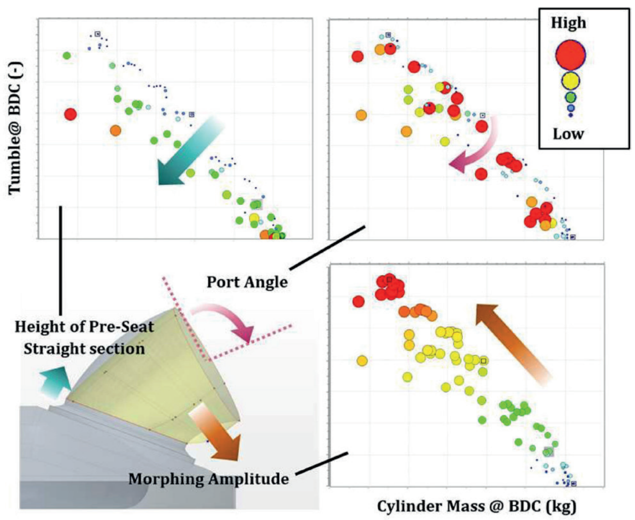

Fig. 27 Effect of intake port parameters on Tumble and charge mass (Seeley and Fischer, 2019)

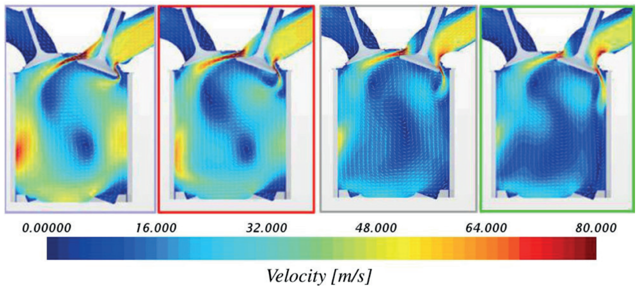

Fig. 28 Velocity fields for different intake port designs (Seeley and Fischer, 2019)

outlined intake port, the maximum tumble with the purple one and the red one is a hybrid of these two. The gray framed port is the original design (Seeley and Fischer, 2019).

6 Summary

The automotive industry is currently facing serious challenges due to ever-tightening emission standards and test methods (RDE). There is limited potential for reduction of pollutants in the engine, so better catalytic reduction is necessary. Stratified (lean) mixture is not recommended due to the associated disadvantages of storage catalyst. Therefore, as of Euro 7, a stoichiometric mixture will be required over the entire engine map to meet the new requirements. In the case of stoichiometric operation, CO and HC formation are low and NO_X is very high. However, a three-way catalyst operates with sufficient efficiency to manage the resulting pollutants. Due to the omission of component protection by fuel enrichment, it is necessary to limit the exhaust gas temperature to 980 °C to protect the components from overheating. Solving the problems requires the introduction of new technologies such as water injection, variable compression ratio, Miller/Atkinson cycles, cooled exhaust gas recirculation (EGR), exhaust gas cooling with integrated exhaust manifold or HCCI combustion process. Select new technologies are detailed in this article.

Water injection allows $\lambda = 1$ operation and CO2 reduction at the same time. The best water injection method based on the present evaluation is by port injection. With variable compression ratio technology, the compression ratio can be changed during operation to the optimum value, in terms of efficiency and knocking. Examples of VCR solutions currently in use include the Nissan Infiniti engine or the connecting rod of the FEV Group. Miller and Atkinson cycles decrease the temperature of the charge during compression, reduce knocking tendency, increase efficiency and specific performance. Combustion development is suitable for reducing emissions (HC, CO, and soot) and fuel consumption. Exploiting secondary air charge motion can also assist. The main secondary air charge motions are the tumble and swirl. High turbulence before and during combustion results in a stable, rapid combustion process, and better mixture formation. In addition to these technical solutions, there are many other options in the hands of engine developers.

Internal combustion engines have reached a high level of development, but serious efforts and large financial investments will still be needed to maintain their competitiveness against electric drives. The present focus of gasoline engine development is to find a technical package of these solutions that is cost efficient and meets future emissions standards.

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