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Analysis the Influences of Combustion Strategies and Technologies on Emission in Gasoline and Diesel Engine

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Abstract: The paper reviews the technologies available to meet the exhaust emissions regulations for Internal Combustion Engine for light-duty and heavy-duty vehicles, non-road mobile machinery and motorcycles. This includes fast light-off catalysts, more thermally durable catalysts, improved substrate technology, diesel particulate filters, selective catalytic reduction, NOx absorbers and lean De NOx catalysts. The stricter worldwide emission legislation and growing demands for lower fuel consumption and anthropogenic CO_2 emission require significant efforts to improve combustion efficiency while satisfying the emission quality demands. Ethanol fuel combined with gasoline provides a particularly promising, and at the same time, a challenging approach. Extensive usage of automobiles has certain disadvantages and one of them is its negative effect on environment. Carbon dioxide (CO_2), carbon monoxide (CO), hydrocarbons (HC), oxides of nitrogen (NOx), sulphur dioxide (SO_2) and particulate matter (PM) come out as harmful products during incomplete combustion from internal combustion (IC) engines. As these substances affect human health, regulatory bodies impose increasingly stringent restrictions on the level of emissions coming out from IC engines. Modern combustion techniques such as low temperature combustion (LTC), homogeneous charge compression ignition (HCCI), premixed charge compression ignition (PCCI) etc., would be helpful for reducing the exhaust emissions and improving the engine performance. However, controlling of auto ignition timing and achieving wider operating range are the major challenges with these techniques.

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Keywords: Engine Emission, Technologies for reducing emissions, Exhaust Gas Recirculation, Homogenous Charge Compression Ignition (HCCI), Stratified EGR

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1. Introduction

The internal combustion engine, in which the combustion of a fuel (normally a fossil fuel) occurs with an oxidizer (usually air) in a combustion chamber that is an integral part of the working fluid flow circuit. In an internal combustion engine (ICE) the expansion of the high-temperature and high-pressure gases produced by combustion apply direct force to some component of the engine [1-10]. The force is applied typically to pistons, turbine blades or a nozzle. This force moves the component over a distance, transforming chemical energy into useful mechanical energy. Emissions of many air pollutants have been shown to have variety of negative effects on public health and the natural environment. Emissions that are principal pollutants of concern include [11-19]:

- **Hydrocarbons:** A class of burned or partially burned fuel which are considered as toxins.

Hydrocarbons are a major contributor to smog, which can be a major problem in urban areas. Prolonged exposure to hydrocarbons contributes to asthma, liver disease, lung disease, and cancer. Regulations governing hydrocarbons vary according to type of engine and jurisdiction; in some cases, "non-methane hydrocarbons" are regulated, while in other cases, "total hydrocarbons" are regulated. Technology for one application (to meet a non-methane hydrocarbon standard may not be suitable for use in an application that has to meet a total hydrocarbon standard. Methane is not directly toxic, but is more difficult to break down in a catalytic converter, so in effect a "non-methane hydrocarbon" regulation can be considered easier to meet. Since methane is a greenhouse gas, interest has been rising on how to eliminate these emissions.

- Carbon Monoxide (CO): A product of incomplete combustion which reduces the blood's ability to carry oxygen; overexposure (carbon monoxide poisoning) may be fatal. Exposure of Carbon Monoxide in high concentration is lethal .
- Nitrogen Oxides (NOx): Generated when nitrogen in the air reacts with oxygen at the high temperature and pressure inside the engine. NOx is a precursor to smog and acid rain. NOx is a mixture of NO, N_2O, and NO_2. NO_2 is extremely reactive. It destroys resistance to respiratory infection. NOx production is increased when an engine runs at its most efficient (i.e. hottest) part of the cycle.
- Particulate Matter: Soot or smoke made up of particles in the micro meter size range; particulate matter, causes negative health effects including but not limited to respiratory disease and cancer.
- Sulphur Oxide (SOx): A general term for oxides of sulphur, which are emitted from motor vehicles burning fuel containing sulphur. By reducing the level of fuel sulphur reduces the level of Sulphur oxide emitted from the tailpipe.

Exhaust emissions can be lowered by reducing engine-out emissions through improvements to the combustion process and fuel management, or by adding changes to the type of fuel used or its composition. Emissions control systems, auto catalysts, absorbers and particulate filters in combination with good quality fuel (low sulphur content) and enhanced engine management reduce emissions to very low levels. As well as their application in new vehicles and machinery, many emissions control systems can also be applied in retrofit applications for good effects [1-6].

2. Catalyst Technologies for Emissions Control

2.1 The Catalyst

Catalysts are needed to reduce emissions to acceptable levels without dramatically reducing performance and fuel economy. This is true for HC, CO and NOx, but NOx is the emission that is most dependent on the catalyst for emissions compliance. There are actually two types of catalysts. Reduction catalysts causes NOx to be reduced into O_2 and N_2. Meanwhile, oxidation catalysts causes HC and CO to oxidize, with any available oxygen, into CO_2 +H_2O. Unfortunately oxidation will only occur when there is enough free oxygen, and reduction will only occur in a relative absence of free oxygen. Rhodium is generally the most efficient reduction catalyst. Platinum and palladium are used for oxidation. 2-way catalytic converters are oxidation catalysts. They oxidize CO and HC, but do not reduce NOx. On the other hand, 3-way catalysts oxidize CO and HC, and reduce NOx. Proper air /fuel mixture control and exhaust oxygen content is required for proper 3-way catalyst performance. In general, oxidation and reduction, are unlikely to occur together at the same time at their highest efficiency. Reduction efficiency is not at its highest unless the oxygen content is very low. This usually does not happen unless the air/fuel mixture is at least a little bit rich. Oxidation only reaches its highest efficiency when the oxygen content is fairly high. That happens when the mixture is at least slightly lean. A dual bed catalyst has two separate chambers. Air can be injected in the middle of the catalyst to increase oxygen content at the back half of the converter. The engine can then also be run slightly rich to improve NOx reduction at the front half of the converter [19-23].

2.2 NOx Control Technologies

With the development of lean burn direct injection gasoline engines, and the increased use of diesel engines in passenger cars, there is an increasing need for the control of NOx in lean combustion systems. Lean burn systems limit CO_2 emissions and reduce fuel consumption, and so are the key technologies for the future [23-24].

2.3 Selective Catalytic Reduction (SCR)

SCR was originally developed and used to reduce nitrogen oxide emissions from coal, oil and gas fired power stations, marine vessels and stationary diesel engines. SCR technology permits the NOx reduction reaction to take place in an oxidizing atmosphere. It is called “selective” because the catalytic reduction of NOx with ammonia (NH_3) as a reductant that occurs preferentially to the oxidation of NH_3 whereby oxygen SCR technology is now fitted to most new heavy-duty (i.e., truck and bus) diesel engines. Systems are also being introduced on light-duty diesel vehicles and on non-road mobile machinery such as construction equipment. It allows diesel engine developers to take advantage of the trade-off between NOx, PM and fuel consumption, and calibrate the engine in a lower area of fuel consumption than if they had to reduce NOx by engine measures alone [10-15].

2.4 **NOx Absorbers or Lean NOx Traps (LNT)**

Lean NOx traps adsorb and store NOx under lean conditions. A typical approach is to speed up the conversion of nitric oxide (NO) to nitrogen dioxide (NO_2) using an oxidation or three-way catalyst mounted close to the engine so that NO_2 can rapidly be stored as nitrate. The function of the NOx storage element can be fulfilled by materials that are able to form sufficiently stable nitrates within the temperature range determined by lean operating engine points. Thus, especially alkaline, alkaline earth and to a certain extent also rare-earth compounds can be used [25-27].

3. **Methodology**

3.1 **Hydrogen as an ignition-controlling agent for HCCI combustion engine by suppressing the low-temperature oxidation**

It is expected that the use of homogeneous charge compression ignition (HCCI) combustion in internal combustion engines will result in higher thermal efficiency and lower NOx emissions than with conventional combustion systems. However, difficulties in controlling the ignition timing in accordance with the engine load prevent HCCI combustion from practical application in vehicle engines. Adjusting the proportion of two fuels with different ignition properties has been reported as an effective technique to control the ignition timing and load in HCCI combustion [3]. However, this technique has not been practically used in vehicles because of the inconvenience of carrying two kinds of fuels. Dimethyl ether (DME) has been studied as a clean alternative to diesel fuel due to its high cetane number and smokeless combustion characteristics [4-6], and DME can be easily produced from methanol by the dehydration reaction [7]. A report suggests using a small amount of DME produced from methanol as an ignition promoter in a methanol direct-injection diesel engine [4]. Methanol can also be thermally decomposed into Methanol Reformed Gas (MRG) which consists of hydrogen and carbon monoxide. Since both hydrogen and carbon monoxide have good anti-knocking properties [8], MRG has been studied as a fuel for spark-ignition engines [9,10]. With this background, an HCCI combustion engine system that was fuelled with DME and MRG has been proposed [28–30]. Because the ignition properties of DME and MRG are very different, adjusting the proportion of the two fuels can control the ignition timing in an HCCI combustion engine fuelled with the two

In addition to ignition control, production of DME and MRG by onboard reformers utilizing the exhaust gas heat of the engine has also been proposed, with an outline of the arrangement as shown in Figure 1. This is because the reactions to produce DME and MRG from methanol are endothermic, the heating values of the produced DME and MRG can be higher than the primary fuel. Therefore, methanol reformation using the engine exhaust gas heat could be utilized to recover waste heat from the engine. By combining efficient HCCI operation and waste heat recovery, a system based on these processes can achieve a good overall thermal efficiency. The use of a single-liquid fuel methanol, also eliminates the inconvenience of having to carry two fuels and makes the HCCI combustion practically possible in vehicles. It is crucial to avoid a too early ignition to achieve higher load operation in HCCI engines, and hydrogen retards the auto ignition of DME considerably [29-30].

The ignition control effect by MRG is attributed to the hydrogen in MRG. Therefore, DME-reformed gases, which contain hydrogen, are also effective to control auto ignition of DME [31-33]. This report investigates the reaction mechanisms in the ignition control of hydrogen by using chemical kinetics analysis.

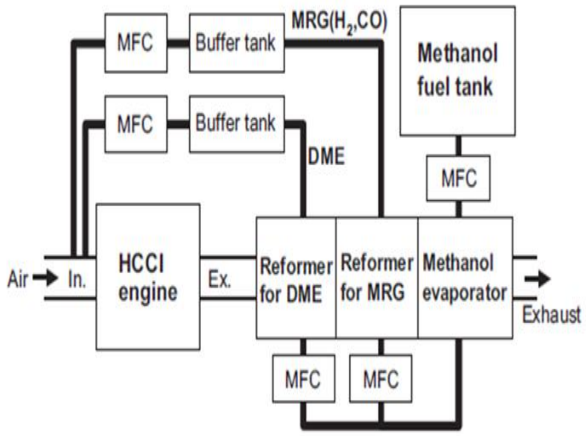


Fig. 1: Concept of HCCI combustion engine system fuelled with DME and MRG onboard-reformed from methanol

3.2 **Experiments and calculations**

The test engine used was a four stroke cycle single -cylinder engine with a bore of 85 mm, a stroke of 88mm and a compression ratio of 9.7. The fuel gases, dimethyl ether (DME; CH_3OCH_3) and hydrogen (H_2), were stored in high-pressure cylinders, and continuously supplied to the intake manifold of the engine as shown in Figure 2. Fuel flow rates were separately controlled using needle valves and measured by mass flowmeters (Oval). The in-cylinder pressure was measured with a piezo electric pressure transducer (AVL GM12D) installed in the cylinder head. For each experimental condition, the pressure data for 100 cycles were averaged and used to calculate the mean in-cylinder gas temperature, the indicated mean effective pressure, the indicated thermal efficiency and the apparent rate of heat release. Concentrations of CO and THC in the exhaust gas were measured with an NDIR analyser and an FID analyser respectively. The engine speed was set at 1000 rpm for all the experiments. The volumetric efficiency was controlled at 75% including fuel gases. The intake air was at room temperature without heating.

A chemical kinetics analysis using CHEMKIN II was performed to analyse the reaction mechanisms in the ignition control effect of hydrogen. The detailed reaction mechanism for DME oxidation reported by Curran et al. [32-33] was employed in SENKIN adiabatic calculations. Volume changes and initial conditions for the calculation were determined according to the engine experiments.

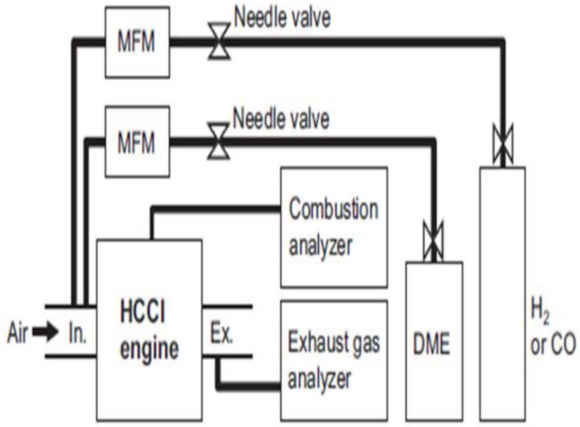


Fig. 2: Schematic diagram of experimental system

4. Results and Discussions

4.1 Experimental results of ignition control by hydrogen in HCCI combustion

Previous research [11] has shown that MRG, which consists of hydrogen and carbon monoxide, has a large ignition control effect on HCCI combustion of DME. Figure 3 shows the effects of hydrogen and carbon monoxide on HCCI combustion of DME. The DME amount was fixed at a value that gives the equivalence ratio of 0.27 without the addition of hydrogen or carbon monoxide. The mole fraction of hydrogen or carbon monoxide to DME was varied from 0 to 1.5. Equivalence ratios which is increased by the addition of hydrogen or carbon monoxide as indicated in the figures were calculated by the total supplied fuels. Figure 4 shows a comparison of results with hydrogen and carbon monoxide at a 40% DME fraction.

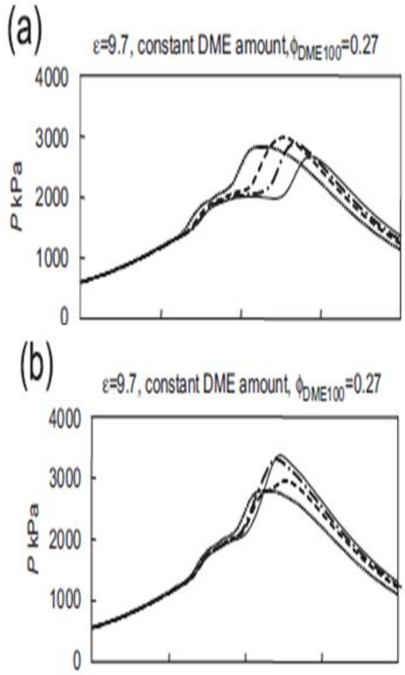


Fig. 3: Effects of hydrogen on auto ignition of DME: (a) hydrogen addition and (b) carbon monoxide addition

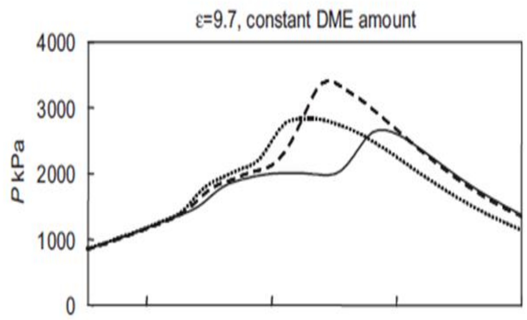


Fig. 4: Ignition control effects by hydrogen and carbon monoxide

Figure 5 shows the crank angles at the onset of the in-cylinder gas mean temperatures at the timings (TLTR, THTR) and cumulative apparent heat release during the low- and high temperature oxidation reactions (QLTR, QHTR). These values are derived from the experimental results shown in Figure 3. The figures show that the addition of hydrogen or carbon monoxide retards the combustion phase in spite of the increased equivalence ratio, and the effect is clearly larger in the hydrogen addition. Though the hydrogen addition retards the timing of the second heat release considerably, there is no remarkable change on the temperature at that timing. Hydrogen makes the first heat release slower and effectively delays the temperature rise during the low-temperature oxidation of DME. Due to the delayed temperature rise, the starting crank angle of the second heat release is considerably retarded in spite of the absence of any remarkable change in the starting temperature of the high-temperature oxidation reactions. There is almost no change in the cumulative apparent heat release during the low temperature oxidation QLTR, while the heat release during the high-temperature oxidation QHTR increases with the equivalence ratio [33-36].

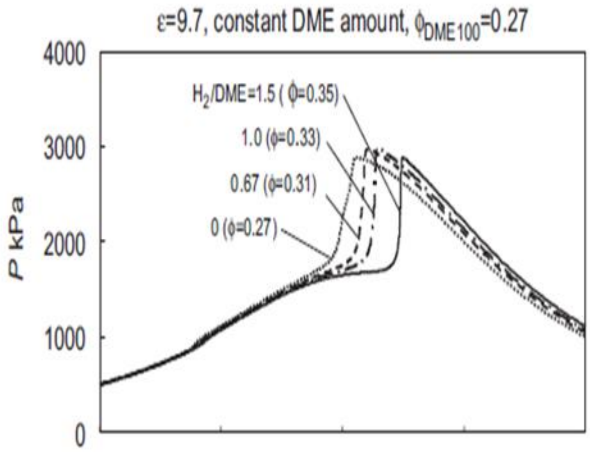


Fig. 5: Crank angles and in-cylinder gas mean temperature at the onset of low- and high-temperature reactions.

4.1.1 Reaction mechanism in the ignition control effect of hydrogen

Figure 6 shows the in-cylinder pressure and the temperature for HCCI combustion of DME and hydrogen that were calculated using CHEMKIN II with the DME oxidation reaction kinetics model by Curran et al. [32-33]. The calculated trends of the combustion phase retardation by the hydrogen addition are similar to the experimental results in Figure 3. The DME oxidation model by Curran et al, is qualitatively consistent with the experimental results in HCCI engines. The first step in the model is the production of CH_3OCH_2 from DME. Though hydrogen abstraction from DME by oxygen is the initiation reaction, but the largest part of the CH_3OCH_2 is produced by the following hydrogen abstraction reaction with OH in the low-temperature oxidation of DME [32-33]:

$$CH_3OCH_3 + OH \rightarrow CH_3OCH_2 + H_2O \quad (1)$$

The CH_3OCH_2 reacts with O_2 or a third body to produce HCHO, OH, and other species. The produced HCHO consumes OH in the following reaction:

$$HCHO + OH \rightarrow CHO + H_2O \quad (2)$$

Here, OH is a chain carrier in the low-temperature oxidation. As a result of the production and consumption of OH, when d[OH]/dt becomes negative, the chain branching terminates. When hydrogen and carbon monoxide are introduced to this low-temperature oxidation process of DME, they consume OH in the following reactions and suppress the consumption of CH_3OCH_3:

$$H_2 + OH \rightarrow H_2O + H \quad (3)$$
$$CO + OH \rightarrow CO_2 + H \quad (4)$$

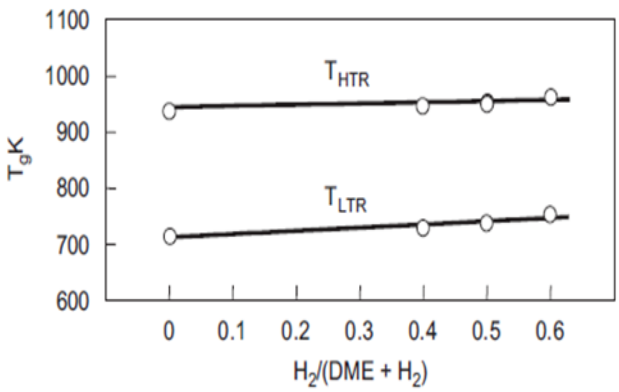


Fig. 6: Calculated effect of hydrogen on auto ignition of DME

Reaction (3) occurs at a 3–6 times higher rate than Reaction (4) in the temperature range from 700 to 1000K, where the low-temperature oxidation reactions occur as shown in Figure 7. This causes the differences in the ignition control effect of hydrogen and carbon monoxide. The hydrogen produced from the reactions quickly combines with O_2 to produce HO_2 and the HO_2 reacts with H to produce H_2O_2:

$$H + O_2 + M \rightarrow HO_2 + M \quad (5)$$
$$HO_2 + H \rightarrow H_2O_2 \quad (6)$$

However, HO_2 and H_2O_2 are much less reactive than OH during the low-temperature oxidation process, while H_2O_2 dissociates to OH in the high-temperature oxidation process. There is another reaction between H and O_2 to produce OH and O:

$$H + O_2 \rightarrow OH + O \quad (7)$$

If this reaction is dominant, the increased OH promotes the low temperature oxidation.

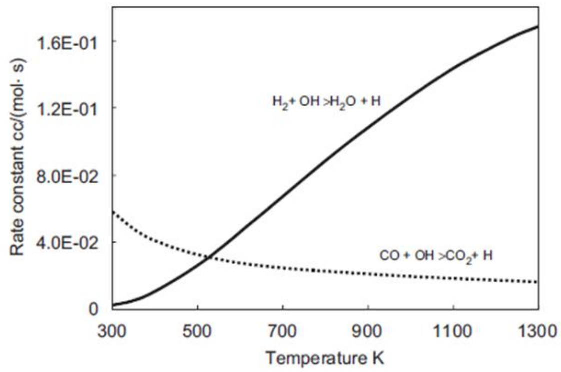


Fig. 7: Rate of constant OH consumption by hydrogen and carbon monoxide.

Figure 8 shows the rate constants for the two reactions between H and O_2 versus temperature. Reaction (5), producing HO_2, is dominant over Reaction (7), producing OH and O, in the low-temperature oxidation regime. Therefore, H radicals produced in Reactions (3) and (4) do not increase the OH concentration. Overall, the introduced hydrogen converts the highly reactive OH into the less reactive HO_2 and H_2O_2 [32-35].

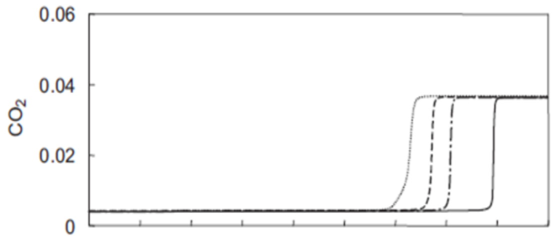


Fig. 8: Rate constants for reaction of H and O_2

Figure 9 shows the influence of hydrogen addition on the mol fractions of chemical species and temperature against crank angle in HCCI combustion of DME. The calculations are for H_2/DME mol fractions of 0, 0.67, 1.0, and 1.5 with a constant DME amount which brings an equivalence ratio of Ø= 0.27 in the neat DME case. The increase in hydrogen decreases the OH concentration remarkably and retards the first peak of the OH concentration, and this subsequently retards the consumption of CH_3OCH_3 and formation of HCHO. An increase in the hydrogen fraction increases the peak concentrations of HO_2 and just before the high-temperature oxidation. The high-temperature oxidation starts with the dissociation of H_2O_2 to 2OH and oxidizes intermittent species to H_2O and CO_2. The H_2O_2 decreases and OH increases rapidly at around 1000K regardless of the hydrogen fraction; along with the abrupt increase in the OH concentration, CH_3OCH_3, CO, and H_2 are rapidly consumed producing H_2O and CO_2 here. The cases with higher hydrogen fraction tend to have more OH during the high-temperature oxidation. This is because of the higher concentration of H_2O_2 during the low-temperature oxidation and combustion process of hydrogen itself. Almost all the introduced hydrogen is consumed during the high-temperature oxidation, and the consumption of it during the low-temperature oxidation is quite small. Here, the combustion processes of hydrogen can be represented by the following reactions:

$$H_2 + OH \rightarrow H_2O + H \quad (3)$$
$$H + O_2 + M \rightarrow HO_2 + M \quad (5)$$
$$H + O_2 \rightarrow OH + O \quad (7)$$
$$O + H_2 \rightarrow OH + H \quad (8)$$

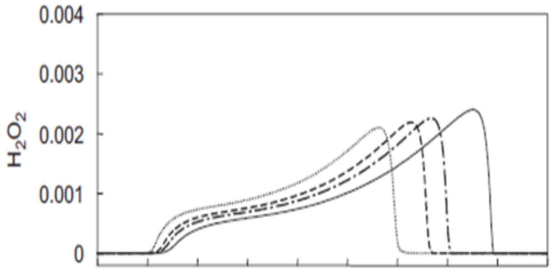


Fig. 9: Calculated mole fraction of species in DME oxidation with different hydrogen fractions

Reaction (7) producing OH and O is slower than Reactions (3) and (8), and is the rate-determining process. However, Reaction (5), another reaction between H and O2, is dominant over Reaction (7) in the temperature range below 1250K as shown in Figure 8, and the combustion process of hydrogen does not proceed during the low-temperature oxidation of DME. The strongly retarded low-temperature oxidation brought about by the addition of hydrogen in the experiments, can be attributed to the consumption of OH by hydrogen. The reduced OH concentration retards the H abstraction from CH_3OCH_3 by OH and subsequent exothermic reactions during the low temperature oxidation. While hydrogen does not affect the starting temperature of the high temperature oxidation, the onset of the second heat release is retarded due to the delayed rise in temperature [35-38].

4.2 **Gasoline engine exhaust gas recirculation**

4.2.1 **EGR system**

Cylinder charge dilution with exhaust gas can be classified into internal EGR and external EGR. With external EGR, exhaust gas is taken from the exhaust port and supplied into the inlet port. Internal EGR is achieved by increasing NVO during exhaust stroke, which requires an improved cam that can rapidly switch cam profiles to achieve any variable valve timing, otherwise it is impossible to independently and effectively control EGR ratio. This, greatly limits the application of internal EGR. As a result, external EGR has become widely used on today's automobile engines. External EGR has a relatively low cost. It only needs to use dedicated EGR control valve, which can control EGR rate effectively under all work conditions of engine [28-32]. Only external EGR will be discussed in this paper.

External EGR system consists of EGR pipe, EGR valve and EGR cooler (cooled EGR). Exhaust goes through EGR valve and EGR cooler, and then enter intake manifold. Constant coolant goes through EGR cooler. EGR valve can be adjusted to get various EGR rates. Pipe material is stainless steel to avoid transfer of engine vibration to exhaust system, and then to measuring instruments [28].

For multi-cylinder engine, important parameters in designing EGR systems are good homogeneous EGR distribution to each cylinder and good dynamic response.

The recirculated exhaust gas can be supplied to the engines either centralized or decentralized as schematically shown in Figure 10. For centralized exhaust gas recirculation systems, EGR valve is far away from inlet valves, normally at the entrance to intake manifold, where collector provides a good mixture of exhaust gas and fresh air, and thus optimal distribution to each cylinder.

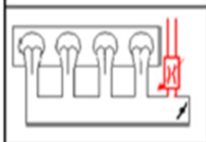
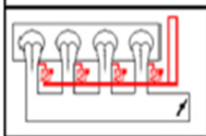
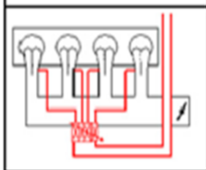
	EGR supply	Number of EGR valves
	central	1
	decentral	1 per cylinder
	decentral	1 multiple valve

Fig. 10: Centralized and decentralized EGR

The decentralized EGR system directly supplies exhaust gas to inlet valves. Two different applications are possible: one application is one valve for each cylinder, and the application is using a multiple valve. With the two applications, it is ensured that, with closed EGR valve, crossflow is impossible between intake manifold runners. The benefits of decentralized EGR system are optimized EGR distribution and good dynamic response. However, the location and orientation of EGR port and valve may influence the in-cylinder EGR distribution significantly, and thus influence fuel stratification by producing an extra flow of recirculated exhaust gas within combustion chamber. This influence must be carefully evaluated when using this application. In addition, the sealing of EGR valve or port and assembly constraints in the vicinity of cylinder cover is also an important factor that constrain the application of this EGR strategy [30]. When EGR is applied, engine intake consists of fresh air and recycled exhaust gas. EGR (%) usually represents the percentage of the recirculated exhaust gas. The percentage of exhaust gas recirculation is defined as the percentage of recirculated exhaust in total intake mixture [31]. Whereby, [mi = ma + mf + mEGR] is the mass of total intake mixture and [mEGR] is the mass of the EGR.

$$\text{ERG (\%)} = (m_{EGR}/m_1 \times 100)$$
$$\text{EGR (\%)} = \frac{(CO_2)_{intakegas} \_ (CO_2)_{ambient}}{(CO_2)_{exhaustgas} \_ (CO_2)_{ambient}}$$

The calculation of recirculation degree was defined by relating the amount of CO_2 in the intake manifold with that in exhaust pipe, where the CO_2 concentrations used in the expression are such that $(CO_2)_{\text{ambient}}$; that was measured in the environment, $(CO_2)_{\text{intakegas}}$ was checked in the intake manifold and $(CO_2)_{\text{exhaust}}$ gas was measured in the exhaust gases [32-33].

4.2.2 **EGR vs. NOX**

As it is well known, NOx is generated under the condition of high-temperature and oxygen–rich environment. The NOx emission of internal combustion engine is mainly NO, but it will be oxidized into NO_2 quickly after entering air. The formation of NO is increased dramatically in exponential function with temperature increase. When temperature is below 1800 K, the rate of NO formation is very low, but it will reach a very high speed when the temperature reaches 2000 K. It can be generally considered that for every 100 K increase of temperature, NO formation rate will nearly double. The formation mechanism of NO is different from that of HC and CO. NO is not the result of incomplete combustion of mixture. Its generation relates with the proliferation of mixed combustion, concentration distribution of flame and heat transfer, thus reaction mechanism is very complex. Basically, nitrogen hardly exists in gasoline. The formation of NOx originates from thermal reaction of N_2 and O_2 in the air in high temperature combustion. NO is formed inside combustion chamber in post-flame combustion process in high temperature region. NO formation and decomposition inside combustion chamber can be described by extended Zeldovich Mechanism [32-39]. The principal reactions near stoichiometric fuel–air mixture governing NO formation from molecular nitrogen are:

$$\boldsymbol{N_2} + \mathbf{O} \leftrightarrow \mathbf{NO + N}$$
$$\mathbf{N} + \boldsymbol{O_2} \leftrightarrow \mathbf{NO + O}$$

N + NH ↔ NO + H

$$\frac{[NO]}{dt} = \left(\frac{6 \times 10^{16}}{T^{0.5}}\right) \exp\left(\frac{-69,096}{T}\right) [O_2]_e^{0.5} [N_2]_e \text{ mol s/cm}^3$$

The initial rate, controlling NO formation can be described in the expression, [NO] denotes molar concentration, and $(O_2)_e$ and $(N_2)_e$ denote equilibrium concentrations [34-38]. The sensitivity of NO formation rate to temperature and oxygen concentration is evident from this equation. Hence, in order to reduce NOx formation inside combustion chamber, the temperature and the oxygen concentration in combustion chamber need to be reduced.

In gasoline engine, EGR is the most principal technique in reducing NOx emission. The main constituents of EGR are N_2, H_2O, O_2 and CO_2. It is well known that CO_2 exerts three effects when being introduced into combustion process. They are:

(1) Thermal effect: Exhaust gas consists of gases of two-atom and three-atom, and the heat capacity of three-atom gas increases faster in combustion process. Due to the increase in heat capacity of the oxidizer, flame temperature is reduced;

(2) Dilution effect, which is the results from the reduction of oxygen concentration in the main stream of the oxidizer and from the reduction of reactive species in combustion process, which, in turns, reduces their collision frequency;

(3) Chemical effect, since CO_2 is an active species and thus participates chemically in the combustion process; these three effects have been studied in the literature [37-39]. Engine tests have demonstrated that NOx is greatly suppressed when the O_2 concentration in combustion chamber is reduced. Figure 11 shows that the reduction in O_2 concentration in the cylinder suppresses NOx emission. The figure shows the combustion in two GDI fuel sprays, one with and the other without EGR. In GDI engine, when using EGR, some O_2 in the cylinder is replaced with exhaust as, and thus local O_2 concentration in the cylinder becomes lower. With the local O_2 concentration reduced, a given amount of fuel will have to diffuse over a wider area before encountering sufficient O_2 for a stoichiometric mixture to be formed. Now, for a given amount of fuel, this larger area contains not only stoichiometric mixture but also an additional quantity of CO_2, H_2O and N_2. These additional gases absorb the energy released by the combustion, leading to lower flame temperature and then lower NO_x generation [9,40].

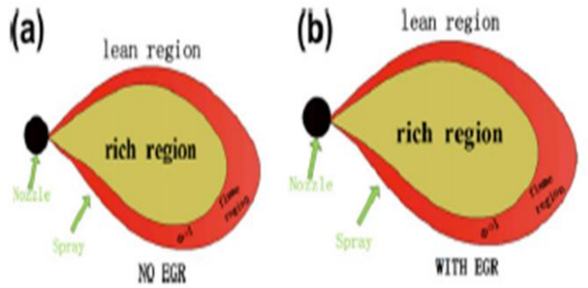


Fig. 11: Increase in volume occupied by spray flame with use of EGR.

4.2.3 **EGR in GDI engines and in PFI engines**

PFI engines work under stoichiometric condition, and EGR is employed primarily to reduce throttling loss at part load range, thus to reduce fuel consumption, and secondarily, to reduce NOx emission levels. In order to keep the same torque and power output after introducing EGR in gasoline engine, further opening of engine throttle is necessary to raise the trapped charge density. This can reduce pumping loss and improvement of fuel economy compared with that when no EGR is used. The improvement in fuel consumption with increasing EGR is due to three factors: first, reduced pumping work, as EGR increases at constant brake load (fuel and air flows remain almost constant, hence, intake pressure increases); second, reduced loss of heat transferred to cylinder wall since burned gas temperature is decreased significantly; third, a reduction in the degree of dissociation in the high temperature burned gases, which allows more fuel's chemical energy to be converted to sensible energy near TDC [41-42]. It is worth to mention that when PFI engine works at full load, the throttle has reached the WOT condition. In this case, the throttle cannot be opened more to increase intake density. Therefore, boosting intake pressure is necessary to gain the same level of torque and power output. If supercharge is not used to increase intake density, the power loss will increase with the increase of EGR ratio at full load [4].

Furthermore, in a conventional PFI engine, EGR trends to influence combustion stability and investigations have confirmed that both ignition delay and combustion period are extended with EGR due to the associated decrease in laminar flame speed. Consequently, it may be appropriate to use EGR in combination with other techniques [42-44]. In general, a larger NOx reduction can be realized with EGR in GDI engines than that in either PFI engines or diesel engines. For GDI engines, the available fuel–air mixing time is comparatively longer than that of diesel engines, and, as a result, EGR role is more effective and NOx emission can be further reduced. In a GDI engine, since there is no throttling effect, the engine works with lean or even ultra-lean mixtures and the in-cylinder AFR is higher, and the introduction of exhaust gas replaces part of fresh air directly. For a given torque and power output, the amount of fuel that the engine supplies must be constant. Therefore, the in-cylinder AFR decreases with the increase of exhaust gas, which can reduce NOx emission effectively. Although in GDI engine, lean-NOx after-treatment technologies can be used to reduce NOx emission, EGR was considered the only feasible way to reduce NOx [7]. In addition, EGR ratio as high as possible are required to keep GDI engine working under stoichiometric condition, so that TWC can work normally, and then we can use it to reduce HC and CO emission.

Table 1 shows the basic differences between GDI engine, EGR system and PFI engine system for different applications. The EGR application ranges, of PFI engines and of GDI engines are different. Figure 12 shows the ranges of conventional PFI engines in comparison with the various working modes of GDI engines. In Figure 12, in homogeneous mode of GDI engines, the use of recirculated exhaust gas corresponds to that of conventional PFI engines.

Table 1: The basic differences between EGR systems for the different applications

	EGR System for GDI Engines	EGR System for PFI Engines
Target	First, reduction in nitrogen oxides Second, reduction in fuel consumption	First, reduction in fuel consumption Second, reduction in nitrogen oxides
Max. EGR rate	• 50%, Stratified mode • 25%, Homogeneous mode	25%
Max. exhaust temperature in the operating range	• 450 °C • (650 °C, homogeneous mode)	650 °C
EGR cooling	Under discussion	Required
Other requirements	• Good dynamics • Good resolution capability • Good distribution	Reduction in power losses at high EGR ratios

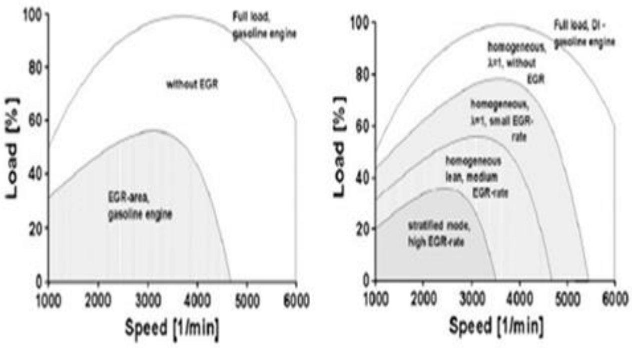


Fig. 12: EGR ranges within the characteristic map

Due to the higher EGR tolerance of the engine, EGR rates are also higher in the case of homogeneous work. In stratified mode, EGR rates comparable to those of diesel engines, and are attainable at much higher rate than that of PFI engines. For a GDI engine using charge stratification, the mixture near spark gap is ideally either stoichiometric or slightly richer. Furthermore, mixture preparation can be improved due to EGR heating effects. Robust combustion is therefore possible with a much higher level of EGR than that with homogeneous combustion. However, there is an associated compromise between a significant NOx reduction and a simultaneous degradation in both HC emission and in fuel consumption due to the degradation of combustion quality [30]. Moreover, in a good EGR system, dynamic response is mandatory to avoid drivability problems in transition from one work mode to the other. When EGR is used in GDI engine, providing appropriate amount of EGR has always been a design challenge, since a relative higher EGR mass flow rate must be metered and the flow must be distributed uniformly to individual cylinders under a much lower pressure difference with that of traditional PFI engines [42-45]. The flow of excessive EGR may require a moderate level of intake vacuum, which will cause great pumping loss, while GDI engines is supposed to reduce such loss substantially

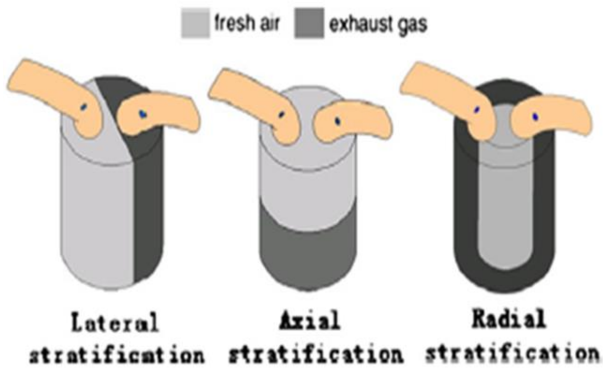


Fig. 13: EGR-stratification modes

4.2.4 **Stratified EGR**

Whether in PFI engine or in GDI engine, homogeneous EGR system reduces laminar flame speed, which will cause reduce of burning speed, increase of HC emission, cycle-by-cycle variations aggravation, and difficulties in achieving steady-state combustion, and even causing fire [31,45]. High-diluted stratified EGR charge to separate air/fuel mixture in intake and compression strokes can be a good practice to generally overcome the above difficulties. The stratified exhaust gas recirculation is characterized by separating EGR air and fresh air in combustion chamber. Due to a minimized exhaust gas concentration at spark plug region, flame propagation is improved compared with homogeneous EGR, EGR compatibility is increased. However, the flow structure in combustion chamber is extremely complex, so to realize complete separation of air and exhaust gas is very difficult. Another challenge in this system is how to realize the stratification in intake stroke and how to maintain the stratification in compression stroke prior ignition.

Different possibilities of exhaust gas stratification are shown in Figure 13, including radial stratification, lateral stratification and axial stratification. Lateral EGR stratification, consisting of air/fuel mixture at the side of cylinder intake, and of only EGR at the exhaust

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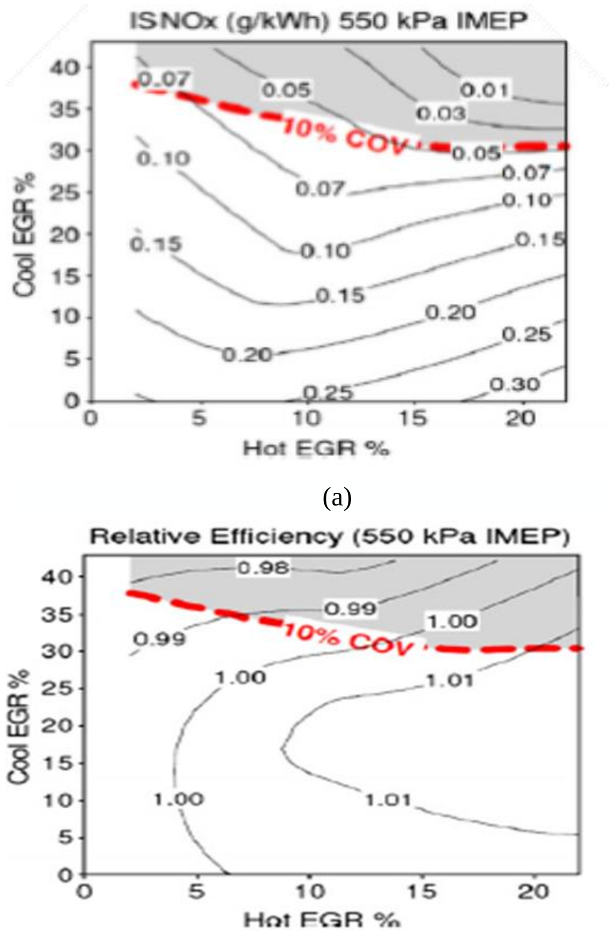
side may separate them well. However, this method requires strong tumble intensity to maintain vertical flow momentum. When the piston moves upwards to TDC, the tumble flow is easily destroyed by turbulent and/or squeeze flow. Axial EGR stratification divides the cylinder into top air zone and bottom EGR zone. The injected fuel can penetrate into the pure air zone, mix with the air, and then be ignited and burned at this zone. In this method, a vertical flow, such as tumble flow or squeeze flow, may make the gases easily mix. As a contrast, radial EGR stratification seems to be the most appropriate flow structure for compression stroke. Since the central air–fuel cylinder and the outer EGR tubular cylinder are concentric with the engine cylinder, as the piston moves upwards in compression stroke, the two cylinders will be compressed in axial direction. Particularly, if both EGR and air are swirling in the same direction, the conservation of angular momentum may make the two zones readjust their interface location to reach new force balance. Radial stratification can sustain much longer time towards the end of compression stroke [46-50]. In summary, radial stratification is the most appropriate method for getting stratification of intake and exhaust.

4.2.5 **Comparison between cooled EGR and hot EGR**

The operation, that exhaust gas is recycled to intake directly, is called hot EGR, and the operation, that EGR after cooling is applied to recycled exhaust, is called cooled EGR [15]. The engine using hot EGR can use the high temperature exhaust to heat the intake, promote combustion, and thus improve the thermal efficiency. While cooled EGR increases intake density, thereby increases volumetric efficiency of engine. At the same time, the decreased temperature can further reduce NOX emission, but HC emission and cycle-by-cycle variations are increased compared with that of hot EGR. The comparison of the characteristics between cooled EGR and hot EGR is given in Table 2 [51-52].

Table 2: Comparison of the characteristics between cooled EGR and hot EGR

	Cooled EGR	Hot EGR
Characteristics	• Lower NO_X values • Better knock suppression • Complex structure • Higher cost	• Lower combustion duration • Lower HC values • Simple structure



(a)

(b)

Fig. 14: Comparison between hot EGR and cold EGR (the maximum EGR tolerance is indicated by the highlighted 10% COVIMEP contour. All data is for lean conditions with constant fuel flow and constant diluents (air + EGR) low) [32]

The heating effect of hot EGR improves the combustion temperature and then gains improvements in fuel–air mixing, therefore HC emission will be reduced. With further increment of EGR ratio, HC emission began to deteriorate, as the introduction of EGR narrows the flammability limits. Figure 14a shows NOx emission decreases significantly with increasing percentages of cooled EGR as a result of a lower combustion temperature presented in this case. While increasing percentage of hot EGR tends to lead to slight increase in NOX emission. Figure 14b shows relative efficiency as a function of cooled and hot EGR in percentage. The efficiency decreases with increased cooled EGR. This can be attributed to the changes in combustion temperature, and reduced temperature leads to a reduction in combustion speed and then in thermal efficiency. At present, despite of the advanced combustion systems and emission control systems, cooled EGR is still the most effective measure to reduce NOx emission [50-53]. When using EGR, the intake composition changes to a large extent. Since the exhaust temperature reduces the intake density, the exhaust gas must be cooled to maintain a high volumetric efficiency to avoid the increase of heat loss.

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4.2.6 **Knock suppression using cooled EGR**

In engine revolution, EGR technique was firstly adopted in diesel engines in order to limit thermal NOx formation rate by reducing combustion chamber temperature thanks to the dilution of fresh charge with a certain amount of exhaust gases recycled at engine intake. Some studies also suggest that appropriate EGR ratio can reduce combustion noise by lower the pressure rise rate and the pressure high frequency oscillation magnitude [34,35]. But with the growing energy and environmental issues, various countries in the world have developed more stringent regulations on vehicle emission and fuel economy to drive gasoline engine to be developed towards downsizing direction. Turbo-charged spark ignition engines are becoming increasingly popular in the world market due to their compactness and high power density. However, due to the high power density of turbo-charged engine, knock combustion and high exhaust gas temperatures constitute problems at high loads [19]. As a result, knock control is becoming increasingly important. In the past century, many scholars conducted studies on knock [36–39]. Knocking combustion is generally accepted as the effect of auto-ignition of the portion of cylinder charge where propagating turbulent flame has not reached. This view had been proved by experimental evidence. The pressure/temperature of end-gas end is controlled by the displacement of piston and the expansion of combusted gases. With advanced facing of the combustion, end gas temperature increases, and finally, the reaction rate in end gas reaches a level at which spontaneous ignition occurs. Another view is based on the interpretation of flame propagation theory. It’s saying that knock is the result of flame acceleration in homogeneous mixture. When flame speed is higher than sound speed, knock occurs [40]. When knock occurs, pressure and heat transfer increase sharply, which will cause serious damage to engine. Normally, the number of knocking cycles is on-line determined using band-pass filtered cylinder pressure signal. For each cycle, the amplitude of the signal from oscillating cylinder pressure is compared with a reference voltage. If the amplitude exceeds reference voltage, the cycle indicates knocking [48-52]. The paper presented here does not care about the causes of knock, but only about knock suppression methods.

Several knock suppression methods have been studied in related research. These methods can be divided into three categories [5-57]. The first is to decrease effective compression ratio or delay ignition timing to limit cylinder pressure. The second is to inject excessive fuel in the mixture to decrease AFR. The third, an alternative and efficient solution, emerging as a promising technology to limit knocking, is through dilution, such as cooled EGR. To reduce exhaust temperature and to limit knock, the most commonly used method is excessive fuel to decrease combustion temperature. Under the condition of fuel enrichment, HC and CO emissions will increase, and TWC can only work under stoichiometric condition, which causes problems of fuel economy and emission. However, using EGR as a diluents allow the use of an overall stoichiometric composition of the charge, thus insures a good conversion of all emissions. Therefore, using cooled EGR instead of excessive fuel to inhibit knock

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and to reduce emission is an effective measure in gasoline engine [41–43].

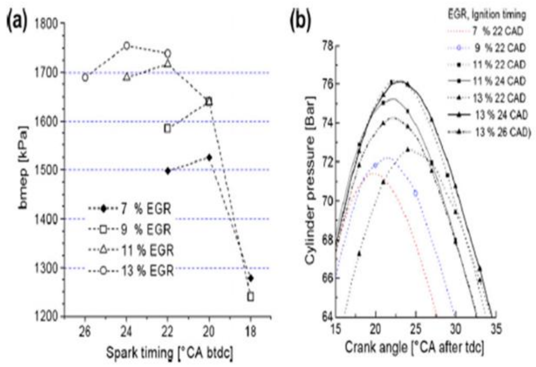

Fig. 15: Maximum BMEP and cylinder pressure as a function of EGR and the ignition angle (engine speed of 4000rpm, EGR temperature of 120±4°C, charge air temperature of 40±2°C) [48-52]

In Figure 15a, maximum BMEP, as a function of the number of the amount of EGR and ignition angle, is presented. For these tests, maximum BMEP was limited by knocking combustion and exhaust temperature. The knock detection was set to detect cycles with amplitudes of the oscillating cylinder pressure above 250 kPa, and the exhaust temperature was limited to 960°C [39]. We can see clearly that after increasing EGR ratio from 7% to 13%, ignition angle and maximum BMEP are increasing. But with further advance ignition, BMEP will not increase, which is limited by knocking combustion. The increase of EGR ratio will reduce this limitation. This is because, increase of EGR, increases the mass in the cylinder, and therefore lowers gas temperature, and subsequently, more fuel can be combusted before reaching temperature limit. Therefore, it can further widen ignition advance angle. Figure 15b shows cylinder pressure traces limited by knock as a function of EGR ratio. In the figure, all ignition angles are the angle before top dead center. The peak pressure tolerance of the engine is slightly increased with the increase in EGR. The possible reason is that the dilution and thermal effect of EGR reduce combustion temperature and inhibit knocking combustion, therefore allow a greater cylinder pressure.

With high load, the major role of EGR is to replace fuel enrichment to inhibit knock. When EGR ratio is 10% with a k of 0.9, the engine has BMEP values of 1645 kPa and 1673 kPa respectively (calculated from Figure 16). That is to say, if knock does not occurs, using high EGR ratio can get the same level of power output compared with that of fuel enrichment, and by using EGR, it can ensure stoichiometric combustion, so this will greatly improve fuel economy and lower emission.

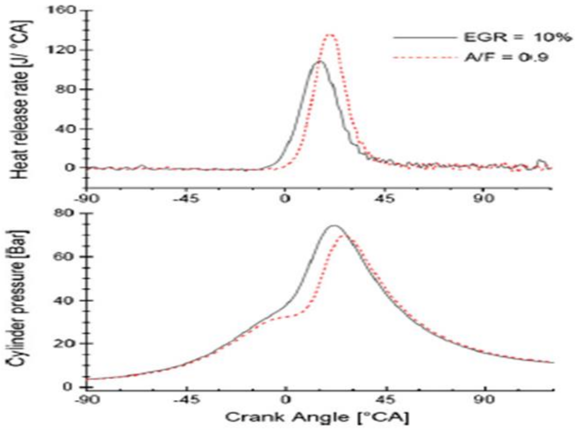

Fig. 16: Comparison of combustion characteristics for 11% fuel enrichment and 10% EGR (engine speed of 4000 rpm, boost pressure of 90kpa, spark ignition timing of 6.6°CA BTDC) [48-52]

Figure 17 shows CO, HC and NOx emissions before catalyst at various EGR rates. The values from fuel enrichment case are in the upper part of the figure. The major effect using EGR instead of fuel enrichment is that CO is significantly decreased from over 4% with fuel enrichment to a more reasonable value of 0.8% with a stoichiometric mixture. The negative aspect is that NOx is increased with a stoichiometric mixture. But with the increase of EGR rate, NOx decreases gradually.

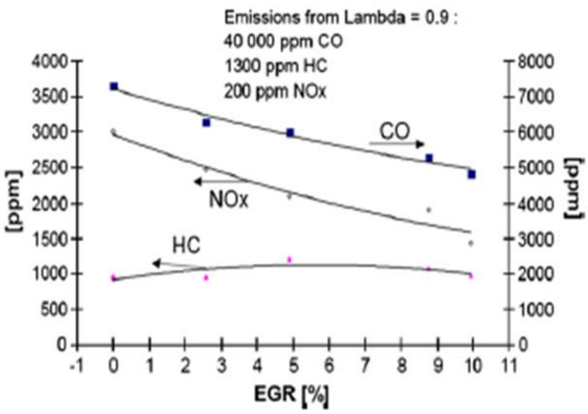

Fig. 17: Engine emission before catalyst from various EGR rates (engine speed of 4000 rpm, stoichiometric fuel-air mixtures) [48-52]

Figure 18 shows intensity of knocking cycles at 5000 rpm and under full load conditions. The fuel rich cases show consistently high average values of knock intensity, while the EGR cases display quite a number of high knock intensity, but average knock intensity is low. The possible explanation is that, the greater instability of the cycles has a rapid initial combustion phase, leading to higher cylinder pressure and consequently to higher end-gas temperature.

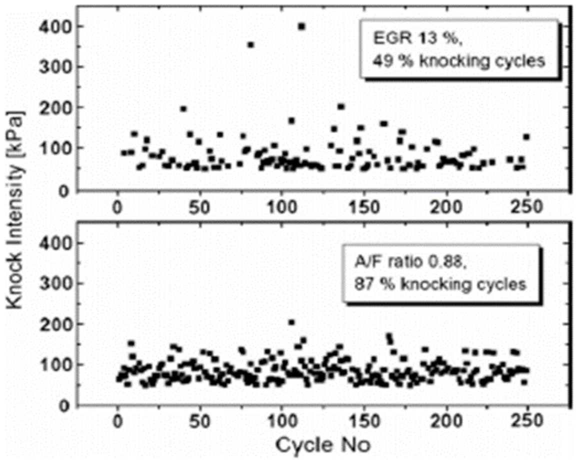

Fig. 18: Knock intensity of knocking cycles (engine speed of 5000 rpm, full load condition, 8ºCA BTDC spark ignition timing of λ = 0.88 and 20ºCA BTDC spark ignition timing of 13% EGR) [48-52]

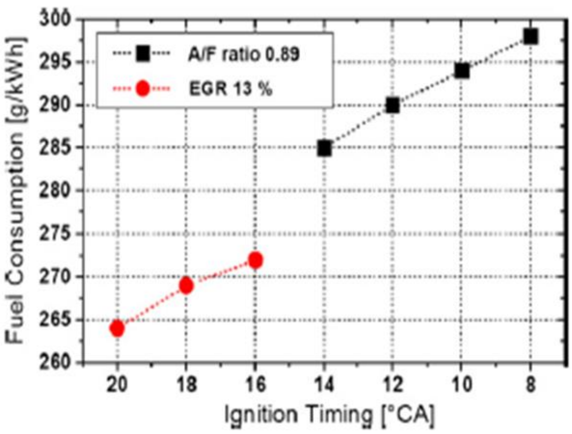

Fig. 19: Comparison of fuel consumption at high load (IMEP ≈ 1650kPa, knock intensity 62kPa at ignition timing 20ºCA BTDC in case of EGR, and 81kPa at ignition timing 8ºCA BTDC in case of fuel enrichment) [48-52]

When knock limit is set to 50 kPa, the average knock intensity of the knocking cycles in the two cases are the same, but in the fuel rich cases, 87% of the cycles are knocking, while only 49% exhibits knock with EGR. Consequently, using EGR to replace fuel enrichment achieves relatively good results. In the full load cases with a rich mixture, fuel consumption is very high. Naturally the consumption is reduced when fuel enrichment is replaced by EGR. This is shown in Figure 19, where fuel consumption in a case with 13% EGR is compared with in a case with k = 0.9. With EGR, the fuel consumption at a comparable knocking intensity reduces by more than 10%.

4.2.7 **Implementations of EGR for turbocharged gasoline engines**

The implementation of EGR is simple for naturally aspirated gasoline engines, since exhaust tailpipe backpressure is normally higher than intake pressure.

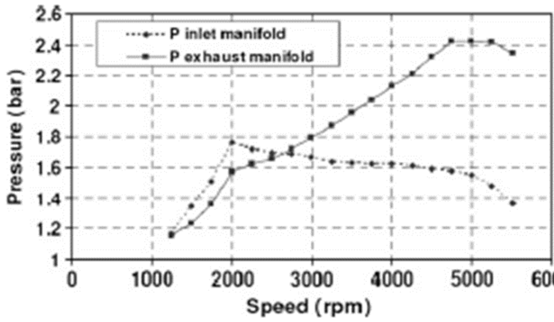

Fig. 20: Low pressure loop EGR and high pressure loop EGR.

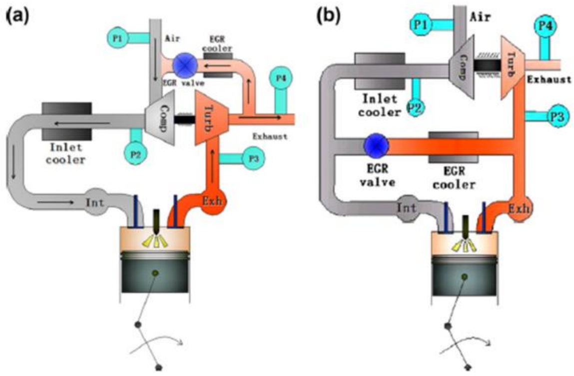

Fig. 21 : Clean vs. dirty EGR circuits

With the development of downsized gasoline engine, turbocharged gasoline engine is becoming increasingly popular in the market. The implementation of EGR is, therefore, more difficult, which leads to a coexistent problem in engines. There are two different schemas are shown in Figure 20 [48-52].The first structure is LPL EGR, as shown in Figure 20a. Exhaust gas goes into turbine at first, and then into compressor together with fresh air. Tailpipe pressure, P4 is larger than compressor inlet pressure, P1, so a positive difference between turbine outlet and compressor inlet is generally available. In the structure with low pressure, pressure difference in EGR circuits is very low, therefore a high permeability for the EGR valve and EGR cooler is required to ensure sufficient EGR flow. Furthermore, EGR cooler must have high efficiency to ensure the efficiency of compressor, otherwise it will cause intake air overheat and reduce volumetric efficiency. However, conventional compressors and inter-coolers are not designed to ensure the temperature and fouling of exhaust, which limits the application of LPL EGR.

LPL EGR is divided into two structures: clean EGR and dirty EGR, as shown in Figure 21a and b. The impact of the two structures on compressor is quite different. When using clean EGR, exhaust goes into TWC and then goes through compressor together with fresh air. The ceramic particles from TWC may corrode the compressor wheel, therefore additional coating is required to protect the wheel. When using dirty EGR, the major difference is the presence of sticky hydrocarbons, which might result in additional necessity of component protection. On the other hand, dirty EGR will also bring some additional benefit [45]:

- Improved rate and efficiency of combustion due to re-burn of hydrocarbons, H_2 and CO in the cylinder, and improved fuel economy.
- Reduced pumping loss as a result of improvement of pressure drop across the turbine.
- Improved EGR cooling due to lower enthalpy pre-catalyst (based on exothermic reactions in the catalyst).

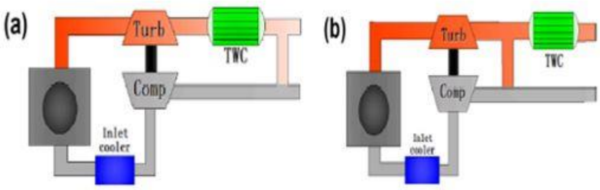

Fig. 22: Inlet and exhaust pressure vs. speed of the engine [48-52]

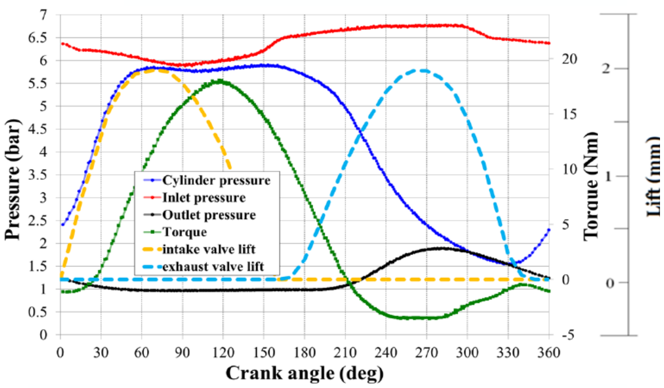

Fig. 23: Cylinder pressure, inlet and outlet pressure, torque variation and valve movement in one engine cycle [58]

In order to ensure compressor durability, it is necessary to conduct more in-depth theoretical and experimental studies to resist corrosion in compressor. The one that has wide applications is the second structure, as in Figure 21b. In this structure, exhaust gas is from the upstream of turbine to the downstream of compressor or of intercooler. However, such HPL EGR must ensure that turbine upstream pressure is sufficiently higher than boost pressure. That is to say, P3 > P2, otherwise we cannot acquire enough exhaust gas being introduced into intake air. Especially under low-speed and high-load condition, exhaust backpressure is lower than intake pressure, as shown in Figure 22. Therefore, in this case, exhaust cannot go into intake naturally, thus remedies must be made by either increasing turbine upstream pressure P3 or reducing boost pressure P2. The first method is to control the intake throttling or exhaust throttling by installing a throttle valve in the downstream of compressor or upstream of turbine. Another widely used method is to use VGT to provide desired EGR driving pressure without substantially sacrificing the performance of turbocharged engine. The shrinking of the flow passage of turbine nozzles will increase turbine upstream pressure P3 and reduce boost pressure P2.

4.3 **Diesel engine exhaust gas recirculation**

4.3.1 **Implementations of EGR**

The implementation of EGR is straightforward for naturally aspirated Diesel engines because the exhaust tailpipe backpressure is normally higher than the intake pressure. When a flow passage is devised between the exhaust and the intake manifolds and regulated with a throttling valve as shown in Figure 24, exhaust gas recirculation is established. The pressure differences generally are sufficient to drive the EGR flow of a desired amount, except during idling whilst a partial throttling in the tailpipe itself can be activated to produce the desired differential pressure. If the exhaust gas is recycled to the intake directly, the operation is called hot EGR. If an EGR cooler is applied to condition the recycled exhaust, it is called cooled EGR.

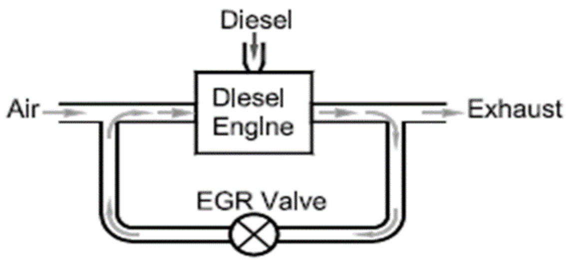

Fig. 24: Exhaust gas recirculation.

Modern Diesel engines, however, are commonly turbocharged, and the implementation of EGR is, therefore, more difficult. A low pressure loop EGR, as shown in Figure 25, is achievable because a positive differential pressure between the turbine outlet and compressor inlet is generally available, ðP4 _ P1Þ > 0. Furthermore, tailpipe pressure P4 can be elevated by partial throttling, that ensures sufficient driving pressure for the EGR flow. However, conventional compressors and inter-coolers are not designed to endure the temperature and fouling of Diesel exhausts. In general, the low pressure loop approach of EGR is not applicable except for exhaust gas designated compressors. Efforts have also been made to route exhaust from the turbine outlet to the inter-cooler outlet directly, by-passing the compressor [28]. Although it circumvents the exhaust fouling problem, an independent EGR pump becomes imperative to counteract the boost pressure. Special EGR pumps are needed to withstand the exhaust heat and fouling, in addition to the substantial pumping power requirements[50-54].

Although options are available, the preferred practice is to recycle the exhaust gas from upstream of the turbine to downstream of the compressor (or downstream of the inter-cooler if applicable), i.e., a high pressure loop EGR as shown in Figure 26. The compressor and inter-cooler are, therefore, not exposed to the exhaust. However, such high pressure loop EGR is only applicable when the turbine upstream pressure is sufficiently higher than the boost pressure, i.e., if ðP3 _ P2Þ > 0 prevails. In case the pressure difference cannot be met with the original matching between the turbocharger and the engine, remedies must be made by either increasing the turbine upstream pressure or reducing the boost pressure

Even though a variety of measures can be taken, the leading contender is to use a variable geometry turbine (VGT) that can effectively provide the desired EGR driving pressure without substantially sacrificing the performance of the turbocharged engine[50-54].

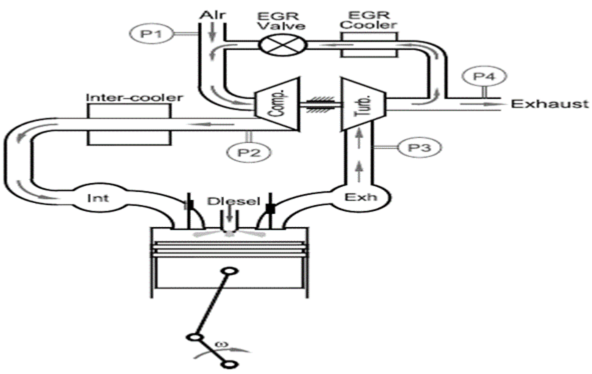

Fig. 25: Low pressure loop EGR

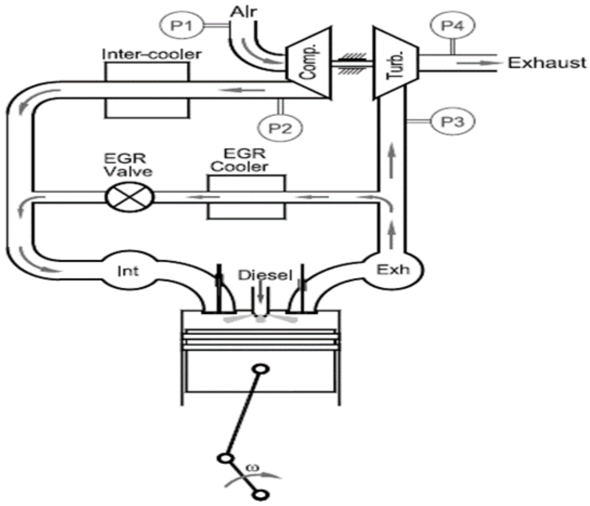

Fig. 26: High pressure loop EGR.

4.3.2 **Laboratory simulated EGR**

In addition to actual EGR, the effect of EGR can be simulated empirically with gas add-on or synthetic gas methods, Figure 27 and Figure 28, are especially useful for fundamental EGR studies. In a simulated EGR operation, an EGR like intake mixture, is actually synthesized with fresh air and/or external storage gases. Such simulated approaches can reproduce the essential characteristics of EGR consistently without actually using exhaust gases that vary in temperature, pressure, concentration and flow rate transiently. The influences of EGR can be efficiently simulated with added CO_2 that comes from an external storage, such as compressed CO_2 gas bottles as shown in Figure 29. In most cases, air is still the major component of the engine intake. The composition of CO_2 can be arbitrarily assigned through a CO_2 flow regulating device. As the added molar concentration of CO_2 increases, the molar concentrations of O_2 and N_2 of the intake mixture decrease linearly [50-54].

The CO_2 add-on method simulates both the thermodynamic and dilution effects of EGR. As the intake CO_2 increases, the cylinder compression pressure reduces. The compression temperature also reduces, which is governed by the quasi-adiabatic compression process. Adding the effect of O_2 dilution, the ignition delay increases substantially. This is indicated by the progressively delayed combustion pressure rise as in Figure 28. As the CO_2 increases further, cycle to cycle variation of the combustion process also increase [30–34]. However, the resulting variations in the exhaust do not affect the consistency of such simulated EGR

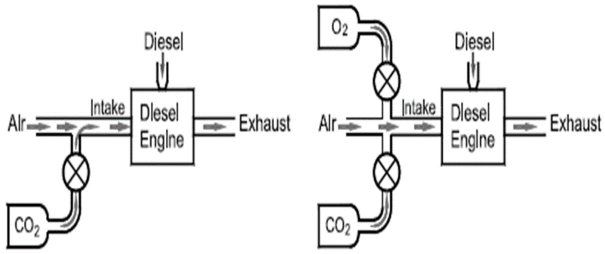

Fig. 27: Gas add-on method for simulated EGR operation

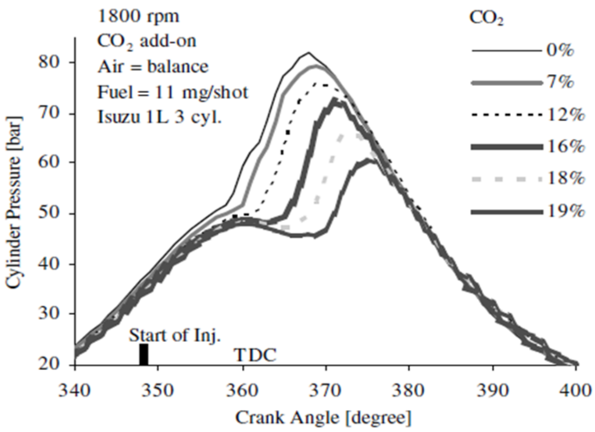

Fig. 28 : Effect of CO_2 add-on

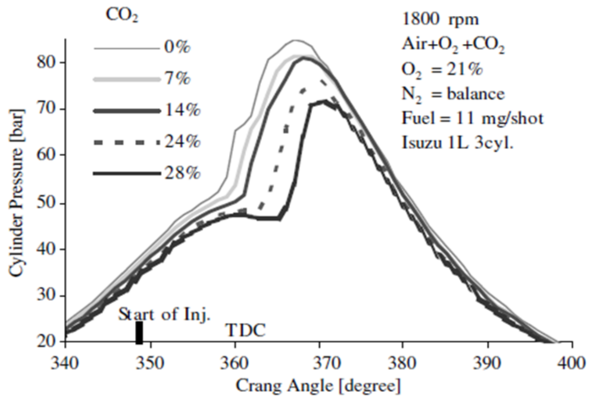

Fig. 29: Effect of CO_2 addition with constant, O_2.

The single gas add-on method can be complemented by adding additional gases. When O_2 is used as a secondary add-on gas, the O_2 level can be held constant while the CO_2 concentration varies against the balance gas, N_2. Thus, isolated effects of CO_2 addition on engine operations, such as the prevailing thermodynamic influences, can be demonstrated. For instance, the prolonging of ignition delay is less significant than without O_2 dilution, comparing Figure 28 and Figure 29. Alternatively, this method can also be used to study the influences of O_2 variations when CO_2 is held constant, which demonstrates the O_2 dilution effects on burning, and emission characteristics in the presence of CO_2. More testing results and detailed analyses on ignition delay and heat release were reported by Zheng and Reader previously [47-49].

Furthermore, a synthetic atmosphere approach can be adapted for comprehensive EGR researches [51-52]. Although there is an obvious increase of costs of bottled gas, but the synthetic gas method can produce arbitrarily assigned intake pressure, temperature and compositions that are independent of ambient and engine operating conditions. Additionally, such simulated EGR contains no combustible substances, while in a severe unstable condition, actual recycled gases do contain a high concentration of combustibles. The absence of combustibles is a major departure from actual EGR systems, which, however, helps to operate the engine stably with extremely high extent of CO_2 addition and O_2 dilution[50-54] .

A synthetic atmosphere engine test rig is shown in Figure 30 [51], is capable of utilizing a number of inert gases to study extreme operating conditions of EGR. Among the inert gases used, argon has the highest specific heat ratio and is immune from oxidation or dissociation during combustion. In contrary, carbon dioxide has the lowest specific heat ratio and is likely to dissociate into lighter molecules under high temperatures. Argon can be used to compensate the thermodynamic property changes produced by CO_2. Nitrogen gas has similar thermodynamic properties to air and may be oxidized under high temperatures to generate oxides of nitrogen.

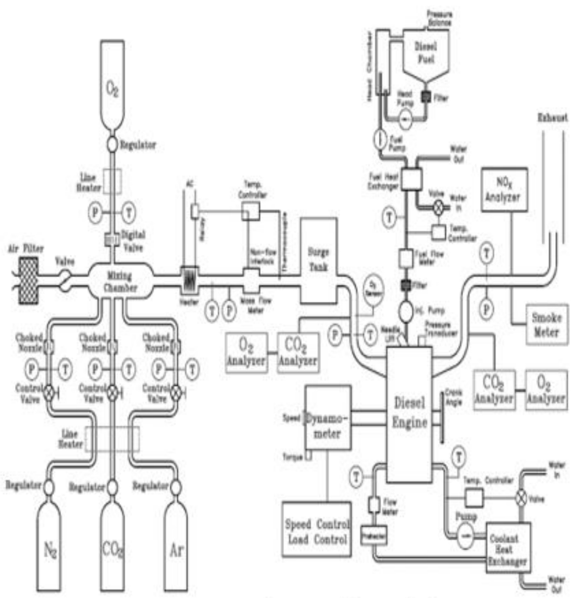

Fig. 30: Synthetic atmosphere method

By studying the isolated influences of each inert gas, the mechanisms of EGR on engine operation and emission control can be quantified. This is part of the ongoing researches at the author's laboratories. However, any results obtained from simulated EGR should be verified with water vapor addition and eventually with actual EGR tests. The consecutive influences between a previous cycle and a current cycle must be included. Extensive experiments indicated that synthesized EGR allows extremely higher ratios of EGR, than actual EGR allows [48-52]. Figure 31 shows the power curves obtained from the test rig when high CO_2 is applied, which operation cannot be produced by actual EGR. The results indicate that power loss alone may tolerate high ratios of EGR.

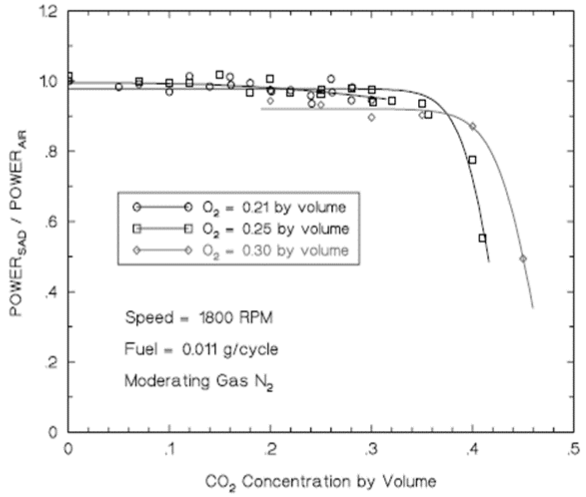

Fig. 31: Non-dimensional power of synthetic atmosphere Diesel engine.

4.3.3 **EGR versus NOx**

Diesel exhaust contains CO_2, H_2O, N_2 and O_2 in thermodynamically significant quantities and CO, THC, NOx and soot in thermodynamically insignificant, but environmentally harmful quantities. In modern Diesel engines, the combination of the former quantities normally comprise more than 99% of the exhaust, while the latter combination, the pollutants, accounts for less than 1% in quantity. Thus, the challenge is to minimize the pollutants by manipulating the thermodynamic properties and the oxygen concentration of the cylinder charge, whilst keeping minimum degradations in power and efficiency, which is the principal reason to apply Diesel EGR.

The load levels of a Diesel engine affect the exhaust composition and the temperature significantly, which is in stark contrast to exhausts from stoichiometric burning engines that largely remain constant irrespective of load variations. Notably, load levels are adjusted by fuelling rate in Diesel engines but by air–fuel mixture charging rate in SI engines. Thus, exhaust oxygen concentrations of Diesel engines vary significantly with engine load. In contrary, only a trace of oxygen remains in the exhaust of stoichiometric burning engines. Without applying EGR, energy efficient Diesel engines normally produce an exhaust that contains oxygen from 5% at full load to 20% during idling. As the excessiveness of exhaust oxygen diminishes with the increase in engine load, the specific heat of the exhaust rises because of the increase in the combustion product CO_2 [48-52].

Thus, the effectiveness of NOx reduction by EGR also varies with load. The heat capacity of the cylinder charge increases with the increase in CO_2 that is brought in by EGR. The flame temperature, and thus, the maximum temperature of the working fluid will be lowered with the increase in CO_2. Test results indicate that high ratios of EGR need to be applied at low load, but low ratios of EGR are sufficient for high load, Figure 32 and 33. When operating at lower loads, Diesel engines generally tolerate a higher EGR ratio because the exhaust contains a high concentration of O_2 and low concentrations of combustion products like CO_2 and H_2O. At high loads, however, the exhaust oxygen becomes scarce and inert constituents become dominating.

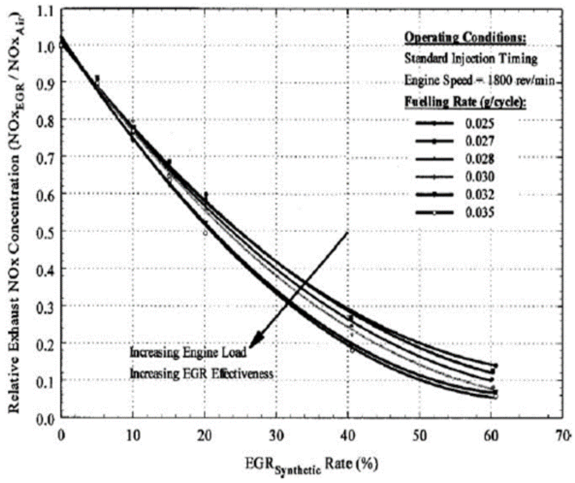

Fig. 32: NO_x reduction versus synthetic EGR rate

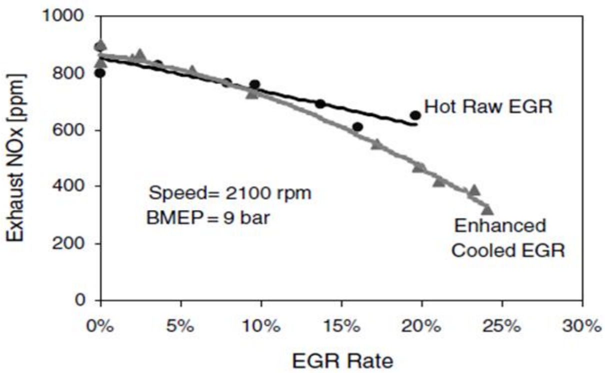

Fig. 33: Comparison between cooled and hot EGR.

If hot exhaust is directly recirculated, the cylinder charge temperature will be afloat with the influx of the EGR heat, especially at high loads, which will raise the working fluid temperature. Test results demonstrated that cooled EGR reduces NOx more effectively than hot EGR [35]. The tests shown in Figure 32 were conducted with synthetic atmosphere as intake, which was equivalent to thoroughly cooled EGR. The intake mixture temperature was maintained at 30 °C, referring Figure 30. The synthetic EGR rate follows the CO_2 definition discussed previously. The test results in Figure 33 were obtained with laboratory enhanced EGR cooling that kept the EGR cooler outlet temperature below 80 °C. In the same figure, a comparison was also shown with hot EGR, and it was apparent that the NOx reduction was less effective.

As load increases, diesel engines tend to generate more smoke because of reduced access to oxygen. Employing EGR, although it is effective to reduce NOx, but it also further aggravates the scenario, i.e., the prevailing NOx and PM trade-off, as shown in Figure 34 [26,27,35,36]. Testing results indicate that low load operations are commensurate with the high rates of EGR, while high loads indicate low or no EGR [35,43, 46]. More importantly, the trend of increased PM formation commonly hinders the application of EGR at full loads. However, since NOx generation is severe at full loads, extended fuel injection retarding could be implemented in lieu of EGR [44].

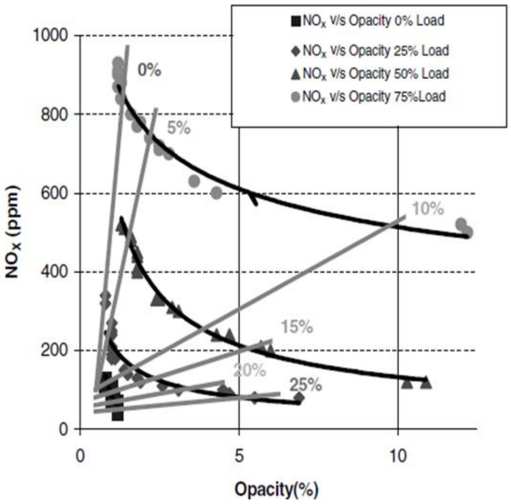

Fig. 34: Trade-off between exhaust NO_x and opacity (smoke) when hot EGR is applied

4.4 **Control of EGR**

An ideal control strategy should collate EGR rate with NOx generation rate transiently. Short of viable fast response lean NOx sensors, the resort is to use look-up tables to command the EGR valve opening. The look-up tables comprise a primary command table that is based on engine speed and fuelling rate (engine load), and a number of modification tables that refer to the operating parameters, such as engine block temperature, boost pressure, injection timing etc. Such static tables are calibrated by the engine manufacturers and reside in the engine control module (ECM).

The impact of EGR on engine operation is similar to turbocharging, both of them affecting the equilibrium states of the entire system. Although appropriate control strategies are capable of setting up consistent EGR operations initially, any drifts in engine operation will affect the initial setup when EGR feedback is not available. In order to achieve feedback control, a common practice is to estimate the EGR rate via measuring the fresh intake air with a mass air flow (MAF) sensor. By assuming a mass flow rate of the cylinder charge, the mass flow rate of EGR could be determined by mass conservation on a steady operating condition.

4.5 **Treatment of EGR**

Because of the vitality of EGR in NOx reduction, it is prudent to explore the applicable limits of EGR. Notably, heavy uses of EGR could degrade the energy efficiency and mechanical durability of the engine [43-45]. Besides, excessive uses of EGR also cause operational instabilities that further aggravate the engine efficiency and durability [51]. However, such instabilities can be reduced by modifying the EGR stream thermally and/or chemically, i.e., through EGR treatments [33].

4.6 **EGR cooling**

EGR cooling increases the density and, therefore, the mass flow rate of the intake charge, which is as important as boost inter-cooling. It is known that the inter-cooler plays an important role in improving engine performances and emissions. In order to prevent fouling, the recirculated exhaust is normally introduced downstream of the inter-cooler, as shown in Figure 27. Without intercooling, the boost temperature can reach 80 °C frequently and over 160 °C occasionally, for moderately turbocharged engines. Effective inter-coolers, which use ambient air as the cooling medium (air cooled), can bring down the boost temperature to only 5–20 °C higher than the ambient. Obviously, if the engine jacket coolant, which normally has a temperature of 85–95 °C, is used as the cooling medium (water cooled), the inter-cooling would be less effective.

In case of hot EGR is applied in conjunction with boost inter-cooling, no matter how effective the air-cooler is, the intake air will be heated by the recirculated exhaust that sets back the intake cooling. Thus, it is imperative to implement sufficient cooling on the EGR. Normally, the engine jacket coolant is used as the cooling medium to remove heat from the EGR stream. Such liquid EGR coolers are compact and easy to install. A cooled exhaust temperature approximating 120 °C is preferred [41].

Furthermore, it is more effective to reduce NOx by cooled EGR, which shares the same scenario with boost inter-cooling. A comparison between hot and cooled EGR is shown in Figure 35. The testing engine has been described by Zheng et al. [50] and Patel [53] previously. A custom built large EGR cooler using tap water was used to maintain the EGR cooler outlet temperature below 120 °C. During the tests, the EGR cooler commonly kept the recycled gas below 70 °C.

EGR cooling also has the potential to stabilize the engine operation by holding the temperature of the recirculated exhaust, and act as a grounding effect in the feedback loop, because the exhaust temperature variations are isolated from the engine intake. An EGR cooler also inserts a plenum in the EGR loop that helps pressure pulsation damping, which effect is also enhanced by the flow restrictions associated with the EGR plumbing [51].

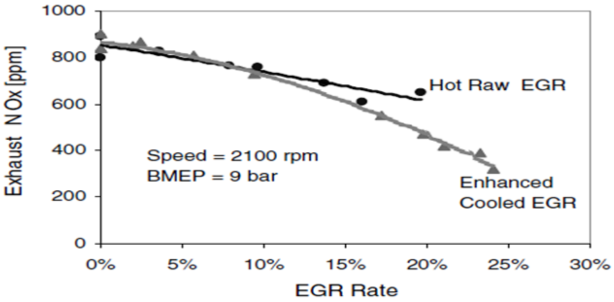

Fig. 35: The effect of EGR cooling on NO_x production

4.7 **EGR oxidation**

Although excessive EGR causes dramatic NOx reduction, the engine operation also approaches zones with higher cyclic variations. Such instabilities are largely associated with prolonged ignition delay and incomplete combustion, which are caused by increased CO_2 and decreased O_2 in the engine intake [47-51]. The deterioration in combustion efficiency results in fluctuations in the combustion products that may escalate the consecutive cyclic variations of the cylinder charge in terms of temperature, pressure and composition [50-53].

In a conventional EGR system, the EGR flow rate is adjusted with an EGR valve, while the EGR temperature is preferably reduced with an EGR cooler. However, the constituents of the EGR stream are generally left intact. Uncontrolled EGR components, such as combustibles, are commonly introduced to the engine combustion chamber. The approach is to eliminate the influences of recycled combustibles on such instabilities, by applying oxidation with a catalyst in the high pressure EGR loop [51]. The elimination of recycled combustibles showed significant effects on stabilizing the cyclic variations, so that the EGR applicable limits are effectively extended. The attainability of low NOx emissions with the catalytically oxidized EGR is shown in Figure 36.

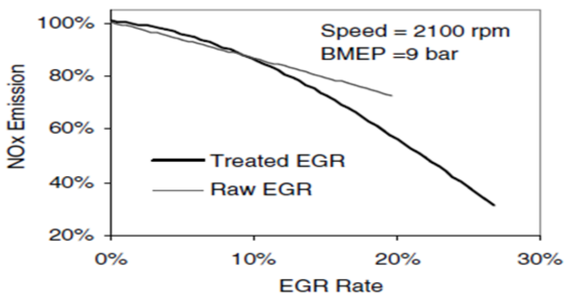

Fig. 36: The effect of oxidation treated EGR.

From medium to high load operations, the exhaust temperatures are above 350 °C for the test engine. At such temperature levels, a satisfactory conversion rate of CO and reactive HC can be obtained reliably with modern catalyst technologies [38, 39]. Since high load operation was targeted in the present work, the oxidation catalyst showed over 90% efficiency in destroying the recycled combustibles.

The oxidation catalytic converter (Figure 37) oxidizes unburned combustibles into CO_2 and H_2O. Although the fuel in the EGR stream was sacrificed by oxidation, it provided a necessary safety margin to run aggressive EGR stably. This is critical when considering the inconsistencies in practical operations. Figure 38 indicates that the oxidized EGR, extended the limit of EGR.

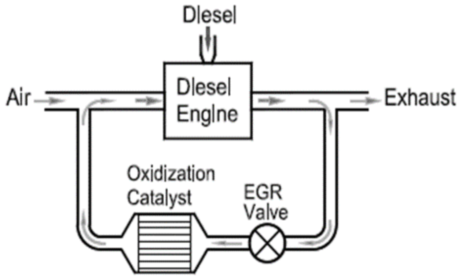

Fig. 37: The layout of an oxidation catalytic EGR operation.

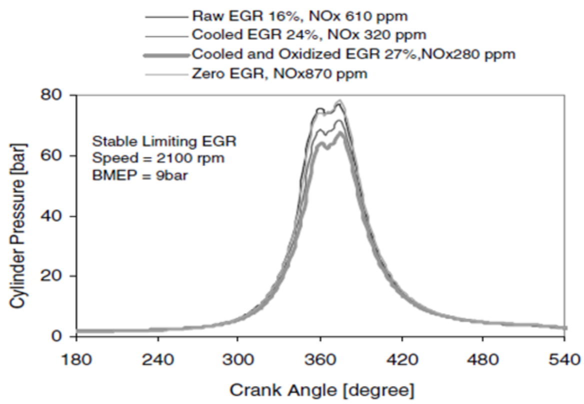

Fig. 38: Cylinder pressure traces of limiting stable EGR operations

4.8 **EGR fuel reforming**

Diesel exhaust temperatures normally range from 120 to 720 °C for non-turbocharged systems and 100 to 600 °C for turbocharged systems. The exhaust oxygen concentration is in the range from 19% to 4% for naturally aspirated engines and 19% to 7% for turbocharged engines, from idle to full load. Due to the significant amounts of surplus oxygen in the exhaust, a method is proposed here to suppress PM production with EGR by fuel reforming in the EGR loop. The heat of the exhaust can be utilized simultaneously. In comparison, exhausts from stoichiometric combustion engines are not suitable for fuel reforming because of the obvious lack of oxygen. A research has been planned at the author's laboratory to incorporate a catalytic rich combustor into the EGR loop as in Figure 39, so that gaseous fuels can be generated on demand. Gaseous

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fuels will be generated in the EGR loop, in which a controlled amount of diesel fuel is reformed to produce hydrogen gas and carbon monoxide in a catalytic rich combustor.

The EGR reformer will produce H_2 and CO, so that in-cylinder premixed combustion will be enhanced. Such an engine operation is similar to dual fuel engines that use a diesel pilot to ignite a gaseous fuel [57]. A conceptual design is proposed in Figure 40 when implementing on a turbocharged engine. If the gaseous fuel follows a super lean turn process, for instance kgas> 1:35, low NOx operations could be achieved. If the diesel pilot quantity is minimized to let the gaseous fuel dominate, the cycle will share the advantages of a homogeneous charge compression ignition (HCCI) engine system. HCCI systems improve fuel economy through nearly instantaneous combustion of a super lean homogeneous fuel/air mixture, which produces very low NOx and particulate matter (PM) emissions. However, breakthroughs are needed to enhance the ignition consistency and to expand the load levels in order to make HCCI operations practical [52-57].

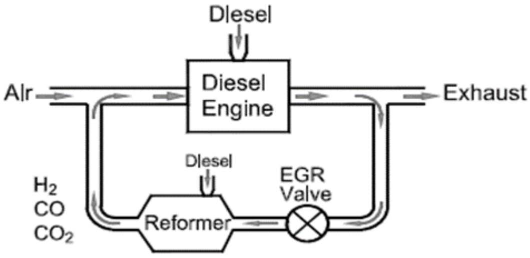

Fig. 39: The layout of a proposed EGR fuel reformer

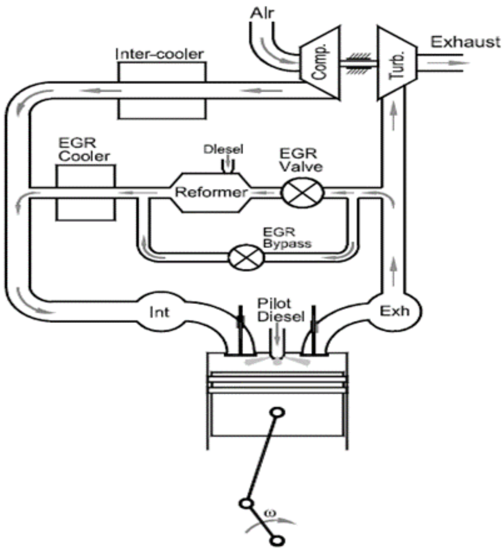

Fig. 40: The layout of a proposed EGR fuel reformer when implemented in a turbocharged system

5. Conclusion

Methods exist as a control measure of CO, HC, NOx, CO_2 PM and PN emissions, for stoichiometric and lean-burn gasoline engines and diesel engines. They are used and proven in many different applications. Continuous improvement in substrate and coating technologies, as part of an integrated system comprising electronic control and fuel quality, allows meeting more and more stringent combustion engines emissions legislations. Besides, hydrogen is an effective ignition controller for HCCI combustion of DME. Hydrogen makes the first heat release slower and delays the rise in temperature during the low temperature oxidation of DME. Due to the delayed rise in temperature, the starting crank angle of the second heat release is greatly delayed. There is no large change in the starting temperature of the high-temperature oxidation. Hydrogen addition prevents a too early ignition and enables higher load operation in HCCI engines. EGR technique can reduce fuel consumption of gasoline engine and meet more stringent emission regulations in the future together with other advanced techniques. The engine using hot EGR can use exhaust to heat intake, promote combustion, and thus improve thermal efficiency. While cooled EGR increases intake density, thereby increases volumetric efficiency of engine. At the same time, decreased temperature can further reduce NOX emission, but HC emission and cycle-by-cycle variations are increased compared with that of hot EGR. Knock is a main problem to improve BMEP of gasoline engine. Commonly used method to inhibit knock is fuel enrichment. Using cooled EGR instead of excessive fuel to inhibit knock and to reduce emission is an effective measure in gasoline engine. With the development of downsized gasoline engine, turbo-charged gasoline engine is becoming increasingly popular in the market, and the implementation of EGR is, therefore, more difficult, which lead to a coexistence problem in engines. Furthermore, by using EGR in a turbocharged gasoline engine, we must also consider turbo matching problem.

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