Effect of High EGR with \( O_2 \) Enrichment on the Exhaust Emissions of a Diesel Engine

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ABSTRACT

The purpose of the present study is to reveal the effect of high EGR with \( O_2 \) enrichment on the exhaust emissions in a direct-injection diesel engine. Through EGR with \( O_2 \) enrichment, the \( N_2 \) component in the intake is replaced with \( CO_2 \) from the recirculated exhaust gas. It is expected that this would give lower NOx emissions. When the EGR ratio is lower, the tradeoff between smoke and NOx is clearly shown. However, in the region of higher EGR ratio at both medium and high load conditions, no tradeoff is recognized. This could be due to the reduction of the intake \( N_2 \) concentration. Comparison of the tradeoff curves for EGR, timing retard, and high EGR with \( O_2 \) enrichment procedures show that high EGR with \( O_2 \) enrichment gives lower smoke and NOx emissions. The general trend of NO estimated using a simple thermodynamic adiabatic calculation, is consistent with the NOx behavior in the experiment.

INTRODUCTION

Increasing global awareness of the exhaust emissions problem has progressively led to more stringent emission regulations. The future directions of emission regulations in the U.S. and Japan are poised towards still lower NOx while in Japan where the NOx regulations are considered to be one of the strictest, is towards lower PM (particulate matter) [1]. The main target is the simultaneous reduction of smoke or particulate and NO emissions.

Exhaust gas recirculation EGR and \( O_2 \) enrichment are the two of the most widely known methods of diesel emission control methods. The EGR reduces NOx at the expense of smoke [2], [3], [4], while \( O_2 \) enrichment reduces smoke but increases NOx [5], [6]. Poor combustion results with increasing the EGR ratio due to the insufficiency of \( O_2 \) available for combustion. On the other hand, too much \( O_2 \) addition results in extremely high combustion temperatures resulting in greater NOx emissions.

If it is possible to recirculate large amounts of the exhaust gas without minimizing the \( O_2 \) content in the intake charge, perhaps the limitation of the unavailability of \( O_2 \) to support combustion could be eradicated. This combination procedure could give either of the two effects, NOx reduction or smoke reduction, which depends on which is predominant, the EGR effect or the \( O_2 \) enrichment effect. However, by increasing the EGR ratio and supplying \( O_2 \) gas, the intake air is greatly reduced, decreasing the intake \( N_2 \) concentration in the charge, potentially reducing the NOx emissions without the attached smoke increase.

To give a clearer picture of the concept of the combination procedure of EGR with \( O_2 \) enrichment, the volume flow rates for the recirculated exhaust gas, the supplied \( O_2 \), and the intake (fresh) air for the case of 80\% EGR with 40 L/min of supplied gas are shown in Fig. 1.

EXPERIMENTAL SET-UP AND PROCEDURE

A schematic of the experimental set-up is shown in Fig. 2. The test engine is a single cylinder, direct-injection diesel engine. Specifications are listed in Table 1.

The engine is coupled to an eddy current type dynamometer. \( O_2 \) gas is supplied from gas bottles. For safety reasons, the crankcase is purged with \( N_2 \) gas. This is done to prevent explosion in the case when the intake gas concentration is high. Provisions were made to recirculate a portion of the exhaust gas back to the intake system by increasing the back pressure. A valve controls the amount of exhaust gas recirculated.

In this study, the EGR ratio is defined as:

\[
\text{EGR\%} = \left( \frac{G_{\text{EGR}}}{G_a} \right) \times 100
\]

where,

\( G_{\text{EGR}} \) is the recirculated exhaust gas mass flow rate,

\( G_a \) is the intake air mass flow rate for the case of no EGR.
Table 1. Test engine specifications.

<table>
<thead>
<tr>
<th>Engine name:</th>
<th>ROBIN DY41D (Fuji Heavy Industries Co., Ltd.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type:</td>
<td>Single cylinder, 4-stroke, direct-injection, OHV, air-cooled</td>
</tr>
<tr>
<td>Compression ratio:</td>
<td>21</td>
</tr>
<tr>
<td>Bore × stroke:</td>
<td>82 × 78 mm</td>
</tr>
<tr>
<td>Stroke volume:</td>
<td>412 cm³</td>
</tr>
<tr>
<td>Maximum power / rpm:</td>
<td>6.3 kW (8.5 PS) / 3600 rpm</td>
</tr>
<tr>
<td>Injection pump type:</td>
<td>Bosch type</td>
</tr>
<tr>
<td>Injection nozzle:</td>
<td>4-hole nozzle</td>
</tr>
<tr>
<td>Nozzle hole diameter:</td>
<td>0.22 mm</td>
</tr>
<tr>
<td>Plunger diameter:</td>
<td>5.5 mm</td>
</tr>
<tr>
<td>Valve opening pressure:</td>
<td>19.1 MPa</td>
</tr>
<tr>
<td>Injection nozzle:</td>
<td>(Injection nozzle)</td>
</tr>
<tr>
<td>Injection Timing:</td>
<td>variable (17 deg BTDC constant in this study)</td>
</tr>
</tbody>
</table>

Fig. 2. Schematic of the experimental set-up.

G\text{EGR} is determined from the next equation as:

\[ G_{\text{EGR}} = G_{\text{a}} - (G_{\text{aEGR}} + G_{\text{O}_2}) \]  \hspace{1cm} (2)

where,

- \( G_{\text{aEGR}} \) is the intake air mass flow rate for the case of EGR,
- \( G_{\text{O}_2} \) is the supplied \( \text{O}_2 \) gas flow rate.

Cylinder pressure is measured with a water-cooled piezoelectric type pressure transducer coupled to a charge amplifier. TDC, crank angle, and needle lift signals are taken, digitized, and processed by a computer. The NOx is measured with portable chemiluminescence type analyzer, while smoke is evaluated using a Bosch type gas sampler in conjunction with a smoke meter.

OPERATING CONDITIONS

The test engine was run at a constant speed of 2800 rpm, over a medium (BMEP = 0.34 MPa), and a high load (BMEP = 0.50 MPa) conditions with the injection timing held constant at 17 deg BTDC.

(a) EGR with \( \text{O}_2 \) Enrichment Case (60% EGR with Supplied \( \text{O}_2 \))

(0-0), (60-0), (60-10), (60-20), (60-30), (70-30), (80-30)

In this condition the engine was run at the original
condition at medium load with no EGR and no supplied O₂. The EGR was then applied at 60%, and then O₂ was gradually added at 10, 20, and 30 L/min at the same EGR ratio. The EGR ratio was then increased from 60 to 70, and then to 80% with the supplied O₂ flow rate kept constant at 30 L/min.

(b) EGR with O₂ Enrichment Case (80% EGR with Supplied O₂)

( EGR% - Supplied O₂ Flow Rate L/min )
(0-0), (80-30), (80-35), (80-40)

As with condition (a) the engine was run at medium load conditions, starting with the original condition (0-0), after which EGR was applied at 80% with the supplied O₂ flow rate initially at 30 L/min which was then gradually increased to 35, and then to 40 L/min.

(c) EGR with O₂ Enrichment Case (80% EGR with Supplied O₂)

( EGR% - Supplied O₂ Flow Rate L/min )
(0-0), (80-40), (80-50), (80-60)

Here, the engine was run at high load engine conditions with the applied EGR at 80% with the supplied O₂ initially at a flow rate of 40 L/min which was later increased to 50, and 60 L/min.

RESULTS AND DISCUSSION

Figure 3 shows the effect of EGR with O₂ enrichment on smoke and NOx emissions. Typical results of EGR (point B) where NOx reduction is attained but with a corresponding increase in smoke can be clearly recognized. On the other hand, the effect of O₂ enrichment is predominant at the condition of EGR with O₂ enrichment (point E) where the smoke decreases slightly compared with the original condition (point A). However, it also resulted in a steep increase in NOx. For all the conditions tested here, the smoke - NOx tradeoff exists.

Figure 4 shows the results corresponding to the conditions in (b) where the applied EGR ratio was increased to 80% with the O₂ flow rate increased from 30 to 40 L/min. This was done in order to further decrease the intake air amount in order to reduce the amount of N₂ in the intake charge for possible NOx emissions reduction. At (80-40) condition (point D) the NOx level is below the 100 ppm level or a 95% reduction in NOx emissions without significantly affecting smoke. Even with the increase in the intake O₂ concentration from 16% at (80-30) to about 19% at (80-40), no significant increases can be found. This may suggest that the lower N₂ concentration, which is roughly about 30%, contributes to the lower NOx values.

Figure 5 shows the effect of EGR with O₂ enrichment on the brake specific fuel consumption (BSFC) for the medium load. At 60% EGR, the BSFC increases to about 23%. The BSFC increase is due to the lower combustion efficiency with the application of EGR. With the EGR held constant at 60%, the improvement in the BSFC can be recognized with the addition of O₂ gas. However, once the EGR ratio is increased up to 80% while the supplied O₂ flow rate is maintained at 30 L/min, the BSFC again increases. This can be attributed to the lowering of the combustion efficiency which is due to the reduction in the intake O₂ concentration. On the other hand, even with the gradual addition of O₂ to 40 L/min at 80% EGR, the BSFC is still higher than point A.

Figure 6 shows the results for smoke and NOx.
emissions for the high load condition described in (c). This figure shows that the application of high EGR with O₂ enrichment at the described conditions reduces tremendously the amount of NOx emissions. At (80-60) condition, the NOx emissions are reduced to about 90%, while the smoke BSU value was only 1 BSU higher than that of the original condition (0-0). At (80-40) condition it is reasonable to expect such low NOx values since at this point (point B), the O₂ concentration is low at 16%. At (80-60) condition, even with the intake O₂ concentration at 27%, the NOx emissions only slightly increases to a little above 100 ppm. On the other hand, at this condition, the intake N₂ concentration is very low which is estimated to be less than 10%. Again, this may suggest the impact of the low intake N₂ concentration on NOx emissions even with the increase in the O₂ concentration. Another factor may be the large intake CO₂ concentration brought about by applying higher EGR ratios.

The effect of high EGR with O₂ enrichment at high load on the BSFC is shown in Fig. 7. At (80-40) condition with the increase in the CO₂ concentration and with the decrease in the intake O₂ concentration, the BSFC increases. However, even with increasing the intake O₂ gas flow rate to 60 L/min (here, the intake O₂ concentration is 27%) the BSFC is still higher than that of the original condition (0-0). This suggests that there are other factors aside from the O₂ concentration which have an effect on the behavior of the BSFC.

The smoke-NOX tradeoff is plotted in Fig. 8 for the EGR with O₂ enrichment, and compared with the simple EGR at 20, and 40%, and retarded injection timings of 6 deg and 11 deg BTDC. It is interesting to note that the results of retarded injection timing fall near the tradeoff curve of simple EGR. Comparing with 40% EGR, the (80-60) condition gives more or less the same NOx value with 40% EGR, while the smoke is half the smoke value at (40-0). The tradeoff curve of higher EGR with O₂ enrichment is “weak”. In a previous paper, the authors [7] have also considered the reduction of the polytropic exponent “n” as one of the factors which may also contribute to the lower NOx emissions. This “n” decrease is attributed to the decrease in the specific heat ratio “k”.

Figure 9 shows the plot of the calculated adiabatic flame temperature and the calculated NO mole fraction using a simple adiabatic flame temperature calculation program [8].

The input parameters are listed just above the figure. The points A, B, C, and D correspond to the high load conditions stated earlier. The excess air ratio is 1.67 based on the original condition (0-0). The NO characteristic formation times considered are 1, 2, and 3 ms. From point A, the calculated adiabatic flame temperature decreases 416 degrees at B which is 19.47% lower than A. With the addition of more O₂, the flame temperature increases to point C from point B. With further addition of O₂, the flame temperature at point D (80-60), approaches that of point A. Considering only the behavior of the flame temperature, the calculated NO mole fraction would be roughly equal or even higher than the NO mole fraction of point A. However, as can be seen in the figure, for all the NO characteristic times considered, the NO mole fraction of (80-60) condition is roughly 78%.
Fig. 7. Effect of High EGR with O₂ enrichment on the BSFC for a relatively higher EGR ratio at high load.

lower than that of (0-0). This can be attributed to the lower intake N₂ gas concentration at (80-60).

Thus the feasibility to control NOx emissions by the intake N₂ concentration can also be verified by this simple thermodynamic model. The discrepancy between the NO reduction ratio of 78%, and the 95% reduction obtained in the experiment could be due to the simplicity of the model. It may also suggest other factors such as the lower combustion efficiency, and the decrease in the pressure and temperature at the time of injection due to the decrease in the specific heat ratio.

CONCLUSIONS

The concept of high EGR with O₂ enrichment was tested on a single cylinder, direct-injection diesel engine. A simple thermochemical calculation was utilized to verify the effect of EGR with O₂ enrichment on NO.

1. With the application of high EGR with O₂ enrichment, a condition was found where the so-called “smoke-NOx tradeoff” was almost non-existent. No significant increases in the NOx was found even with the increase in the intake O₂ concentration. This could be explained by the very low intake N₂ concentration.

2. A simple thermodynamic model was used to calculate the adiabatic flame temperature and the corresponding NO mole fraction. It was shown that the reduction in the intake N₂ concentration does reduce NO formation.

Fig. 8. Smoke and NOx relationships for various methods of emissions control.

\[ \text{Condition}\]

\[\begin{array}{c|ccc|c|}
\hline
\text{Intake Gas Composition} & \text{O}_2 & \text{N}_2 & \text{H}_2\text{O} & \text{CO}_2 \\
\hline
\text{A} & 20.73 & 79.27 & 0.00 & 0.00 \\
\text{B} & 16.30 & 34.01 & 33.49 & 16.20 \\
\text{C} & 22.60 & 21.41 & 27.27 & 28.70 \\
\text{D} & 27.20 & 8.30 & 22.70 & 41.80 \\
\end{array}\]

Fig. 9. Estimation of NO by a simple adiabatic thermodynamic calculation.
NOMENCLATURE

\[ G_{EGR} = \text{recirculated exhaust gas mass flow rate, \, kg/s} \]
\[ G_a = \text{intake air mass flow rate for the case of no EGR, \, kg/s} \]
\[ G_{aEGR} = \text{intake air mass flow rate for the case of EGR, \, kg/s} \]
\[ G_{O_2} = \text{supplied O}_2 \text{ gas flow rate, \, kg/s} \]
\[ (0-0) = \text{0\% EGR with 0 L/min supplied O}_2 \]
\[ (80-60) = \text{80\% EGR with 60 L/min supplied O}_2 \]

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REFERENCES