

Diesel engine exhaust gas recirculation—a review on advanced and novel concepts

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Abstract

Exhaust gas recirculation (EGR) is effective to reduce nitrogen oxides (NO_x) from Diesel engines because it lowers the flame temperature and the oxygen concentration of the working fluid in the combustion chamber. However, as NO_x reduces, *particulate matter* (PM) increases, resulting from the lowered oxygen concentration. When EGR further increases, the engine operation reaches zones with higher instabilities, increased carbonaceous emissions and even power losses. In this research, the paths and limits to reduce NO_x emissions from Diesel engines are briefly reviewed, and the inevitable uses of EGR are highlighted. The impact of EGR on Diesel operations is analyzed and a variety of ways to implement EGR are outlined. Thereafter, new concepts regarding EGR stream treatment and EGR hydrogen reforming are proposed. © 2003 Elsevier Ltd. All rights reserved.

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1. Introduction

Diesel engines have inherently high thermal efficiencies, resulting from their high compression ratio and fuel lean operation. The high compression ratio produces the high temperatures required to achieve auto-ignition, and the resulting high expansion ratio makes the engine discharge less thermal energy in the exhaust. The extra oxygen in the cylinders is necessary to facilitate complete combustion and to compensate for non-homogeneity in the fuel distribution. However,

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Nomenclature

BMEP	break mean effective pressure
CA	crank angle
CO	carbon monoxide
ECM	engine control module
EGR	exhaust gas recirculation
MAF	mass air flow sensor
HCCI	homogeneous charge compression ignition
NO _x	oxides of nitrogen
PM	particulate matter
SI	spark ignition
TDC	top dead center
THC	total hydrocarbon
VGT	variable geometry turbine
x	molar concentration
λ	air excess ratio
\dot{m}	mass flow rate

high flame temperatures predominate because locally stoichiometric air–fuel ratios prevail in such heterogeneous combustion processes [1]. Consequently, Diesel engine combustion generates large amounts of NO_x because of the high flame temperature in the presence of abundant oxygen and nitrogen [2,3].

Diesel engines are lean burn systems when overall air–fuel ratios are considered, commonly with an air excess ratio $\lambda = 1.5$ – 1.8 on full loads and higher λ values as load reduces. During idling, for instance, the air to fuel ratio of a modern Diesel engine can be 10-fold higher than that of stoichiometric engines ($\lambda > 10$). However, diffusion controlled Diesel combustion is predominately stoichiometric burn, in a microscopic sense, because the flames are prone to localize at approximately stoichiometric regions within the overall fuel lean but heterogeneous mixture. The prevailing flame temperature can be estimated with adiabatic stoichiometric flame temperature calculations [1,4]. For a given engine speed, it is obvious that the NO_x *generation rate* is closely related to the *fueling rate*, the engine load level. On a power generation basis, therefore, the decrease in overall mixture strength will not drastically reduce the specific rate of NO_x generation.

Unlike Diesel engines, homogeneously charged engines, such as spark ignited gasoline engines or other gaseous fuel engines, can actually use λ control to reduce NO_x effectively. To a homogeneous charge, the weakening in mixture strength can effectively reduce the flame temperature and propagation speed. An excessively fuel lean mixture, $\lambda > 1.2$ – 1.4 (depending on the type of fuel), could produce substantially lowered NO_x emissions [4–8]. The trend in NO_x reduction enhances with further weakening of the cylinder charge until sustainable flame propagation becomes unreliable and unburned combustibles intolerable. When an extremely lean mixture is used, for instance when $\lambda \approx 1.8$, a *homogeneous charge compression ignition* (HCCI) concept could be applied, where the engine operation improves fuel economy through nearly instantaneous com-

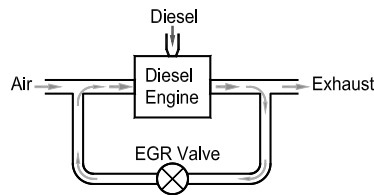


Fig. 1. Exhaust gas recirculation.

bustion that normally produces very low NO_x and PM emissions simultaneously. Although the concept is highly promising, to date, a viable model of an HCCI engine has yet to be fully developed [6,7,9].

On Diesel engines, the benefits of HCCI operation can be partially reproduced by enhancing premixed combustion that can be achieved with fuel delivery control in the injection schedule, spray pattern and air movement matching. The improved premixed burning can suppress PM generation effectively and reduce NO_x generation moderately [7,10]. However, such NO_x reduction effects are not as strong as when injection retarding is applied, although the latter procedure commonly is associated with smoke and power deteriorations [11]. The application of turbo-charging with inter-cooling also has moderate effects in reducing NO_x and PM simultaneously, resulting from enhanced fuel–air mixing and lowered temperatures of the combustion products [2,8].

Considering the prevailing stoichiometric burning of Diesel engines, it would be more efficient to lower the *specific heat capacity ratio* of the working fluid in order to lower the flame temperature. The introduction of CO_2 into the engine intake, which can be achieved by recycling a fraction of the exhaust gas into the engine intake as shown in Fig. 1, can increase the specific heat capacity effectively. Concurrently, the EGR dilutes the O_2 concentration of the working fluid. Thus, NO_x generation can be drastically lowered [10–12], which is the primary reason for Diesel EGR. However, diffusion controlled Diesel combustion is also associated with fuel rich pockets that are always struggling to find oxygen at the late stages of combustion, especially when the engine operates on high loads. The application of EGR worsens the scenario that increases the difficulties to burn smoke free.

In contrary, homogeneous charge engines produce little PM as long as the charge is not fuel rich, largely irrespective of EGR applications. For stoichiometric or lean burn SI engines, the flame sweeps over a homogeneously distributed fuel that does not lack access to oxygen, even when EGR is applied.

2. Implementations of EGR

2.1. Actual engine 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, Fig. 1, 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*.

Modern Diesel engines, however, are commonly turbocharged, and the implementation of EGR is, therefore, more difficult. A low pressure loop EGR, as shown in Fig. 2, is achievable because a positive differential pressure between the turbine outlet and compressor inlet is generally available, $(P_4 - P_1) > 0$. Furthermore, tailpipe pressure P_4 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 [12]. 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.

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, Fig. 3. 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 $(P_3 - P_2) > 0$ prevails. In case the pressure difference cannot be met with the original matching between the

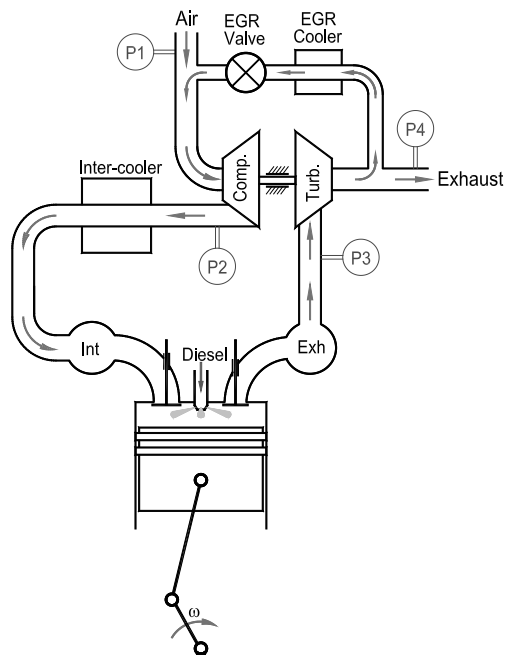


Fig. 2. Low pressure loop EGR.

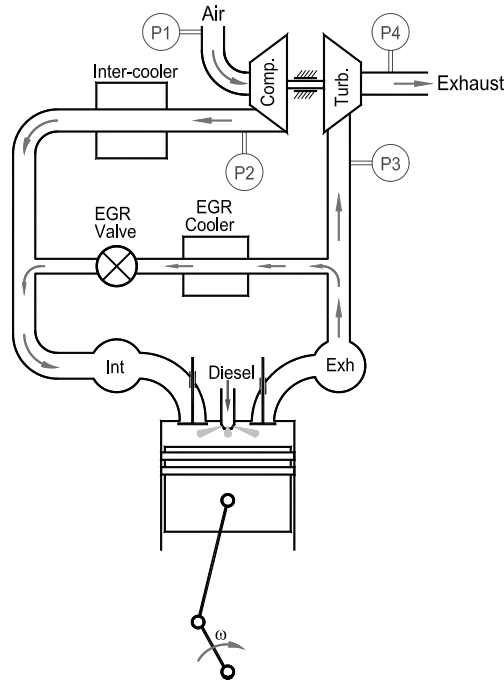


Fig. 3. High pressure loop EGR.

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. In such systems, the EGR control is closely tied to the VGT control [8,13]. The shrinking of the flow passage of the turbine nozzles will increase the turbine upstream pressure (P_3) and reduce the boost pressure (P_2).

It should be noted that the EGR flow components, ducts and valves, need to withstand the boost pressure (commonly 1–2 bar gauge pressure) whilst being leak free. The section of duct from the engine exhaust to the inter-cooler should also be resistant to exhaust temperatures that are commonly in a range of 100–600 °C. In order to absorb the thermal expansion and to tolerate the mechanical vibration, the duct should be made with a flexible structure, such as with stainless steel bellows. In order to control the EGR flow rate, the EGR valve opening should be modulated with an electronically controlled vacuum or pressure diaphragm actuator, for instance.

When EGR is applied, the *engine intake* consists of fresh air and recycled exhaust. The percentage of recycled gases is commonly represented by an *EGR ratio*, i.e. the mass ratio of recycled gases to the whole engine intake. The fresh air intake contains negligible amounts of CO_2 while the recycled portion carries a substantial amount of CO_2 that increases with EGR flow rate and engine loads. Notably, CO_2 is merely a combustion product. Thus, it is intuitive and practical, to measure *EGR ratio* by comparing the CO_2 concentrations between the exhaust and intake of the engine:

$$\text{EGR ratio} = \frac{\text{intake CO}_2 \text{ concentration}}{\text{exhaust CO}_2 \text{ concentration}}$$

From a dynamic point of view, the constituent of intake CO_2 is affected by the EGR valve opening and the exhaust CO_2 concentration. A higher exhaust CO_2 concentration leads to a higher intake CO_2 concentration and vice versa. It appears that the intake CO_2 will continuously drift upward until the engine is stalled. However, the *incremental concentration of CO_2* from the intake to the exhaust is merely resulting from the burning of the fuel supplied to the engine. Because the fueling rate is independent of the EGR ratio, stabilized EGR does prevail in steady state engine operations.

It should be noted, however, that the cylinder charge consists of engine intake replenishment and cylinder residual gases so that the CO_2 concentration of the *engine intake* is normally lower than that of the *cylinder charge*. If the residue amount is purposely boosted to dilute the fresh engine intake, the operation is known as *prompt EGR*. Obviously, the EGR ratio defined above, counting on the concentration of intake CO_2 alone, neglected the effects of prompt EGR.

2.2. Laboratory simulated EGR

In addition to actual EGR, the effect of EGR can be simulated empirically with *gas add-on* or *synthetic gas* methods, Figs. 5 and 7, which 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, Fig. 4. 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:

$$x_{\text{O}_2} = 0.21(1 - x_{\text{CO}_2})$$

$$x_{\text{N}_2} = 0.79(1 - x_{\text{CO}_2})$$

Because of being externally synthesized, the intake mixture is independent of engine operating conditions, which effectively cuts off the intrinsic relationship between in-cylinder burning quality and EGR composition [16–19]. Thus, the cyclic variations of exhaust properties will not cause

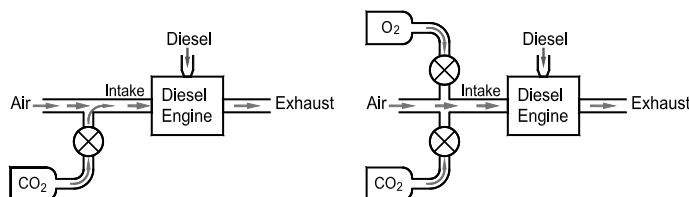


Fig. 4. Gas add-on method for simulated EGR operation.

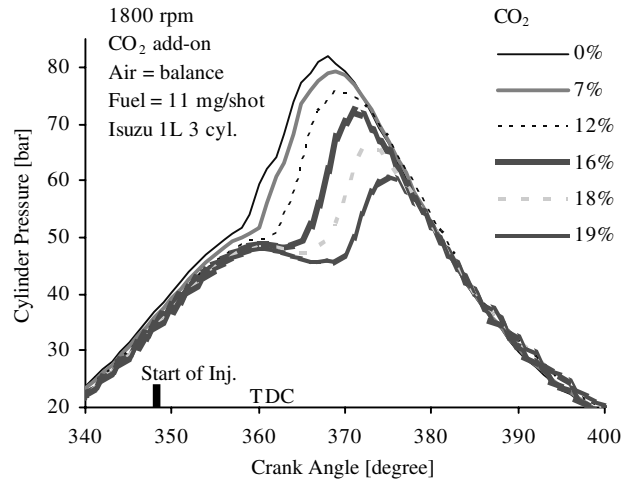


Fig. 5. Effect of CO₂ add-on.

corresponding variations in engine intake so that intensified effects of CO₂ addition or O₂ dilution can be investigated independently. The system operation is under much better controlled conditions even with extremely high ratios of the equivalent EGR, which is ideal to study the limits of EGR in terms of high CO₂ and low O₂ compositions at the intake.

The CO₂ add-on method simulates both the thermodynamic and dilution effects of EGR. As the intake CO₂ increases, the cylinder compression pressure reduces. The compression temperature also reduces, which is governed by the quasi-adiabatic compression process. Adding the effect of O₂ dilution, the ignition delay increases substantially, which is indicated by the progressively delayed combustion pressure rise in Fig. 5. As the CO₂ increases further, cycle to cycle variations of the combustion process also increase [14–18]. However, the resulting variations in the exhaust do not affect the consistency of such simulated EGR.

The single gas add-on method can be complemented by adding additional gases. When O₂ is used as a secondary add-on gas, the O₂ level can be held constant while the CO₂ concentration varies against the balance gas N₂. Thus, isolated effects of CO₂ 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 O₂ dilution, comparing Figs. 5 and 6. Alternatively, this method can also be used to study the influences of O₂ variations when CO₂ is held constant, which demonstrates the O₂ dilution effects on burning and emission characteristics in the presence of CO₂. More testing results and detailed analyses on ignition delay and heat release were reported by Zheng and Reader previously [14–16].

Furthermore, a synthetic atmosphere approach can be adapted for comprehensive EGR researches [18,19]. The synthetic gas method can produce arbitrarily assigned intake pressure, temperature and compositions that are independent of ambient and engine operating conditions, although with obviously increased costs in bottled gases. 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

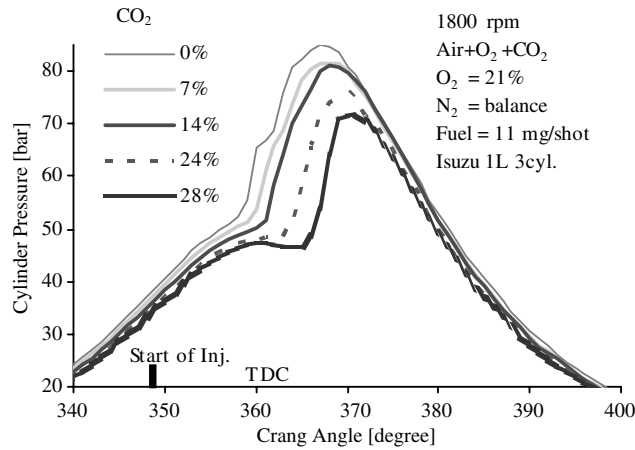


Fig. 6. Effect of CO₂ addition with constant O₂.

from actual EGR systems, which, however, helps to operate the engine stably with extremely high extent of CO₂ addition and O₂ dilution.

A synthetic atmosphere engine test rig is shown in Fig. 7 [18], which 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

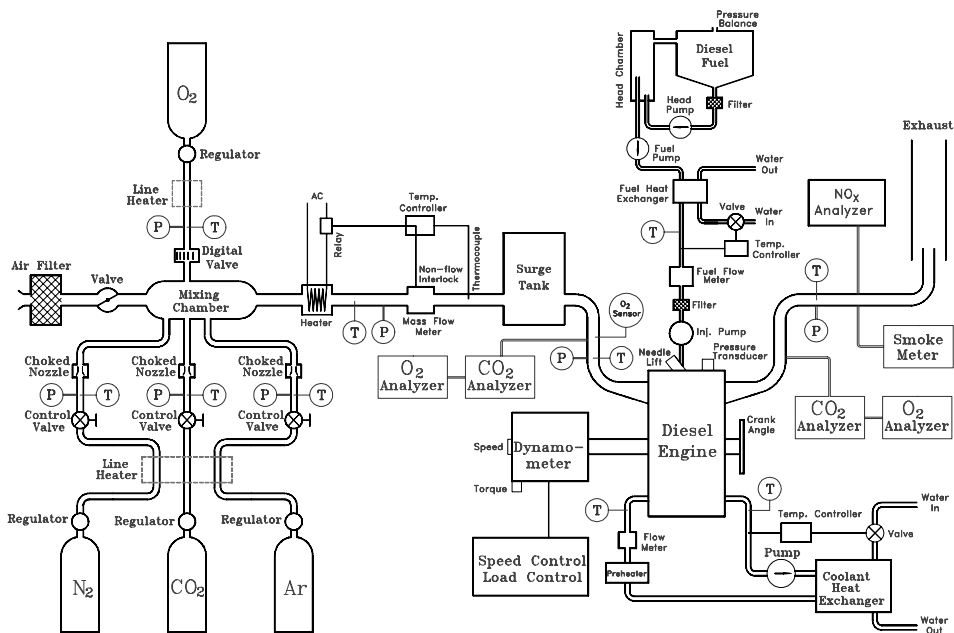


Fig. 7. Synthetic atmosphere method.

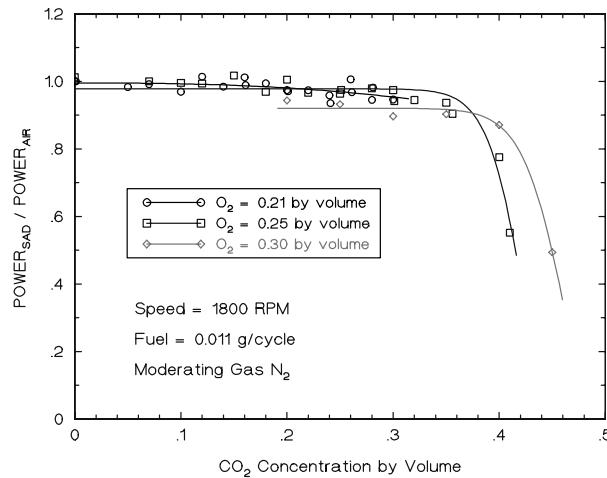


Fig. 8. Non-dimensional power of synthetic atmosphere Diesel engine.

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.

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 on going researches at the authors' 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 [15,19]. Fig. 8 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.

3. EGR versus NO_x

Diesel exhaust contains CO_2 , H_2O , N_2 and O_2 in thermodynamically significant quantities and CO , THC , NO_x 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 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 fueling 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 .

Thus, the effectiveness of NO_x 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, Figs. 9 and 10. 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 CO_2 and H_2O . At high loads, however, the exhaust oxygen becomes scarce and inert constituents become dominating.

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 NO_x more effectively than hot EGR [2]. The tests shown in Fig. 9 were conducted with synthetic atmosphere as intake, which was equivalent to thoroughly cooled EGR. The intake mixture temperature was maintained at 30°C , referring Fig. 7. The synthetic EGR rate follows the CO_2 definition discussed previously. The test results in Fig. 10 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 the NO_x reduction was less effective.

As load increases, Diesel engines tend to generate more smoke because of reduced access to oxygen. Employing EGR, although effective to reduce NO_x , further aggravates the scenario, i.e. the prevailing NO_x and PM trade-off, as shown in Fig. 11 [10,11,19,20]. Testing results indicate that low load operations are commensurate with high rates of EGR, while high loads indicate low or no EGR [2,10,13]. More importantly, the trend of increased PM formation commonly hinders

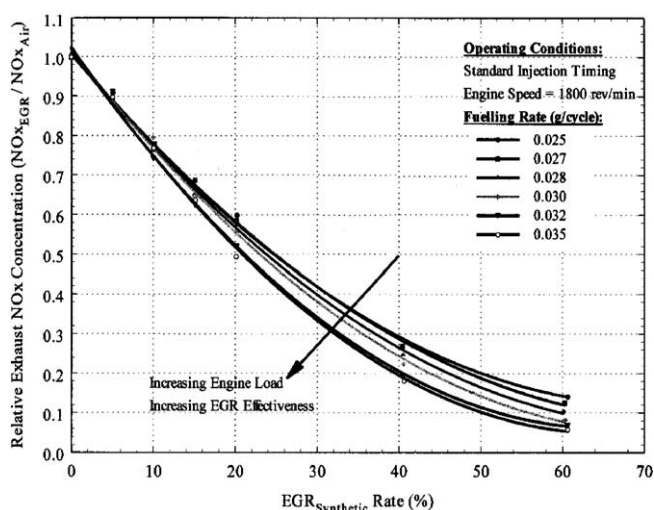


Fig. 9. NO_x reduction versus synthetic EGR rate.

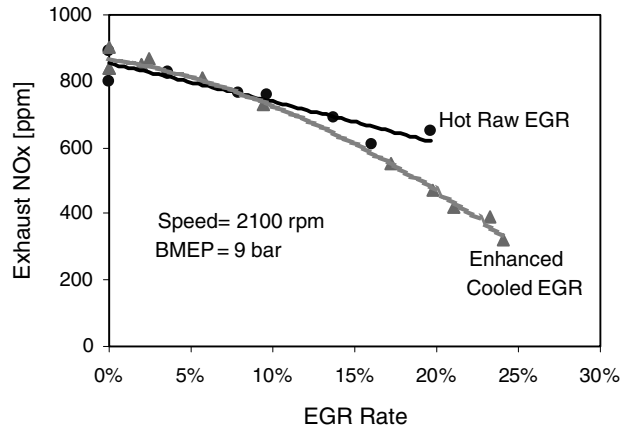


Fig. 10. Comparison between cooled and hot EGR.

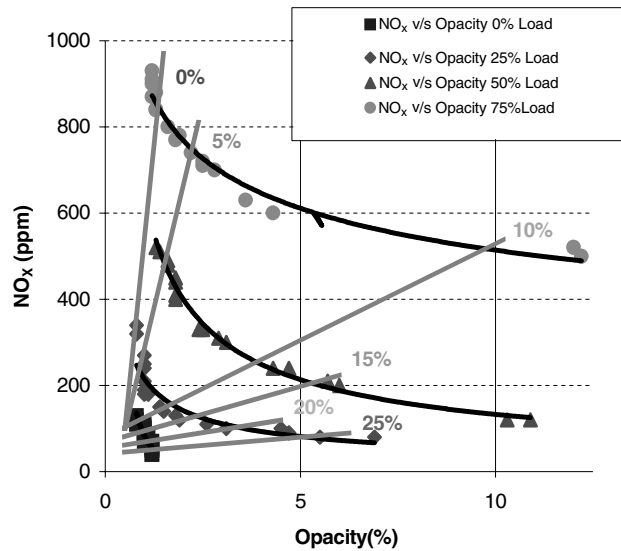


Fig. 11. Trade-off between exhaust NO_x and opacity (smoke) when hot EGR is applied.

the application of EGR at full loads. However, since NO_x generation is severe at full loads, extended fuel injection retarding could be implemented in lieu of EGR [11].

4. Control of EGR

An ideal control strategy should collate EGR rate with NO_x generation rate transiently. Short of viable fast response lean NO_x 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 fueling 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:

$$\dot{m}_{\text{EGR}} = \dot{m}_{\text{Int}} - \dot{m}_{\text{MAF}}$$

However, the estimation on the mass flow of the cylinder charge is hindered by a number of transient operating parameters that include EGR temperature, engine block temperature, after-cooler temperature and boost pressure, while most of these parameters are not monitored by the ECM. Additionally, if a variable geometry turbine (VGT) is employed to ensure sufficient EGR, by raising the turbine upstream pressure (P_3 in Fig. 3), adjustment of the nozzle areas of the VGT is commonly based on static look-up tables even when the pressure and temperature of the exhaust drift significantly [8,12,13].

Furthermore, a sufficient EGR control needs real time EGR rate and combustion quality monitoring, but viable sensor technologies associated with lean burn systems are yet to be developed. For instance, although the boost pressure is normally monitored by an engine ECM, the turbine upstream conditions are commonly not monitored. Without sufficient feedback control, the setup of EGR has to compensate any discrepancies of implementation. Consequently, the maximum EGR ratios in use are generally lower than the maximum allowable EGR that is optimized on well controlled conditions. In general, the EGR operation effects engine operating stabilities through the following parameters:

- EGR valve opening,
- EGR loop differential pressure,
- EGR cooler cooling efficiency,
- in cylinder combustion efficiency,
- exhaust and intake temperatures.

The above parameters directly affect the quantity and/or quality of EGR that, in turn, affects the equilibrium states of engine operation. For instance, the characteristics of in-cylinder fuel burning affect the exhaust temperature that, in turn, affects the exhaust backpressure. The backpressure directly affects the rate of EGR that again affects the in-cylinder burning processes. Additionally, the functioning of the following sub-systems also affects EGR through their influences on exhaust backpressure:

- turbocharging,
- exhaust brake,
- exhaust aftertreatment systems.

5. Treatment of EGR

Because of the vitality of EGR in NO_x 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 [10]. Besides, excessive uses of EGR also cause operational instabilities that further aggravate the engine efficiency and durability [18]. However, such instabilities can be reduced by modifying the EGR stream thermally and/or chemically, i.e. through EGR treatments [17].

5.1. 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 Fig. 4. Without inter-cooling, 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 has a temperature of 85–95 °C commonly, is used as the cooling medium (water cooled), the inter-cooling would be less effective.

In case 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 [8].

Furthermore, it is more effective to reduce NO_x by cooled EGR, which shares the same scenario with boost inter-cooling. A comparison between hot and cooled EGR is shown in Fig. 12. The testing engine has been described by Zheng et al. [17] and Patel [20] previously. A custom built

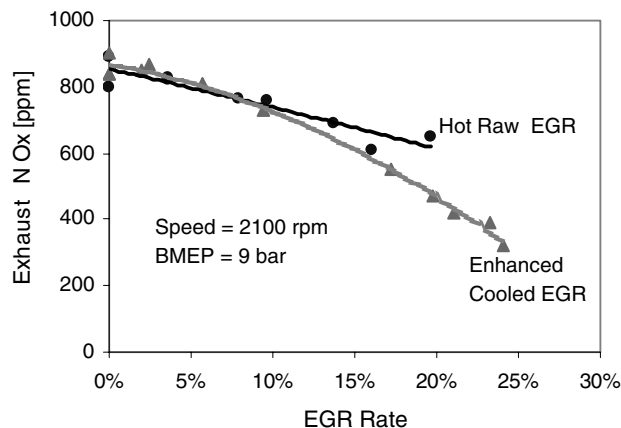


Fig. 12. The effect of EGR cooling on NO_x production.

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, 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 [21].

5.2. EGR oxidation

Although excessive EGR causes dramatic NO_x 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 [14,18]. 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 [17–19].

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 [18]. 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 NO_x emissions with the catalytically oxidized EGR is shown in Fig. 13.

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 [22,23]. Since high load operation was targeted in the present work, the oxidation catalyst showed over 90% efficiency in destroying the recycled combustibles.

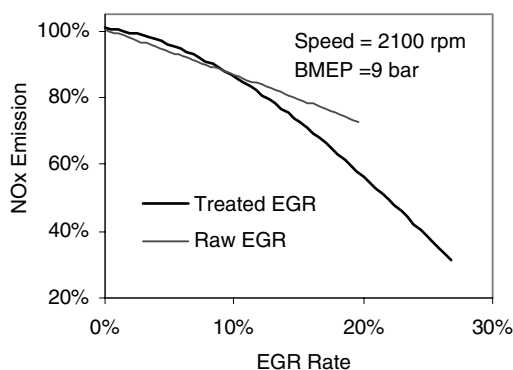


Fig. 13. The effect of oxidation treated EGR.

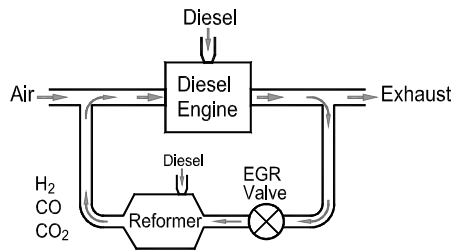


Fig. 16. The layout of a proposed EGR fuel reformer.

into the EGR loop so that gaseous fuels can be generated on demand, Fig. 16. Gaseous 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 [24]. A conceptual design is proposed in Fig. 17 when implementing on a turbocharged engine. If the gaseous fuel follows a super lean burn process, for instance $\lambda_{\text{gas}} > 1.35$, low NO_x 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

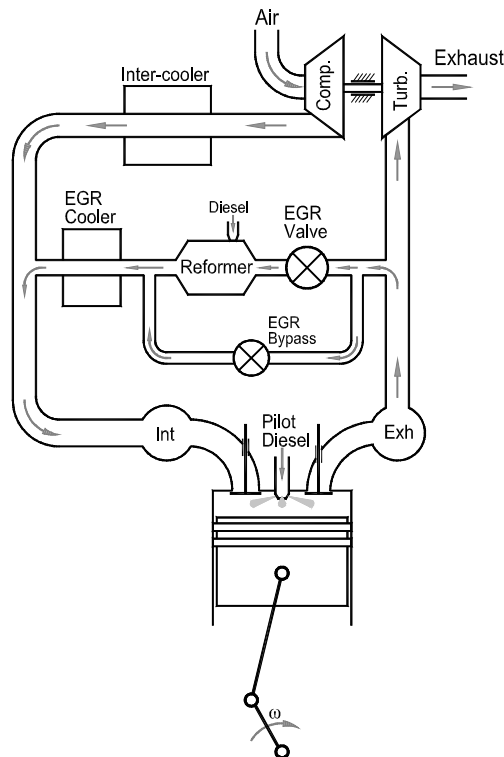


Fig. 17. The layout of a proposed EGR fuel reformer when implemented in a turbocharged system.

(HCCI) engine system. HCCI systems improve fuel economy through nearly instantaneous combustion of a super lean homogeneous fuel/air mixture, which produces very low NO_x 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.

6. Final comments

Diesel exhaust contains sulfuric salts and other abrasive and corrosive substances. It has been argued whether EGR should be applied to Diesel engines because of the increased piston-cylinder wearing [25]. Heavy uses of EGR could also deteriorate the energy efficiency, operational stability and PM generation of the engine. However, the concern over increased wearing and deteriorated performance has soon given way to stringent emission regulations. In stark contrast, the current concern is on how aggressively EGR should be applied to all speeds and all loads, although EGR increased wearing continues to be a problem affecting engine durability and performances.

To date, EGR is still the most viable technique that can reduce NO_x dramatically. Energy efficient aftertreatment systems dealing with NO_x and PM simultaneously are still in the early development stages. The inability of available catalytic aftertreatment technologies further encourages aggressive uses of EGR.

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