

DEDICATED EXHAUST GAS RECIRCULATION IN SPARK IGNITION ENGINES

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ABSTRACT

The impact of high levels of exhaust gas recirculation (EGR) has been well documented to decrease fuel consumption and reduce emissions of spark ignition engines. But there are also many limitations associated with this technology like EGR control and tolerance, which can reduce the potential efficiency improvements. A new concept called D-EGR has been presented in which the exhaust from a sub group of power cylinders is channeled back to the intake of all the cylinders. In this literature review both experimental and numerical analysis of this technology is shown. In the former case experiments were performed on 2.0 L PFI engine with gasoline as a fuel in part and high load conditions and the results show that at part loads the D-EGR engine can lead to lower Brake Specific Fuel Consumption, lower HC and CO emissions and higher brake thermal efficiency. At high load operations the results show improved combustion stability and superior knock tolerance. In the numerical studies it shows comparable thermal efficiency with conventional spark ignition (SI) engines and reduction in NO_x emissions.

Keywords: exhaust gas recirculation, NO_x emissions, spark ignition engines.

INTRODUCTION

The ever going race of improving fuel economy and reducing emissions has led to number of new technologies being invented and tested worldwide. Current SI engine technologies which are in production include but not limited to direct injection, downsizing with turbochargers or superchargers and hybridization. One alternate way is to dilute the incoming charge with fresh air or exhaust gases from the engine. It is well proven that lean burn in SI engines offer more efficiency but this may lead to more NO_x emissions. Cooled EGR is a promising technology which can lead to higher efficiency as it leads to increased indicated cycle efficiency and reduced heat transfer and throttling losses. It is because of the fact that EGR gases have a high specific heat capacity and it also works a diluent in combustion process. This lead to lower in-cylinder gas temperatures. It further improves efficiency by increasing the

ratio of specific heats during cycle. With high levels of EGR reduction in NO_x, CO, PM and PN emissions has also been noticed.

There are also some challenges associated with cooled EGR which include but not limited to misfire, stability of combustion, EGR flow control and engine component durability. With increasing the dilution levels of EGR burn velocities are decreased which can lead to unstable engine operation or misfire. To overcome these shortcoming while maintaining or exceeding the efficiency Dedicated EGR has been introduced.

D-EGR

D-EGR is an engine configuration in which the exhaust from one sub group of power cylinders is channeled back to the intake of all the cylinders i.e. creating EGR for all cylinders with the exhaust of a few. In the case of four cylinder

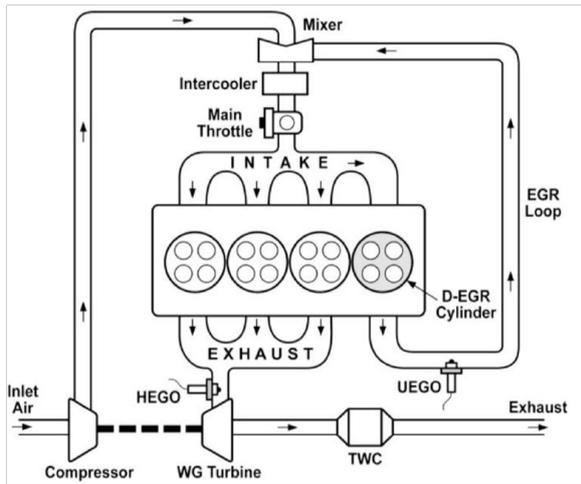


Fig. 1. Layout of turbocharged 4-cylinder engine in D-EGR configuration

engines, the exhaust of one cylinder is used as an EGR for the entire test engine which leads to stable EGR value of 25%. So, the need for active EGR control across the speed load range is minimized. This also increases the range of equivalence ratio of the dedicated EGR cylinder to run as its exhaust will never exit the engine before passing through other cylinders. So, D-EGR cylinder can be run rich and produce more H_2 and CO due to incomplete combustion. These two gases act as reformat for other cylinders and also helps to recover some waste heat which is generally rejected to atmosphere while still generating power in the dedicated cylinder. The presence of reformat enhances the flame speed, combustion and knock tolerance of all the cylinders and therefore can overcome the limitations of SI engines at higher EGR levels. Even in the low pressure loop EGR, by adding bottled reformat, the performance of D-EGR can be simulated. Other cylinders in the meanwhile can be run on stoichiometric air-fuel ratios which means that a three-way catalyst can be utilized to reduce pollutant emissions. Figure 1 shows the schematic of a D-EGR engine set up.

LPL EGR AND D-EGR AT PART LOAD OPERATION

Experimental Set up

In this study a turbocharged four cylinder 2.0 L engine with port fuel injection is used. In its baseline configuration, it produced a peak torque of 300 N-m at 2250 rpm and maximum power of 180 hp at 5500 rpm. The original compression ra-

tio of the engine was 9.1:1 which was increased to 12.5:1 and 13.5:1. The engine was made to run on different EGR values in low pressure loop and also with dedicated EGR with various equivalence ratios for dedicated cylinder which led to a nominal EGR value of 25% for rest of the cylinders. To make it possible to compare the above mentioned configurations both of these set up used identical engine long block, boosting and fueling hardware as well as ignition system.

Brake Specific Fuel Consumption

In Figure 2, Brake specific fuel consumption is shown for different cases of LPL EGR and D-EGR at 2000 rpm/2 bar BMEP. The labels inside the columns represent the cyclic variability of IMEP for each engine configuration. The equivalence ratio was kept constant for dedicated cylinder at $\Phi_D = 1.3$. Reduction in BSFC was mainly due to improved combustion which in turn could depend on various factors. There was substantial decrease in BSFC when 25% cooled EGR was used in baseline engine configuration as is clear from the difference between first and second column. The next major reduction was for D-EGR case which was due to significantly higher combustion stability due to high quality EGR. The increase in BSFC for D-EGR in fourth column was because no EGR mixer was used which lead to uneven supply of EGR to different cylinders which was also the reason for increase in CoV.

Emissions

The variation in emissions of various exhaust gases with respect to Φ_D is shown in figure 3. It is evident from the graph that with increase in Φ_D ,

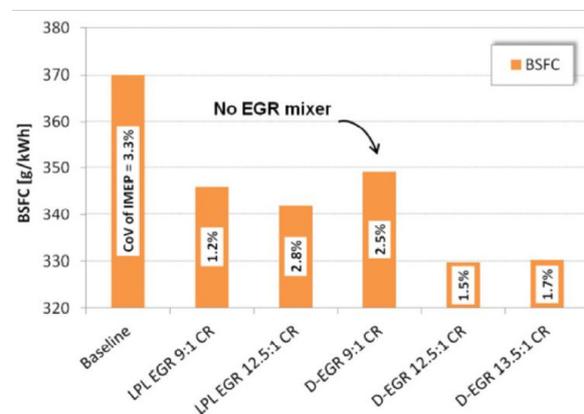


Fig. 2. BSFC for various LPL and D-EGR configuration at 2000 rpm/2 bar BMEP with $\Phi_D = 1.3$

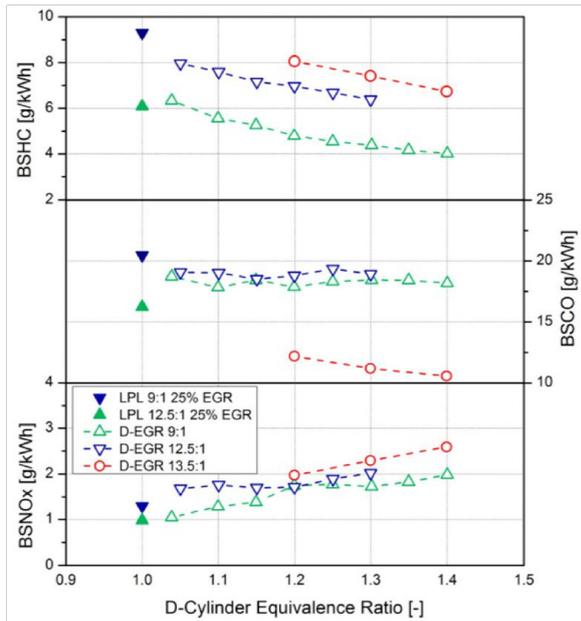


Fig. 3. D-EGR engine out emissions as a function of Φ_D at 1250 rpm/3.5 bar BMEP

BSHC was reduced for all compression ratios. In general, higher CR test showed higher BSHC due to crevice loading effect. Brake Specific CO was relatively unaffected with change in Φ_D but it was significantly lower for 13.5:1 CR. BSNO_x emissions were almost doubled owing to higher in-cylinder temperatures which was due to increase in amount of H₂ and CO in the exhaust of dedicated cylinder with increase in Φ_D but these were still ¼ that of a non-dilute engine.

Brake Thermal Efficiency

The variation in BTE and CoV at different part load conditions is shown in figure 4. Generally, the main efficiency improvements were due to cooled EGR and higher CR in LPL configuration.

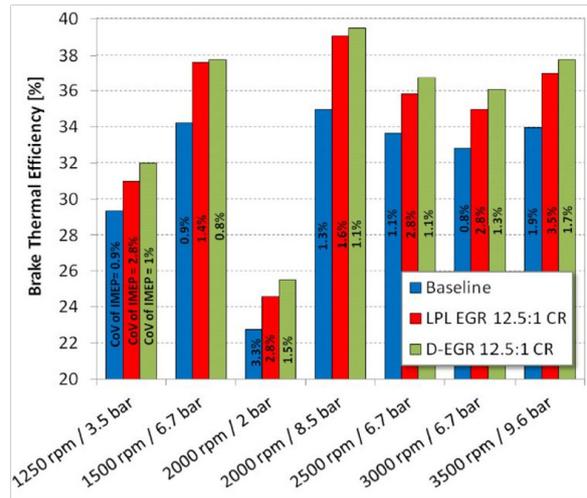


Fig. 4. Brake thermal efficiencies comparison at various part load conditions

Although, D-EGR set up showed higher BTE for all cases but its impact was less significant as compared to LPL EGR. Also, CoV was always lower in D-EGR due to stable EGR supply.

LPL EGR AND D-EGR AT HIGH LOAD OPERATION

Experimental Set up

The same engine set up was used in this case but the operations were conducted for higher loads.

Combustion Stability

The BSFC for D-EGR and LPL EGR configurations at 2500 rpm is shown in figure 5. All the tests were conducted for a constant CR of 12.5:1. As BSFC primarily depend upon combustion, it is

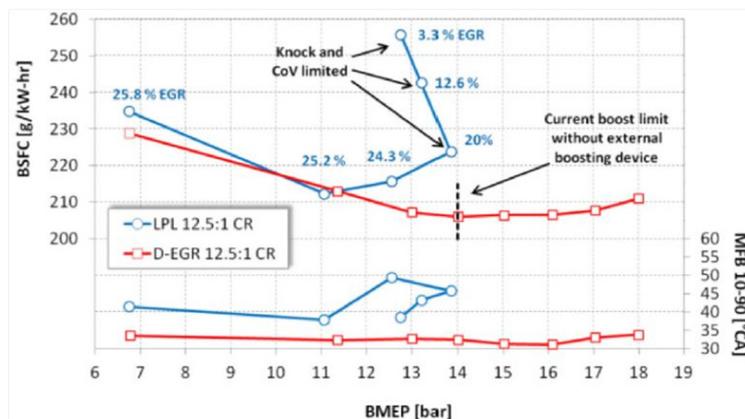


Fig. 5. Load sweep at 2500 rpm with 12.5:1 CR

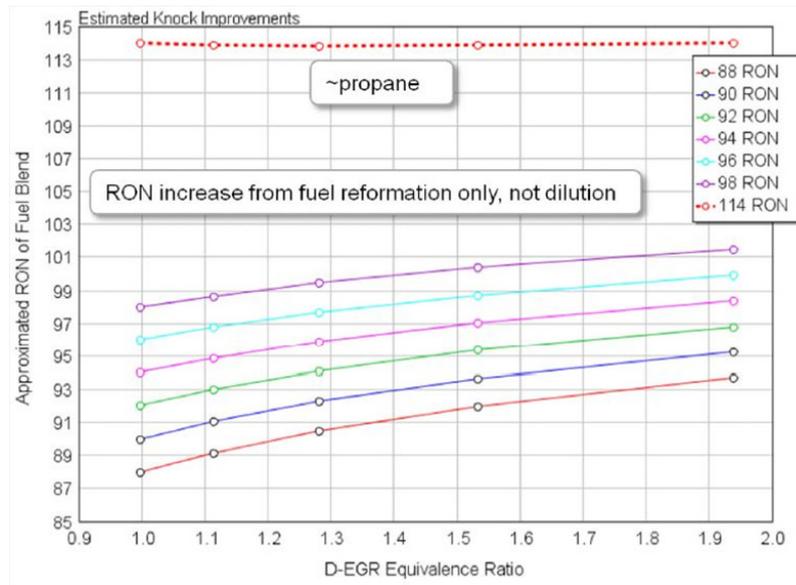


Fig. 6. Predicted RON increase with Φ_D based on component blends

apt to analyze combustion stability on a BSFC graph. The stability requirement in this case was to maintain the cyclic variation of less than 5% in IMEP. It can be seen that at low loads i.e. 6-7 bar, D-EGR engine configuration had better BSFC as compared to LPL EGR due to improved engine stability and combustion efficiency. At mid loads i.e. 11 bar BMEP there was small difference between the BSFC values of both cases as the engine was running at optimal combustion phasing with good stability and combustion efficiency in both LPL EGR and D-EGR. The most notable difference was found at higher engine loads i.e. above 11 bar where D-EGR improved combustion efficiency whereas LPL EGR engine came to its inherent combustion limitations. As is clear from the graph at 12.5 bar BMEP, EGR has to be reduced from less than 25% because of stability limit with spark retard. However, D-EGR engine could sustain 18 bar load due to high quality of EGR.

Impact of Φ_D on RON

The main reason why D-EGR leads to higher knock tolerance and better combustion can be seen by comparing the RON of the exhaust gas coming out of the dedicated cylinder. This is the result of a different simulation study conducted as part of same research by Gingrich et al. In figure 6 the impact of Φ_D on RON is shown. As can be seen with increase in the Φ_D , RON increased this was because of the fact that H_2 and CO both have 120 and 108 RON (approx.) ratings respectively

as compared with gasoline which has 91 RON. As Φ_D is increased more H_2 and CO was produced in dedicated cylinder due to incomplete combustion and these gases acts as a fuel reformat for other cylinders. Apart from this, H_2 also leads to higher laminar flame burning velocities and higher flame temperature thus avoiding the main problem with EGR i.e. partial burn and misfire.

Engine Operating Map LPL EGR vs D-EGR

To graphically represent the improvements possible with D-EGR concept, full operating map of LPL EGR and D-EGR configuration with 12.5:1 CR is shown in Figure 7. The red region at the top of the figure represents the region that

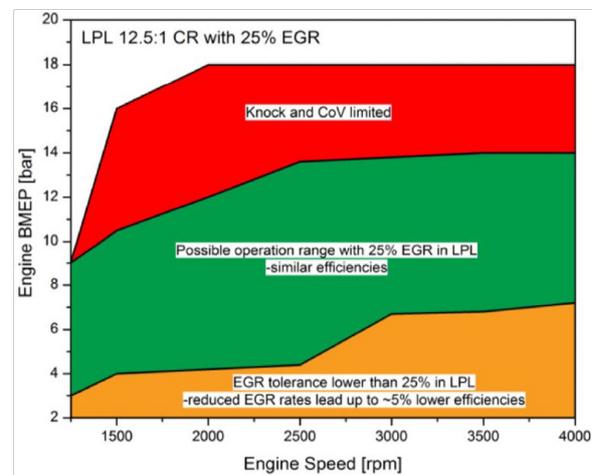


Fig. 7. Engine operating map in the LPL EGR setup with 25 % EGR

could not be accessed in LPL EGR case due to knock and CoV of IMEP limitations but was accessible in D-EGR configuration with proper boost system. The yellow zone at the bottom represents the area where operation with LPL EGR was not possible due to combustion instability which increases potential of partial burn or mis-fire. This region can be accessed with lower EGR but with BSFC penalty of up to 5%. Only in the middle green part LPL EGR configuration was capable of operating with 25% EGR and comparable efficiency to D-EGR.

Due to limitation of current review paper, quantitative impact of H₂ and CO on in-cylinder combustion process is not discussed in detail. However, it is evident beyond doubt that these gases improve combustion, the direct result of which can be seen in improved BSFC and lesser CoV.

D-EGR NUMERICAL ANALYSIS WITH NATURAL GAS

Analysis Method

In this study a numerical analysis of a SI engine was performed. The assumed SI engine had bore of 82mm, stroke length of 64mm and CR of 10.76. The calculation of laminar burning velocity was done at adiabatic conditions. The Φ for D-cylinder was assumed to vary from 0.6-3.4 and number of engine cylinders were assumed to vary from 3 to 6 that were run with the exhaust from dedicated cylinder and fresh air and fuel equivalence ratios of 0.6-1. The fuel selected was methane which had 90% natural gas. To calculate the engine work and heat release rate Weibe function was used. Combustion duration in Weibe function was assumed by function of turbulent burning velocity that was proportional to laminar burning velocity. Calculation of heat loss was done by Woschini function. Turbulent flow was not considered in this case and the temperature of cylinder wall was assumed to be 498.15k. Due to brevity of space detailed equations are not shown.

Thermal Efficiency

Total thermal efficiency of all cylinders i.e. 1-4 as a function of equivalence ratio of cylinder 1 is shown in figure 8. Cylinders 2-4 were run on stoichiometric conditions while the Φ_D was varied from 0.6-3.4. Inlet Temperature was assumed to be 298.15k and inlet pressure was considered

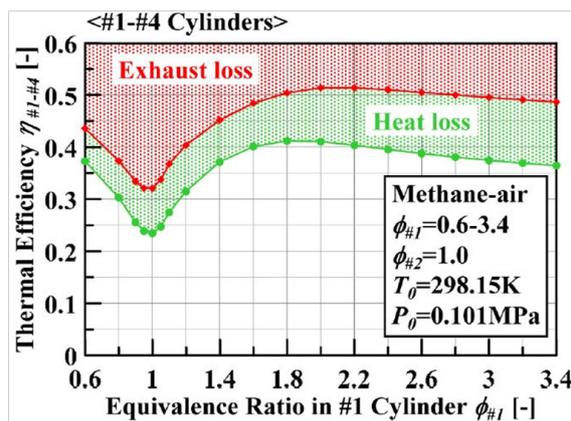


Fig. 8. Total thermal efficiency in the SI engine

as 0.101 MPa. It is evident from the graph that the net thermal efficiency increased when Φ_D was increased from 0.6 to 1.6. At $\Phi_D = 2.2$ thermal efficiency of the concept engine was highest. This seemed to be due to increase in H₂ concentration in the dedicated exhaust stream with increase in Φ_D . However, flame temperature also increased because of CO that promotes laminar burning velocity like H₂ does but also causes little increase in temperature which leads to heat losses. Therefore, maximum thermal efficiency after considering heat losses occurred at $\Phi_D = 1.8$. It is reasonable to expect that if there is a way to reduce heat loss to the cylinder walls maximum thermal efficiency will occur at $\Phi_D = 2.2$.

NO_x Concentration in Exhaust Gases

To show the impact of D-EGR on NO_x, six different cases of D-EGR composition were simulated to compute the NO_x concentration in the final exhaust gases coming out of the engine. These six cases are shown in table 1. All the cylinders were run on stoichiometric conditions except cyl-

Table 1. Calculation Conditions

Case	D-EGR ratio (number of cylinders)	Equivalence ratio		Thermal efficiency (%)
1*	no EGR (4)	$\Phi = 1.0$		36.0
2	no EGR (4)	$\Phi = 0.8$		41.7
3	$\gamma = 0.2$ (6)	$\Phi_{\#1} = 1.6$	$\Phi_{\#2-\#6} = 1.0$	41.9
4	$\gamma = 0.25$ (5)	$\Phi_{\#1} = 1.6$	$\Phi_{\#2-\#5} = 1.0$	41.7
5	$\gamma = 0.33$ (4)	$\Phi_{\#1} = 1.8$	$\Phi_{\#2-\#4} = 1.0$	41.2
6	$\gamma = 0.5$ (3)	$\Phi_{\#1} = 2.0$	$\Phi_{\#2-\#3} = 1.0$	38.4

* reference case

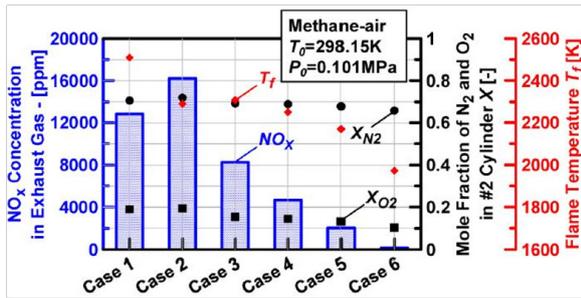


Fig. 9. NOX concentration in exhaust gas varied by the calculation

inder 1 where Φ_D was varied from 1.6 to 2. Also in case 1 and 2, no EGR was used.

The NOx emissions along with flame temperature for the six cases are shown in figure 9. The flame temperature was maximum for case 1 as in this case stoichiometric A/F ratio was provided which lead to complete combustion whereas in case 2 although flame temperature was lower than case 1 but it had higher NO_x emissions due to excessive O₂ which reacted with N₂ to form NO_x.

For cases 3-6 reduction in NO_x concentration was observed with increasing D-EGR, this was due to the fact that dilution effect of CO₂ and H₂O was becoming more prominent with increasing EGR. At the same time the amount of premixed air and fuel intake reduced, which lead to decrease in heat release rate and hence lower temperatures. The decrease in the inlet mixture meant decrease in N₂ and O₂ intake in each cycle which further decreased the NO_x concentration and this phenomenon intensified as the increase of EGR ratio. It is noteworthy that thermal efficiencies were more than 41% and NO_x emissions less than 8100 ppm when nominal D-EGR was less than 0.33. The last case with 3 cylinders although showed least NO_x but it was estimated that the flame temperature becomes too low to operate engine stably.

CONCLUSIONS

As this is still a nascent technology to corroborate the same type of results from different research institutions is not entirely possible but it can be said with confidence that with D-EGR technology some of the limitations of EGR can be corrected. There is one area where study number 1 and 3 do not fully agree, the first study shows overall reduction in exhaust emission but little increase in NO_x emissions whereas the third study

shows large reduction in NO_x emissions with increasing the D-EGR values. This discrepancy may be due to the fact that the latter work is a simulation study where many hypothetical cases like $\Phi_D > 1.7$ are considered which may not be practical due to physical limitations of the engine hardware.

- All the above studies confirm that with $\Phi_D > 1.2$, load range for stable combustion can be enhanced by using D-EGR concept.
- The HC and CO emissions are reduced with increasing Φ_D values as is predicted by the simulation as well as experimental studies.
- The main point at which all the studies agree is that thermal efficiency is improved by increasing D-EGR values.
- Second study shows wider range of engine BMEP operation with increasing values of D-EGR.
- Second study also proves that H₂ and CO acts as fuel reformat which is seen by increase in the RON ratings.

Abbreviations

- AFR - Air Fuel Ratio
- BMEP - Brake Mean Effective Pressure
- BSCO - Brake Specific Carbon Monoxide
- BSFC - Brake Specific Fuel Consumption
- BSHC - Brake Specific Hydrocarbons
- BSNO_x - Brake Specific Oxides of Nitrogen
- BTE - Brake Thermal Efficiency
- CO - Carbon Monoxide
- CoV - Coefficient of Variance
- CR - Compression Ratio
- D-EGR - Dedicated Exhaust Gas Recirculation
- Φ_D -Dedicated Cylinder Equivalence Ratio
- EGR - Exhaust Gas Recirculation
- HC - Hydrocarbon
- IMEP - Indicated Mean Effective Pressure
- LPL - Low-Pressure Loop
- NO_x - Oxides of Nitrogen
- PFI - Port Fuel Injected
- PM - Particle Matter
- PN - Particle Number
- RON - Research Octane Number
- SI - Spark Ignited

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