DIRECT INJECTION DIESEL ENGINES: WHAT IS THE LIMIT FOR NOx REDUCTION?

D.W. Dickey and T. Ryan
Southwest Research Institute

INTRODUCTION

The purpose of this paper is to review in-cylinder NO_x reduction strategies and to investigate the lower limit for NO_x emissions for a direct-injected diesel engine. NO_x and particulate emissions have decreased steadily for the past decade while engine thermal efficiency has simultaneously increased. The question is, how long can we continue to refine the current direct injection combustion system and still realize further reductions in NO_x emissions? Is there some fundamental limit to in-cylinder NO_x reduction, and if so, what is it?

In the United States, the Statement of Principles (SOP) agrees to NO_x emissions of 2.0 g/hp-hr in the year 2004 for on-highway heavy-duty engines. At the same time, the SOP research goals are 1.0 g/hp-hr. Engine manufacturers can refine current technology to meet the 2.0 g/hp-hr NO_x level for 2004. The research goal of 1.0 g/hp-hr NO_x is much more challenging. There are potentially two options to meet the 1.0 g/hp-hr level. One option is to use NO_x aftertreatment which is now the subject of intense research. Lean NO_x catalysis, $DeNO_x$ systems, and thermal plasmas are some of the technologies being studied. The other option is to attack the problem at the source (i.e., in-cylinder). It will be shown in this paper that if we change the combustion process in the diesel engine from a diffusion flame to a premixed lean combustion system, then orders of magnitude reduction in NO_x are possible.

BACKGROUND

In the late 1980's, meeting the 1991 and 1994 U.S. NO_x standard of 5.0 g/hp-hr was viewed as a real challenge. Now just a few short years later we are faced with meeting 4.0 g/hp in 1998 and 2.0 g/hp in 2004 with SOP research goals of 1.0 gram NO_x. The 1991 and 1994 5.0 gram levels were easily met using turbocharging with intercooling, higher injection pressures, combustion system optimization, and injection timing retard. The 1998 standard of 4.0 g/hp will be met with the addition of injection rate shaping. The 2004 NO_x level of 2.0 g/hp can be demonstrated using the above technology plus the use of exhaust gas recirculation. Unfortunately, EGR has an adverse effect on particulates so careful attention must be paid to developing an EGR system to optimize the NO_x-Particulate-BSFC tradeoffs while not sacrificing engine durability.

Historically then, we have been able to meet future NO_x standards by advancing the state-of-the-art for the diffusion flame combustion system. Now, to meet NO_x emissions standards beyond 2004, we may have to take a more radical approach. We should begin to think about changing the mode of combustion in order to get significant reductions in NO_x without resorting to aftertreatment systems.

IN-CYLINDER NO_x REDUCTION STRATEGIES

NO_x is formed in the combustion chamber when nitrogen and oxygen are present in a high temperature region. In a diesel engine, the adiabatic, stoichiometric, flame temperature is used as the characteristic temperature for a diffusion-controlled combustion process. Fuel mixes with entrained air in the fuel jet and burns near stoichiometry. Correlations between this calculated flame temperature and NO_x emission formation are common in the literature.[(1)*] Methods to reduce NO_x then focus on reducing peak flame temperature and\or reducing the oxygen concentration.

Numbers in parentheses designate entries in the Reference List

Strategies used to reduce NO_x emissions include:

- Injection Timing Retard
- Injection Rate Shaping
- Charge Air Chilling
- Water Fuel Emulsions
- Exhaust Gas Recirculation (EGR)

The effect of injection timing retard is shown in Figure 1. Injection timing retard can be used to reduce peak flame temperature and NO_x emissions but at the expense of fuel consumption. Injection timing retard was important to meet the U.S. 1994 NO_x level but by 1998, the benefit was depleted because further retarding of injection timing can result in poor combustion. The ability to change injection timing as a function of engine speed and load is also important.

Injection rate shaping can be used to tailor the injection event to reduce peak flame temperature. Hydraulic intensifiers and some common rail fuel systems have the capability to change injection rate shape as a function of speed and load as shown in Figure 2 (Herzog, P., IMECHE Conference 1989). Rate shaping can also be accomplished by changing the injection cam profile for in-line pumps and unit injector engines.

Special cases of rate shaping include pilot injection and split injection. The effect of pilot injection on NO_x and BSFC is shown in Figure 3[2]. In this case, the pilot injection event has a main injection that is retarded four degrees (compared to the no pilot case) with a pilot that leads the main injection by 15 degrees crank angle. As shown in Figure 3, NO_x is reduced at this 50 percent load condition by 16 percent with no effect on BSFC.

Split injections are another possibility to reduce NO_x emissions. Split injections are compared with a single injection in Figure 4[3]. The split injection provides a lower particulate level at retarded injection timings which allows the engine to operate at a lower NO_x level. Note that the improvements shown in Figure 4 were accomplished on an engine that may not be representative of more advanced technology engines. The objective of split and pilot injection is to tailor the heat release to minimize peak combustion temperature where NO_x is formed.

Air-to-air intercooling was effectively used in 1991 to reduce NO_x emissions. Charge air chilling is an extension of this same principle. A vapor cycle refrigeration system was added to the engine intake to lower intake manifold temperature well below the 40°C level used for conventional intercooling. The effect of manifold temperature on NO_x emissions is shown in Figure 5. NO_x reduction varied from 0.5 - 0.67 percent per °C reduction in intake manifold temperature. Unfortunately, chilling the intake air is not a viable solution due to the large BSFC penalty (shown in Figure 6) and the increase in cost and package size.

Water fuel emulsions are another technology used to reduce NO_x emissions. Water fuel emulsions are somewhat unique because they also have the potential to reduce particulates and fuel consumption. The effect of water content in the fuel on NO_x emission is shown in Figure 7. At a high load, the NO_x reduction is 1.0 - 1.3 percent per 1 percent water by volume in the fuel. A production water emulsion system must have the ability to control the water content in the fuel as a function of engine speed and load on a cycle-by-cycle basis.

EGR is now the most popular technology being investigated to meet the 2004 NO_x level of 2.0 g/hp-hr. A well-developed EGR system can reduce NO_x by 40 percent or more over the U.S. transient cycle. EGR can also be used to improve engine fuel consumption at intermediate NO_x levels by substituting EGR for injection timing retard. The effect of EGR on NO_x emissions is shown in Figure 8. EGR reduces NO_x by diluting the fuel air mixture with inert mass which has the effect of limiting peak flame temperatures. Unfortunately, EGR also increases particulates (except at idle where EGR reduces particulates) by lowering the temperature late in the combustion cycle where particulate matter is oxidized.

EGR SYSTEM DEVELOPMENT

The development of an EGR system will briefly be described in this section. The EGR system was used as a tool to investigate the lower NO_x limit.

A high pressure loop (HPL) EGR system was developed and installed on a 1991, 11.1 liter, Detroit Diesel, Series 60 engine. A schematic of the HPL EGR system is shown in Figure 9. A unique HPL EGR valve was designed, fabricated, and installed between the exhaust manifold and turbine housing as shown in Figure 10. A unique throttle blade located in the valve simultaneously opened the EGR passage and applied backpressure to drive EGR. Under non EGR conditions, the EGR passage was closed to preserve turbocharger pulse energy. A schematic drawing of the valve and photographs of the valve in the full open and closed positions are shown in Figures 11-13.

The effect of EGR on steady-state NO_x and particulates at rated power is shown in Figure 14. EGR rates are selected to provide the best NO_x-PM tradeoff. As shown in Figure 14, about 8 percent EGR can be used at rated power to reduce NO_x by 40-50 percent with minimal impact on particulate. Steady-state mapping can be used to derive an EGR strategy for the U.S. FTP as shown in Figure 15. No EGR is used at low speed high load due to the air-fuel ratio deficit. Fifty percent EGR is used at idle to reduce both NO_x and PM. At part load, 10-30 percent EGR is used.

An EGR control system is required to make the transition between EGR regions as well as to handle extreme transients. The transient EGR strategy turns off EGR during severe "up-transients". A step transient shown in Figure 16 (taken from a portion of the FTP) was used to tune the transient control parameters. The effect of the transient control strategy on opacity is shown in Figure 17. The opacity with EGR and transient control approaches the no EGR level. Turning off EGR during up transients also has an impact on NO_x emissions as shown in Figure 18.

Several U.S. transient tests were then conducted with the Series 60 engine. Real time NO_x and particulate emissions for the third section (600-900 seconds) of the transient test are shown in Figures 19 and 20. The emissions for the engine with and without EGR are similar during up transients because EGR was turned off during severe up transients. At the other conditions, EGR generally lowers NO_x and increases particulates.

The effect of several different EGR strategies on transient NO_x and particulates is shown in Figure 21. A curve is drawn through the data showing the best tradeoff. The best EGR strategy for this engine produces about a 40 percent reduction in NO_x with a 20 percent increase in particulate. EGR was gradually added until the engine began to fail the transient test statistics due to poor engine performance. It is interesting to note that the curve asymptotically approaches 2 g/hp-hr NO_x at high EGR levels for this 1991 engine. Modern engines with more refined combustion systems can achieve NO_x levels below 2 g/hp-hr with EGR using today's diesel fuel. The question remains, what is the lower limit for NO_x? Can the SOP research goal of 1 g/hp-hr NO_x be acheived?

LIMIT FOR NO_x REDUCTION

What keeps us from reducing NO_x even further? A list of potential factors that limit NO_x reduction are listed below:

- Fuel Composition
- Fuel Injection System
- EGR System
- **Diesel Engine Combustion**

Fuel composition has a definite effect on NO_x emissions. Generally fuels with a higher H/C such as methane produce lower NO_x due to their lower flame temperatures. But for the sake of this argument we will assume that we will use diesel fuel. Advances in fuel injection systems have resulted in lower NO_x emissions through rate shaping and injection timing control. Further advancements in fuel systems will likely result in lower particulates but the NO_x benefit is probably already saturated. EGR systems that are perfectly matched and have perfect transient control could result in an additional 15 percent reduction in NO_x but again this parameter's effect is already saturated. The real limit to further NO_x

reduction is the direct-injection diesel engine combustion process itself.

Diesel engine combustion is shown pictorially in Figure 22. A high pressure fuel jet entrains air during injection into the combustion chamber. While this nonhomogeneous mixture has high global air fuel ratios (ranging from 20 - 120:1) combustion occurs locally near stoichiometry. This localized, stoichiometric, diffusion controlled combustion process results in high peak flame temperatures and high NO_x. The basic problem with the diffusion controlled combustion process is that the combustion zone temperature is much higher than the bulk gas temperature.

The bulk gas temperature is compared with a calculated combustion zone temperature for a diesel engine in Figure 23. The combustion zone temperature is much higher than the bulk gas temperature because of the nonhomogeneous localized burning that takes place with diffusion controlled combustion surrounding the individual fuel jets. NO_x is formed in this high temperature combustion zone. The best we could do to reduce NO_x emissions would be to burn the fuel everywhere in the combustion chamber at the global air fuel ratio. For this combustion scenario, the two curves in Figure 23 would converge; the combustion zone temperature would approach the bulk gas temperature.

Returning to the original question, what is the limit for NO_x reduction? The limit is diesel engine combustion. Stoichiometric diffusion combustion produces high flame temperatures and high NO_x. Diluents can be added to reduce flame temperature up to the point where combustion begins to deteriorate. If the diesel combustion could be changed from a stoichiometric, diffusion flame to a homogeneous, lean mixture combustion, significant reductions in NO_x emissions are possible.

The other important question is what is the lowest engine-out NO_x that can be obtained using the diesel engine? A "NO_x Boundary" is shown in Figure 24 along with the future NO_x and particulate standards for the U.S. The NO_x Boundary is defined as the minimum NO_x that the diesel engine can emit. While the exact value of the NO_x Boundary is not known; it is shown as 1.5 g/hp-hr in Figure 24 assuming today's diesel fuel and current diesel technology. The value of 1.5 was chosen based on experience. Certainly, most engine manufactures have demonstrated engine out NO_x values near or just below this value. The exact value of the NO_x Boundary is not as important as the overall concept that a lower limit exists due to the fundamental limits of diffusion flame combustion. Showing the NO_x Boundary at 1.5 g/hp-hr does however raise the question, can we get to the SOP research goal of 1.0 g/hp-hr without aftertreatment?

The conventional diesel engine (stoichiometric diffusion flame) produces NO_x values above the NO_x Boundary as shown in Figure 24. Retarding the injection timing, injection rate shaping, chilling the intake air, and adding diluents such as EGR or water can only reduce the NO_x to the NO_x Boundary. There are three possible ways to jump over this NO_x Boundary:

- Change the fuel
- Use NO_x aftertreatment
- Change the diesel combustion process

The first two approaches are not considered in this paper. If the diesel combustion process is changed from a stoichiometric diffusion flame to a lean premixed type of combustion system, the flame temperature and NO_x can be reduced. In fact, the area below the NO_x Boundary represents the ideal combustion where the combustion temperature approaches the bulk gas temperature.

A SwRI NO_x model called ALAMO_ENGINE[1] was used to estimate the value of the NO_x Boundary as an analytical check on our experimental observation. First, ALAMO_ENGINE results were compared with experimental data from a Caterpillar 3176 engine equipped with an EGR system, as shown in Figure 25. The air-fuel ratio and EGR rates were measured for an 8-mode cycle. The measured NO_x agreed well with the ALAMO_ENGINE result. More aggressive EGR schedules were then applied using the model while maintaining a minimum air-fuel ratio as shown in Figures 26-29. Finally, the most aggressive EGR schedule was selected as shown in Figure 29 in an effort to determine the NO_x Boundary. The results in Figures 28 and 29 indicate that the NO_x Boundary is somewhere between 1.0 and 1.5 g/hp-hr. Note that this is a best case scenario because the model does not account for combustion deterioration with EGR.

HOMOGENEOUS CHARGE DIESEL ENGINE

As mentioned above, one method to go below the NO_x Boundary is to change the diesel combustion process from a diffusion flame to a lean premixed combustion. SwRI has conducted research with a Homogeneous Charge Compression Ignition (HCCI) combustion system [4]. The HCCI system is shown schematically in Figure 30. Diesel fuel is port injected using an air assist injector into a heated intake air stream. The homogeneous charge is then compression ignited creating many ignition sites throughout the lean mixture. The multiple lean ignition sites produce low peak flame temperatures and extremely low NO_x emissions. HCCI combustion approaches the ideal combustion (below the NO_x Boundary) where the combustion zone temperature approaches the bulk gas temperature.

The NO_x and particulate emissions from a single cylinder diesel engine operating in conventional diesel and HCCI combustion mode are compared in Figure 31. The HCCI engine produces NO_x values that are 98 percent lower than the conventional diesel engine. The particulates for HCCI combustion should also be very low. The particulates for both the diesel and HCCI engine are relatively high in Figure 31 because of the use of a small, single-cylinder, variable-compression-ratio, research engine. In addition, characterization of the particulate emissions obtained during HCCI operations indicted that the particulate was predominantly volatile organic material. A comparison between the heat release rate and cylinder pressure for the diesel and HCCI engines are shown in Figures 32 and 33. HCCI combustion heat release always has a characteristic precombustion, or cool flame reaction, before the main combustion event, as shown in Figure 32.

HCCI engines produce extremely low NO_x emissions compared to the conventional diesel engine, as shown in Figure 31. So when will all the diesel engine manufacturers convert to HCCI combustion? Well there are some technical challenges that must be overcome before HCCI combustion engines are adopted for use in trucks. The main challenge with HCCI combustion is to control the start of combustion. HCCI combustion occurs between limits of misfire on one side and engine knock on the other side. It is also difficult to get the engine to operate in HCCI mode over all of the engine speed and load range. The efficiency of current HCCI engines is not as good as diesel engines because of a lower compression ratio (10-11:1) required to eliminate knock and early start of combustion resulting in poor cycle efficiency. To eliminate the problem with limited speed and load operating range, the first HCCI engines will probably show up as dual-combustion mode engines. The engine would start as a conventional diesel engine and then switch over to HCCI operation (by turning on a port injector and turning off the in-cylinder injector) at predefined speeds and loads. The conventional diesel engine could have a HCCI operating regime that produces extremely low emissions. The "Dual Combustion Mode Engine" would improve the emissions of the conventional diesel engine while solving the limited operating range problem of the HCCI engine[5]. The ultimate goal is to have a diesel engine that operates over the full operating range with HCCI combustion. The "complete HCCI engine" can then eliminate the expensive direct injection fuel system and replace it with a relatively inexpensive port fuel injection system.

CONCLUSIONS

- In-cylinder NO_x reduction may be limited by diffusion flame combustion in the direct-injected diesel engine.
- With today's diesel fuel and current technology, the observed NO_x limit appears to be around 1.0-1.5 g/hp-hr over the U.S. transient cycle.

- The ALAMO_ENGINE NO_x model predicts a NO_x limit of 1.2 g/hp-hr over a simulated transient cycle. The NO_x model assumes no deterioration in combustion with EGR and a minimum air fuel ratio of 24:1. The model does not predict particulate emissions, which may be extremely high.

- NO_x aftertreatment or fuel modifications will most likely be required to meet the SOP research goal of 1.0 g/hp-hr over the transient cycle.

- Homogeneous Charge Compression Ignition (HCCI) combustion is one method to achieve extremely low engine-out NO_x emissions; a 98 percent reduction in NO_x was measured compared to conventional diesel combustion.

- Future diesel engines may operate with a dual-mode combustion system. The engine may start as a conventional diesel engine and then switch over to HCCI combustion mode for predefined speed and load ranges. The dual-mode combustion diesel engine will have emissions that are lower than the conventional diesel engine. The dual-combustion mode diesel engine may be able to meet the SOP research goal of 1.0 g/hp-hr NO_x.

- A "complete HCCI engine" has the potential to eliminate the direct injection fuel system and replace it with a relatively inexpensive, port fuel injection system.

ACKNOWLEDGMENTS

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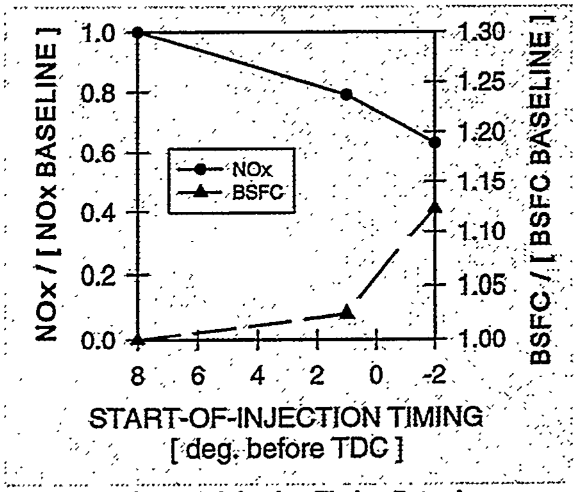


Figure 1. Injection Timing Retard

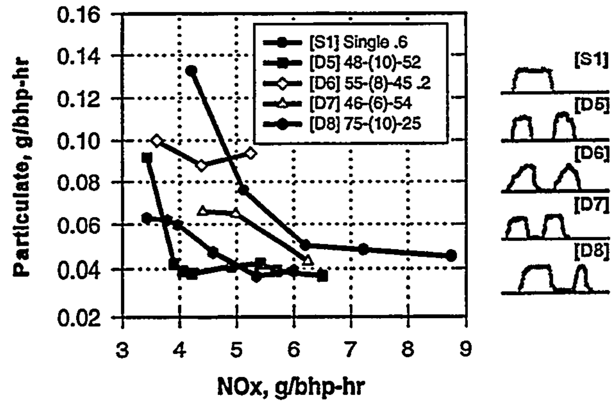


Figure 4. Injection Rate Shape: Split Injection

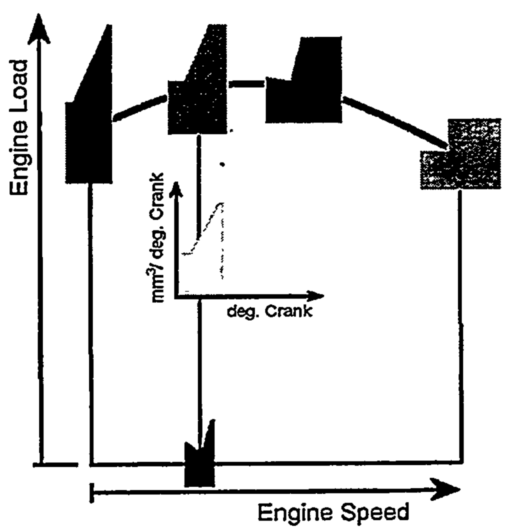


Figure 2. Advanced Injection Rate Shape

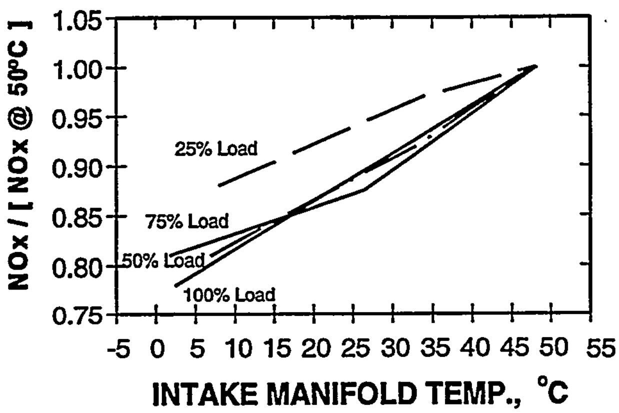


Figure 5. Charge Air Chilling

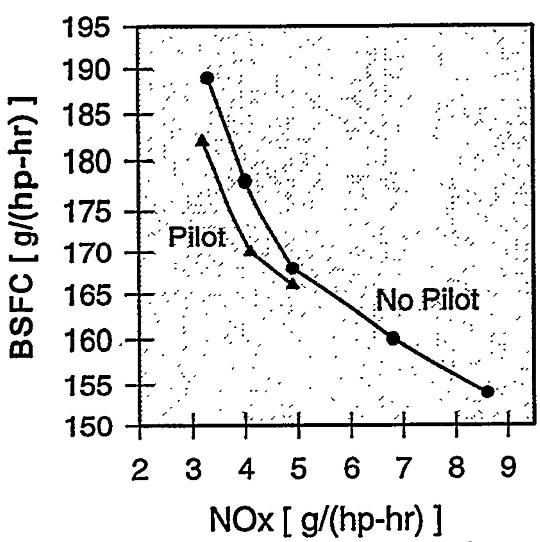


Figure 3. Injection Rate Shape: Pilot Injection

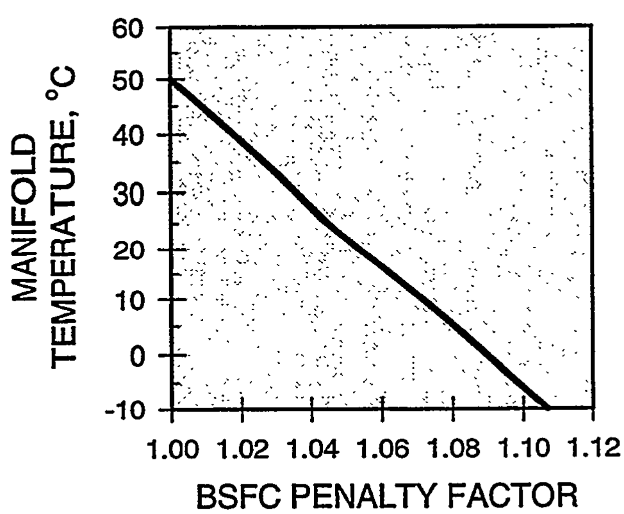
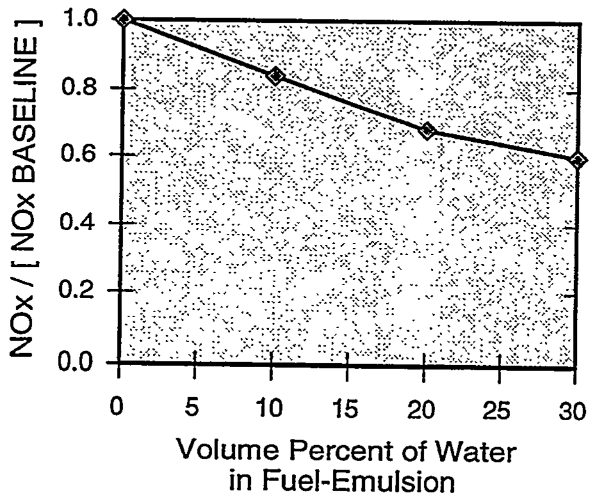


Figure 6. Vapor Cycle Chilling Cost



6 Liter, 2000 rev/min, 80% Load

Figure 7. Water Fuel Emulsion

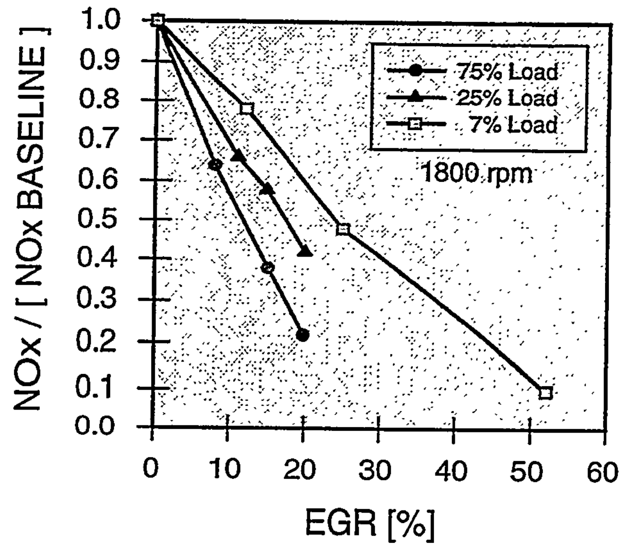


Figure 8. Exhaust Gas Recirculation (EGR)

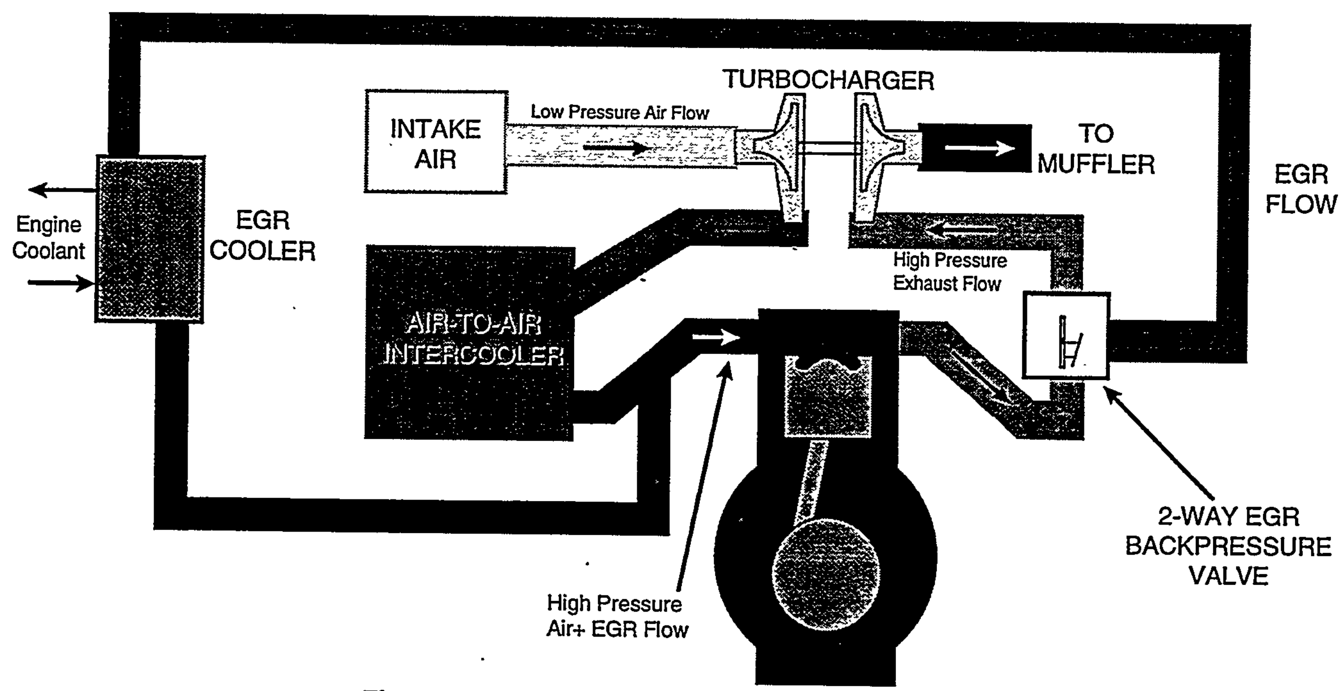


Figure 9. High Pressure Loop (HPL) EGR System

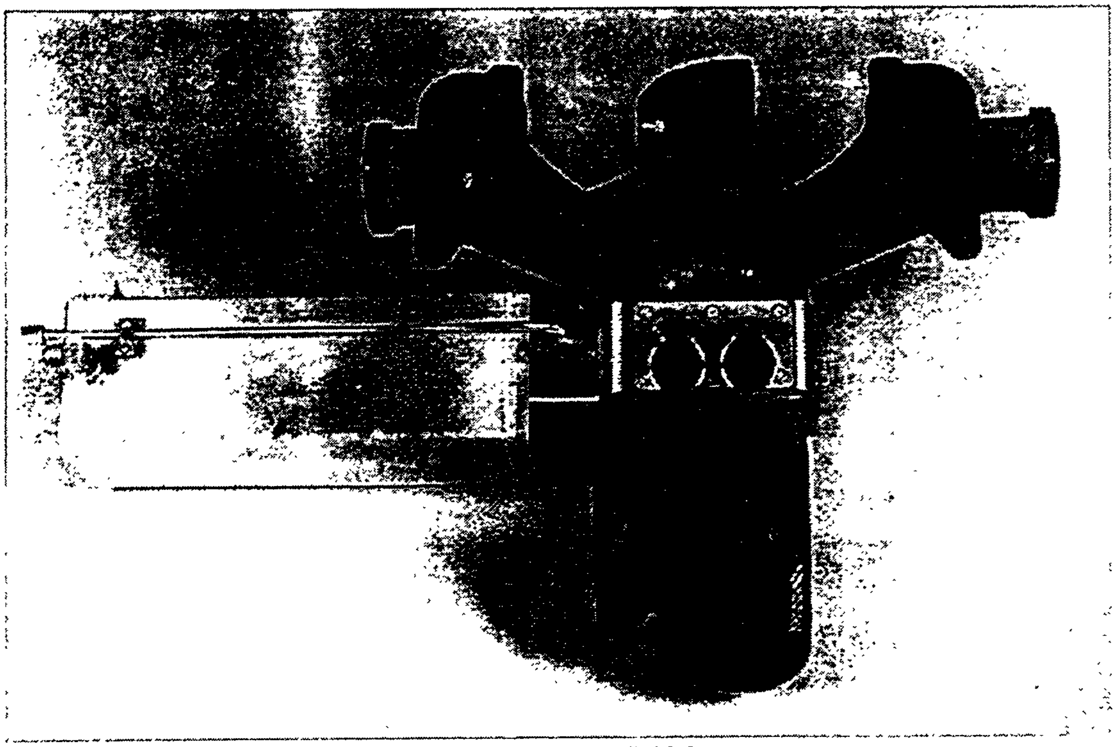


Figure 10. HPL EGR Valve

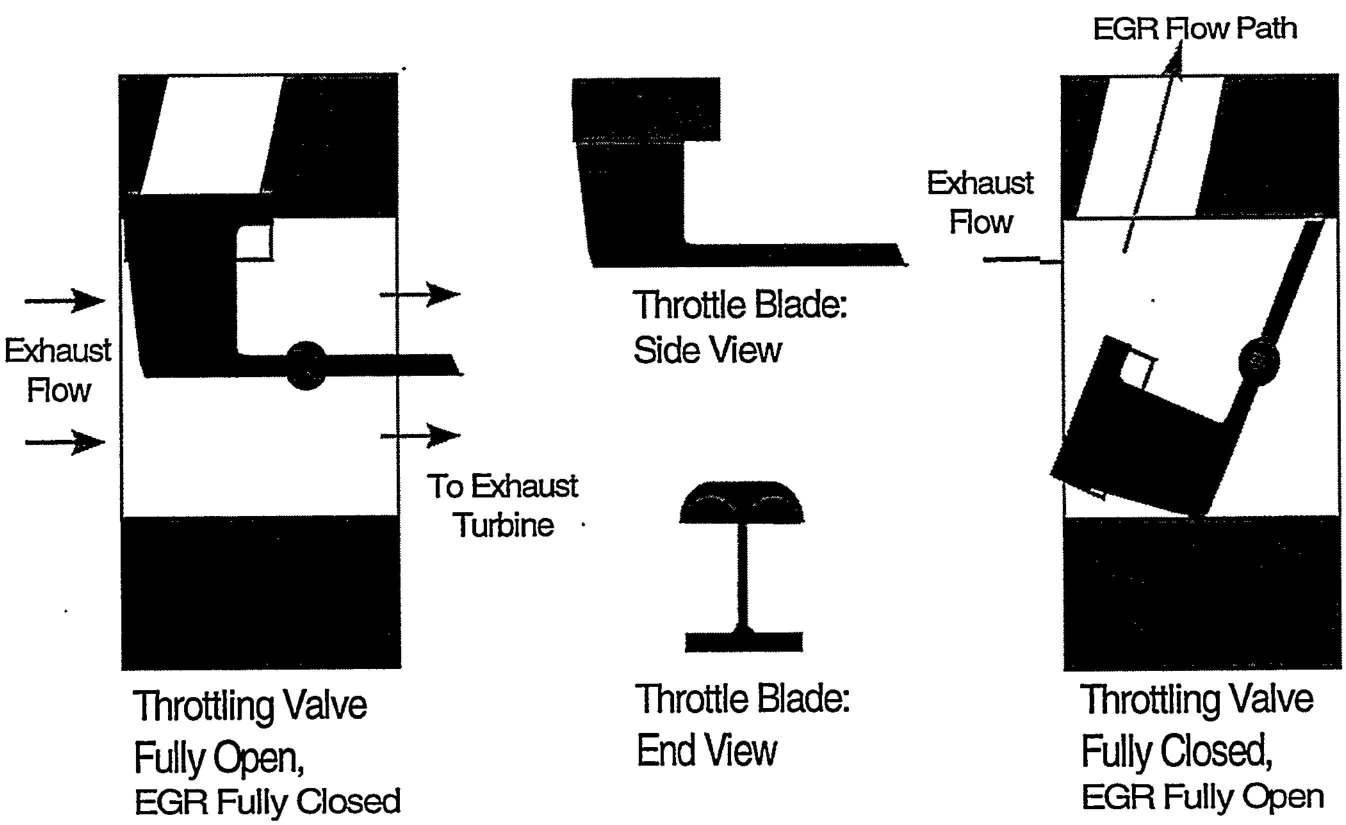


Figure 11. HPL EGR Valve

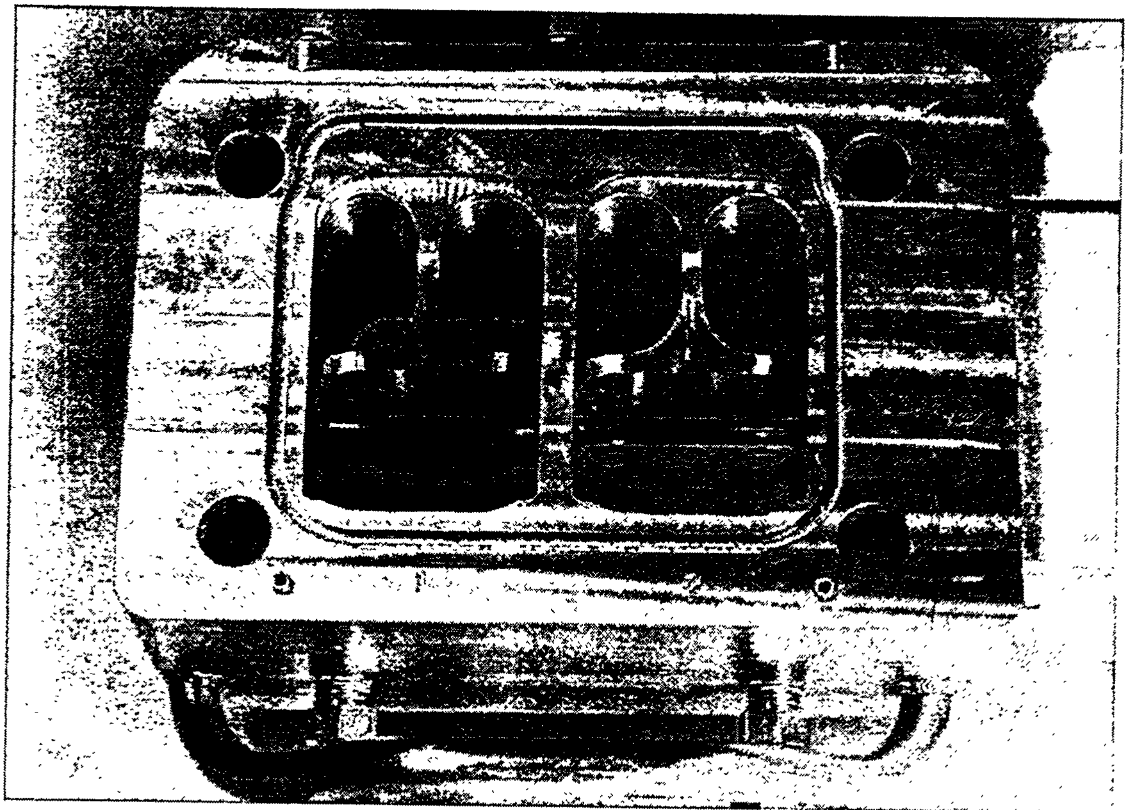


Figure 12. HPL EGR Valve

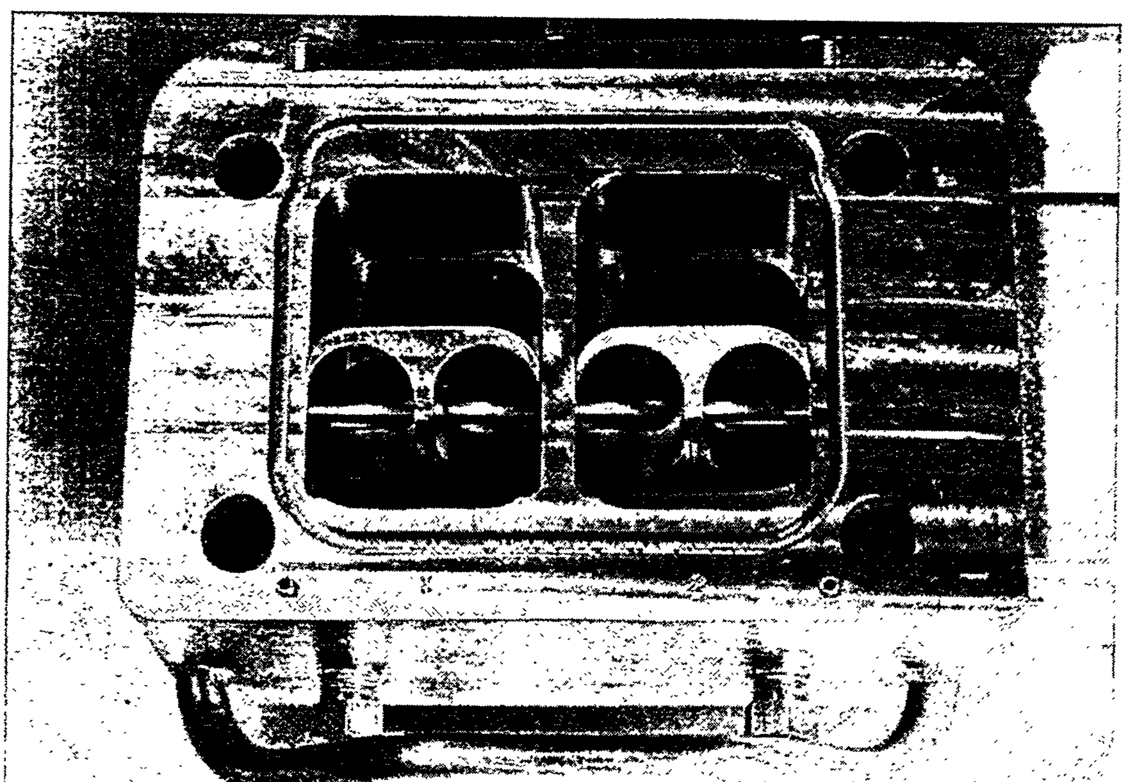


Figure 13. HPL EGR Valve

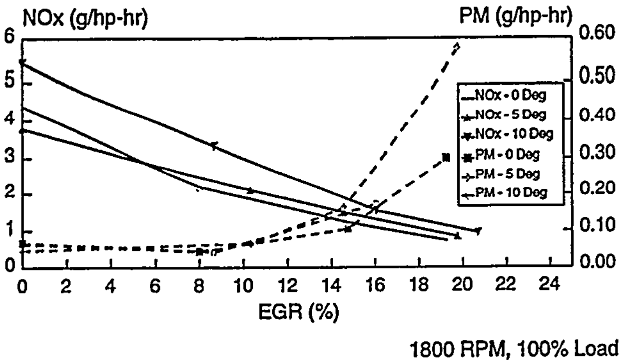


Figure 14. NO_x & PM vs. EGR Rate

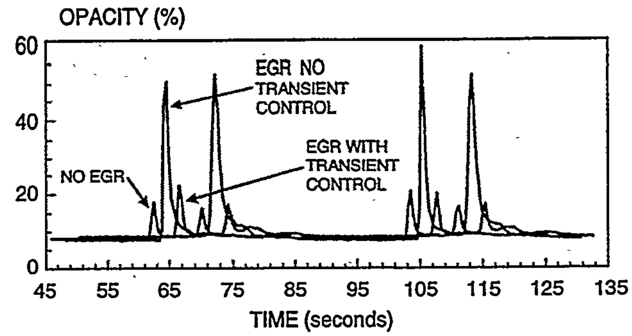


Figure 17. Effect of Step Transient on Opacity

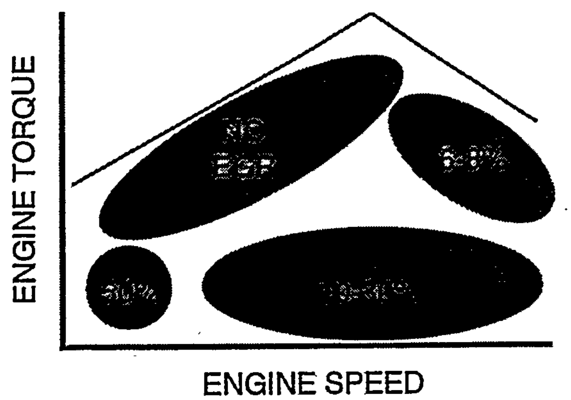


Figure 15. EGR Strategy to Optimize NO_x-PM Tradeoff for U.S. FTP

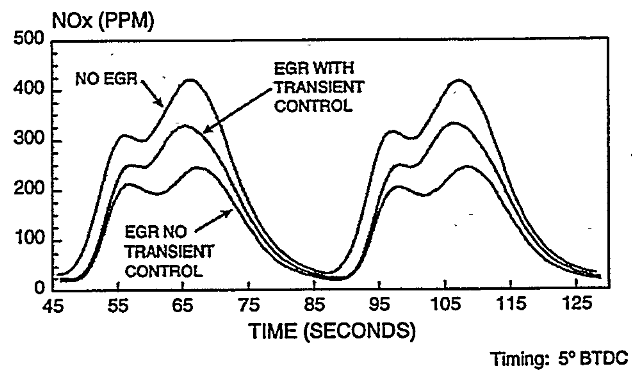


Figure 18. Effect of Step Transient on NO_x

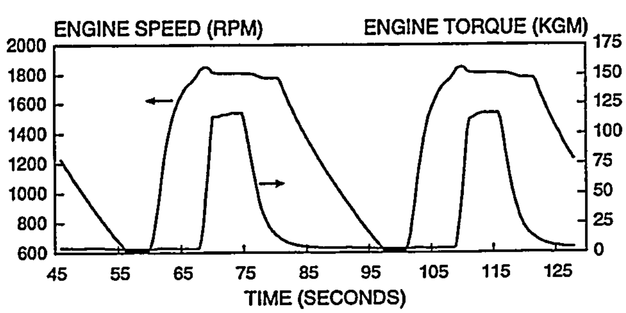


Figure 16. Engine Speed and Load for Step Transient

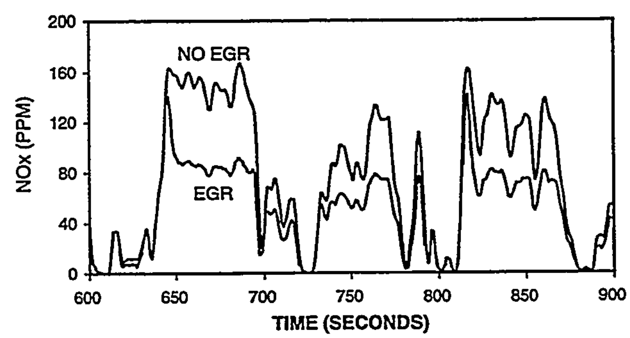


Figure 19. U.S. FTP Transient NO_x

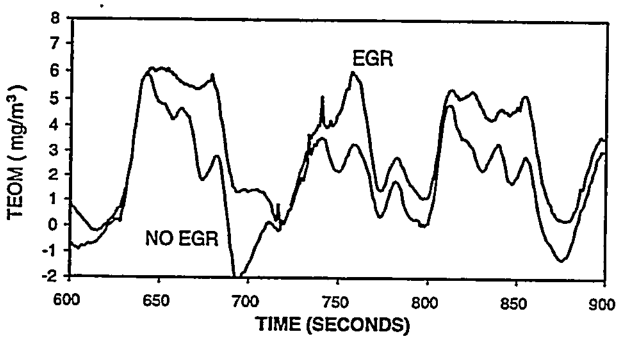


Figure 20. U.S. FTP Transient Particulates

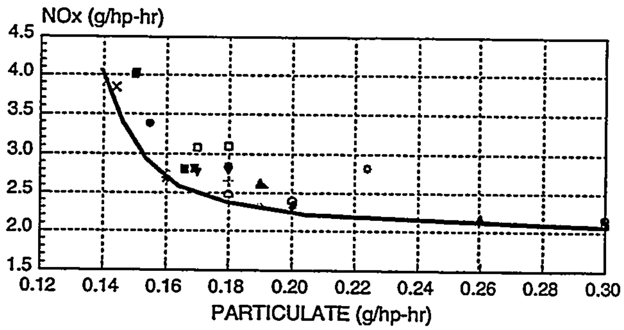


Figure 21. Effect of EGR Strategies U.S. FTP Transient Test Results

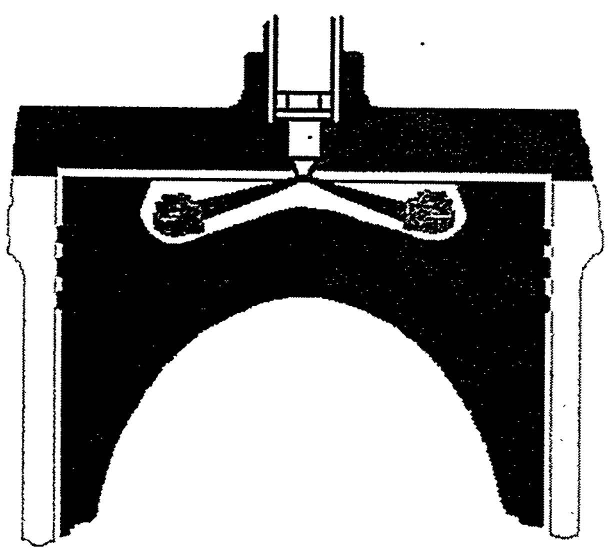


Figure 22. Diesel Engine Combustion

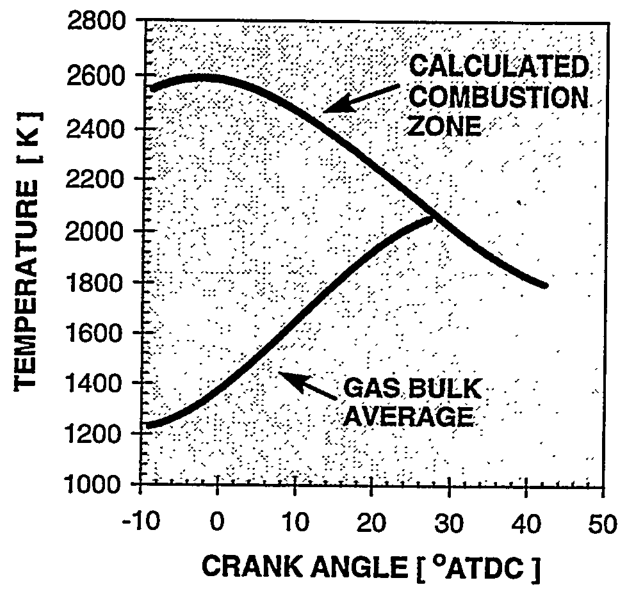


Figure 23. NO_x Formation (Diesel Combustion)

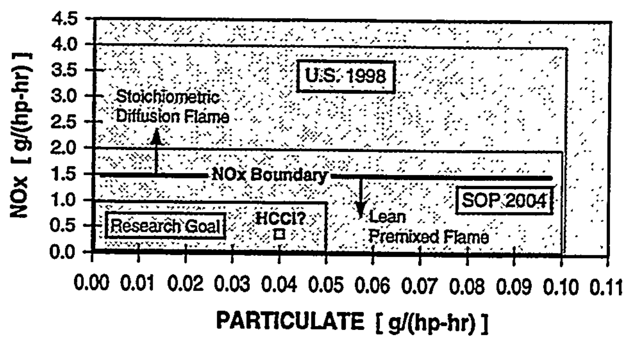


Figure 24. NO_x Boundary

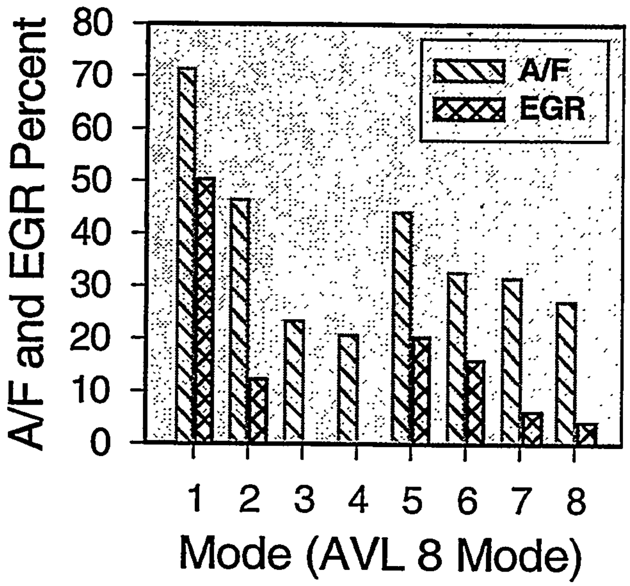


Figure 25. Measured EGR, Air/Fuel, and No_x
(Measured NO_x = 2.73 g/hp-hr and
Modeled NO_x =2.88 g/hp-hr)

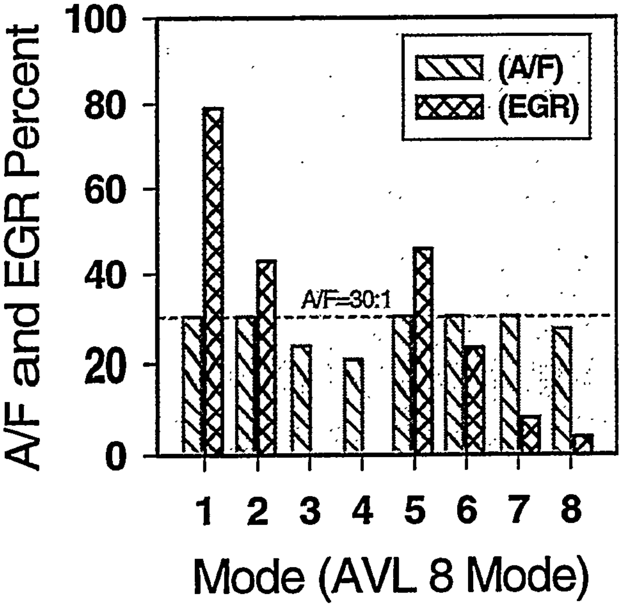


Figure 26. Modeled NO_x (NO_x= 2.3 g/hp-hr)

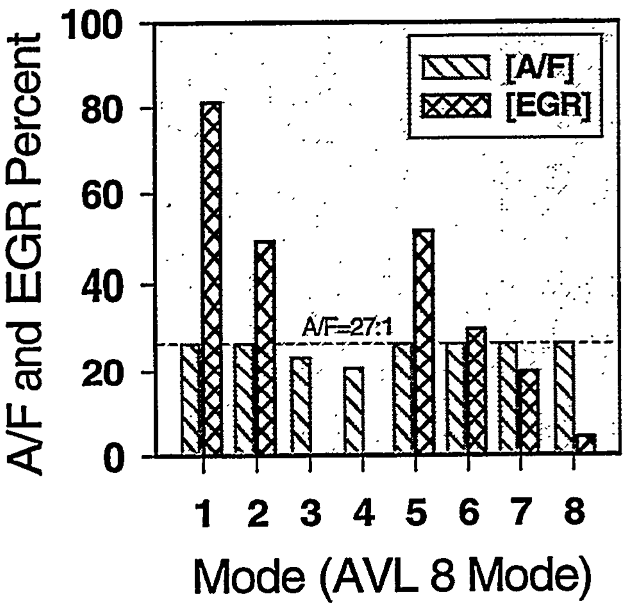


Figure 27. Modeled NO_x (NO_x = 1.9 g/hp-hr)

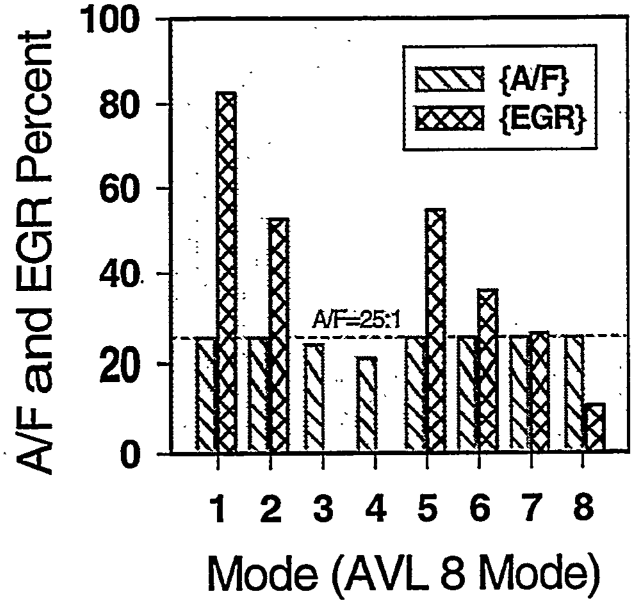


Figure 28. Modeled NO_x (NO_x= 1.4 g/hp-hr)

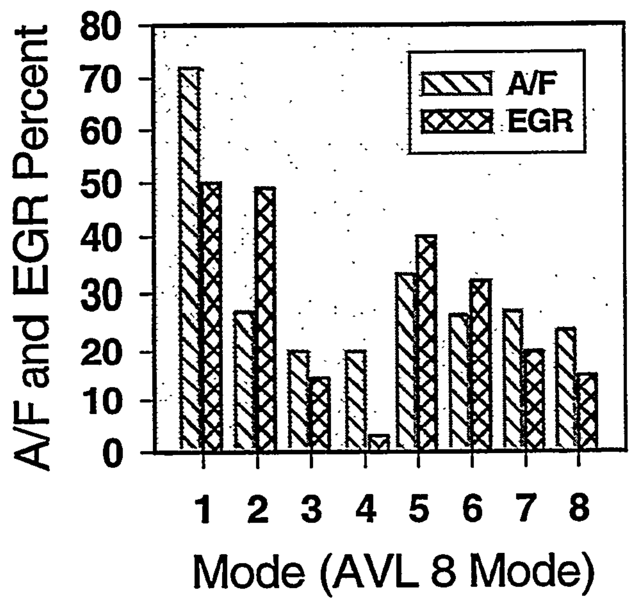


Figure 29. Modeled NO_x (NO=1.2g/hp-hr)

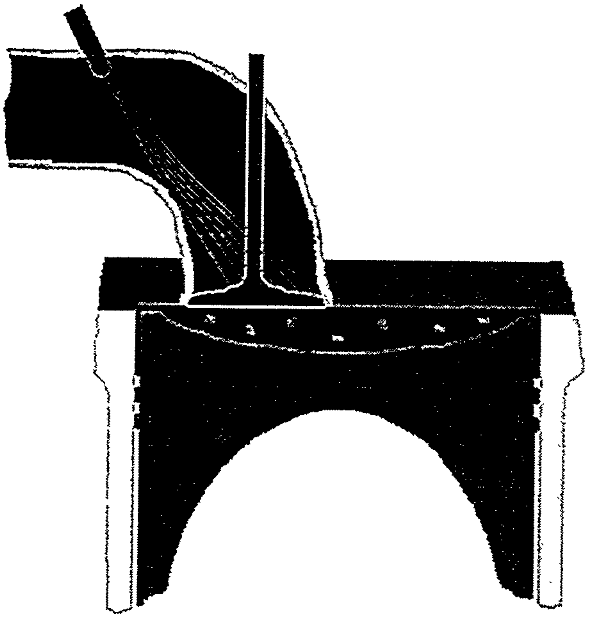


Figure 30. Homogeneous Charge Compression Ignition (HCCI) Combustion

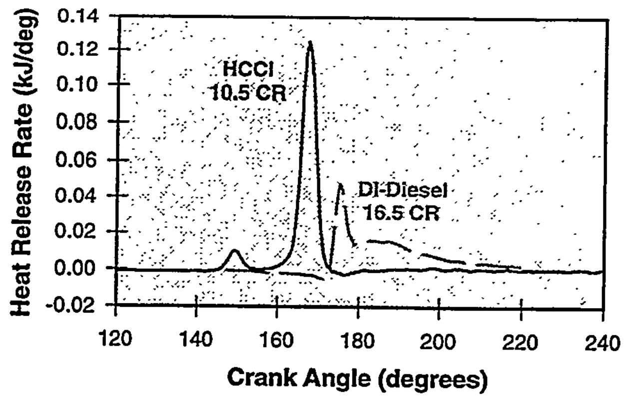


Figure 32. DI-Diesel vs. HCCI Heat Release Rate Comparison

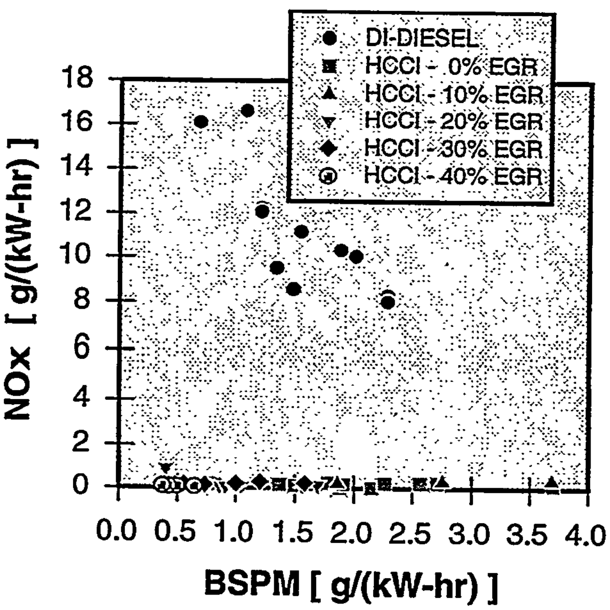


Figure 31. Emissions Comparison: DI-Diesel vs HCCI

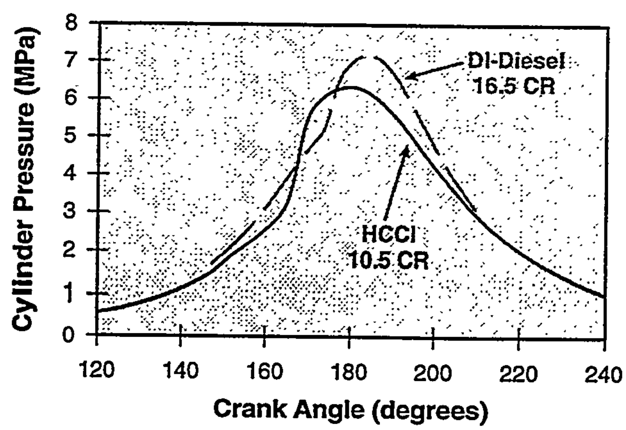


Figure 33. Cylinder Pressure Comparison