Applied Energy 180 (2016) 169–184

Contents lists available at ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

Effect of exhaust gas recirculation on advanced diesel combustion and alternate fuels - A review

J. Thangaraja, C. Kannan *

School of Mechanical Engineering, VIT University, Vellore 632 014, India

HIGHLIGHTS

- Demonstrated EGR effectiveness on advanced diesel combustion and alternative fuels.
- Rendered a comprehensive outlook on EGR designs and measurement methods.
- Extensive classification and comparison of EGR configurations are carried out.
- Examined the engine combustion, emission characteristics with EGR from 0 to 70%.
- Adverse effects of EGR on engine wear and oil contamination are highlighted.

ARTICLE INFO

Article history:
Received 3 March 2016
Received in revised form 17 July 2016
Accepted 25 July 2016

Keywords:
Exhaust gas recirculation
Engine wear
Homogeneous charge compression ignition
Lube oil deterioration
Biodiesel-NO_x

ABSTRACT

Ever since the establishment of the California Air Resources Board (CARB) in 1968 and Environmental Protection Agency (EPA) in 1970, significant strides have been made in diesel engine emission control technology. The diesel emission control is being achieved using strategies involving in-situ and after-treatment techniques and even with their effective combinations. Among these techniques, recirculation of the exhaust gases back to the engine inlet is an in-situ approach for Nitrogen Oxides (NO_x) control. Moreover, exhaust gas recirculation (EGR) has been used for controlling the onset of combustion process. In the current review, the importance of EGR for advanced diesel combustion like homogeneous charge compression ignition (HCCI) or low-temperature combustion (LTC) system and the requirement of EGR with the use of alternate fuels are discussed. In order to facilitate better understanding, the adverse effects of EGR, the impact of EGR on diesel engine wear and lube oil deterioration is also highlighted.

Contents

* Corresponding author.
E-mail addresses: thangaraja.j@vit.ac.in (J. Thangaraja), kannan.chidambaram@vit.ac.in (C. Kannan).

http://dx.doi.org/10.1016/j.apenergy.2016.07.096

Nomenclature

ARC	Active Radical Combustion
ATAC	Active Thermo Atmospheric Combustion
CA	Crank Angle
CARB	California Air Resources Board
CFR	Co Operative Fuel Research
CI	Compression Ignition
CN	Cetane Number
CO	Carbon Monoxide
CO_2	Carbon Dioxide
COMF	Combustible Oxygen Mass Fraction
CRDI	Common Rail Direct Injection
CRT	Continuous Regeneration Trap
DME	Dimethyl Carbonate
DME	Dimethyl Ether
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
DR	Dilution Ratio
EDAX	Energy Dispersive X Ray Analysis
EELS	Energy Electron Loss Spectroscopy
EGR	Exhaust Gas Recirculation
EMS	Electronic Mobility Spectrometer
EPA	Environmental Protection Agency
ESR	Electron Spin Resonance
FTP	Federal Test Procedure
$H_2O_{(v)}$	Water Vapor
HC	Hydrocarbons
HCCI	Homogeneous Charge Compression Ignition
HPL	High Pressure Loop
HRTEM	High Resolution Electron Microscopy
HSDI	High speed Direct Injection
HVO	Hydrogenated Vegetable Oil
IEGR	Internal Exhaust Gas Recirculation
IMEP	Indicated Mean Effective Pressure
Lambda λ	Excess Air Factor
LNT	Low NO_x Trap
LPL	Low Pressure Loop
LTC	Low Temperature Combustion
MK	Modulated Kinetics
NO_x	Nitrogen Oxides
NTU	Number of Transfer Units
PAH	Poly Aromatic Hydrocarbons
PCCI	Premixed Charge Compression Ignition
PCI	Premixed Combustion Ignition
PFI	Port Fuel Injection
PM	Particulate Matter
PPCCI	Partially Premixed Charge Compression Ignition
PSD	Particle Size Distribution
RME	Rapeseed Methyl Ester
SCR	Selective Catalytic Reduction
SEM	Scanning Electron Microscope
SI	Spark Ignition
SOI	Start of Injection
SVO	Straight Vegetable Oil
TDC	Top Dead Centre
TEM	Transmission Electron Microscope
VGT	Variable Geometry Turbocharging
VVT	Variable Valve Timing

1. Introduction

Over the last five decades, the vehicle population across the globe has grown significantly. The concern over vehicular pollution increased steadily after 1960s Los Angeles episode. It is well-known that vehicular emissions vary with engine type, operating conditions and fuel utilized, and all these aspects need to be addressed simultaneously for their abatement. The severity of engine pollutant depends on their concentration and exposure time and severely affects human health. Owing to inherent fuel economy advantage, diesel engines have made inroads into automotive applications beside their usual stationary domain. However, the diesel fuel operated vehicles are the major source of NO_x and particulate emissions and responsible for the deterioration of ambient air quality. Many countries like Brazil, China, Sri Lanka, Denmark and Paris are in the process of eliminating diesel vehicles and enforcing heavy taxes and levies on such vehicles. Recently, Indian government banned the registration of diesel-run private cars with capacity of 2000 CC and above in certain cities [1]. It is imperative that, the sustenance of diesel operated engines is possible with ultra-low emissions and lower fuel consumption. Numerous research works have been carried out on the influence of in-cylinder mixture formation and diesel combustion process [2–4]. The NO_x-PM trade-off in diesel engines is a classical challenge for developing emission control technology and hence stands at the forefront of diesel engine development. The NO_x/PM emission could be restricted through in-cylinder control measures well ahead of their formation, or through after-treatment control devices which involve the conversion of NO_x/PM emission to relatively benign compounds [5]. Achieving lower NO_x and PM emissions, particularly in diesel engines, through in-cylinder technologies presents a formidable challenge, whilst developing after-treatment technologies to handle them individually is economically unattractive. A comprehensive summary of various strategies for controlling diesel NO_x and PM emissions is provided in Fig. 1.

Among various strategies, recirculation of exhaust gases (EGR) is an established and effective technology for NO_x inhibition in diesel engines [6,7]. In the diesel engines, even up to 50% or more of exhaust gas can be recycled. However, in the case of petrol engines, the maximum EGR is limited to 20% without affecting combustion stability [8]. Johnson [9] in a review highlighted that advanced engines will have significant amount of EGR to meet the stringent emission norms.

Abd-Alla [10] reviewed the NO_x reduction potential of EGR in diesel, gasoline and dual fuelled engines. The author recommended that adding EGR along with the intake air of diesel engine has higher NO_x reduction potential than the air displacement method. In the case of gasoline engines, substantial NO_x reduction is reported with 10–25% EGR with a penalty in combustion stability. Simultaneous reduction of NO_x and smoke is observed with EGR operated dual fuelled engine. Further, the author concluded that the usage of EGR is the most effective way in improving exhaust emissions. Zheng et al. [7] reviewed the various ways of implementing EGR and its threshold limits for NO_x reduction. Further, the authors analysed the impact of EGR on diesel operations and proposed conceptual designs for EGR fuel reformer.

Thus, the future diesel vehicles demand engine modifications as well high quality fuels to adhere with stringent emission norms. It is inferred that, EGR has become an essential control strategy for both advanced combustion engines [11] and alternate fuelled engine applications [12,13]. Detailed discussions on the influence

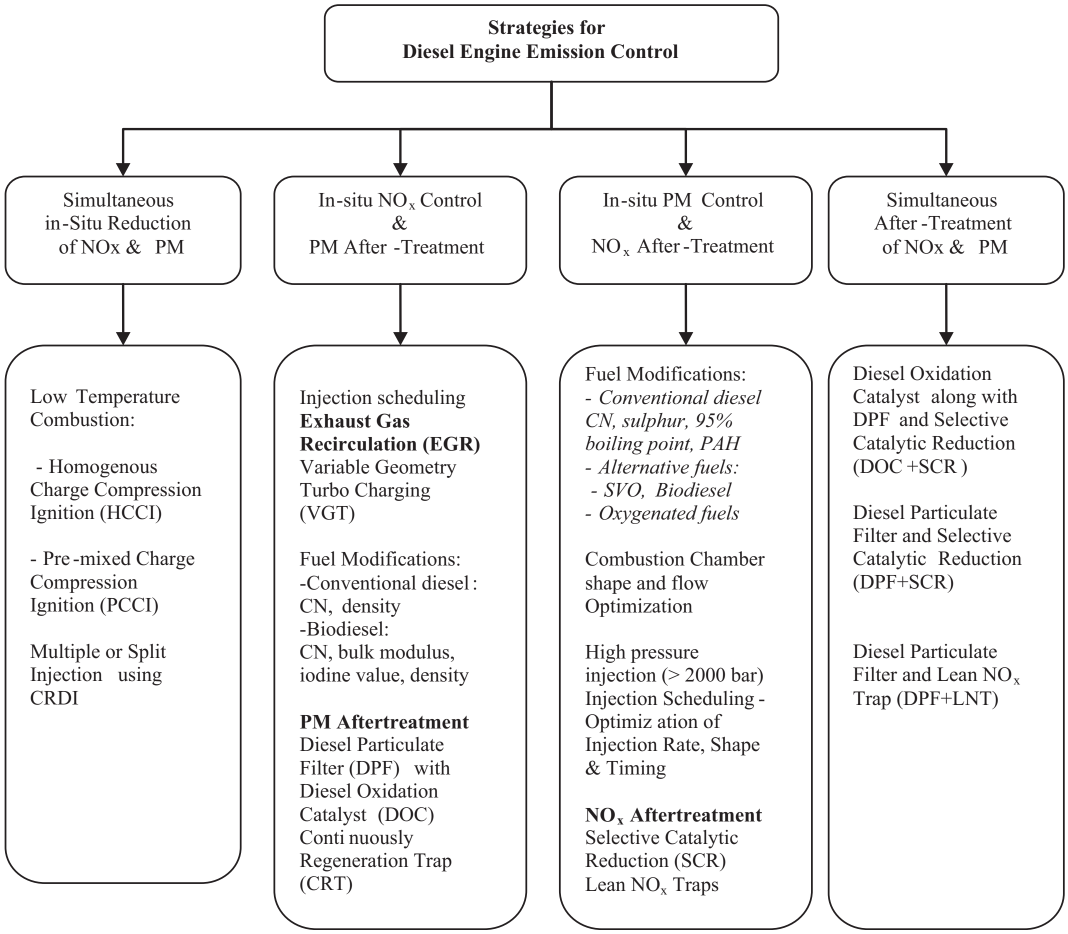

Fig. 1. Diesel emission control – a summary of in-cylinder and after-treatment options [141].

of advanced combustion on performance characteristics of diesel engines [14–17] and with alternate fuels [18–20] are available in the existing literature. Also, there are few review articles discussing the EGR effects on diesel engine combustion and NO_x emissions [7,10]. Nevertheless, there is limited elaboration on the possibilities of EGR on advanced combustion and alternate fuelled engines. Hence, the current review attempts to discuss the requirement and the effects of EGR in advanced combustion concepts and alternate fuelled diesel engines. Further, the review comprehends the EGR designs, configurations and the operating window for advanced diesel engines. The detrimental effects of EGR on engine wear and lube oil characteristics are also highlighted, which would facilitate the necessary engine modifications required for future engine development.

2. Exhaust gas recirculation

Exhaust gas recirculation (EGR) is the method by which a portion of the engine exhaust is re-routed back to the combustion chamber via the inlet system with a control valve. EGR acts as a heat sink and displaces some of the fresh inlet oxygen with diluents (CO_2 and H_2O) to suppress the NO_x formation. The higher heat capacity of these diluents results in lower combustion rate and temperature rise, which reduces the peak cylinder gas temperature [21]. A pictorial representation of the EGR effect on diesel combustion is provided in Fig. 2. Though EGR results in lower NOx emission, it causes higher soot, HC, CO emissions and inferior engine performance [8,22] as shown in Fig. 2. In addition to thermal, dilution and chemical effects; Ladommatos et al. [23] appended two more effects, viz. rise in inlet charge temperature and thermal throttling with hot EGR. Ladomattos et al. carried out an interesting study to assess the effects of EGR on diesel combustion process in a four cylinder, direct injection diesel engine. In this controlled experiments, various diluents such as carbon dioxide [24,25], water vapor [26] and their combinations [27,28] were inducted along with the intake air to simulate EGR.

The combined and individual contribution of dilution, chemical and thermal effects of CO_2 and H_2O on diesel engine NO_x and soot emissions are illustrated in Fig. 3. It is clear from Fig. 3 that, the influence of dilution effect is the major factor responsible for the change in NO_x and soot emission characteristics. The displacement of intake oxygen with EGR implementation leads to reduction in excess-air ratio, which in turn increases the ignition delay. This considerably influences the cylinder gas temperature and soot formation [29]. Further, the NO formation kinetic is affected by the dilution effect through the reduction in the partial pressure of oxygen concentration [30].

To compensate the dilution effect, few researchers have promoted additional EGR by employing a turbocharger with variable nozzle area turbine [31]. In this concept, exhaust gases were recycled and pumped along with the inducted air instead of replacing a proportion of it. The additional charge increases the inlet heat capacity which results in lower combustion temperature and NO_x emission. Moreover, the inlet oxygen concentration remains unaffected; hence soot oxidation mechanism will not get aggravated.

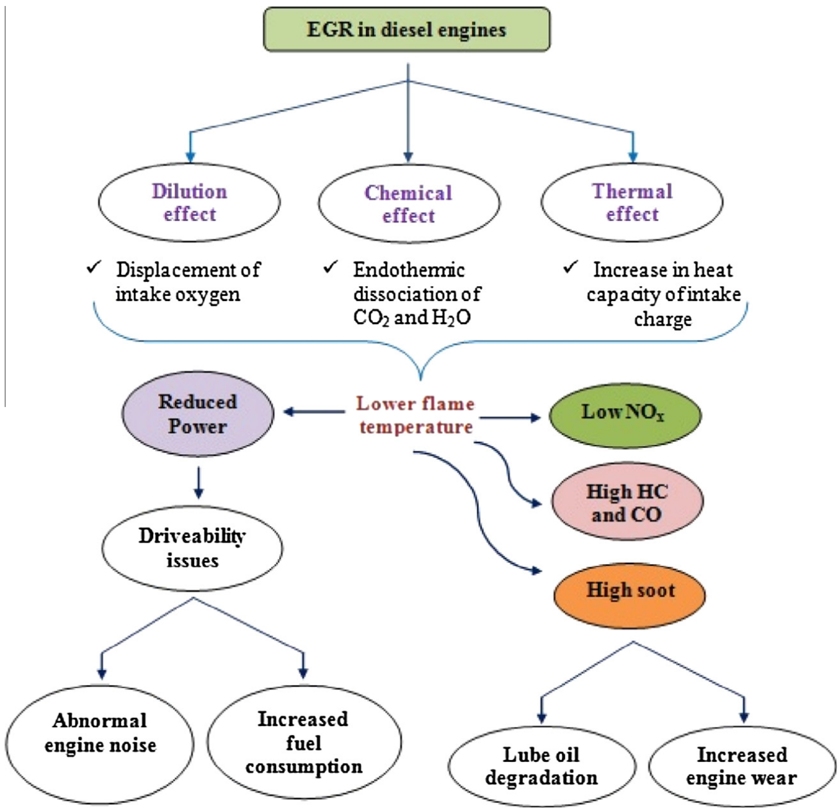

Fig. 2. Effects of EGR on diesel combustion and pollutant formation [30,29,24,26,27,25,22].

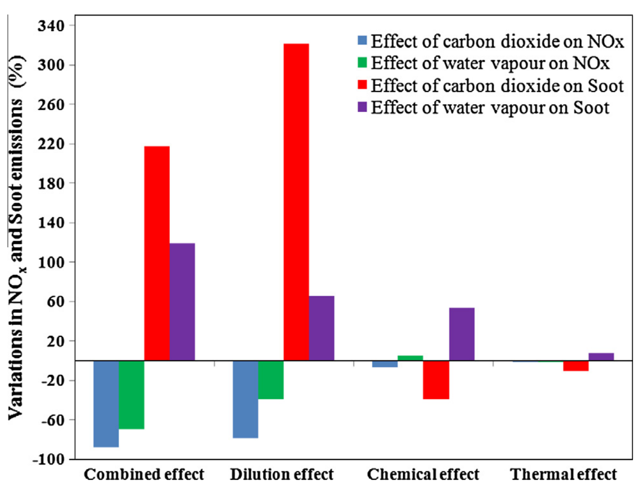

Fig. 3. Effects of CO_2 an H_2O on NO_x and soot emissions [28].

Thus, additional EGR concept has the potential of exhibiting significant reductions in NO_x emissions without any penalty in soot emissions.

2.1. EGR design, configurations and operating window

Exhaust can be re-routed either internally or externally into an engine cylinder. Internal EGR requires variable valve timing (VVT) or other mechanisms to retain a certain fraction of exhaust from a preceding cycle, whereas external EGR could be accomplished by means of the pressure differential between the inlet air and exhaust gas via an external pipe arrangement. In case of turbocharged diesel engines, external EGR is accomplished via short-path/high-pressure loop (HPL) [31–34] or a long-path/low pressure loop (LPL) system [35] as shown in Fig. 4.

In an EGR system, if exhaust residual is flown from the exhaust manifold upstream of turbine directly to the intake manifold i.e. (P_3-$P_2 > 0$), the pipe arrangement is shorter and hence known as short path/HPL EGR (refer Fig.4(a)). With HPL EGR, the exhaust back pressure (P_3) should be higher than the intake pressure (P_2). On the other hand, if (P_3-$P_2 < 0$), the exhaust is drawn downstream of the turbine and re-routed into the fresh air upstream of the compressor known as long path/LPL EGR (refer Fig.4(b)) [7]. In the case of LPL EGR, the excessive exhaust back pressure to the EGR flow is prevented and the gas mixture also gets cooled in the intercooler. Thus the inlet gas temperature could be maintained as low as 25 °C, much less as the temperatures at which the HPL-EGR system operates and thus exhibits enhanced NO_x reduction potential. However, LPL EGR deteriorates the aluminium components of the compressor/intercooler and plumbing arrangement has been often cumbersome. The classical HPL EGR lowers the boost pressure and thus matching of turbocharger is generally required. Further, to enhance the kinetic energy of EGR, an auxiliary device, usually a venturi is employed in HPL EGR system. For convenience and better performance of EGR response, HPL EGR system has been adapted to automotive turbocharged diesel vehicles. Further, detailed analysis of these two EGR systems could be found in the following references [36–38]. Also there have been attempts to develop a hybrid EGR, combining the benefits of both LPL and HPL EGR system [39,40].

Cooling the residual improves fuel economy, engine performance and enables further NO_x reduction. Fig. 5 shows NO_x emission (relative to full load) as a function of calculated oxygen mass

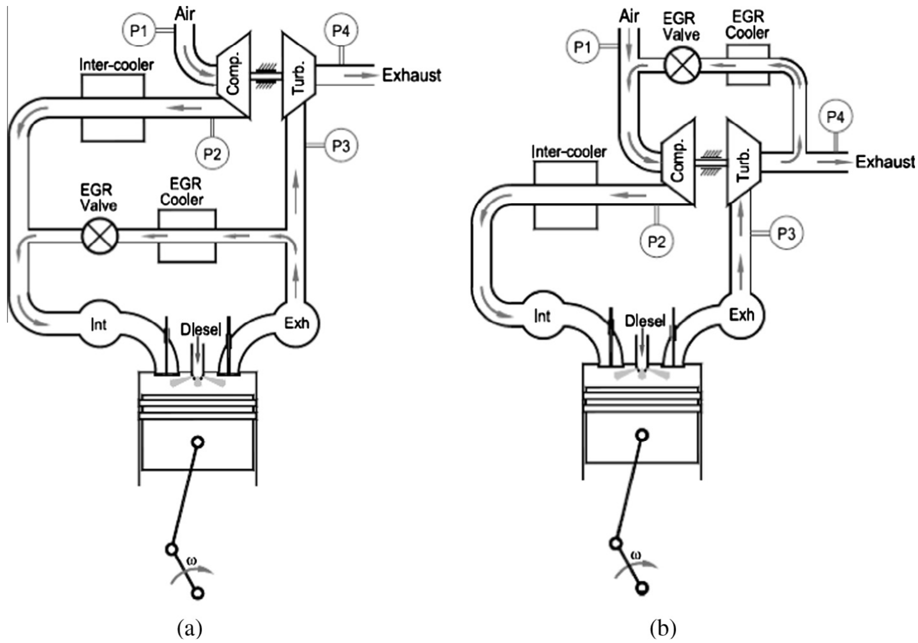

Fig. 4. (a) High pressure and (b) low pressure loop EGR [7].

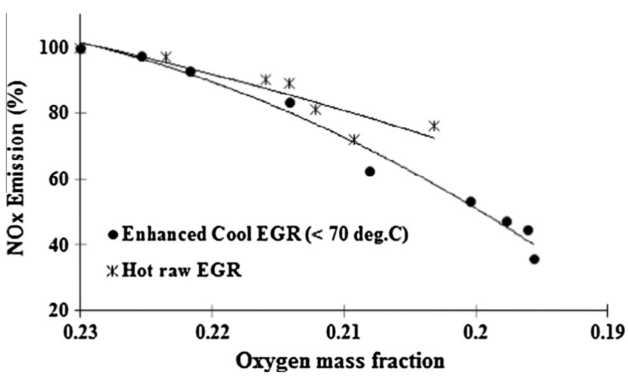

Fig. 5. Comparison between hot and cooled EGR [41].

fraction for cooled and hot EGR. The NO_x emission decreases with decrease in inlet oxygen mass fraction. In the case of hot raw EGR, the exhaust gases are directly recycled, hence the inlet charge temperature will be higher than the cooled EGR. In a comparative study, Zheng et al. [41] demonstrated that cooled EGR reduces NO_x more effectively than hot EGR. Thus, the recycled exhaust gas temperature has to be cooled below 120 deg. C [42] and opted for increased inlet mass flow rate, stable engine operation and dampens the pressure pulsation [7]. In order to prevent fouling, the cooled exhaust is normally introduced downstream of the inter-cooler. However, cooled EGR affects ignition delay, fuel vaporization and mixing rate [8]. Further, it has been shown that, the thermal efficiency could be marginally increased by hot EGR, which is attributed to the higher intake charge temperature [43]. In a recent review, Wei et al. [44] mentioned that cooled EGR has a complex structure and higher cost than hot EGR. Selim [12] opined that the choice between cooled EGR to reduce emissions and hot EGR to improve engine performance has to be made by considering specific engine operating conditions.

Kim et al. [45,46] investigated the effectiveness of EGR coolers using NTU (Number of Transfer Units) method with shell and stack-type heat exchangers. The engine employed in this study was a four cylinder turbocharged diesel engine operated with common rail injection system. Their results indicated that stack-type cooler has 25–50% higher effectiveness and heat transfer performance than the shell and tube type due to an increased surface area and better mixing of the gases. In another study, the authors investigated the flow and heat transfer characteristics of different EGR coolers. Three types of EGR coolers, viz. Plain type, spiral type-1 and spiral type-2 were tested. They concluded that the spiral type-2 cooler exhibits better heat transfer characteristics compared to plain and spiral type-1 coolers.

EGR estimation is carried out, based on direct measurements of volume [32] or mass [47] flow rates of intake charge with and without EGR as:

$$EGR(\%) = \frac{(\text{Mass or volume of air without EGR}) - (\text{Mass or volume of air with EGR})}{(\text{Mass or volume of air without EGR})} \tag{1}$$

On the basis of carbon dioxide concentrations [7,30] in the inlet and exhaust manifold, EGR is measured as:

$$EGR(\%) = \frac{[CO_2]inlet}{[CO_2]exhaust} \tag{2}$$

The EGR configuration, measurement basis and EGR concentration followed by various researchers are provided in Table 1.

While there is no universally established definition for quantifying EGR, the mass-based and the gas concentration-based EGR formulae are widely followed [11]. Derived parameters like oxygen mass fraction [48], combustible oxygen mass fraction [49] and dilution ratio [36] are also employed as suitable indicators of

Table 1
EGR rate as calculated by various researchers.

Investigator(s)	Types of EGR	EGR rate calculation	EGR rate (%)
Ladommatos et al. [32]	Short-path, cooled EGR	Based on volume flow rates $EGR_v(\%) = \frac{(V_{af} - V_{ae})}{(V_{af})} \cdot 100$ Based on mass flow rates $EGR_m(\%) = \frac{m_e}{m_c} \cdot 100$	2,7,8.1,18,22,28,47,49
Jacobs et al. [33]	Short-path, cooled EGR	Not mentioned	5,10,15,20,25
Kohketsu et al. [34]	Long and Short-path, uncooled EGR	$EGR(\%) = \frac{[CO_2]_{inlet} - [CO_2]_{ambient}}{[CO_2]_{exhaust} - [CO_2]_{ambient}}$	10,20,30,40
Baert et.al [31]	Short-path, cooled EGR	$EGR(\%) = \frac{[CO_2]_{inlet} - [CO_2]_{ambient}}{[CO_2]_{exhaust}}$	5,10,15,20
Shin et al. [35]	Long and Short-path, cooled EGR	$EGR(\%) = \frac{[CO_2]_{inlet}}{[CO_2]_{exhaust}}$	2,16,31
Zheng et al. [7]	Hot and cooled EGR	$EGR(\%) = \frac{[CO_2]_{inlet}}{[CO_2]_{exhaust}}$	5,10,15,20,25
Lehto et al. [101]	Simulated EGR (Nitrogen)	$EGR(\%) = \frac{m_{N_2}}{m_{air} + m_{N_2}}$	2.5,5,7.5,10,15,20,25, 30,35,40,45

EGR rate. To calculate oxygen mass fraction (Xo_2) from the EGR mass rate (X_r), d'Ambrosio et al. [48] proposed the following expression:

$$Xo_2 = \frac{\frac{Mo_2(no_2)_{air}}{\sum_j (n_j M_j)_{air}} + \frac{X_r}{1-X_r} \frac{Mo_2(no_2)_{EGR}}{\sum_j (n_j M_j)_{EGR}}}{1 + \frac{X_r}{1-X_r}} \tag{3}$$

where the first and second term in the numerator represents the mass concentration of oxygen in the ambient air and in the EGR gas respectively. The authors correlated the calculated oxygen mass fraction with the diesel NO_x emissions and proved their algorithm is an effective tool to deduce EGR. Oxygen fractions indicate the O_2 fraction in the intake charge and Snyder et al. [49] proposed a control-variable-based accommodation strategy by determining combustible oxygen mass fraction (COMF) as

$$\text{COMF} = \frac{Y_{O2a}\dot{m}_{a+}Y_{O2exh}\dot{m}_{EGR} + Y_{O2f}\dot{m}_f}{\dot{m}_a + \dot{m}_{EGR} + \dot{m}_f} \tag{4}$$

where Y_{O2a} is mass fraction of oxygen in air; $\dot{m}_a$ is air mass flow rate; Y_{O2exh} is mass fraction of oxygen in exhaust and $\dot{m}_{EGR}$ is EGR flow rate; Y_{O2f} is mass fraction of oxygen in fuel and $\dot{m}_f$ is fuel flow rate. Unlike diesel fuel, the oxygenated fuel like biodiesel when injected contribute greater combustible oxygen mass fractions which can be estimated from Eq. (4). The authors observed an exponential relationship between the estimated COMF and measured brake-specific NO_x for the tested fuels at various operating conditions. They attributed it to the increase in fraction of inert species with increase in EGR fraction, resulting in reduced COMF and NO_x.

Maiboom et al. [36] investigated the influence of dilution ratio on an automotive diesel engine equipped with a low pressure loop EGR to study the influence of higher EGR rate on the combustion and NO_x-PM characteristics. The authors defined dilution ratio (DR) as

$$DR(\%) = 100 \cdot \frac{21 - Xo_{2\_in}}{21} \tag{5}$$

where X_{O2_in} is the measured oxygen concentration in the inlet manifold. Higher Ignition delay and lower NO_x emissions were reported with increasing dilution ratio.

The authors observe that the intention of replacing EGR ratio with other parameters (Eqs. (3)–(5)) provides a reliable measurement and mostly intended to enhance the EGR transient measurement for better control approaches. However, the calculated EGR trend by various methods should be consistent over a wide range of engine operating conditions.

3. Advanced engine combustion

3.1. Homogeneous charge preparation strategies

Homogeneous mixture preparation plays a crucial role in the reduction of soot and nitrogen oxide emissions. The local fuel-rich regions, which are responsible for nitrogen oxide emissions could be reduced by the effective preparation of homogeneous charge. The lean homogeneous mixture also results in reduced soot emissions [50,51]. The homogeneous mixture can be achieved through either external mixture formation, or in-cylinder direct injection as shown in Fig. 6. Both strategies have their inherent limitations. The external mixture formation results in reduced volumetric efficiency, while the in-cylinder direct injection leads to dilution of lubricating oil.

3.1.1. External mixture preparation

Different techniques like manifold induction, fumigation, port fuel injection and wide open throttle have been attempted for external homogenous mixture preparation for highly volatile fuels like gasoline and alcohols. Even for low volatile fuels like diesel, the mixture can be externally prepared by using a fuel vaporizer. External mixture formation is the simplest method for mixing gaseous fuel with air in the inlet manifold. This type of external mixture formation was attempted by different researchers with different fuels like acetylene [52,53], hydrogen [54,55] and biogas [56,57].

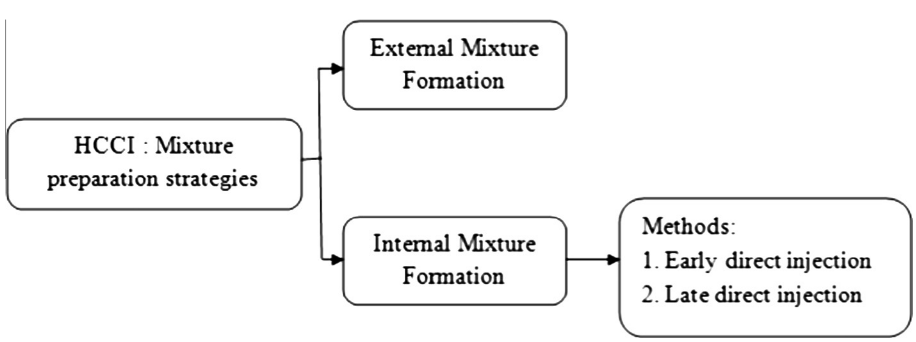

Fig. 6. Strategies for homogeneous mixture preparation [14].

First Homogeneous Charge Compression Ignition (HCCI) combustion process termed as Active-Thermo Atmospheric Combustion (ATAC) was recognized on a two stroke engine, in which the entire mixture in the combustion chamber burns gradually in the meantime [58,59]. Another HCCI combustion system called as Active Radical Combustion (ARC), was successfully demonstrated on two stroke gasoline engines, where the exhaust gas temperature decreased to ~19% during the entire engine load range [60].

To lower the diesel smoke emissions, few researchers have attempted external mixture preparation through an electronically controlled fuel vaporizer [61–63]. Moreover, EGR was used for controlling the combustion phasing and NO_x emissions. For successful implementation, the operating temperature of vaporizer was maintained above the boiling temperature of the less volatile fuel [64].

Another advanced diesel combustion concept is partially premixed charge compression ignition (PPCCI). Several research works have been carried out using diesel fuel with a different premix ratio [65–67]. The premixed ratio (r_p) is expressed as

$$r_p = \frac{Q_p}{Q_t} \tag{6}$$

where (Q_p) is ratio of energy of premixed fuel and (Q_t) is the total energy, given as

$$Q_p = \frac{m_p}{h_{up}} \tag{7}$$

$$Q_t = m_p h_{up} + m_d h_{ud} \tag{8}$$

where (m_p) is the premixed fuel mass, (m_d) is the diesel fuel mass, (h_u) is the lower heating value, and subscripts 'p' and 'd' denotes premixed and diesel fuel, respectively [65]. Among the various methods, port fuel injection (PFI) is the simplest method which yields improvement in volumetric efficiency and fuel distribution over carburetion. High volatile fuels like gasoline and alcohols are preferred for port fuel injection [68–71].

3.1.2. In-cylinder mixture preparation

In order to overcome the wall impingement problems associated with low volatile fuels like diesel in a PFI system, the in-cylinder mixture preparation technique was investigated. There are two strategies being used for in-cylinder mixture preparation: (i) injecting the fuel earlier during compression stroke, called as early direct injection [72–75] (ii) fuel injection after top dead centre, called as late direct injection [76]. Both the strategies are using high injection pressure with multi-hole nozzle geometry to increase spray disintegration and thus the formation rate of homogenous mixture. Table 2 provides the different in-cylinder direct injection strategies employed for diesel HCCI engines.

The motivation behind the possible implementation of HCCI combustion roots from its potential for significant reduction in both NO_x and PM emissions. In comparison to conventional diesel combustion, NO_x can be drastically reduced (~98%) with HCCI [77–79]. In addition, the absence of localized fuel rich regions and diffusion combustion suppresses the soot formation. Low temperature reactions and flameless combustion attained in HCCI lead to low heat rejection rates and thus beneficial in attaining higher thermodynamic cycle efficiencies [80–82]. However, the major hindrances for practical implementation of HCCI includes [14–16]: (i) poor cold start characteristics (ii) higher HC and CO emissions (iii) limited operational range (iv) difficulty in combustion phase control (v) control over auto-ignition and (vi) homogeneous charge preparation. The operating parameters for HCCI control and operating range extension are grouped into two categories: (a) Mixture auto-ignition properties (b) Mixture temperature history [83]. The implementation of EGR in an HCCI engine promotes both mixture auto-ignition and temperature history. Different EGR techniques attempted by various researchers for combustion phase control in HCCI engines are discussed in the following section.

4. EGR for advanced engine combustion concepts

Kanda et al. [84] conducted engine experiment with reduced compression ratio and adopted an injection nozzle with narrow spray angle. Moreover, a large amount of EGR up to 54% was being used to retard the ignition timing towards top dead centre. Reduced NO_x and increased HC, CO emissions were reported. They concluded that early advanced ignition and lean air-fuel mixture lowers the indicated mean effective pressure.

Boyarski et al. [85] achieved premixed combustion ignition (PCI) in a light duty diesel engine by adopting a nozzle with 120° spray angle with modified piston bowl geometry. The authors utilized EGR as high as 68% to control the onset of combustion process and reported 10% penalty in fuel consumption with an increase in HC and CO emissions. Iwabuchi et al. [86] also established PCI following impinged spray nozzle and showed an improvement in fuel consumption (10%). The authors concluded that the impinged spray nozzle characteristics were found to be suitable for early direct injection HCCI system. Xu et al. [87] carried out engine experiments by varying the cam phasing to trap internal EGR for fuel evaporation and charge homogenization. In this experiment, HCCI combustion was accomplished by fuel injection at exhaust stroke top dead centre using electronically controlled fuel injector. This aids in complete evaporation of liquid fuel by hot residuals and thus leading to homogeneous charge formation.

Das et al. [88] attempted HCCI combustion using dual injection strategy and EGR as shown in Fig. 7. In this study, the effects of premixed ratio on engine performance and emission characteristics were studied using different EGR rates. The authors observed 76%

Table 2
Different in-cylinder direct injection strategies for diesel HCCI.

Author	Strategy	Salient features	Merits	Demerits
Nishijima et al. [72]	Early in-cylinder direct injection	Two sprays impinged each other at the cylinder centre	Low NO_x and UBHC emissions	Higher fuel consumption
Hashizume et al. [73]	Early in-cylinder direct injection	Premixed lean combustion followed by diffusion combustion	Reduced NO_x at high loads	Higher UBHC emissions
Hasegawa et al. [74]	Early in-cylinder direct injection	An early injection for fuel diffusion and secondary one as an ignition trigger	Low NO_x and smoke emissions	Limited to low loads
Su et al. [75]	Early in-cylinder direct injection	Multi-pulse injection for limiting spray penetration	Ultra low NO_x and smoke emissions	Design of injection scheme is complicated for high power output
Kimura et al. [76]	Late in-cylinder direct injection	Late fuel injection timing with high swirl ratio, EGR levels with modified piston geometry	Ultra low emissions possible with 5 way catalyst	Operational limitations: 50% max torque and 75% max speed.

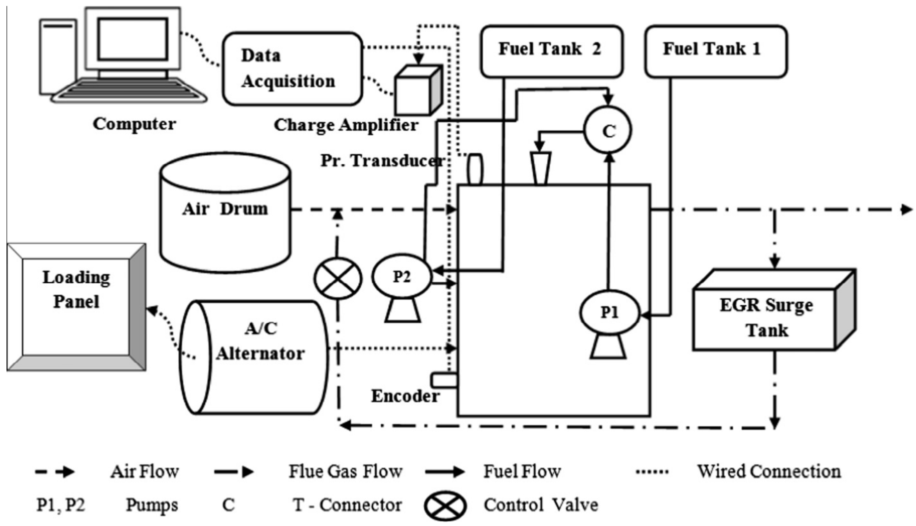

Fig. 7. HCCI experimental set up with dual injection and EGR [88].

reduction in NO_x and 40% lower smoke emissions with 80% premixed ratio and 30% EGR. An increase in premixed ratio also resulted in improved mean effective pressure, specific fuel consumption with penalties on HC and CO emissions.

Shi et al. [89] conducted an engine experiment in a single cylinder engine involving both internal and external EGR techniques to promote fuel vaporization and reduction in in-cylinder temperature respectively. Externally cooled EGR (50%) was used in this experiment to reduce the mixture temperature during the compression stroke. However, higher EGR has found to increase the CO emissions due to incomplete combustion.

Garcia et al. [90] compared the performance characteristics of HCCI combustion over conventional diesel combustion with a maximum EGR of 33%. The authors observed that the HCCI combustion was found to produce ultra-low NO_x emissions with a marginal change in the soot emissions.

Singh et al. [62] conducted an engine experiment in a twin cylinder, direct injection diesel engine, where one cylinder was modified to operate in HCCI combustion while the other cylinder operated in conventional diesel combustion mode. During this engine test, the mixture strength was varied between $\lambda = 4.96$ (lean or misfire limit) to 2.56 (rich and knocking limit) and 20% EGR was used to control the combustion phasing. Efficient HCCI combustion was observed when $\lambda \leqslant 3.70$, while rough engine operation was experienced at high load conditions (i.e. $\lambda < 3.0$).

Zhao et al. [91] studied the effect of external EGR and di-methyl ether (DME) premix ratio on the performance, combustion and emission characteristics of twin cylinder, naturally aspirated diesel engine. In this experimental study, EGR was varied from 0 to 27%, while the premix ratio from 0 to 30%. While increasing the EGR rate, the NO_x emissions were found to be reduced with an increase in smoke, HC and CO emissions. However, a significant reduction in NO_x and smoke emissions were reported with higher premix and EGR rate.

Chen et al. [92] investigated the effect of port fuel injection (PFI) of n-butanol and EGR on the combustion and emission characteristics of a diesel engine. The engine was equipped with an independent intake and exhaust system. In this experiment, the EGR was varied from 0%, 15% and 45% for four mixing ratio of butanol as 0%, 20%, 38% and 47%. By a quantitative examination, it was observed that as EGR rate increased to 45%, the NO_x emissions decreased by 97%, whilst the soot emissions increased drastically. However, with the combination of PFI of butanol (47%) and EGR (45%), the soot and NOx emissions diminished by 88% and 17% respectively. Further, the authors claim that PFI of butanol possess the potential to overcome the traditional diesel NO_x-soot trade-off problem.

Wimmer et al. [93] investigated the late injection strategy of HCCI combustion (MK combustion). Modulated Kinetics (MK) combustion is a type of premixed, low temperature combustion process. Considerable reduction in NO_x is achieved with high EGR, without a substantial increase in smoke. The retarded injection scheme lengthens the ignition delay period and thereby prevents smoke formation. However, fuel injection after top dead centre (aTDC) will lead to a drop in fuel economy and higher hydrocarbon (HC) emissions. In order to overcome these undesirable effects, in-cylinder gas flow is strengthened by an optimized design of combustion chamber and intake port geometry. In this combustion, around 40% EGR was used to reduce the NOx emissions to a considerable level.

Kondo et al. [94] attempted different technologies to suppress combustion noise generated from a diesel engine operating on HCCI mode. By combining EGR (~44%) and retarded injection (9° aTDC), the authors observed that the combustion noise lowered by 10 dB without any penalties on other emissions. Fang et al. [95] studied the air–fuel mixing and combustion phenomena in an optical engine using the single retarded injection scheme. The start of injection (SOI) was retarded by an angle of 2° for the same injection pressure and exhaust gases were recirculated up to 25%. This led to a simultaneous reduction of soot and NO_x emissions with premixed combustion modes under part load conditions. Table 3 shows the overview of EGR implementation in engines of advanced combustion concepts.

The maximum concentration of EGR rates attempted by various researchers for single and multi- cylinder engines are provided in Figs. 8 and 9 respectively. Exhaust gases as high as 70% were recycled in low temperature combustion systems to maintain the combustion temperatures around 1600 K at all operating conditions. The lower combustion temperatures decelerate the NO_x formation, whilst acting as barrier for the soot oxidation. In addition to the lower in-cylinder temperature, the higher charge dilution with EGR limits the soot formation by reducing the precursor formation of soot particles. Further, extended ignition delays are observed with low temperature combustion. It provides more time for fuel evaporation and thus promotes mixture homogeneity, which results in uniform temperature across the combustion chamber [96–99].

It is interesting to observe that, the dilution effect plays a dominant role in NO_x formation in the case of low temperature com-

Table 3
Overview of EGR implementation in Advanced Engine Combustion.

Investigator (s)	Engine type & modification, mode of combustion	Useful inferences
Kanda et al. [84]	Single-cylinder and injector with a narrower cone angle, PCCI system	High EGR (54%) is introduced to retard the ignition timing toward TDC Lower NOx emission with penalties in HC, CO, Soot and IMEP
Boyarski et al. [85]	Light duty diesel engine with 120° spray angle nozzle, modified piston bowl geometry, PCI combustion system	High EGR (68%) is used to control the onset of combustion, which penalized fuel consumption, HC & CO emissions
Iwabuchi et al. [86]	Single cylinder diesel engine with impinged spray nozzle, PCI combustion	Lower HC and smoke emissions with poor transient response and instability in engine temperature characteristics
Xu et al. [87]	Single cylinder diesel engine with negative valve overlap, HCCI combustion system	During mode shifting, too late and early switching of EGR causes unsmooth combustion (CA50)
Das et al. [88]	Four stroke single cylinder diesel engine. HCCI combustion by dual injection	30% of exhaust gases had lowered NOx by 76% and smoke by 40% using 80% promised ratio, with mild knocking combustion at higher loads
Shi et al. [89]	Four stroke single cylinder diesel engine, Diesel HCCI combustion with both internal and external EGR	Internal EGR benefits homogeneous mixture formation and reduces smoke emissions, whereas external EGR helps to expand the load limit of HCCI
Garcia et al. [90]	Four stroke single cylinder diesel engine. HCCI and conventional diesel combustion with external EGR	The optimal injection timing for HCCI combustion mode is 45 deg. CA BTDC. Negligible soot emissions with ultra-low NO_x emissions are observed
Singh et al. [62]	Four stroke two cylinder diesel engine with one cylinder in HCCI combustion mode; and the other in conventional diesel combustion mode	Efficient HCCI combustion was achieved for ($\lambda > 3.70$); while at high load conditions ($\lambda < 3.0$), the HCCI combustion is found to be noisy
Zhao et al. [91]	Four-stroke two-cylinder diesel engine. Premixed combustion with external EGR	PCCI engine with EGR delayed SOC and prolonged combustion duration. With increased EGR, NOx emission decreased, but smoke, CO and HC emissions increased
Chen et al. [92]	Direct injection (DI) diesel engine with independent intake and exhaust system, Port fuel injection (PFI) of n-butanol	Under low EGR rates, soot emission decreases, but NO*x* emissions increased, however, high EGR rate lowered both soot and NO*x* emissions with increased butanol concentration
Wimmer et al. [93]	Late injection MK combustion HCCI system, EGR ~ 40%	A significant reduction in NOx emission is achieved

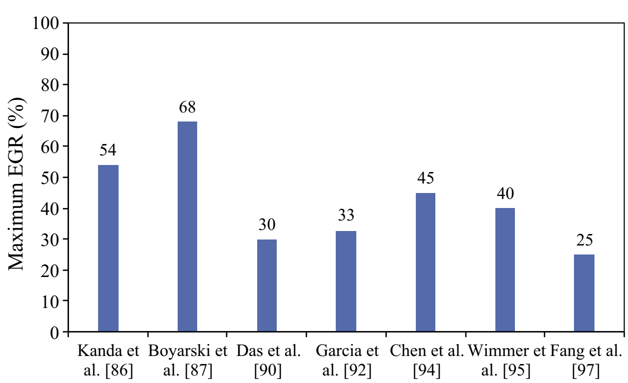

Fig. 8. Maximum EGR usage in single cylinder engines for advanced combustion control.

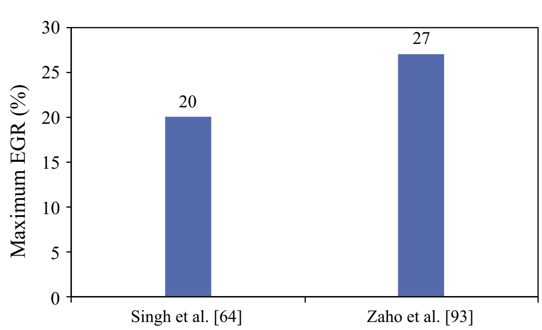

Fig. 9. Maximum EGR usage in multi-cylinder engines for advanced combustion control.

bustion concepts as well [18]. However, auto ignition event is influenced by thermal effect to a larger extent, while chemical and dilution effects are insignificant. To summarize, the dilution effect of EGR is found to be the major responsible contributor for NO_x reduction mechanism in both conventional and advanced diesel combustion process. However, with regard to soot emission, the mixture homogeneity and the increased ignition delay are the prevailing factor in the case of advanced diesel combustion process.

5. EGR for alternate fuelled engines

5.1. EGR for alternate liquid fuelled engines

The application of EGR in alternate fuelled engines is immense and generally employed for emission mitigation. The current review discusses in detail about the application of EGR with liquid alternate fuels like raw oils, processed oils, alcohols and gaseous alternate fuels for compression ignition engines.

Mani et al. [100] experimentally investigated the effect of cooled EGR on a single cylinder diesel engine using 100% waste plastic oil. The NO_x emissions are reported to be higher with waste plastic oil operation. However, recycling 20% exhaust gases lowered the NO_x emissions without significant changes in smoke, CO and HC emissions. Thus the authors recommended the usage of waste plastic oil as an alternate fuel in diesel engines by adopting the EGR strategy.

To avoid the practical problems associated with substitution of 100% raw/ straight vegetable oil in compression ignition engines, the raw oils are processed by means of transesterification, hydrogenation, pyrolysis and dilution etc. Lehto et al. [101] compared the NO_x and smoke emission characteristics of hydrogenated vegetable oil (HVO) to fossil diesel (EN590) in a single-cylinder research engine. The engine test was conducted with simulated EGR using nitrogen gas and it was shown that the NO_x reduction with 30% EGR was similar for diesel and hydro treated oil. However, the lower smoke emissions of HVO enable higher EGR percentages in comparison with fossil diesel.

Liu et al. [102] investigated the combustion and emission characteristics of HVO blends (30 and 60% by vol.) in a turbocharged direct injection diesel engine with maximum EGR concentration of 46%. At high EGR rate, HVO blends resulted in shorter ignition delay and longer combustion duration, whereas the higher combustion temperatures of HVO blends increased the soot formation rate. It was reported that, the NO_x emission decreases linearly with increasing EGR, whereas total hydrocarbon and carbon monoxide emissions increases.

Agarwal et al. [47] investigated the usage of biodiesel and EGR simultaneously to control the regulated pollutants in a two-cylinder diesel engine. They observed the effectiveness of NO_x reduction by EGR varies with load and suggested to use higher EGR rates at low load and lower EGR rates at high load conditions. A 20% biodiesel blend with 15% EGR is reported to be the optimum strategy for biodiesel operation. Kawano et al. [103] tested the effectiveness of HPL and LPL EGR system in a four cylinder turbo charged diesel engine using rapeseed methyl ester (RME). The test results revealed that LPL EGR has the potential for effective NOx reduction than HPL EGR, because it expands the upper limit of EGR rate (45%). However, the HPL EGR with biodiesel drastically reduces NO_x emission, without an increase in PM emission, because soot formation was suppressed by the oxygen in RME. The observations of other available investigations concerning the effects of EGR on biodiesel-NO_x are included in Table 4. Marques et al. [104] evaluated the EGR technique with biodiesel blends in a single and two cylinder diesel engines. The authors varied EGR from 5 to 30% and concluded that with the increase in EGR rate, the NO_x emissions decreases. The authors opined that, the reduction in oxygen concentration and decreased flame temperatures with EGR to be the major cause for NO_x reduction.

The potential of DME as an alternative fuel for compression-ignition engines were reviewed by Arcoumanis et al. [105]. The favourable properties of DME as a candidate fuel in compression ignition engine includes a higher cetane number, lower auto-ignition temperature and higher oxygen content (around 35% by mass) resulting soot-free combustion. However, slight modifications are required in the fuel injection system to handle this less dense and highly evaporative DME. The NO_x emission from DME-fuelled engines is found to be contradictory, further DME engines allow a higher EGR rate for NO_x reduction without being restricted by the NO_x-soot trade-off. Thus, future regulations are satisfied with high EGR in combination with a lean NO_x trap after treatment system. Moreover, Kapus and Ofner [106] optimized the EGR and injection parameters for a DME-fueled engine and achieved lower combustion noise in comparison to diesel operation. Sato et al. [107] observed higher output and lower NO_x for DME operated engine with supercharging and EGR compared to fossil diesel operation.

Murayama et al. [108] carried out steady-state engine testing in a single cylinder research DI diesel engine for studying the combined effects of Dimethyl Carbonate (DMC) with EGR as shown in Fig. 10. For this investigation, the authors employed cooled EGR by submerging the EGR pipeline through a cooling water tank and further wounding it with cooling water coils. However, the condensed water has been allowed to pass along the line of cooled EGR. The EGR test results indicated 50% NO_x reduction for 20% EGR rate. Further, the combined use of EGR (5, 10 and 15%) and DMC (10%) lowered both the NO_x and smoke emissions substantially.

In general, the exhaust gas recycle is an effective technique for reducing NO_x emission and for biodiesel fueled engines provides a better trade-off with HC and CO [106]. The NO_x reduction due to EGR is widely reported in case of both fossil diesel and their alternatives. The major attributes for this effect include:

- Reduced in-cylinder oxygen concentration leading to a temperature drop in the burning zone due to dilution, thermal, and chemical effects [6,47,104,109–111].
- An increased H_2O and CO_2 concentrations in the engine exhaust due to combustion of the differing constituents of biodiesel and their diesel blends [112].

The lower in-cylinder temperature and increased latent heat of evaporation, significantly lowers the NO_x formation in alcohol fuelled engines than diesel-fuelled counterparts. However, few research studies have explored the potential of EGR in alcohol fuelled engines. Haupt et al. [113] studied an exhaust after-treatment system comprising an exhaust gas recirculation (EGR), oxidative catalyst and a diesel particulate filter (DPF) coupled to

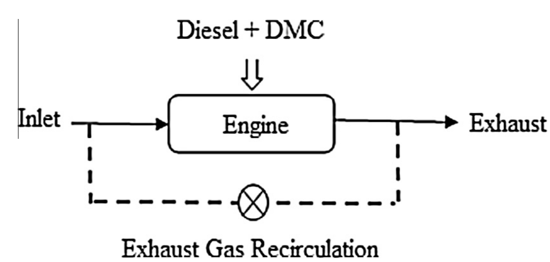

Fig. 10. DMC operated diesel engine with EGR [108].

Table 4
Summary of NO_x control with EGR for biodiesel and biodiesel blends.

Author	Engine – fuel(s)	EGR method & flow rate	NO_x results	Trade-off
Agarwal et al. [47]	Multi-cylinder, naturally aspirated diesel engine - Rice bran methyl ester blends	10, 15 and 20%	Maximum NO_x reduction is observed at full load	15% EGR with 20% biodiesel blend is found optimum
Kawano et al. [103]	Four cylinder turbo charged diesel engine - Rapeseed methyl ester	HPL and LPL EGR system 0–50%	$BSNO_x$ emission steadily reduced from 6 to 0.7 g/kWh	Both HC and CO for RME are higher than diesel at high EGR
Marques et al. [104]	Single and two cylinder engines - 15% biodiesel blend	0–30%	Prominent degree of NOx reduction at higher loads	NO_x emission is slightly lower than diesel fuel at 20% EGR
Tsolakis et al. [112]	Single-cylinder, air-cooled, diesel engine - B20, B50 and B100 RME	10, and 20%	Higher NOx reduction with RME and its blends	Biodiesel constituents changes the exhaust gas composition
Pradeep et al. [142]	Single cylinder, direct injection - Jatropha methyl ester	5, 10, 15%	Full load biodiesel-NO emission at 15% EGR, is found lower than diesel	15% EGR is optimal for NO reduction and lower smoke, CO, HC
Saleh et al. [143]	Multi-cylinder, direct injection-Jojoba methyl ester	0–50%	NOx is lowered by 50% at low load conditions	EGR (5–15%) provided better trade-off between HC, CO and NO_x
Bhaskar et al. [110]	Single cylinder, DI Diesel engine -Fish biodiesel and blends (20,40,60,80)	10, 20 and 30%	NO_x reduction is proportional to EGR rate	NO_x and soot are optimal at 20% EGR
Jiménez-Espadafor et al. [144]	Single cylinder, naturally aspirated direct injection-Colza biodiesel and blends (0,30,65,100)	Varying operating Conditions	62% reduction in NO_x with B100 and 30% EGR	Biodiesel Fuel blends lower NO_x but increases BSFC

a 9-liter ethanol-fuelled diesel engine. The EGR retrofitted in the turbocharged diesel engine is a low pressure loop EGR, where the flow rate of EGR is controlled by a patented EGR throttle. By employing this after-treatment system, the authors claim to significantly lower the HC, CO and NO concentrations from Euro III regime to Euro IV limit at the tested conditions. However, a significant increase in NO_2 / NO ratio was reported. The authors also examined the unregulated emissions like formaldehyde, acetaldehyde, acrolein, etc. Their test results showed a significant reduction in all the aldehydes and mostly all the hydrocarbons with ethanol-operated diesel engine. For low level emissions, the authors suggested to retune the engine system or catalyst replacement.

Li et al. [114] proposed an approach to counterbalance the ignition problem with alcohol fuels (lower cetane number) by the combination of un-cooled internal EGR and diesel pilot injection. The authors demonstrated this approach in a two-stroke single-cylinder diesel engine by varying the scavenging pressure to control the EGR rate with ethanol (95% purity) fuel. However, a small quantity of castor oil (1%) was added to ethanol to improve the fuel system reliability and durability. They concluded that the NO_x-smoke trade-off could be alleviated in an alcohol-fuelled engine by following this approach with an improvement in thermal efficiency of 2–3% than fossil diesel operation in mid and high load conditions.

Ogawa et al. [115] studied the effect of EGR usage in a single cylinder, small diesel engine (0.83 L), equipped with supercharger and common rail injection system. The effects of intake oxygen displacement due to the addition of EGR and the changes in injection pressure were investigated for alcohol and vegetable oil blends (20% ethanol + 40% 1-butanol + 40% vegetable oil blend and 50% 1-butanol + 50% vegetable oil). LPL EGR was preferred to the high pressure one, and the cooled exhaust gas was ducted upstream of the supercharger as shown in Fig. 11. While the intake gas temperature was maintained below 30 °C. It was shown that the NO_x-smoke trade-off improved with all the alcohol blended fuels at lower intake oxygen concentrations. Whereas at 16% intake oxygen concentration with the blend of 20% ethanol + 40% 1-butanol + 40% vegetable oil yielded very low NO_x and smokeless operation.

Thangaraja et al. [116] compared the effect of HPL EGR to methanol blended Karanja biodiesel in a turbocharged direct injection diesel engine. The engine experiments were carried out at maximum torque speed of 1400 rpm and higher load (60%, 80% & 100%) conditions by recycling 5% of exhaust gas and 10% by vol. of methanol to Karanja biodiesel. Their experimental results using the two strategies showed a significant reduction in both NO (35–45%) and smoke (60 and 90%) concentrations relative to fossil diesel operation. The authors suggested using methanol addition relative to the EGR strategy from convenience of fuel economy and smoke/PM standpoint.

5.2. EGR for alternate gaseous fuelled engines

Several researchers have studied the effects of EGR on gaseous fuels like hydrogen, natural gas in a diesel engine. One of the widely adapted methods to utilize natural gas in compression ignition engines is by following the dual-fuel operation [117]. Weavar and Turner [118] presented an exhaustive review article on dual-fuel natural gas engines. The authors defined dual-fuel engine as an internal combustion engine in which the primary fuel (usually natural gas) is inducted along with the intake air, and the air/fuel mixture is ignited near the end of compression stroke by injecting a small amount of secondary/pilot fuel (diesel) as shown in Fig. 12. This review also covered the potential applications, advantages and challenges of dual-fuel engines. Even though the fuel interchangeability and ease of retro-fitment are considered advantageous; poor part load performance is a major hindrance with dual fuel mode of operation. Among the various strategies, EGR has also been considered to overcome the poor performance at low load operating conditions [13,119].

Daisho et al. [13] carried out an parametric study in a dual-fuel operated direct-injection diesel engine admitting hot and cold EGR. EGR ratios of 20 and 40% were attempted to improve the part load (25 and 50%) performance. With hot EGR, the rise in intake charge temperature promoted better combustion of premixed natural gas mixtures. This in turn enhanced the thermal efficiency and reduced the unburned hydrocarbon emissions. However, the cooled exhaust gas (to room temperature) by means of a water-cooled heat exchanger yielded better NOx emission characteristics. It is interesting to notice that with higher natural gas fraction (above 40%), the smoke level diminished with hot EGR and thus the authors opined that EGR to be the most effective in improving exhaust emissions.

Similar experimental study was carried out by Selim [12] in an indirect injection diesel engine with the pre-combustion chamber. The author studied the effect of EGR on combustion noise and thermal efficiency of a dual fuel engine operated with diesel and compressed natural gas. Both hot and cold exhaust gases were recycled within lower concentrations (<15%). By insulating the EGR pipes, the author attempted hot EGR and raised the recycled exhaust gas temperature from 30 to 55 °C. The author concluded that a marginal quantity of EGR (5%) is beneficial in terms of thermal efficiency, combustion noise and NO_x emission standpoint.

Ishida et al. [120] developed a natural gas engine for medium-duty truck applications comprising a catalytic converter and a cooled EGR system. Apart from NO_x, CO, HC & PM emissions, the authors also measured the CO_2 emission following D13-mode operating conditions. The greenhouse gas emission from the

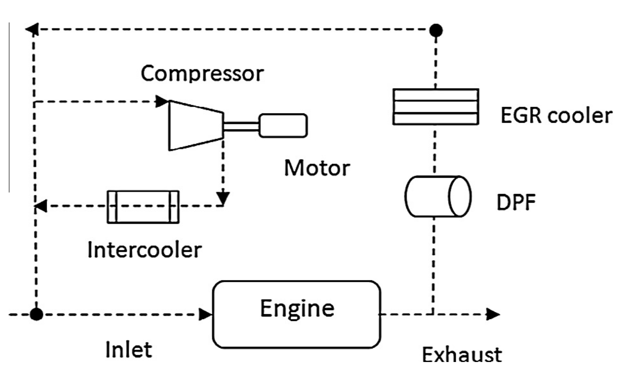

Fig. 11. Cooled EGR layout in a supercharged diesel engine [115].

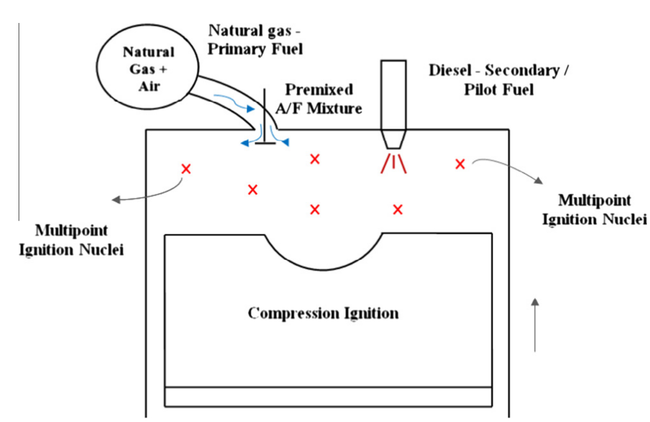

Fig. 12. Configuration of dual fuel operation in a compression ignition engine.

natural gas engine is reported to be 15% lower than conventional diesel operation and thus the authors proposed natural gas engine as the most suitable for vehicle applications. In a recent experimental study by Besch et al. [121], similar reduction in brake-specific CO_2 emissions was observed using dual-fuel operation.

Lee et al. [122] systemically modified the inlet charge with propane gas and inducted EGR into the inlet port of a high-speed direct-injection (HSDI) single-cylinder diesel engine. The authors evaluated the characteristics of dual fuel combustion for high and low air-fuel ratios. The EGR rate was calculated based on the CO_2 approach and was varied from 20, 40 and 45%. They showed that the dual-fuel combustion with EGR reduces both NO_x and PM with a decrease in IMEP in comparison to diesel combustion mode. The authors also highlighted that the lower cetane number of propane causes dual-fuel combustion to be more sensitive to EGR rates than diesel combustion.

In a controlled laboratory study, Saravanan et al. [123] enriched the intake charge with gaseous hydrogen and adopted cooled EGR technique to study the performance and emission characteristics of a single-cylinder diesel engine. The flow rates of hydrogen gas and EGR employed in this study were 20 L/min, 15% and 25% respectively. Though the authors observed a sharp reduction in NO_x emission, other regulated emissions viz. CO, HC and smoke began to increase with EGR addition. However, the enrichment of hydrogen has increased the brake thermal efficiency by 6%.

To summarize the effectiveness of EGR in alternate fuelled engines, it has been observed that EGR remains as an effective method to decrease NO_x emission and to improve the engine performance. Especially hot EGR is preferred to promote better combustion and a marginal quantity of cooled EGR is sufficient for suppressing the biodiesel-NO_x penalty. Further, oxygenated fuels allow higher EGR concentration without being restricted by the conventional NO_x-soot trade-off.

6. Effect of EGR on engine oil contamination and engine wear

Even though EGR has proven to be effective for light duty diesel engines, it results in a sharp increase in in-cylinder soot formation. This increase in engine soot emission has augmented the wearing of engine parts such as: cylinder liner, piston rings, valve-train system and bearings. The engine wear is majorly influenced by lubricant additive depletion. The lubricant ceases to perform its intended function due to changes in its chemical properties, which was mainly effected by the accumulation of soot in engine oil. Abrasion, adhesion, corrosion, scuffing and additive depletion are the most important wear mechanisms in a diesel engine. Out of these mechanisms, abrasion, adhesion and scuffing entail mechanical damage to surfaces; while corrosion and additive depletion necessitate a series of chemical reactions and finally resulting in wear. The different engine operating conditions such as cold start, warm up, idling, high acceleration can destroy the lubricant contact zone and thus results in breakdown of the full oil film permitting contact between soot particles and the engine surfaces. In general, the lubricant boundary layers have the thickness of 0.001 μm to 0.05 μm and hence the soot particles with their diameters ranging from 0.01 μm to 0.8 μm could cause abrasive wear.

Gautam et al. [124] studied the effect of soot contamination on engine oil using a three body wear machine. With the help of a statistical tool, the author performed the experiments with different engine oil blends (oil + additives). The engine oil blends were formulated with different intensities of phosphorous, dispersant and sulfonate and also with two different levels. Wear scars and surface chemical analysis was carried out using scanning electron microscope (SEM) and energy–dispersive X-ray analysis (EDAX) respectively. It was found that engine wear was increasing in proportion to soot concentration, whereas it gets decreased with high intensities of phosphorous. Moreover, the SEM images confirmed that that the diesel soot was abrasive.

George et al. [125,126] continued similar experimental study to examine the effect of diesel soot on the viscosity of lubricant oil using rotary viscometer. In addition to different oil blends, three levels of soot concentration (low - 0%; medium - 2% and high - 4%) was also considered in this study within the range of temperatures (40–90 °C). The viscosity of oil blends was found to increase with higher level of soot concentration and dispersant. High viscosity oils could result in increased engine wear due to pumping problems.

Antusch et al. [127] characterized the soot particles by high resolution electron microscopy (HRTEM), electron spin resonance (ESR) and energy electron loss spectroscopy (EELS). Further, the authors assessed the tribological behaviour of soot by means of a pin-on-disk tribometer coupled to a high resolution wear measurement system. The experimental results revealed that the wear was primarily due to the soot reactivity rather than their mechanical properties. The authors concluded that factors such as morphology, surface chemistry and reactivity of soot particulates were playing a major role in the wear of engine components.

Hu et al. [128] investigated the effect of soot particles on the tribological behaviour of engine oils namely formulated engine lubricant (CDSAE15 W-40) and base oil (150SN). Soot particle contamination was simulated using carbon black. The friction and wear were measured using a four-ball tribometer. It was showed that the anti-wear and anti-friction properties of the CDSAE 15W-40 were better than 150SN base oil, which was attributed to the dispersed carbon black and presence of additives. Moreover, it was highlighted that the wear resistance and frictional properties of 150SN base oil could be improved by addition of the dispersant.

In general, particles produced from compression ignition engines are of major concern to diesel engine manufacturers due to their impact on engine performance and engine wear. Further, the usage of EGR in diesel engines also influences the particle formation and its severity intensifies as the engine becomes older [129]. Thus to better understand how EGR affects the diesel particles of conventional, advanced combustion and alternate fuelled engines, the following section is provided.

7. Effect of EGR on diesel particle characteristics

In the spectrum of particle size distribution, particles emitted from diesel engines are mostly in the nano scale i.e. $D_p < 50$ nm (nucleation mode). These nucleation mode particles typically comprised of volatile organic or sulfur compounds and solid carbon or metal compounds. On the other hand, accumulation mode particles ($50\text{ nm} < D_p < 1000\text{ nm}$) mainly composed of carbonaceous soot agglomerates and contributes to a greater extent to the formation of particle mass [129]. While smaller particle with diameter less than 17 nm have negligible effect on the total particle mass; larger particles (17–170 nm) have significant effect on particle mass [130]. The following equation (Eq. (9)) represents the cubed dependence of particle mass (M_p) on the particle diameter (d).

$$M_{p=}\rho_p \frac{\pi d^3}{6} \tag{9}$$

where ρ_p is the average particle density. However, the fine particles (PM2.5) are more harmful than the coarse particles (PM10) [131]. Especially diesel engines are the major sources of fine particles. Ntziachristos et al. [132] carried out a comparative study between heavy duty and light duty diesel engines on their exhaust particle characteristics (mass, number of solid and total particles). The authors reported that the heavy duty engines emits lower solid particle emissions in comparison with light duty vehicles and attributed it to the rapid combustion, higher EGR rate and higher PM emission limits with light duty engines. They concluded that the diesel particle emissions are not essentially proportional to the engine size.

For a non-optimized EGR system, Kreso et al. [133] studied the effects of recirculating cooled exhaust gases (up to 16.8%) on heavy duty diesel engines at steady-state test conditions. An increase in brake specific total particulate matter (BSTPM) with 16% EGR at both the operating modes was reported. The authors related the TPM emissions to the sum of soluble organic fraction (SOF), solids (SOL) and the filter collected sulfate (SO_4^{2-}) as:

$$TPM = SOF + SOL + SO_4^{2-} \tag{10}$$

The solids and sulfate portion of TPM are found to be increased with EGR; while the SOF emission decreased with increasing EGR rate. A higher number and volume of particles under accumulation mode were reported in comparison to the nuclei mode with EGR. The authors concluded that EGR even at relatively lower concentrations affects the diesel particle size distributions (PSD). The effects of EGR on particulate morphology were investigated by Zhu and Lee [134] in a light-duty diesel engine by using a transmission electron microscope (TEM). The dilution and thermal effect of EGR on the morphological characteristics, such as primary particle size, aggregate particle size and fractal geometry (which represent the complex geometry of particulates) were examined with different EGR rates from 0 to 19%. A significant increase (>70%) of the primary particle diameter is encountered with EGR. The authors reported that the thermal effect has more detrimental cause to the increase of primary particle sizes than the dilution effect at low EGR rate (6.6%). However, the dilution effect became dominant contributor at higher EGR rates (13%). Further, the particle agglomeration was found to be enhanced due to the combustion deterioration with EGR. The authors concluded that the fractal geometry of particles is sensible to the variation in EGR rate.

Kolodziej et al. [130] conducted an experimental study to characterize the diesel particle from conventional combustion and low temperature combustion (LTC) mode in a single cylinder research engine. EGR as high as 65% with an early injection timing of 38.5 deg. bTDC was adopted for LTC, whereas in conventional diesel combustion, standard injection timing of 12.8 deg. bTDC with EGR rates of 30 and 50% was attempted. While smaller average particle size and high organic content was observed with LTC; the conventional diesel combustion led to larger average particle size with lower organic content. As shown in Fig. 13, the authors symbolized the particles formed during conventional and low temperature combustion modes.

Zhou et al. [135] examined the effect of hot EGR (EGR temperature maintained around 65 °C) on the ultra-fine particle emissions using an electrical low pressure impactor analyser in a diesel pilot-ignited natural gas premixed charge compression ignition (DPING-PCCI) engine. The authors observed an unimodal shape of particle size distribution with EGR and advanced injection strategies. Also, a significant reduction (∼57%) in the total ultrafine particle concentration with hot EGR was reported. Bullock et al. [136] performed experiments on HCCI engine with port injected compressed natural gas to characterize the exhaust particles based on the size and density. A single cylinder CFR engine was operated under three different modes viz. motoring, spark ignition and HCCI mode for two compression ratios (10 & 17:1). For the compression ratio of 17:1, particles of bigger mean diameter and higher concentration were reported. Moreover, lower particulate mass emission was noticed with HCCI mode than during the motoring operation. A substantial quantity of particle mass was observed under all modes of operation including the motoring condition. This clearly indicates that the lubricating oil is another major source of particulates and could contribute up to 40 percent of the total [8].

Tsolakis [137] analysed the effects of externally cooled EGR (10% & 20% by vol.) on the particle size distribution in a naturally aspirated diesel engine operated with neat rapeseed methyl ester (RME). The oxygen content in RME yielded a significant reduction in the exhaust smoke and particle mass emissions. However, the number of smaller particulate emissions (aerodynamic diameter < 0.2296 μm) increased significantly in comparison to the fossil diesel combustion. In addition, the use of EGR has further aggravated the total particle number and mass for both the test fuels. However the increase in the total particle mass is more significant in the diesel case and hence, the author recommended the usage of EGR with neat RME combustion. Recently, Labecki et al. [138] used the electrostatic mobility spectrometer (EMS) to characterize PSD in a turbo-charged engine operated with diesel and RME fuels. The test results showed that the addition of EGR caused the particles to agglomerate, forming larger size particles. The authors opined to the sooting tendency of EGR, creating a conducive ambience for coagulation, accumulation, condensation of volatile fractions on the particles and surface growth.

Qi et al. [139] investigated the effects of injection pressure, injection timing and EGR (up to 30%) on the particle size distribution (PSD) of fossil diesel and biodiesel (waste cooking oil) in a CRDI engine. In this experimental study, the exhaust PSD was measured from 5.4 nm to 358 nm by employing a scanning mobility particle sizer. Though the biodiesel emitted lower particle concentration than fossil diesel, the addition of EGR increased the particle number concentration and size distribution due to the formation of larger size particles. Further, the authors studied the EGR influence on the PSD for two different injection timings (25 and 5 deg bTDC) and reported that the effects of EGR are similar for both injection timings. They recommended higher injection pressure in conjunction with EGR addition. Nord et al. [140] investigated the particulate emitting potentiality of ethanol fuel in multi cylinder diesel engines equipped with EGR, CRT and DPF. The test results revealed that ethanol fuelled engine generally emits lower particle mass and smaller particle size than the diesel operated engine. The authors recommended combining DPF with EGR for effective reduction of particulate emissions from both ethanol and diesel fuelled engines.

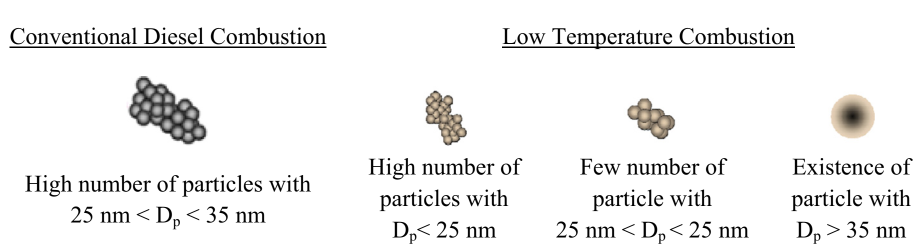

Fig. 13. Symbolic representation of particles from conventional and LTC mode [130].

8. Concluding remarks and future directions

In diesel literature, the effects of EGR on advanced diesel combustions and alternate fuels have been reported. In general, it is concluded that the usage of EGR is a necessary strategy for implementing advanced combustion concepts and to control emissions particularly NO_x from alternate fuelled diesel engines. Further, it is noticed that the magnitude of recycled gases varies widely depending upon the choice of application. As high as 68% of exhaust gases has been recycled for advanced combustion control studies. The major conclusions drawn from the usage of EGR are as follows;

Internal and External modes of EGR are found to promote the homogeneous mixture formation and ignition timing control in advanced diesel combustion systems.

- o The lower exhaust gas temperatures and oxygen displacement via EGR tend to produce less NO_x, but more soot, CO and HC's are generated by alternate fuelled diesel engines.
- o Few research studies have shown that the EGR generated soot have a harmful effect on the engine lube oil properties which leads to increased engine wear.

EGR mitigates the NO_x formation at the expense of smoke emissions, particularly at higher loads. Hence at high load conditions, it is recommended to use other strategies or a combination of techniques like low EGR and injection retardation, which could prove useful in mitigating NO_x without penalizing the smoke emissions. Further, the following aspects need to be addressed for successful implementation of EGR in future engines:

- An accurate measurement of EGR during transient conditions and seamless transition is essential for controlling the recycled gas concentration (low to high) and temperatures (hot/cold) depending upon the engine operating conditions.
- Control aspects during transient operation needs to be addressed for the implementation of LPL EGR.
- Advanced combustion systems operating with substantial amount of EGR demands high quality fuels. Hence, future combustion systems pertain to tailor-made fuels.
- Further long-term studies are warranted for optimal usage of EGR in both alternate fuelled operation and advanced combustion mode considering their detrimental effect on engine tribology and other emission characteristics.
- Few research studies have shown that the EGR generated soot have a harmful effect on the engine lube oil properties which leads to increased engine wear. While on the other hand, EGR causes increased particle number concentration in the accumulation mode and particles of larger size range in the diesel exhaust.

References

[1] National Green Tribunal, Southern Zone Bench, Government of India. <http://www.greentribunal.gov.in/Writereaddata/Downloads/137-2016%28KOCHI%29OA23-5-2016.pdf> [accessed on 15-05-2016].
[2] Tsurushima T, Miyamoto T, Akagawa H, Aoyagi Y, Lee JH, Lee D, et al. Effects of initial in-cylinder flow field on mixture formation in a premixed compression ignition engine. SAE paper no. 2000-01-0331; 2000.
[3] Gao J, Matsumoto Y, Namba M, Nishida K. An investigation of mixture formation and in-cylinder combustion processes in direct injection diesel engines using group- hole nozzles. Int J Engine Res 2009;10:27–44.
[4] Moon S, Tsujimura T, Oguma M, Chen Z, Huang Z, Saitou T. Mixture condition, combustion and sooting characteristics of ethanol–diesel blends in diffusion flames under various injection and ambient conditions. Fuel 2013;113:128–39.
[5] Johnson TV. Diesel Emission Control in Review. SAE paper no. 2006-01-0030; 2006.
[6] Yu RC, Shahed SM. Effects of injection timing and exhaust gas recirculation on emissions from a DI diesel engine. SAE paper no. 811234; 1981.
[7] Zheng M, Reader GT, Hawley JG. Diesel engine exhaust gas recirculation––a review on advanced and novel concepts. Energy Convers Manage 2004;45:883–900.
[8] Sher E. Handbook of air pollution from internal combustion engines. USA: Academic press; 1998.
[9] Johnson TV. Diesel emission control in review. SAE paper no.2009-01-0121; 2009.
[10] Abd-Alla GH. Using exhaust gas recirculation in internal combustion engines: a review. Energy Convers Manage 2002;43:1027–42.
[11] Asad U, Zheng M. Exhaust gas recirculation for advanced diesel combustion cycles. Appl Energy 2014;123:242–52.
[12] Selim MY. Effect of exhaust gas recirculation on some combustion characteristics of dual fuel engine. Energy Convers Manage 2003;44:707–21.
[13] Daisho Y, Yaeo T, Koseki T, Saito T, Kihara R, Quiros EN. Combustion and exhaust emissions in a direct-injection diesel engine dual-fueled with natural gas. SAE paper no. 950465; 1995.
[14] Bendu H, Murugan S. Homogeneous charge compression ignition (HCCI) combustion: Mixture preparation and control strategies in diesel engines. Renew Sust Energ Rev 2014;38:732–46.
[15] Gan S, Ng HK, Pang KM. Homogeneous charge compression ignition (HCCI) combustion: implementation and effects on pollutants in direct injection diesel engines. Appl Energy 2011;88:559–67.
[16] Yao M, Zheng Z, Liu H. Progress and recent trends in homogeneous charge compression ignition (HCCI) engines. Prog Energ Combust Sci 2009;35:398–437.
[17] Hasan MM, Rahman MM. Homogeneous charge compression ignition combustion: advantages over compression ignition combustion, challenges and solutions. Renew Sust Energy Rev 2016;57:282–91.
[18] Vallinayagam R, Vedharaj S, Yang WM, Roberts WL, Dibble RW. Feasibility of using less viscous and lower cetane (LVLC) fuels in a diesel engine: a review. Renew Sust Energy Rev 2015;51:1166–90.
[19] Kumar P, Rehman A. Bio-diesel in homogeneous charge compression ignition (HCCI) combustion. Renew Sust Energy Rev 2016;56:536–50.
[20] Imtenan S, Varman M, Masjuki HH, Kalam MA, Sajjad H, Arbab MI, et al. Impact of low temperature combustion attaining strategies on diesel engine emissions for diesel and biodiesels: a review. Energy Convers Manage 2014;80:329–56.
[21] Majewski WA, Khair M. Diesel emissions and their control. USA: SAE International; 2006.
[22] Brehob WM. Mechanisms of pollutant formation and control from automotive sources. SAE paper no. 710483; 1971.
[23] Ladommatos N, Abdelhalim SM, Zhao H, Hu Z. The effects on diesel combustion and emissions of reducing inlet charge mass due to thermal throttling with hot EGR. SAE paper no. 980185; 1998.
[24] Ladommatos N, Abdelhalim SM, Zhao H, Hu Z. The dilution, chemical and thermal effects of exhaust gas recirculation on diesel emissions-part 2: effects of carbon dioxide. SAE paper no. 961167; 1996.
[25] Ladommatos N, Abdelhalim SM, Zhao H, Hu Z. The effects of carbon dioxide in exhaust gas recirculation on diesel engine emissions. Proc IMechE: Part D-J Automob Eng 1998;212:25–42.
[26] Ladommatos N, Abdelhalim SM, Zhao H, Hu Z. The dilution, chemical and thermal effects of exhaust gas recirculation on diesel emissions-part 3: effects of water vapour. SAE paper no. 971659; 1997.
[27] Ladommatos N, Abdelhalim SM, Zhao H, Hu Z. The dilution, chemical, and thermal effects of exhaust gas recirculation on diesel engine emissions-part 4: effects of carbon dioxide and water vapor. SAE paper no. 971660; 1997.
[28] Ladommatos N, Abdelhalim SM, Zhao H, Hu Z. The effects of exhaust gas recirculation on diesel combustion and emissions. Int J Engine Res 2000;1:107–26.
[29] Ladommatos N, Abdelhalim SM, Zhao H, Hu Z. The dilution, chemical, and thermal effects of exhaust gas recirculation on diesel engine emissions-part 1: effect of reducing inlet charge oxygen. SAE paper no. 961165; 1996.
[30] Maiboom A, Tauzia X, Hétet JF. Experimental study of various effects of exhaust gas recirculation (EGR) on combustion and emissions of an automotive direct injection diesel engine. Energy 2008;33:22–34.
[31] Baert RS, Beckman DE, Veen A. Efficient EGR technology for future HD diesel engine emission targets. SAE paper no. 1999-01-0837; 1999.
[32] Ladommatos N, Balian R, Horrocks R, Cooper L. The effect of exhaust gas recirculation on combustion and NOx emissions in a high-speed direct-injection diesel engine. SAE paper no. 960840; 1996.
[33] Jacobs T, Assanis D, Filipi Z. The impact of exhaust gas recirculation on performance and emissions of a heavy-duty diesel engine. SAE paper no. 2003-01-1068; 2003.
[34] Kohketsu S, Mori K, Sakai K, Hakozaki T. EGR technologies for a turbocharged and intercooled heavy-duty diesel engine. SAE paper no. 970340; 1997.
[35] Shin B, Cho Y, Han D, Song S, Chun KM. Hydrogen effects on NOx emissions and brake thermal efficiency in a diesel engine under low-temperature and heavy-EGR conditions. Int J Hydrogen Energy 2011;36:6281–91.
[36] Maiboom A, Tauzia X, Shah SR, Hétet JF. Experimental study of an LP EGR system on an automotive diesel engine, compared to HP EGR with respect to PM and NOx emissions and specific fuel consumption. SAE paper no. 2009-24-0138; 2009.
[37] Zamboni G, Capobianco M. Experimental study on the effects of HP and LP EGR in an automotive turbocharged diesel engine. Appl Energy 2012;94:117–28.
[38] Cornolti L, Onorati A, Cerri T, Montenegro G, Piscaglia F. 1D simulation of a turbocharged Diesel engine with comparison of short and long EGR route solutions. Appl Energy 2013;111:1–15.

[39] Shutty J, Benali H, Daeubler L, Traver M. Air system control for advanced diesel engines. SAE paper no. 2007-01-0970; 2007.
[40] Mao B, Yao M, Zheng Z, Li Y, Liu H, Yan B. Effects of Dual Loop EGR on Performance and Emissions of a Diesel Engine. SAE paper no. 2015-01-0873; 2015.
[41] Zheng M, Irick DK, Hodgson J. Stabilizing excessive EGR with an oxidation catalyst on a modern diesel engine. ASME paper no. ICES2002-455; 2002.
[42] Ishida A, Nishimura A, Uranishi M, Kihara R, Nakamura A, Newman P, et al. Development of ECOS-DDF natural gas engine for medium duty trucks––exhaust gas emission reduction against base Diesel engine. JSAE paper 20005001; 2000.
[43] Dishy A, has hi You T, Iwashiro Y, Nakayama S, Kihara R, Saito T. Controlling combustion and exhaust emissions in a direct-injection diesel engine dual-fueled with natural gas. SAE paper no. 952436; 1995.
[44] Wei H, Zhu T, Shu G, Tan L, Wang Y. Gasoline engine exhaust gas recirculation–a review. Appl Energy 2012;99:534–44.
[45] Kim HM, Lee DH, Park SK, Choi KS, Wang HM. An experimental study on heat exchange effectiveness in the diesel engine EGR coolers. J Mech Sci Technol 2008;22:361–6.
[46] Kim HM, Park SK, Choi KS, Wang HM, Lee DH, Lee DK, et al. Investigation on the flow and heat transfer characteristics of diesel engine EGR coolers. Int J Automot Technol 2008;9:149–53.
[47] Agarwal D, Sinha S, Agarwal AK. Experimental investigation of control of NOx emissions in biodiesel-fueled compression ignition engine. Renew Energy 2006;31:2356–69.
[48] d'Ambrosio S, Finesso R, Spessa E. Calculation of mass emissions, oxygen mass fraction and thermal capacity of the inducted charge in SI and diesel engines from exhaust and intake gas analysis. Fuel 2011;90:152–66.
[49] Snyder DB, Adi GH, Hall CH, Shaver G. Control-variable-based accommodation of biodiesel blends. Int J Engine Res 2011;12(23):564–79.
[50] Assanis DN, Najt PM, Dec JE, Eng JA, Asmus TN, Zhao F. Homogeneous charge compression ignition (HCCI) engines. Warrendale, PA: Society of Automotive Engineers; 2003.
[51] Zhao H. HCCI and CAI engines for the automotive industry. Elsevier; 2007.
[52] Nathan SS, Mallikarjuna JM, Ramesh A. Effects of charge temperature and exhaust gas re-circulation on combustion and emission characteristics of an acetylene fuelled HCCI engine. Fuel 2010;89:515–21.
[53] Lakshmanan T, Nagarajan G. Experimental investigation of port injection of acetylene in DI diesel engine in dual fuel mode. Fuel 2011;90:2571–7.
[54] Ibrahim MM, Ramesh A. Experimental investigations on a hydrogen diesel homogeneous charge compression ignition engine with exhaust gas recirculation. Int J Hydrogen Energ 2013;38:10116–25.
[55] Saravanan N, Nagarajan G. Performance and emission studies on port injection of hydrogen with varied flow rates with diesel as an ignition source. Appl Energy 2010;87:2218–29.
[56] Nathan SS, Mallikarjuna JM, Ramesh A. An experimental study of the biogas–diesel HCCI mode of engine operation. Energy Convers Manage 2010;51(7):1347–53.
[57] Morsy MH. Ignition control of methane fueled homogeneous charge compression ignition engines using additives. Fuel 2007;86:533–40.
[58] Oguma H, Ichikura T, Iida N. A Study on adaptability of alternative fuels for lean- burn two-stroke ATAC engine. SAE paper no. 972091; 1997.
[59] Esterlingot E, Guibert P, Lavy J. Thermodynamical and optical analyses of controlled autoignition combustion in two-stroke engines. SAE paper no. 972098; 1997.
[60] Al-Kaf SA, Suhaimi A, Abbass HA. Radical combustion: new concept for two stroke engines. ASEAN J Sci Technol Develop 2000;17:91–9.
[61] Ganesh D, Nagarajan G, Ibrahim MM. Study of performance, combustion and emission characteristics of diesel homogeneous charge compression ignition (HCCI) combustion with external mixture formation. Fuel 2008;87:3497–503.
[62] Singh AP, Agarwal AK. Combustion characteristics of diesel HCCI engine: an experimental investigation using external mixture formation technique. Appl Energy 2012;99:116–25.
[63] Ganesh D, Nagarajan G. Homogeneous charge compression ignition (HCCI) combustion of diesel fuel with external mixture formation. Energy 2010;35:148–57.
[64] Ganesh D, Nagarajan G. Performance and emission analysis on mixed-mode homogeneous charge compression ignition (HCCI) combustion of biodiesel fuel with external mixture formation. SAE paper no. 2011-01-2450; 2011.
[65] Lee CS, Lee KH, Kim DS. Experimental and numerical study on the combustion characteristics of partially premixed charge compression ignition engine with dual fuel. Fuel 2003;82:553–60.
[66] Bhaskar K, Nagarajan G, Sampath S. The effects of premixed ratios on the performance and emission of PPCCI combustion in a single cylinder diesel engine. Int J Green Energy 2013;10:1–11.
[67] Bhaskar K, Nagarajan G, Sampath S. The performance and emission characteristics of fish oil methyl esters (FOME) and diesel blends in a partially premixed charge compression ignition engine. Int J Green Energy 2014;11:389–403.
[68] Soloiu V, Duggan M, Harp S, Vlcek B, Williams D. PFI (port fuel injection) of n-butanol and direct injection of biodiesel to attain LTC (low-temperature combustion) for low-emissions idling in a compression engine. Energy 2013;52:143–54.
[69] Panão MR, Moreira AL. Interpreting the influence of fuel spray impact on mixture preparation for HCCI combustion with port-fuel injection. Prog Combust Inst 2007;31:2205–13.
[70] Padala S, Le MK, Kook S, Hawkes ER. Imaging diagnostics of ethanol port fuel injection sprays for automobile engine applications. Appl Therm Eng 2013;52:24–37.
[71] Brusstar M, Stuhldreher M, Swain D, Pidgeon W. High efficiency and low emissions from a port-injected engine with neat alcohol fuels. SAE paper no. 2002-01-2743; 2002.
[72] Nishijima Y, Asaumi Y, Aoyagi Y. Impingement spray system with direct water injection for premixed lean diesel combustion control. SAE paper no. 2002-01-0109; 2002.
[73] Hashizume T, Miyamoto T, Hisashi A, Tsujimura K. Combustion and emission characteristics of multiple stage diesel combustion. SAE paper no. 980505; 1998.
[74] Hasegawa R, Yanagihara H. HCCI combustion in DI diesel engine. SAE paper no. 2003-01-0745; 2003.
[75] Su W, Lin T, Pei Y. A compound technology for HCCI combustion in a DI diesel engine based on the multi-pulse injection and the BUMP combustion chamber. SAE paper no. 2003-01-0741; 2003.
[76] Kimura S, Aoki O, Ogawa H, Muranaka S, Enomoto Y. New combustion concept for ultra-clean and high-efficiency small DI diesel engines. SAE paper no. 1999-01-3681; 1999.
[77] Gray AW, Ryan TW. Homogeneous charge compression ignition (HCCI) of diesel fuel. SAE paper no. 971676; 1997.
[78] Nakagome K, Shimazaki N, Niimura K, Kobayashi S. Combustion and emission characteristics of premixed lean diesel combustion engine. SAE paper no. 970898; 1997.
[79] Mase Y, Kawashima JI, Sato T, Eguchi M. Nissan's new multivalve DI diesel engine series. SAE paper no. 981039; 1998.
[80] Suzuki H, Koike N, Ishii H, Odaka M. Exhaust purification of diesel engines by homogeneous charge with compression ignition part 1: Experimental investigation of combustion and exhaust emission behavior under pre-mixed homogeneous charge compression ignition method. SAE paper no. 970313; 1997.
[81] Aoyama T, Hattori Y, Mizuta JI, Sato Y. An experimental study on premixed-charge compression ignition gasoline engine. SAE paper no. 960081; 1996.
[82] Harada A, Shimazaki N, Sasaki S, Miyamoto T, Akagawa H, Tsujimura K. The effects of mixture formation on premixed lean diesel combustion engine. SAE paper no. 980533; 1998.
[83] Stanglmaier RH, Roberts CE. Homogeneous charge compression ignition (HCCI): benefits, compromises, and future engine applications. SAE paper no. 1999-01-3682; 1999.
[84] Kanda T, Hakozaki T, Uchimoto T, Hatano J, Kitayama N, Sono H. PCCI operation with early injection of conventional diesel fuel. SAE paper no. 2005-01-0378; 2005.
[85] Boyarski NJ, Reitz RD. Premixed compression ignition (PCI) combustion with modeling-generated piston bowl geometry in a diesel engine. SAE paper no. 2006-01- 0198; 2006.
[86] Iwabuchi Y, Kawai K, Shoji T, Takeda Y. Trial of new concept diesel combustion system-premixed compression-ignited combustion. SAE paper no. 1999-01-0185; 1999.
[87] Xu M, Gui Y, Deng KY. Fuel injection and EGR control strategy on smooth switching of CI/HCCI mode in a diesel engine. J Energy Inst 2015;88:157–68.
[88] Das P, Subbarao PM, Subrahmanyam JP. Control of combustion process in an HCCI- DI combustion engine using dual injection strategy with EGR. Fuel 2015;159:580–9.
[89] Shi L, Cui Y, Deng K, Peng H, Chen Y. Study of low emission homogeneous charge compression ignition (HCCI) engine using combined internal and external exhaust gas recirculation (EGR). Energy 2006;31:2665–76.
[90] García MT, Aguilar FJ, Lencero TS. Experimental study of the performances of a modified diesel engine operating in homogeneous charge compression ignition (HCCI) combustion mode versus the original diesel combustion mode. Energy 2009;34:159–71.
[91] Zhao Y, Wang Y, Li D, Lei X, Liu S. Combustion and emission characteristics of a DME (dimethyl ether)-diesel dual fuel premixed charge compression ignition engine with EGR (exhaust gas recirculation). Energy 2014;72:608–17.
[92] Chen Z, Liu J, Wu Z, Lee C. Effects of port fuel injection (PFI) of n-butanol and EGR on combustion and emissions of a direct injection diesel engine. Energy Convers Manage 2013;76:725–31.
[93] Wimmer A, Eichlseder H, Klell M, Figer G. Potential of HCCI concepts for DI diesel engines. Int J Vehicle Des 2006;41:32–48.
[94] Kondo M, Kimura S, Hirano I, Uraki Y, Maeda R. Development of noise reduction technologies for a small direct-injection diesel engine. JSAE Rev 2000;21:327–33.
[95] Fang T, Coverdill RE, Chia-fon FL, White RA. Air–fuel mixing and combustion in a small-bore direct injection optically accessible diesel engine using a retarded single injection strategy. Fuel 2009;88:2074–82.
[96] Ogawa H, Li T, Miyamoto N. Characteristics of low temperature and low oxygen diesel combustion with ultra-high exhaust gas recirculation. Int J Engine Res 2007;8:365.
[97] Akihama K, Takatori Y, Inagaki K, Sasaki S, Dean AM. Mechanism of the smokeless rich diesel combustion by reducing temperature. SAE paper no. 2001-01-0655; 2001.

[98] Alriksson M, Rente T, Denbratt I. Low soot, low NOx in a heavy duty diesel engine using high levels of EGR. SAE paper no. 2005-01-3836; 2005.
[99] Zhao H, Xie H, Peng Z. Effect of recycled burned gases on homogeneous charge compression ignition combustion. Combust Sci Technol 2005;177:1863–82.
[100] Mani M, Nagarajan G, Sampath S. An experimental investigation on a DI diesel engine using waste plastic oil with exhaust gas recirculation. Fuel 2010;89:1826–32.
[101] Lehto K, Elonheimo A, Häkkinen K, Sarjovaara T, Larmi M. Emission reduction using hydrotreated vegetable oil (HVO) with Miller timing and EGR in diesel combustion. SAE Int J Fuels Lubr 2012;5(1):218–24.
[102] Liu D, Ghafourian A, Xu H. Phenomenology of EGR in a light duty diesel engine fuelled with Hydrogenated Vegetable Oil (HVO), Used Vegetable Oil Methyl Ester (UVOME) and their Blends. SAE paper no. 2013-01-1688; 2013.
[103] Kawano D, Ishii H, Goto Y, Noda A, Aoyagi Y. Optimization of engine system for application of biodiesel fuel. SAE paper no. 2007-01-2028; 2007.
[104] Marques A, Monteiro E, Moreira NA, Malheiro S. NOx Emissions Reduction in a Biodiesel Engine by Means of EGR Technology. SAE paper no. 2007-01-0078; 2007.
[105] Arcoumanis C, Bae C, Crookes R, Kinoshita E. The potential of di-methyl ether (DME) as an alternative fuel for compression-ignition engines: a review. Fuel 2008;87:1014–30.
[106] Kapus P, Ofner H. Development of fuel injection equipment and combustion system for DI diesels operated on dimethyl ether. SAE paper no. 950062; 1995.
[107] Sato Y, Noda A, Sakamoto T, Goto Y. Performance and emission characteristics of a DI diesel engine operated on dimethyl ether applying EGR with supercharging. SAE paper no. 2000-01-1809; 2000.
[108] Murayama T, Zheng M, Chikahisa T, Oh YT, Fujiwara Y, Tosaka S et al. Simultaneous reductions of smoke and NOx from a DI diesel engine with EGR and dimethyl carbonate. SAE paper no. 952518; 1995.
[109] Yoshimoto Y, Tamaki H. Reduction of NOx and smoke emissions in a diesel engine fueled by biodiesel emulsion combined with EGR. SAE paper no. 2001-01-0649; 2001.
[110] Bhaskar K, Nagarajan G, Sampath S. Optimization of FOME (fish oil methyl esters) blend and EGR (exhaust gas recirculation) for simultaneous control of NO_x and particulate matter emissions in diesel engines. Energy 2013;62:224–34.
[111] Qi D, Leick M, Liu Y, Chia-fon FL. Effect of EGR and injection timing on combustion and emission characteristics of split injection strategy DI-diesel engine fueled with biodiesel. Fuel 2011;90:1884–91.
[112] Tsolakis A, Megaritis A, Wyszynski ML, Theinnoi K. Engine performance and emissions of a diesel engine operating on diesel-RME (rapeseed methyl ester) blends with EGR (exhaust gas recirculation). Energy 2007;32:2072–80.
[113] Haupt D, Nord K, Egebäck KE, Ahlvik P. Hydrocarbons and aldehydes from a diesel engine running on ethanol and equipped with EGR, catalyst and DPF. SAE paper no. 2004-01-1882; 2004.
[114] Li X, Qiao X, Zhang L, Fang J, Huang Z, Xia H. Combustion and emission characteristics of a two-stroke diesel engine operating on alcohol. Renew Energy 2005;30:2075–84.
[115] Ogawa H, Setiapraja H, Hara K, Shibata G. Combustion and emissions with bio- alcohol and non esterified vegetable oil blend fuels in a small diesel engine. SAE Int J Fuels Lubr 2012;5:1395–403.
[116] Jeyaseelan T, Mehta PS. Reducing NO in a biodiesel fueled compression ignition engine-an experimental study. SAE paper no. 2015-24-2483; 2015.
[117] Karim GA. A review of combustion processes in the dual fuel engine—the gas diesel engine. Prog Energy Combust 1980;6:277–85.
[118] Weaver CS, Turner SH. Dual fuel natural gas/diesel engines: technology, performance, and emissions. SAE paper no. 940548; 1994.
[119] Selim MY. A study of some combustion characteristics of dual fuel engine using EGR. SAE paper no. 2003-01-0766; 2003.
[120] Ishida A, Nishimura A, Uranishi M, Kihara R, Nakamura A, Newman P. The development of the ECOS-DDF natural gas engine for medium-duty trucks: exhaust emission reduction against base diesel engine. JSAE Rev 2001;22:237–43.
[121] Besch MC, Israel J, Thiruvengadam A, Kappanna H, Carder D. Emissions characterization from different technology heavy-duty engines retrofitted for CNG/Diesel dual-fuel operation. SAE Int J Engines 2015;8:1342–58.
[122] Lee J, Choi S, Kim H, Kim D, Choi H, Min K. Reduction of emissions with propane addition to a diesel engine. Int J Automot Technol 2013;14:551–8.
[123] Saravanan N, Nagarajan G, Kalaiselvan KM, Dhanasekaran C. An experimental investigation on hydrogen as a dual fuel for diesel engine system with exhaust gas recirculation technique. Renew Energy 2008;33:422–7.
[124] Gautam M, Chitoor K, Durbha M, Summers JC. Effect of diesel soot contaminated oil on engine wear—investigation of novel oil formulations. Tribol Int 1999;32:687–99.
[125] George S, Balla S, Gautam V, Gautam M. Effect of diesel soot on lubricant oil viscosity. Tribol Int 2007;40:809–18.
[126] George S, Balla S, Gautam M. Effect of diesel soot contaminated oil on engine wear. Wear 2007;262:1113–22.
[127] Antusch S, Dienwiebel M, Nold E, Albers P, Spicher U, Scherge M. On the tribochemical action of engine soot. Wear 2010;269:1–12.
[128] Hu E, Hu X, Liu T, Fang L, Dearn KD, Xu H. The role of soot particles in the tribological behavior of engine lubricating oils. Wear 2013;304:152–61.
[129] Kittelson DB. Engines and nanoparticles: a review. J Aerosol Sci 1998;29:575–88.
[130] Kolodziej C, Wirojsakunchai E, Foster DE, Schmidt N, Kamimoto T, Kawai T et al. Comprehensive characterization of particulate emissions from advanced diesel combustion. SAE paper no. 2007-01-1945;2007.
[131] Englert N. Fine particles and human health—a review of epidemiological studies. Toxicol Lett 2004;149:235–42.
[132] Ntziachristos L, Mamakos A, Samaras Z, Rexeis M, Hausberger S. Diesel particle exhaust emissions from light duty vehicles and heavy duty engines. SAE paper no. 2006-01-0866; 2006.
[133] Kreso AM, Johnson JH, Gratz LD, Bagley ST, Leddy DG. A study of the effects of exhaust gas recirculation on heavy-duty diesel engine emissions. SAE paper no. 981422; 1998.
[134] Zhu J, Lee K. Effects of exhaust gas recirculation on particulate morphology for a light-duty diesel engine. SAE paper no. 2005-01-0195; 2005.
[135] Zhou L, Liu Y, Sun L, Hou H, Zeng K, Huang Z. Effect of hot exhaust gas recirculation on the combustion characteristics and particles emissions of a pilot- ignited natural gas engine. SAE paper no. 2013-01-1341: 1116–25.
[136] Bullock DS, Olfert JS. Size, volatility, and effective density of particulate emissions from a homogeneous charge compression ignition engine using compressed natural gas. J Aerosol Sci 2014;75:1–8.
[137] Tsolakis A. Effects on particle size distribution from the diesel engine operating on RME-biodiesel with EGR. Energy Fuels 2006;20:1418–24.
[138] Labecki L, Lindner A, Winklmayr W, Uitz R, Cracknell R, Ganippa L. Effects of injection parameters and EGR on exhaust soot particle number-size distribution for diesel and RME fuels in HSDI engines. Fuel 2013;112:224–35.
[139] Qi D, Lee CF, Lin Y. Effect of injection parameters and EGR on the particle size distributions and exhaust emissions for diesel and biodiesel fuels in CRDI engine. SAE paper no. 2014-01-1612; 2014.
[140] Nord K, Haupt D, Ahlvik P, Egebäck KE. Particulate emissions from an ethanol fueled heavy-duty diesel engine equipped with EGR, catalyst and DPF. SAE paper no. 2004-01-1987; 2004.
[141] Mehta PS, Pundir BP, Sharma GK. NOx control technologies for automotive vehicles, Technology Information Forecasting and Assessment Council Report. Collaborative Automotive R&D, Department of Science and Technology, Government of India; 2011.
[142] Pradeep V, Sharma RP. Use of hot EGR for NO_x control in a compression ignition engine fuelled with bio-diesel from Jatropha oil. Renew Energy 2007;32:1136–54.
[143] Saleh HE. Effect of exhaust gas recirculation on diesel engine nitrogen oxide reduction operating with jojoba methyl ester. Renew Energy 2009;34:2178–86.
[144] Jiménez-Espadafor FJ, Torres M, Velez JA, Carvajal E, Becerra JA. Experimental analysis of low temperature combustion mode with diesel and biodiesel fuels: a method for reducing NO_x and soot emissions. Fuel Process Technol 2012;103:57–63.