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LOW PRESSURE LOOP EGR SYSTEM ANALYSIS USING SIMULATION AND EXPERIMENTAL INVESTIGATION IN HEAVY-DUTY DIESEL ENGINE

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ABSTRACT−EGR (Exhaust Gas Recirculation) systems are extensively used to reduce NOx emissions in light duty diesel engine but its application to heavy duty diesel engines is yet to be widely implemented. In this study, the simulation model for a EURO 3 engine was developed using WAVE and then its performance and emission levels were verified with experimental results. The possibility of operating a EURO 3 engine with LPL EGR system to satisfy the EURO 4 regulation was investigated. Each component of the engine was modeled using CATIA and WaveMesher. The engine test mode was ESC 13, and the injection timing and fuel quantity were changed to compensate for the reduction of engine power caused by applying EGR. As a result of the simulation, it was found that EURO 4 NOx regulation could be satisfied by applying an LPL EGR system to the current EURO 3 engine.

KEY WORDS : EGR (Exhaust Gas Recirculation), Simulation, Diesel engine, Emission

NOMENCLATURE

BSFC : brake specific fuel consumption (g/kWh)
CRDI : common rail direct injection
DPF : diesel particulate filter
EGR : exhaust gas recirculation
ESC 13: european stationary cycle 13
HPL : high pressure loop
LNT : lean NOx trap
LPL : low pressure loop
PM : particulate matter
SCR : selective catalytic reduction
VGT : various geometry turbine

1. INTRODUCTION

The global warming and carbon dioxide are issues of great concern in our world. In an effort to mitigate their impact, emission regulations are becoming more stringent. In recent years, much of this regulation is targeted towards reducing pollution from diesel engines, which are widely considered to be a significant source of hazardous emissions. The demand and use of diesel engines have been continuously growing in our society since they provide better fuel efficiency and less carbon dioxide emissions than conventional gasoline engines (Morgan *et al.*, 1999; Valaszaki and Jouannet, 2000). Conventional diesel engines were mainly applied to heavy duty vehicles or machinery because of soot and PM production in low engine speeds. Recent improvements in fuel injection systems and electrical control devices allow diesel engines to be widely used for light duty vehicles and machines (Bravo *et al.*, 2005). For the heavy duty diesel engines, the emission standards were relatively generous compared with light duty engines, and therefore readily met with today's technology. But with the emission standards for heavy duty vehicles recently strengthened, there is an urgent need for Common Rail Direct Injection, EGR, and exhaust after treatment systems on heavy duty diesel engines to meet the new emission standards (Valaszaki and Jouannet, 2000; Zhu and Lee, 2005; Takada *et al.*, 2005).

Currently, NOx emission is the biggest issue in diesel engines. Emission from gasoline engines can be readily controlled by EGR and three way catalysts. However, three way catalysts are not applicable to diesel engine exhaust. As an effort to reduce NOx emission from diesel engines, several aftertreatment techniques such as Selective Catalytic Reduction (SCR) using urea and Lean NOx Trap (LNT) have been developed, But these techniques have all face technical limitations and are not applicable for broad operating conditions. For instance, urea SCR

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for heavy duty diesel engines require urea filling stations and on board urea injection systems (Kitamura *et al.*, 2005).

EGR is widely used in heavy duty diesel engines to reduce NOx emissions but EGR has disadvantages on BSFC and PM (Hawley *et al.*, 1999). Most of the light duty diesel engines currently use some form of EGR, and it is the primary emission control scheme which reduces NOx emission from the source. EGR is designed to mix fresh air with the exhaust gas and supply mixture into cylinders. It is known that the EGR rate should be increased to a certain level and the EGR gas should be distributed evenly to each cylinder to maximize NOx reduction rate. Decreased oxygen concentration inside the cylinders reduces NOx production. In addition, high thermal mass of the EGR gas lowers the temperature of the gas in the cylinder, which leads to a reduction of the NOx production rate (Zeldovich, 1946).

There are two different types of EGR. One is a High Pressure Loop (HPL) EGR and the other is a Low Pressure Loop (LPL) EGR. Generally, High Pressure Loop EGR pumps high pressure exhaust gas, after the exhaust manifold and before the turbine, into the path after compressor (Pfeifer *et al.*, 2002). The disadvantage of a HPL EGR is that it requires a separate induction system since the pressure difference between the exhaust and the intake ports are too small to increase the EGR rate sufficiently (Charles *et al.*, 2005). In the High Pressure Loop EGR, pressures at which exhaust gas is extracted and injected are both high, and supplemental devices are required due to the small pressure difference between them. Studies on the EGR pump, venturi and throttle valve to enable a more smooth gas flow are ongoing.

Another way to apply EGR on a heavy duty diesel engine is by using a low pressure loop EGR (LPL EGR). LPL EGR is often used on systems with aftertreatment. LPL EGR can enable a large amount of EGR gas to flow easily because the pressure of the exhaust gas after the exhaust manifold, turbine and DPF are significantly lower than the pressure of the exhaust manifold and it is aspirated before the compressor. (Kapparos *et al.*, 2005) HPL EGR can contaminate the intake system because the unfiltered exhaust gas enters directly into the intake manifold. On the other hand, LPL EGR can be applied without any contamination of the intake system due to the filtered exhaust gas. LPL EGR has a higher turbo charger efficiency compare to HPL EGR, and therefore, LPL has an advantage on BSFC. The engine durability increases for LPL since DPF causes less intake system contamination occurs with the LPL EGR (Charles *et al.*, 2005). In addition, the exhaust gas is cooled while passing through the DPF, therefore requiring only a low capacity EGR cooler.

HPL EGR has the advantage on BSFC over LPL EGR due to the lower pumping loss at low rpms. On the other hand, HPL EGR requires additional devices such as venturi and intake throttling which force exhaust gas into the intake manifold (Tao *et al.*, 2005). HPL EGR easily contaminates the intake system and compressor due to the unfiltered HC and soot while LPL EGR is known as a relatively clean EGR due to its use of filtered exhaust gas. Additionally, LPL EGR has nearly perfect EGR distribution even at high EGR rates. However, LPL EGR suffers from limited pressure ratio due to the increased temperature of compressor inlet gas (Heywood, 1988).

For this study, WAVE is used for the Engine system simulation. WAVE is a useful simulation software for engine performance and 1D gas dynamics used in many industry sectors including motor sport, automotive, motorcycle, truck, agricultural, locomotive, marine and power generation. WAVE enables performance simulations to be carried out based on virtually any intake, combustion and exhaust system design, and a new driveline model allows complete vehicle simulations.(Ha *et al.*, 2006)

ESC (European Stationary Cycle) 13 Mode was used as the operating condition in the analysis of engine cycles. As shown in Table 1, modes 1 and 2 are the additional points, and the others correspond to the ESC 13 mode. Four different load conditions 100%, 75%, 50% and 25% for each engine speed were selected under the given operating condition. Changes for each engine speed and load must be completed within the first 20 seconds, and the engine speed and load must be maintained within ±50 rpm and ±2% torque, respectively (Chen and Yanakiev, 2005). In addition to this ESC 13 mode, two other arbitrary modes (within the control area) were also selected.

Table 1. ESC 13 Mode operating condition.

Mode NO	Speed (rpm)	Torque (N·m)	Load (%)	Weighting factor
1	1000	1264	100	0
2	2200	1019	100	0
3	599	37	0	0.15
4	1285	1327	100	0.08
5	1610	649	50	0.1
6	1610	970	75	0.1
7	1285	661	50	0.05
8	1285	995	75	0.05
9	1285	330	25	0.05
10	1610	1290	100	0.09
11	1610	324	25	0.1
12	1935	1155	100	0.08
13	1935	287	25	0.05
14	1935	860	75	0.05
15	1935	573	50	0.05

The engine simulations were carried out for the following reasons. First of all, performance, BSFC and emission level of the engine can be predicted and improved based on its hardware parameters. It can potentially minimize the engine development period, cost and manufacturer's risk. New NOx strategies, which satisfy EURO 4 and EURO 5 emission standards for the heavy duty diesel engines can be developed by creating the virtual EGR Loop and validating its performance and emission levels. In addition, important parameters for NOx generation can be predicted.

The generation of PM is studied under the assumption that the vehicle is equipped with a DPF. Eventually, the system is designed to filter and regenerate PM using DPF, because applying EGR to diesel engines increase PM generation.

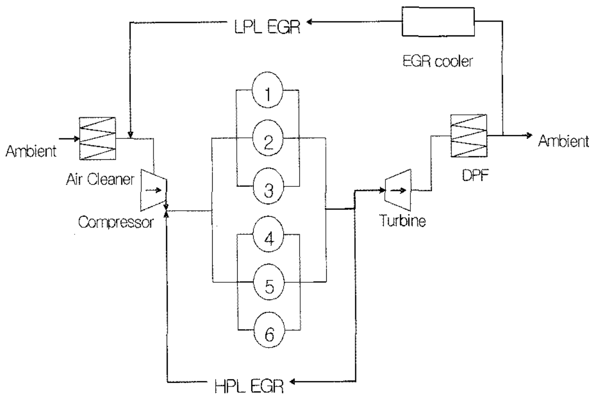

Figure 1. Schematic diagram of high pressure loop EGR and low pressure loop EGR system.

2. EXPERIMENTAL SETUP

In this experiment, a heavy duty diesel engine with Common Rail Direct Injection system is used. A 7.6L six cylinder engine, which has a bore of 108 mm, stroke of 139 mm and compression ratio of 16.6, produces maximum power of 320PS at 2200 rpm and maximum torque of 135 kgf·m at 1200 rpm. This engine is only equipped with a turbo charger and operates without EGR and VGT.

3. SIMULATION METHOD

3.1. Analysis Methods

To provide accurate information regarding the dimension and geometry of the intake manifolds for WAVE, CAD designs were implemented using CATIA. WaveMesher was also used to convert geometric data for the intake manifolds to the WAVE format data. The actual compressor and turbine map was used for WAVE simulations. The pressure of the fuel in the common rail, intercooler temperature of its inlet and outlet, the fuel injection rate and timing, and the intake valve lift are simulated to match real engine operating conditions. Prior to installing experiment device, we performed the model simulation to find the maximum EGR rate by changing EGR duct diameter. Based on this simulation results, we decided the EGR duct diameter for the experiment. Therefore, the diameter and length of the EGR pipe were determined and various engine conditions were chosen to recover the power loss caused by the EGR. EURO 3 engine was simulated using WAVE and its actual experimental data were compared with the simulation data.

3.2. Simulation Model Setup

Figure 1 shows the model of an actual engine used in the WAVE simulation. A WAVE model is created by connecting components such as tapped pipes, multiple ducts, junctions, orifice, and the atmospheric condition blocks to analyze the general or complex-compressive flow. In the WAVE simulation, it is very important to model the exact dimensions of the experimental engine. The mesh of the intake manifold was generated using CATIA. For the simulated structure of the intake manifold, each duct and junction were modeled separately and created exactly the same as the actual structure of the intake manifold. The dimensions of elements such as ducts and junctions can be easily modified. The compressor, turbine, air cleaner and DPF were also simulated exactly the same to the actual engine specifications. The actual engine data was implemented in WAVE as input parameters. For the air cleaner and DPF, the characteristics of pressure drop were considered and applied to WAVE. The actual maps of the compressor and turbine were also used.

4. RESULT AND DISCUSSION

4.1. Performance Validation

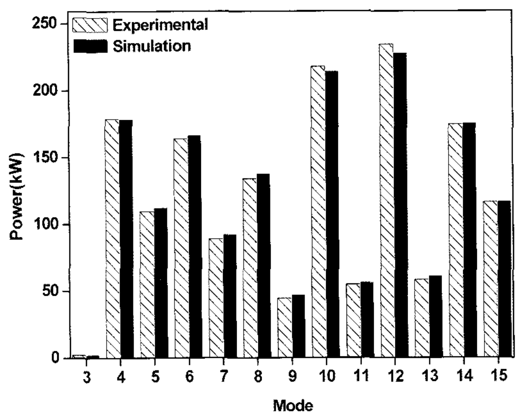

Figure 2. Power comparison between simulation and experimental results.

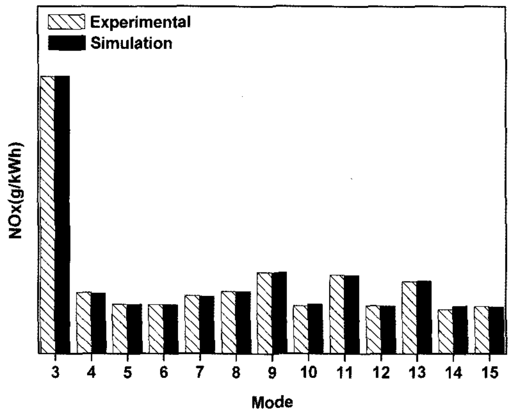

Figure 3. NO_x emissions comparison between simulation and experimental results.

Experimental and simulation results are shown in Figure 3. The differences of each point are within 5%.

From this figure, it can be confirmed that the simulation results are in good agreement with experimental results.

4.2. NO_x Emission Validation

Figure 3 shows the results of the NOx Emissions. Differences between the experimental and simulation results are within 5%. The Zeldovich model was used for the NOx prediction.

The simulation without the EGR system has exactly the same experimental condition as the real engine. This will help to understand the upcoming simulation tendency. To match the real value of experimental engine, the NOx value of each mode for the simulation was tuned by adjusting the NO_x scaling factor which is included in Diesel jet combustion model of WAVE.

4.3. EGR Duct Diameter for Maximum EGR Rate

The diameter of the EGR duct must be selected before applying EGR to the engine, since EGR rates are varied for the different load condition under the ESC 13 mode operating conditions. Sometimes, NOx production needs to be minimized by fully opening the EGR valve. The maximum applicable EGR rate depends on the EGR duct diameter; therefore the maximum EGR rates for each ESC 13 mode were analyzed as shown in Figure 4 when the EGR valve is fully opened. The applicable EGR rates differ between the hot and cooled EGR system when determining EGR rate.

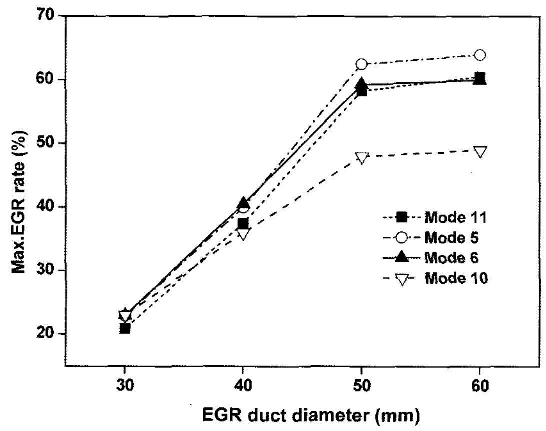

Figure 4. The decision of maximum EGR rate by simulation method for 1610 rpm operating condition.

4.4. The Validation of Simulation and Experimental Results

Figures 5–7 show the comparison between experimental and simulation results under cooled EGR condition. Both results are well matched for all load conditions. The experimental data and simulation results show deNOx efficiency of 90% or higher for 25% load condition. Also, torque has not decreased much even with an EGR rate of 35% because of enough O_2 in the EGR gas as shown in Figure 5 (Heywood, 1988). Figure 6 also shows deNOx efficiency of up to 50 % with 50% load and EGR rate of 20%. Under 75% load condition in Figure 7, maximum 5% of EGR was used due to relatively large torque drop compared to 25% load condition, but when the injection timing was advanced by 3° (degree) the EGR rate could be increased up to 10% (Choi *et al.*, 2004).

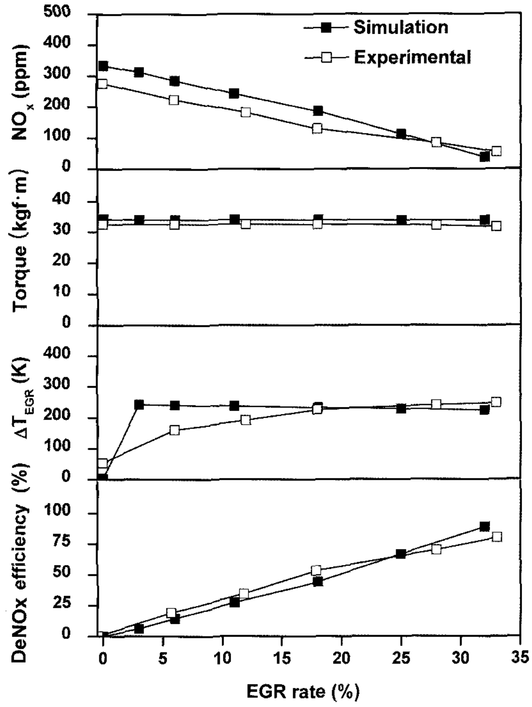

Figure 5. Effects of different EGR rate on $deNO_x$ efficiency, NO_x, torque and EGR temperature at 1610 rpm, 25% load condition.

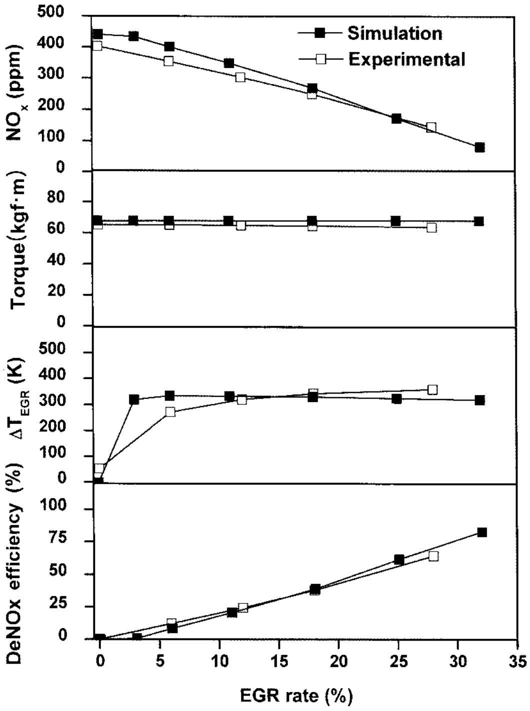

Figure 6. Effects of different EGR rate on deNO$_x$ efficiency, NO$_x$, torque and EGR temperature at 1610 rpm, 50% load condition.

4.5. NOx Emissions in ESC 13 Mode

The calculations results in Figures 8 and 9 show the relationships between EGR rate, NOx emissions level, NOx reduction efficiency in applying cooled EGR and hot EGR. The 35–40% of Hot EGR rate and 50% of Cooled EGR rate are applied to the Modes 9, 11, and 13, which are all 25% load conditions. It is assumed that the Hot EGR system does not have the EGR cooler system , and the Cooled EGR condition is similar to the Experiment condition in Figures 5–7. The NOx reduction efficiency is higher than 80% in Modes 5, 7, and 15 at which load is 50%. As previously mentioned, these simulation results are in good agreement with the experimental results. The Maximum NOx reduction efficiency can be achieved under the load conditions lower than 50%.

It has been shown that the cooled EGR rate is higher than hot EGR rate (Lapuerta *et al.*, 2000; Gao and Schreiber, 2002). When the injection timing was advanced by an average of 3° (degree) in order to apply maximum EGR rates in both Hot EGR and Cooled EGR systems, the power loss was regained regardless of the load conditions. The cooling range of the Cooled EGR gas was limited to a maximum of approximately 120°C. Regardless of the engine load condition, reasonable temperatures of Cooled EGR gas were found.

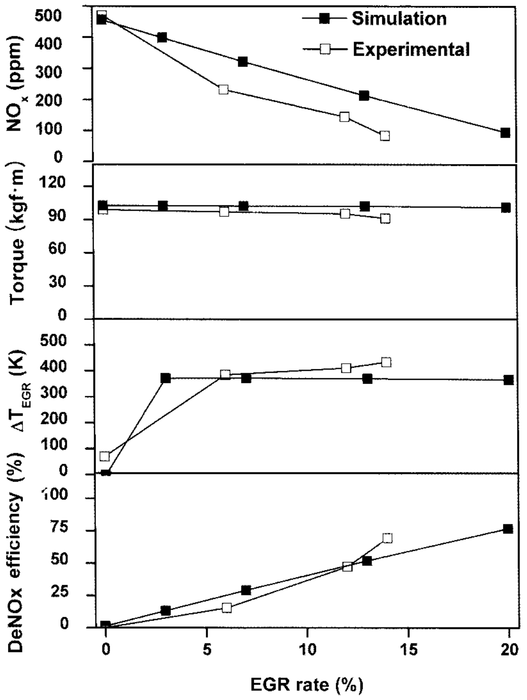

Figure 7. Effects of different EGR rate on deNO$_x$ efficiency, NO$_x$, torque and EGR temperature at 1610 rpm, 75% load condition.

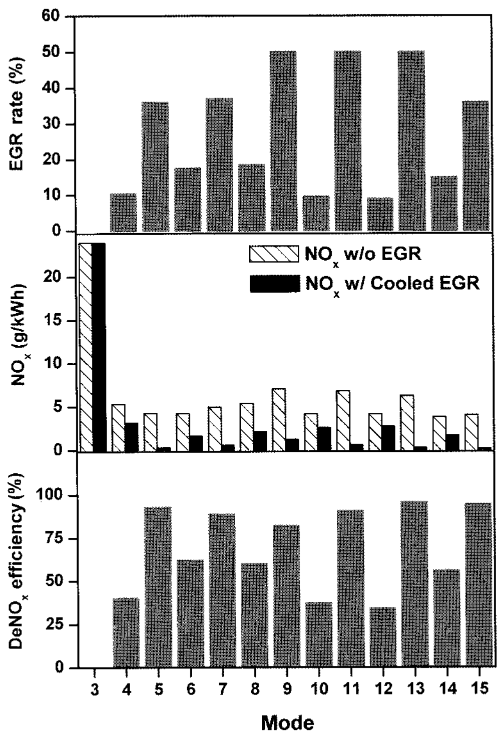

Figure 8. Cooled EGR effect on NO$_x$ emissions in ESC 13 Mode with EGR rate, deNO$_x$ efficiency.

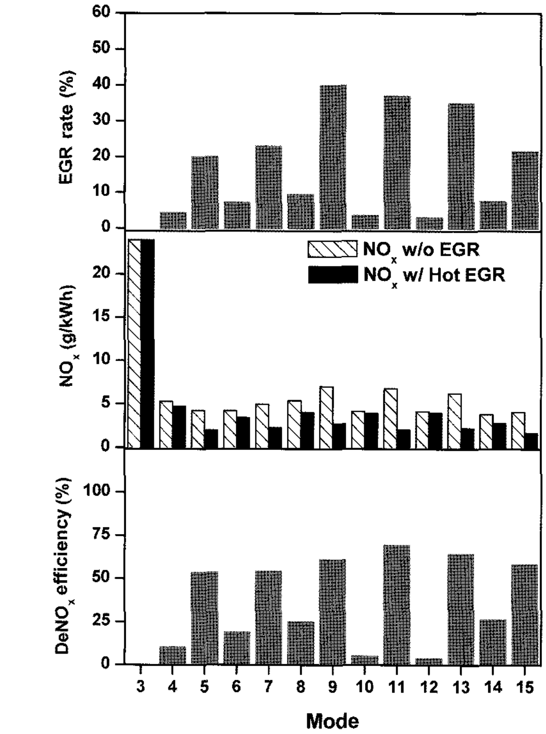

Figure 9. Hot EGR effect on NO_x emissions in ESC 13 Mode with EGR rate, $deNO_x$ efficiency.

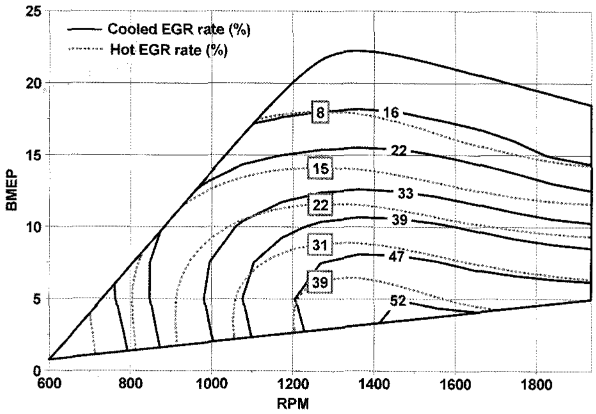

Figure 10. The cooled and hot EGR rate contour map with different engine speed and bmep.

4.6. The Optimized EGR Map

In Figure 10, Hot EGR rate and Cooled EGR rate are plotted as a function of engine speed and load conditions. In the ESC 13 mode which includes the main operating range from 1285 rpm to 1935 rpm, the maximum applicable Cooled EGR is 35% for load conditions of 50% or lower and 25% for load conditions higher than 50%. With a constant power output and fuel supply, the applicable EGR rate was optimized. As shown in the figure above, an additional 8% EGR could be applied for each

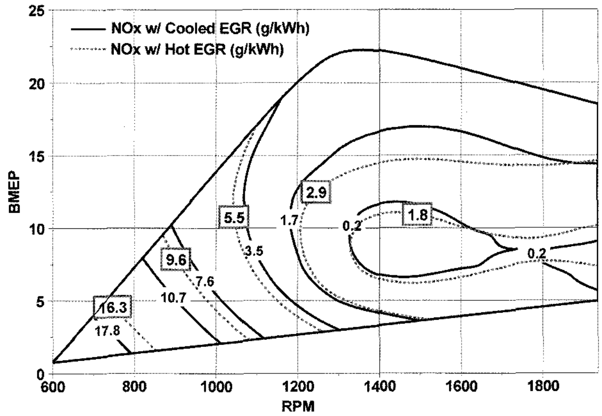

Figure 11. The cooled and hot EGR NO_x emissions contour map with different engine speed and bmep.

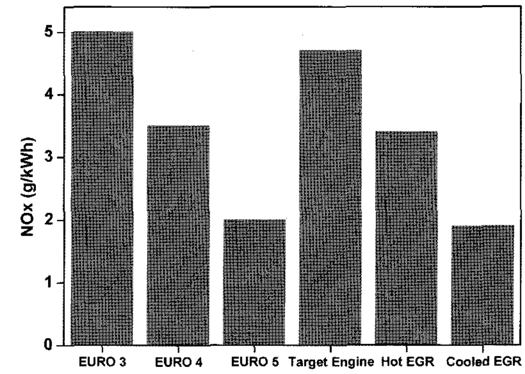

Figure 12. The comparison of NO_x emissions with EURO legislation and simulation results.

operating conditions with Cooled EGR. For example, with bmep of 5 at 1200 rpm, the applicable EGR rates were found to be 47% for Cooled EGR and 39% for Hot EGR. Also, with bmep of 17 at 1200 rpm the applicable EGR rates were found to be 16% for Cooled EGR and 8% for Hot EGR.

4.7. The Map of NO_x Emissions

Figure 11 shows NOx emissions from the simulation results. The Cooled EGR rate is higher than the Hot EGR rate under the fixed rpm and BMEP operating conditions. Therefore, lower NOx emissions level will be achieved in Cooled EGR condition. And NOx production for Cooled EGR and Hot EGR system shows a large deviation with various operating conditions (Tennison and Reitz, 2001; Danov and Gupta, 2004). At low rpm area, it shows 30% NOx reduction efficiency regardless of bmep. But it shows eight times better NOx reduction efficiency at the pressure around 10bar and rpm between 1400 and 1800, because the cooling effect of the Cooled EGR is relatively larger in this area.

4.8. Target Strategy and EURO Legislation

EURO emission legislations are as shown Figure 12, which also includes NOx emissions of the experimental engine.

The experimental engine used in this study is not yet equipped with the EGR system, which satisfies only EURO 3 emission legislations. The engine with hot EGR system will satisfy EURO 4 emission legislation of 3.5 g/kWh. To satisfy the EURO 5 legislation, it will be necessary to apply a Cooled EGR system. This experimental engine will satisfy the EURO 5 legislation of 2.0 g/kWh. The simulation method will be useful to predict emission levels for the design of an EGR system.

5. CONCLUSION

Simulations were performed using WAVE to fulfill the EURO 4 and the EURO 5 legislations for an experimental engine, currently configured to satisfy the EURO 3 legislation.

Cooled EGR strategy was selected as the primary NOx emission control method. The calibrated simulations show a maximum of 5% difference with experimental results. To determine the maximum EGR rate of the experimental engine, EGR valve opening ratios for each section of the ESC 13 mode was maximized and the EGR duct diameter was changed. The applicable maximum EGR rate for each point was found to be linearly proportional to the EGR duct diameters between 30 to 50 mm.

A Cooled EGR system could apply 8% more EGR rate than the Hot EGR system regardless of its operating condition. The production rate of NOx varies for different operating conditions, but always exhibited at least a 30% NOx reduction efficiency.

The applicable ranges of both Hot EGR and Cooled EGR were simulated and determined for each ESC 13 Mode. The LPL EGR was simulated in WAVE to verify the potential of the experimental engine. The results show that EURO 3 certified experimental engine has the potential to satisfy EURO 4 and EURO 5 legislations by applying Hot EGR and Cooled EGR, respectively.

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