EXPERIMENTAL INVESTIGATIONS OF EFFECTS OF COOLING/NON-COOLING OF EGR ON TWO LEVEL OF COMPRESSION RATIO IN A COMMON RAIL DIESEL ENGINE

by

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Exhaust gas recirculation (EGR) cooling is followed for many years and proven as an efficient technique for reducing nitrogenoxide-particle mater (NO_x-PM) emissions. The EGR cooling helps in emission reduction of NO_x and PM. But, it brings associated issues like cooler fouling; misfiring in cold condition, if cooling is not bypassed, during cold start. Also, it increases HC and CO emissions thus leading to increased loading of diesel oxidation catalyst.

In the present study, two engine operating points were chosen from extra urban driving cycle part of modified new European driving cycle for India. The two set of compression ratios were prepared. The NO_x-PM along with HC and CO emissions were measured with 18.5 compression ratio. The emissions with cooled EGR were taken first which form the base optimization. The cooler was removed from the EGR circuit and same measurements were done. In the third step, the increased NO_x due to non-cooled EGR was brought to the original value by increasing the injection rail pressure and EGR rates simultaneously. In this process, the PM emission was found to be increasing marginally from its optimized value. The same experimentation was repeated for the 16.5 compression ratio.

Both the compression ratios exhibit the similar trends of emissions. The magnitude of NO_x-PM, HC, and CO differs for both the compression ratios. In order to meet the baseline optimized NO_x-PM emissions with prime objective to get rid of EGR cooler and gain cost saving, reduced compression ratio found to be promising solution. There was marginal increase in CO and HC emission with this approach.

Key words: *compression ratio, exhaust gas recirculation, cooling of EGR, new European driving cycle*

Introduction

Because of their high thermal efficiency and durability, Diesel engines are widely used in commercial vehicles and play an important social and economic role. At the same time, they are subject, in light of the need for environmental protection, to demands for ever-greater reductions in NO_x, PM or (soot/smoke) emissions. However, decreases in NO_x emission often cause increases in PM emission, and *vice versa*. Reduced NO_x and PM are difficult to achieve simulta-

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neously through combustion improvement. In order to overcome these problems, many researchers have investigated various strategies to overcome this demand from compression-ignited engine [1-3, 4, 5-7].

High compression ratio (CR) increases theoretical thermal efficiency, but decreases mechanical efficiency. The maximal pressure within a cylinder, and mechanical loses, increases with an increase of both engine load and CR [8]. Lowering the CR is being used in view if the increasing requirement of power output and stringent emission requirements [9, 10]. Lowering of the CR gives the chance to increase the power by adding more fuel for the same peak firing pressure [9]. This is due to the fact that when the CR is lowered, compression temperature and pressure at top dead center (TDC) decreases. Consequently, ignition takes longer even when fuel is injected near TDC, enabling better mixture of air and fuel [8, 11]. This alleviates the formation of NO_x and PM because the combustion becomes more uniform without localized high-temperature areas and oxygen insufficiencies. Furthermore, injection and combustion close to TDC result in a highly-efficient diesel engine, in which a larger amount of actual work (or, a higher expansion ratio) is obtained than in a high compression ratio diesel engine [2]. It shifts the NOx-PM trade off dramatically. However, the CO and HC emissions are increased. [1, 11]

The EGR cooling is also being used as a measure to control the NO_x-PM emission trade off. Also, cooling of EGR has been reported to increase HC and CO emissions [12]. Removal of cooling helps to improve the CO and HC at the cost of increase in NO_x-PM. The increase in EGR and injection pressure is being used to suppress the NO_x emissions [13]. The ability of modern common rail injection system allows the higher injection pressures. So, combining the benefits of each of this, overall benefits can be achieved.

Mahr [13] has explained in their article the trends to go with higher injection pressure and higher EGR rates coupled with multiple injection system. Beside the injection system and the EGR-rate, the shape of the combustion chamber, the CR, the air motion, and the air-fuel-ratio are also the important measures to improve the combustion to reach low raw emissions of exhaust gas to meet the stringent exhaust gas limits of the future.

In the present article, with and without EGR cooling, emissions were measured with 18.5 CR which forms a base at two set of engine operating condition from extra urban driving cycle (EUDC) part of new European driving cycle (NEDC). Later, the vehicle was updated with an engine which has a modification in CR from 18.5 CR to 16.5 CR.

From the experiments, it is found that the trends of emission were found to be similar with varying magnitude. However, a combination of 16.5 CR and no EGR cooling has a potential to meet the same emissions what 18.5 CR with EGR cooling. Thus, a potential cost saving of 3% can be achieved.

Table 1. Engine specification under testing

Engine parameter	Specification
Engine capacity [L]	2.2
Bore [mm]	85
Stroke [mm]	96
Compression ratio [–]	18:5:1 and 16.5:1
Rated power kW at rpm	88 at 4000
Max. torque (N-m) at rpm	290 at 1800-2600
Firing order	1-3-4-2
Aspiration	Turbocharged (VGT) with inter cooling
EGR and EGR cooling	Yes, without bypass

Experimental set-up and test description

The engine that was selected for the testing activity is a 2.2L direct injection (DI) common rail engine. The engine specifications are mentioned in tab.1. The engine was equipped with common rail injection system with the potential to inject pilot injection.

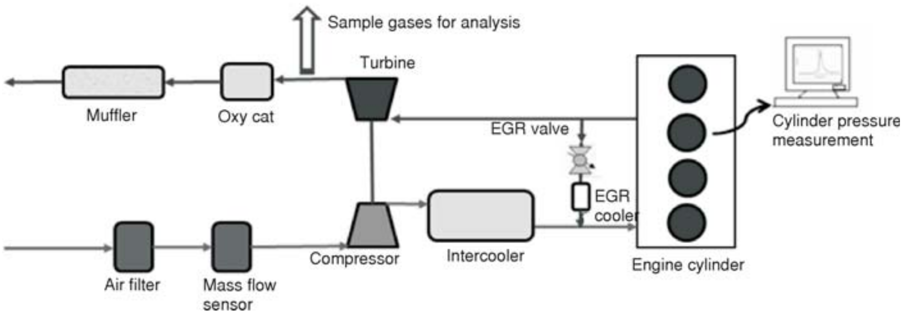

Figure1. Experimental set-up for the engine

The maximum possible rail pressure (RP) was up to 1600 bar. The boost pressure was controlled by pneumatically controlled variable geometry turbine (VGT) vanes. The line drawing of the set up is given in fig. 1.

The analysis of the gaseous emissions was performed using AVL AMA i60 emission system [14] which contains:
(1) non-dispersive infra red (NDIR) analyzer for the CO, and CO_2 measurements, (2) paramagnetic analyzer for the O_2 measurements, (3) chemiluminescence detector (CLD) analyzer for the NO_x measurements, (4) flame ionization detector (FID) analyzer for the HC measurements, and (5) smoke was measured by AVL415S smoke meter.

(1) AVL NDIR i60 analyzers are designed for the measurements of the concentrations of various infrared active compounds like CO_2, CO, and others in the exhaust gas. The measuring principle of the NDIR is the non-dispersive infrared absorption process. The instrument posses noise: ≤1% of range full scale, drift: ≤1% range full scale/24 hours and reproducibility: ≤0.5% range full scale. Linearity: ≤2% of measured value (10-100% of range full scale).
(2) AVL paramagnetic detector (PMD) i60 analyzers are designed for measurements of the O_2 concentration in the exhaust gas. The underlying measuring principle of the PMD is the paramagnetic property of O_2 molecules in a magnetic field. The instrument posses range of 0-1% for low measurement and 0-25% for highest possible measurement with drift: ≤1% range full scale/24 hours (at typical laboratory conditions, *e. g*. ambient temperature fluctuations within ±5 °C/41 °F), and reproducibility: ≤0.5% range full scale.
(3) AVL CLD analyzers are designed for the measurements of NO_x concentrations in the exhaust gas. The NO_x is defined as the sum of nitrogen oxide (NO) and nitrogen dioxide (NO_2). The measuring principle of the CLD is the emission of light from exited NO_2 molecules returning to a lower energy state. The analyzer posses a noise: ≤1% of range full scale, drift: ≤1% of range full scale *e. g*. ambient temperature, reproducibility: ≤0.5% of range full scale, linearity: ≤2% of measured value (≤1% range full scale whichever is smaller).
(4) AVL FID analyzers are designed for the measurements of the total hydrocarbon (THC) concentrations in the exhaust gas. The measuring principle is ionization of organic carbon atoms in hydrogen flame burning in an electric field. The instrument posses noise: ≤0.5% of range full scale, linearity: ≤2% of measured value (10-100% of range full scale) ≤1% range full scale whichever is smaller. The drift: ≤1% range full scale/24 hours (CH_4: additionally 1% of measured THC value/hours) (At typical laboratory conditions, *e. g*. ambient temperature fluctuations within ±5 °C/41 °F) and reproducibility: ≤0.5% range full scale (CH_4: additionally 1% of THC value possible).

(5) Smoke was measured with the help of AVL415S. The following tab. 2 gives the measurement.

Table 2. Measurement principal with measurement accuracy of AVL415S

Measurement principle	Measurement of filter paper blackening
Measured value output	Filter smoke number (FSN) or mgm^{-3} (soot concentration)
Measurement range	0 to 10 FSN
Detection limit	0.002 FSN or ~ 0.02 mgm^{-3}
Resolution	0.001 FSN or 0.01 mgm^{-3}

The soot flow rate was calculated based on the formula given in eq. (1).

$$\text{Soot} = [\text{Constant} \times \text{Smoke} \times \exp(0.31\times \text{smoke}) \times G_{\text{exh}}] \div e \tag{1}$$

where soot [gh^{-1}] is soot flow, smoke – smoke measured by smoke meter in FSN, G_{exh} [kgh^{-1}] – exhaust gas flow rate, and e [kgm^{-3}] – exhaust gas density.

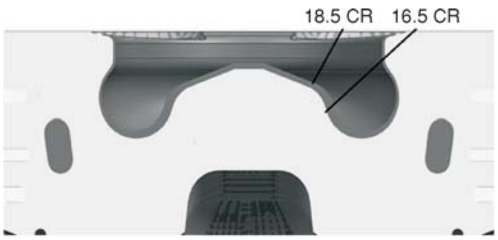

Figure 2. Piston bowl geometry modification to achieve 16.5 CR

The experiments were carried out in a four cylinder common rail diesel engine with two set of CR. The 18.5:1 CR forms a base and 16.5:1 was derived by changing the bowl shape. The CR reduction was achieved by increasing the bowl volume by 14.8% as seen in fig. 2 without disturbing the squish. The injector spray pattern was changed in order to suit the CR of 16.5:1. Due to the change in the bowl shape, an injector cone angle was changed from 152° (in case of 18.5 CR) to 148° (in case of 16.5 CR) keeping the through flow constant.

The tests were performed on eddy current dynamometer. The engine speed and torque was maintained as per the set of operating points. The intake air temperature and pressure were maintained at 1 bar and 25 °C. Both oil and coolant engine-in temperatures were set at 90 °C with flows maintained constant. The emissions were measured with the commercial analyzers from AVL mentioned above. A glow plug mounted in-cylinder pressure transducer was used for high cylinder pressure measurement. It is a commercially available with product name of GH13P was in combination with the M8 glow-plug adaptor AG04 a nearly flush mounted solution for diesel engines. It allows measurements without pipe oscillations and pressures range of up to 250 bar. The sensor has sensitivity of 16 pCbar^{-1} nominal and a linearity ≤ ±0.3% [14]. The cylinder pressure was acquired by AVL INDICOM Mobile [15]. The acquired data of cylinder pressure was used for calculating the heat release. The heat release calculations are based on the cylinder pressure measurement [15] and expressed by:

$$\frac{\mathrm{d}Qn}{\mathrm{d}t} = \frac{\gamma}{\gamma - 1} p \frac{\mathrm{d}V}{\mathrm{d}t} + \frac{1}{\gamma - 1} V \frac{\mathrm{d}p}{\mathrm{d}t} \tag{2}$$

where, dQn/dt is the heat release rate, γ – the ratio of specific heats, p – the pressure, dV/dt – the rate of change of volume, v – the volume, dp/dt – the rate of change of pressure.

Test methodology

Table 3 gives the two operating points chosen for the study along with the other injection and boosting parameters. These speeds were chosen as they are part of EUDC cycle of modified NEDC cycle for India. The EGR cooling effect will be dominant in these two points.

Table 3. Various parameters at chosen vehicle speed

Engine [rpm]	1800	2300
Torque [Nm]	55	66
Vehicle speed [kmh^{-1}]	70	90
Air fuel ratio	29.9	30.5
Fuelling (mg per stroke)	13	15.9
No of pilot	1	1
Pilot 1 separation [μs]	1690	1480
Pilot 1 quantity [mg per stroke]	1.04	1.04
Main injection timing (° bTDC)	1.43	3.5
Boost pressure [mbar]	1046	1210

Table 4 gives the set of parameters varied during the experiment. Here, the EGR is mentioned in term of air flow in mg per stroke. For 1800 rpm/55 Nm load, with no EGR, air flow was 584 mg per stroke. So, 380 mg per stroke of air flow indicate the EGR flow rate of 204 mg per stroke, *i. e*. 34% of EGR flow. The RP were measured with the help of rail pressure sensor and changed through electronically controlled electronic controlled unit (ECU).

Table 4. Test parameters

	EGR cooling on		EGR cooling off	
	18.5 CR/16.5 CR		18.5 CR/16.5 CR	
Engine (rpm) /load (Nm)	Rail pressure [bar]	EGR [mg per stroke]	Rail pressure [bar]	EGR [mg per stroke]
1800/55	590	380	630	340
2300/66	860	460	900	420

The first step in the experimentation involved the emissions measurement with base setting *i. e*. EGR cooling was on at selected points *i. e*. 1800 rpm /55 Nm and 2300 rpm/66 Nm from the optimized calibration. These points achieve the level of NO_x and PM such that the overall cycle emissions were met comfortably. These measurements were repeated for both the CR to see the change in the emissions level. In order to see the effect of cooling for EGR, EGR cooler was removed from the EGR circuit as second step. To bring the emission of NO_x back to its original value in the optimized setting, simultaneous increase of EGR and RP was carried out. This forms the step three. The increase in EGR rate with the increase RP allows increase of engines digesting capacity for EGR and helps in reducing the NO_x without/with marginal increase of soot emission [6]. Cylinder pressure was measured with the help of AVL INDISCOPE during all the measurement. The heat release rate and mass burned fraction was calculated from cylinder pressure measured using equations mentioned in previous section. This data is presented in result and discussion to support the effect and analyze the situation.

Discussion of results

Effect at 1800 rpm/55 Nm load

Figure 3 gives the comparison of soot *vs*. NO_x emissions for both 18.5 and 16.5 CR ratio in all three steps.

– Step 1: As it setting of EGR rate and RP with EGR cooling.

– Step 2: As it is setting of EGR rate and RP with no EGR cooling.
– Step 3: Increased EGR rate and RP with no EGR cooling.

It can be seen that the emissions, NO_x for 16.5 CR was not significantly lower as expected due to reduction of CR from base 18.5 CR. The observation confirms the finding by Laguitton *et al.* [3] NO_x emission under pre-mixed combustion is mainly controlled by other factors such as air fuel ratio, EGR rate rather than temperature and pressure. The removal of EGR cooler leads to the increase in the NO_x emission for both the CR and the magnitudes is also comparable. When EGR rate and RP was increased simultaneously, the value of NO_x was brought down to a value slightly lower than base optimized setting. As far as soot emissions are concerned, the soot was found to be increased with step 3 with both CR from its base value. However, there was a drastic shift in the soot emission for 16.5 CR than 18.5 CR.

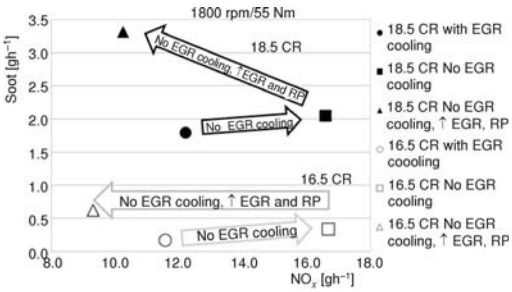

Figure 3. PM *vs.* NO_x emissions for three iteration and both CR at 1800 rpm/55 Nm

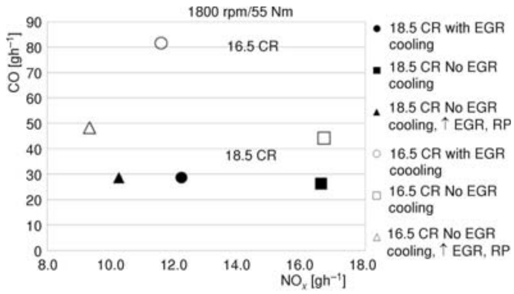

Figure 4. CO *vs.* NO_x emissions for three iteration and both CR at 1800 rpm/55 Nm

From fig. 4, it can be seen that the 18.5 CR shows no much change in the CO emissions. However, the 16.5 CR shows increase in the CO emission for base itself and hence for the other two iterations. The lowering of the CR leads to the lower combustion temperature and leads to the increase in HC (though not discussed in the article but exhibit the same trend as CO) and CO emissions. From fig. 5, it can be seen that the 16.5 CR has negative heat release rate. Since, the temperature is lower, most of the heat is being utilized for the vaporizing the fuel admitted inside the cylinder. There is no pilot injection burning seen in 16.5 CR as against the 18.5 CR. Due to this, the fuel burns in pre-mixed part of the combustion and lead to similar NO_x levels. The removal of EGR cooling brings CO emissions down as close as half (see fig. 4). This is due to the fact that cold EGR has reduced levels of O_2 concentration and improper combustion due to reduced overall cycle temperature and increased EGR rates. But, in order to meet the soot emissions, iteration three was tried out without cooling. Thus, in the base calibration, if we reduced the CR alone, though it brings the soot keeping same level of NO_x, but increase the CO and HC emissions. If we remove the EGR cooling in 18.5 CR, the NO_x can be brought down by use of increase in RP and EGR, but leads to increase PM (soot).

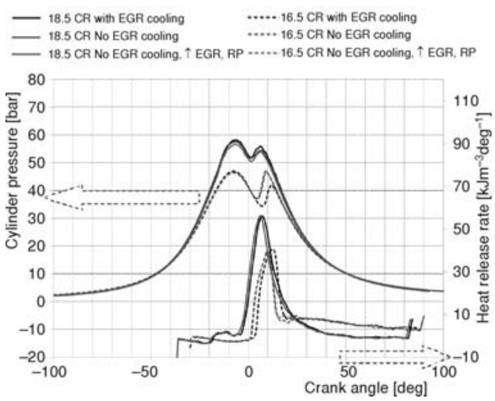

Figure 5. Cylinder pressure and heat release rate comparison at 1800 Rpm/55Nm

Thus, using 16.5 CR and removal of EGR cooling can meet the base emission level *i. e.* 18.5 CR and with EGR cooling, achieved with some compromise. Soot reduction was mainly achieved by lowering the CR and CO reduction by non-cooled EGR. Here, only CO was compromised *i. e.* 66.7%. (from 30 gh^{-1} to 50 gh^{-1}) with a gain in soot emission of 70% (2 gh^{-1} to 0.6 gh^{-1}). The loss in the CO and HC emissions were happening at the EUDC where light off of diesel oxidation catalytic convertor was already attained and can be negotiated with minimal effort.

Effect at 2300 rpm/66 Nm load

Figure 6 gives the similar comparison of soot *vs.* NO_x and CO *vs.* NO_x for 2300 rpm/66 Nm load. From fig. 6, it can be seen that the base emissions for soot for both the CR differs. The NO_x emissions were found to be slightly lower with 16.5 CR compared to 18.5 CR. Unlike 1800 rpm/55 Nm, 2300 rpm/66 Nm load operates at higher speed and at marginally higher load, the temperature and pressure influences the NO_x formation. When cooling of EGR was removed, the NO_x was increased for both the cases. In the iteration three, the NO_x was brought to its base calibration. In this process, the CO was matched very close to the base calibration with 18.5 CR. This can be seen in fig. 7.

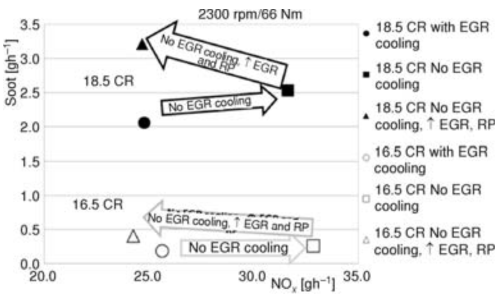

Figure 6. Soot *vs.* NO_x emissions for three iterations and both the CR at 2300 rpm/66 Nm

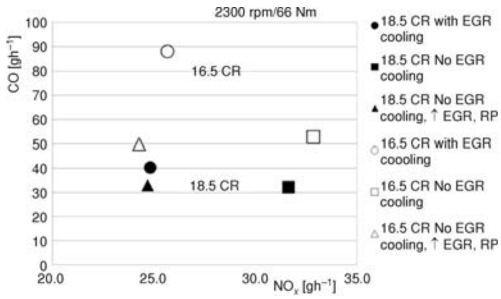

Figure 7. CO *vs.* NO_x emissions for three iterations and both the CR at 2300 rpm/66 Nm

In this case, the increase in CO was 23.6% with a reduction in soot emission of 80.9%. Higher CO can be attributed to the reasoning mentioned for the 1800 rpm/55 Nm point. From fig. 8, similar trend was observed for heat release and heat release rate.

Conclusions

From the experimental investigations, it was revealed that going with the trend of lower CR, significant soot reduction, and NO_x reduction is possible. The NO_x reduction was more prominent in higher loads. Such reduction in CR brings associated HC and CO increase. These emissions aggravates further with the use of heavy EGR cooling. No cooling of EGR reduces CO and HC and reduction of CR helps for better NO_x-soot trade off. Therefore, combining the benefits of each of the two combinations, we can meet the baseline calibration *i. e.* 18.5 CR

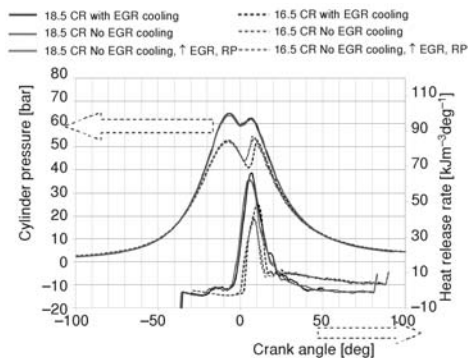

Figure 8. Cylinder pressure and heat release rate comparison at 2300 rpm/66 Nm load

and EGR cooling. In this process, EGR cooler can be eliminated and cost can be optimized with careful optimization of associate after treatment.

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Acronyms

bTDC – befor top dead centre
CLD – chemiluminescence detector
CO – carbon monoxide
CR – compression ratio, [–]
DI – direct injection
ECU – electronic controlled unit
EGR – exhaust gas recirculation
EUDC – extra urban driving cycle
FID – flame ionization detector
FSN – filter smoke number
HC – hydrocarbon
NDIR – non-dispersive infra red
NEDC – new European driving cycle
NO_x – oxides of nitrogen
PM – particle matter
PMD – paramagnetic detector
PPM – parts per millions
RP – rail pressure
rpm – revolution per number
TDC – top dead center
THC – total hydro carbon
VGT – variable geometry turbine

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