DIESEL ENGINES: ONE OPTION TO POWER FUTURE PERSONAL TRANSPORTATION VEHICLES

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INTRODUCTION

As we stand at the threshold of a new millennium, we wonder about the shape of personal transportation beyond 2000 AD. To start, we consider the societal, environmental, economic, and technological forces that are constantly applying pressure for change. These forces are not numerous, but they are powerful: the explosion in world population and its ravenous appetite for improved living standards are straining the earth's natural resources, its urban and agricultural space, and its environmental health. Translated into terms germane to automotive transportation, the most important forces are urban congestion, air pollution, global warming, and petroleum depletion. Responses to these driving forces can be considered in terms of scenarios [1], an artifice that distances the engineer from prediction while allowing contemplation of a wide range of alternatives without embarrassment. Such an analysis has led us to the conclusion "that the ultimate goals for personal transportation would be zero vehicle emissions in polluted areas and zero fossil fuel consumption, if the most dire predictions of climatologists and resource experts materialize" [2].

To move from the present toward these ultimate goals, a number of pathways involving various technologies exist. A useful schematic representation of this journey from the present into the future is shown in Figure 1, where various technologies are positioned with respect to their effects on fossil fuel consumption and vehicle emissions (in terms of U.S. and California standards). A third, and very important dimension, cost, is not shown, but generally, cost would increase as we move from the technologies of the present to those of the future. Note also, that as a benchmark, Figure 1 shows the position of the Honda Civic equipped with a VTEC engine.

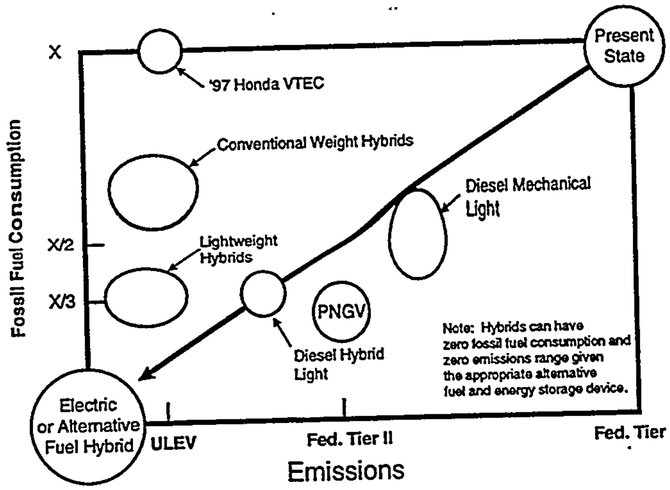

Figure 1. Options for Approaching Fuel Consumption and Emissions Goals.

Figure 1 is not a prediction. Rather, it helps organize the technological direction in personal transportation and it illustrates some important points: there are no silver bullets; many options and their combinations can be used to construct alternative pathways to the future; all aspects of transportation vehicles must be addressed (mass, resistance to motion, powertrains, fuels, and exhaust aftertreatment where applicable). Furthermore, Figure 1 illustrates that despite many years of research and the big push for development that resulted from the two petroleum crises of the 70's, the auto industry is still dealing with an undiminished range of options for meeting future challenges, options that span a broad spectrum of technological and commercial maturity.

Gasoline engines, diesel engines, fuel cells, electric vehicles, hybrid vehicles, and an assortment of alternative fuels still populate both the imagination and continuing research and development. These options are actively pursued by automakers worldwide, as well as by governmental and academic laboratories. Over the years, more and more of the technologies are being tried out in demonstration or limited market situations. Witness the General Motors electric vehicle EV-1, the PNGV program in the U.S., the BMW hydrogen-fueled car, and the Honda CNG Civic, as a few examples. No clear choice exists as yet, and even the diesel engine, which some had written off years ago, remains as one of the options.

Figure 1 also shows that diesel engines, either as stand-alone vehicle powerplants or as auxiliary power units in hybrid vehicles, could be viable choices for meeting future needs, depending on the requirements that must be satisfied and on progress in diesel technology compared to progress in the other technologies depicted in the figure.

Figure 1 suggests that some of the other options depicted are much preferable to the diesel engine. Such a conclusion, however, ignores two additional important dimensions not included in this two-dimensional figure: demonstrated technological performance and cost. In theory, more efficient powerplants than the diesel engine exist, but no powerplant has the demonstrated high efficiency of the diesel engine. The diesel engine's cost is higher than that of a comparable gasoline engine, but this cost is known and it is likely to be considerably smaller than that of some of the more exotic solutions suggested in Figure 1.

The Achilles' heel of diesel engines is exhaust emissions. If engine-out emissions could be reduced and if exhaust aftertreatment devices could be developed to further reduce emissions, the position of the diesel engine options in Figure 1 would be improved dramatically. Efforts to do both are under way in industrial, academic, and governmental laboratories worldwide. This paper will describe recent efforts in our laboratory, which have concentrated primarily on reducing engine-out emissions without sacrificing efficiency while using conventional diesel fuel. On our way to describing this work, however, we will also touch upon a number of related subjects. We will start by placing the diesel engine in perspective with respect to other powerplants, so as to add some detail to the broad outline already presented and summarized in Figure 1. This broadly based material will be expanded with detailed accounts of the following: the efficiency and emissions of state-of-the-art diesel engines; the research that we have done to advance the state of the art; and the technologies that, based on our work and that of others, could improve diesel engines. We will illustrate some of these technologies using examples taken from the new General Motors ECOTEC diesel engine. The topics of alternative fuels, exhaust aftertreatment devices, and application of diesels to hybrids will also be covered, but in a more general way, as they have not been our prime research areas in the recent past. For extensive research on some of these topics we rely more on the appropriate supplier industry, academia, and government. Furthermore, as our concern is personal transportation, we will limit the scope of this paper to light-duty vehicles.

We are not prepared to declare the diesel engine either the winner of the race to the future or a last-place finisher. We believe it to be one of the still open options, and we are intrigued by its technological challenges. Consequently, it is but one of several technologies that we are actively pursuing within our R&D portfolio.

THE DIESEL IN PERSPECTIVE

In North America, the contact of the public with diesel engines is largely through trucks, buses, and locomotives, many of which are older designs. Thus, the image of diesel engines in the minds of much of this public, and perhaps the public of other parts of the world as well, is of engines that are slow to start, slowly accelerating, noisy, and dirty. In the 1970's, two "energy crises" led to the introduction of new diesel powered passenger cars into the North American market and to a surge in their sales. Following a return to pre-energy crises levels of fuel availability and prices, diesel passenger car sales fell again to very low levels. In Europe on the other hand, where public policy has kept fuel prices high through taxes, diesels have historically captured a more substantial market share. In some countries, favorable diesel-to-gasoline price ratios have further contributed to high diesel sales. Motivated by the steadier market, auto manufacturers and engine consulting firms have worked continuously on the shortcomings of diesel engines and have made tremendous progress in correcting them. As a result of these efforts, contemporary European diesel passenger cars are difficult to distinguish from those with gasoline engines. These latest European diesel cars start nearly as rapidly, have similar noise characteristics, emit no visible exhaust smoke or perceptible exhaust odor, and provide equal or better acceleration, especially in traffic, compared to their gasoline-engined brethren. The effect of these improvements on the market has been a dramatic increase in diesel market share for new vehicles in the European Union (EU) from about 14% in 1990 to about 27% in 1996.

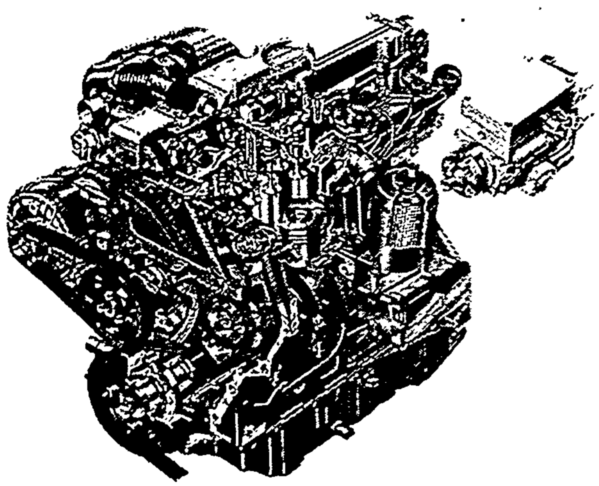

Figure 2. ECOTEC 2.0-L DI Diesel Engine [3].

Recent advanced engine development in Europe has focused on high-speed direct-injection (HSDI) configurations. For example, General Motors Europe introduced for the 1997 model year [3] a new four-valve per-cylinder HSDI diesel engine of 2.0-L displacement called the ECOTEC DI (see Figure 2). This engine is the first to reach the market of an expected wave of new HSDI diesel engines that combine a number of attributes: four valves per cylinder, high-pressure electronically controlled fuel injection, and high levels of EGR (exhaust gas recirculation). The motivation for choosing these attributes is that they are enablers for minimizing exhaust emissions from HSDI diesel engines. These and other enablers will be discussed in more detail later.

To place the diesel in perspective, we must compare it to other available options or competitive technologies.

COMPETITIVE TECHNOLOGIES

The baseline engine that new engine technologies are compared to is the SI (spark ignition) piston engine. Its most common form is port fuel injected, homogeneous stoichiometric charge with modest EGR, and a three-way-catalyst exhaust-aftertreatment system. While this engine has been the "standard" for several years, it is being continually improved in all its aspects including efficiency and emissions.

For all engines, minimizing heat transfer losses helps to improve engine efficiency. Changes that lower the combustion chamber temperature or minimize the surface area exposed to hot gases will reduce the fraction of fuel energy lost as heat. However, attempts to simply insulate engines have not been successful. Various characteristics of the engine cycle also influence efficiency, and engine modifications that improve the cycle efficiency are used (e.g. increased compression ratio). Anything that influences the efficiency of the combustion process, either in the fraction of fuel burned usefully or in optimizing the times and phasings of the combustion event, is a target for improvement. Minimizing losses due to friction is also sought during the evolution of the standard PFI (port fuel injection) engine.

Efficiency also depends on how the engine is operated. The baseline engine is less efficient at converting fuel into mechanical energy at light load than it is at heavy load. Load is controlled by throttling the flow of inlet air, and energy is dissipated to pump the inlet air across this pressure drop. Anything that increases the average loading (increased BMEP) during a typical driving cycle (the FTP cycles are of prime concern) will increase the fuel economy of the vehicle. This leads to strategies involving engine downsizing, perhaps accompanied by boost devices, cylinder deactivation, variable displacement, etc. The alternative to changing the operating range is to broaden the efficiency islands (i.e., contour islands of constant efficiency when plotted on a load-

TABLE 1. Qualitative Comparison of Light Duty Vehicle Engines

	Fuel Economy	Emissions	Cost	Comments
SI Piston Engine				
Baseline PFI engine	0	0	0	Homogeneous stoichiometric, modest EGR, 3-way catalyst
Variable Valve Timing (VVT)				Many mechanizations possible
Performance enhancing	+	+	-	Broader torque curve, more power, internal EGR
Load control	++	0	--	Elimination of throttle and consequent losses
Cylinder deactivation	++	-	--	Operates on fewer cylinders to increase load
Enhanced Dilution PFI				Basically homogeneous operation, improved combustion stability
EGR	+	+	0	Uses three-way catalyst
Lean	+	-	-	Probably needs lean NOx catalyst
EGR and Lean	+	0	0	Mixture of above two strategies
Direct Injection Stratified	+++	-	--	Dilution enhancement obtained with stratified operation. Operate with both EGR and air (lean) dilution. Will need lean NOx catalyst.
Downsized boosted				With intercooler
Turbocharged	++	0	---	Mostly used today as a power enhancement.
Supercharged	++	0	---	Probably will evolve towards Miller cycle.
Modified Cycles				
Overexpansion	++	0	---,U	Needs a viable mechanism. (Except for Miller cycle as above)
2-stroke cycle	+	-	0	May end up in niche markets.
SI Engine With Different Mechanization	0	0	U	Examples are Wankel and swash plate designs. Gains must be in areas of more compact/lightweight design, cheaper, and less friction. Surface to volume ratio influences combustion performance. Likely requires major tooling investment.
DI Diesel Engine	+35%	-	---	Turbocharged and intercooled with lean NOx aftertreatment. The only engine with demonstrated large fuel economy gains.
Stirling Engine	U	+	---,U	Not a likely near term contender; considerable demonstration/ development work needed. Most suitable for hybrids.
Gas Turbine	U	+	---,U	Not a likely near term contender; considerable demonstration/development work needed. Most suitable for hybrids. Low cost ceramics probably needed to make it viable.

+ = better, 0 = baseline, - = worse; U = unknown (speculative). The number of +'s and -'s indicates a rough measure of the advantage or disadvantage respectively.

speed graph) shifting larger portions of a driving cycle to high efficiency locations on the engine load-speed map. These changes typically involve operation with greater charge dilution using air, EGR, or both.

Table 1 lists some engine types/features that will provide competition to the **DI Diesel Engine** in delivering value to the customer. The taxonomy must always be arbitrary, and the listing is not exhaustive, but the table provides an overview of the engine types that many people believe (or wish) will be available in the near future. Many variations are shown for the baseline **SI Piston Engine** because improved versions of the baseline engine will likely be the most viable light duty vehicle engines for many years to come. Many of the SI Piston Engine features act to control losses from throttling and heat transfer. Hence, one cannot view the list as a "shopping cart" and simply sum up the features.

The fuel economy potentials of the **Stirling Engine** and **Gas Turbine** options are listed in the table as "Unknown" even though there are many numbers available from developers of these engines. As there are no demonstrated, viable, automotive Stirling engines or gas turbines, their efficiency gains are highly speculative. Peak thermodynamic efficiencies of a refined Stirling engine could approach 50%, and those for a high temperature (i.e., ceramic) gas turbine could be in the range of 40%. The thermodynamic efficiency for converting fuel into mechanical energy of a good, contemporary baseline SI engine is approximately 35% at its best operating condition. This compares to approximately 42% for a diesel engine. Obviously, operation away from the peak efficiency point degrades efficiency, and average efficiencies over a driving cycle are less than the maximum efficiency. In hybrid vehicle operation, the power range over which the engine oper-ates is reduced, but it seems very unlikely that any hybrid will operate only at the engine's peak efficiency. Efficiencies and operating characteristics of all components of the hybrid vehicle system will dictate the optimum strategy for engine operation and for overall vehicle efficiency.

Fuel cells have not been added to this chart as information about them would be even more speculative than that of gas turbines or Stirling engines. However, fuel cells are clearly a long-time-frame contender, as they offer great potential for both fuel efficiency and emissions. Fuel cells need breakthroughs in both technol.-go capability and costs in order to achieve commercial success. They would be incorporated into an electric hybrid drive train.

Table 1 shows that there are no clear winners among these engines and that the diesel faces several challengers.

DIESEL ATTRIBUTES AND CHALLENGES

The major challenges for continued growth of diesel engines in the market, as already noted, are exhaust emissions and cost. The exhaust emissions challenge derives from current and future exhaust emissions standards shown in Table 2.

TABLE 2. Future Diesel Light Duty Vehicle Emissions Standards

US	Year	Emissions - g/mi NOx	CO	NMHC or NMOG	PM
Federal Tier I	Current	0.4	3.4	0.25	0.08
Federal Tier II	2004	0.2	1.7	0.125	0.08
California* LEV	--	0.2	3.4	0.075	0.08
California* ULEV	--	0.2	1.7	0.04	0.04
EUROPE		**Emissions - g/km NOx**	**CO**	**HC+ NOx**	**PM**
Stage III	2000	0.5	0.64	0.56	0.05
Stage IV	2005	0.25	0.5	0.30	0.025

*California standards are for the fleet average and they are phased in over a period of 10 years (1993-2003).

For light-duty vehicles, compliance with expecked U.S. Federal standards (Federal Tier II) following expiration of the current diesel NOx emissions exemption of 1.0 g/mi will require intensive efforts. The most difficult to achieve standards are NOx and particulate matter (PM), which tend to trade off with each other. Compliance with HC (i.e., NMHC or NMOG) and CO emissions will be challenging as well, but less so than for NOx/PM. On the bright side, the long standing unregulated emissions issue of diesel exhaust odor has been virtually eliminated by the use of exhaust oxidation catalysts.

Other challenges for diesel engine developers are cost, vehicle mass increase, and both the characteristics and the quality of fuels. Diesel engines are more costly to produce than gasoline engines due primarily to the high-pressure electronically controlled fuel injection systems required. These systems can approach half the cost of the engine. Additional expenses also come from higher cylinder pressures and the need for intake boosting. Higher cylinder pressures require stronger and, therefore, more costly reciprocating parts (e.g., piston, connecting rod, bearings, crankshaft, etc.). Boosting, which is also required to provide acceptable performance, needs turbochargers (in most cases) and intercoolers (in some cases), both of which add cost. High output current and future diesels also require oil coolers. Vehicle costs also increase due to requirements for increased battery power (for cold cranking against high compression ratios).

The increased engine strength required to control high cylinder pressures without NVH (noise, vibration, and harshness) problems contributes to increased vehicle weight as do the boost and intercooler devices. Diesel fuel characteristics influence cold startability, combustion noise, and exhaust emissions as will be noted later.

Against these challenges, diesels offer the highest proven fuel economy of any engine, performance advantages (due to their torque characteristics), and increased vehicle driving range. From an environmental point of view, diesels offer CO_2 reductions proportional to their energy efficiency gains and minimal organic emissions from evaporative losses, running losses, and refueling. These issues will be addressed in more detail with focus on the HSDI diesel. To do this, however, we need to examine data that will help project the characteristics of future diesel engines. Such characteristics include performance, efficiency, and emissions.

PERFORMANCE AND EFFICIENCY OF FUTURE DIESELS

POWER DENSITY AND PERFORMANCE

The peak power density of diesel engines has been increasing steadily. A snapshot of data from a group of current and advanced development engines is shown in Figure 3. On the left is data from a group of IDI (indirect injection) and DI turbo diesels with power densities ranging from 35 to 50 kW/L. Also shown are spark ignition (SI) engine data from a two-valve engine in NA (naturally aspirated) and boosted form (33 to 44 kW/L, respectively), and a group of NA four-valve SI engines (42 to 46 kW/L). A high-performance four-valve gasoline SI engine is also shown with 72 kW/L. In spite of having the lowest peak power engine speeds, the diesel engines have comparable power density to the supercharged two-valve SI engine and only slightly lower values than the typical NA four-valve SI engines.

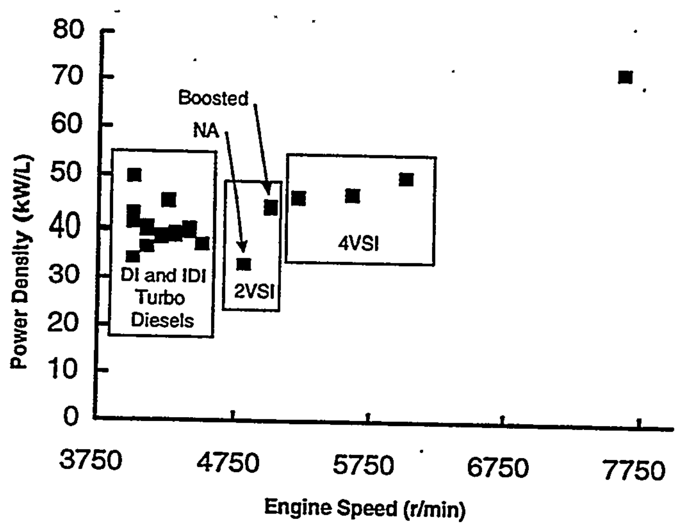

Figure 3. Peak Power Density for DI Diesel, IDI Diesel, and SI PFI Engines.

However, vehicle performance depends not only on peak power, but also on the peak torque and torque curve shape. The peak torque density values corresponding to the data of Figure 3 are shown in Figure 4, which shows that the diesels have the highest peak torque densities and their peak torque values occur at lower engine speeds than those for the SI engines. This characteristic results from the shape of the diesel torque-speed (or BMEP-speed) curve, as shown in Figure 5.

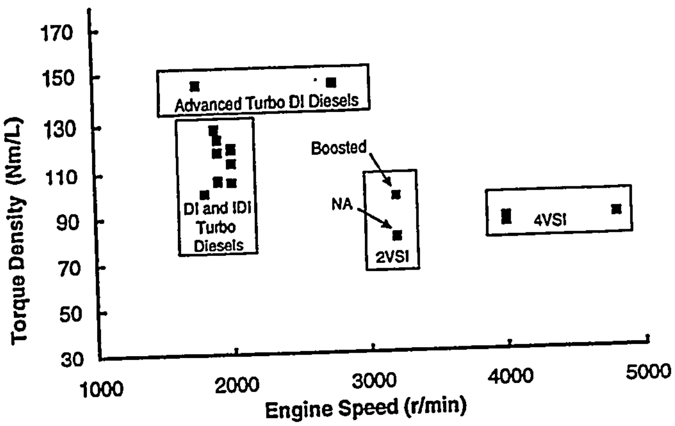

Figure 4. Peak Torque Density for DI Diesel, IDI Diesel and Spark Ignited Engines.

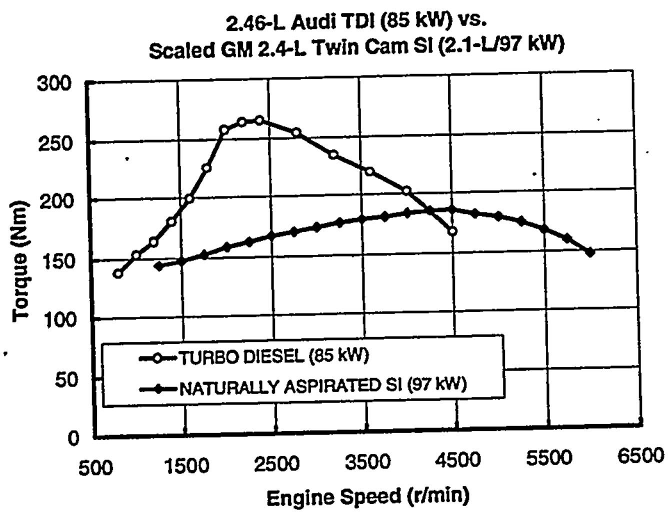

Figure 5. Torque Versus Eengine Speed for HSDI and Four Vlve P.F.I. SI.

The effect of these high torque values and low peak torque speeds is to provide vehicle accelerations equivalent to SI engine vehicles with higher peak power, as expected from the following equation, which provides a crude estimate of the relationship between maximum engine power and vehicle acceleration [4]

$$Peak\,Power\,(kW) = C \cdot \frac{Vehicle\,Mass\,(kg)}{Acceleration\;(0-60(s))} \qquad (1)$$

The coefficient C depends primarily on three things, the shape of the torque curve, the final drive ratio, and the transmission ratio range. Based on comparisons of Equation 1 with simulation calculations for a vehicle with a 2.1-L L-4 SI engine with a four-speed automatic transmis-sion, the coefficient value is 0.75. For a 2.46-L turbo HSDI diesel with the same transmission, the coefficient for the diesel is 0.64. Thus, because of the torque-curve shape and peak torque value of the diesel engine, the maximum power required is about 14% less than that for a SI engine for equal vehicle mass and equal 0-60 mph performance. In addition, because the diesel peak torque speed occurs at 1800-2500 r/min compared with 3500-5000 for the SI en-gine, diesel powered vehicles respond well to traffic speed variation, thus requiring fewer transmission shifts for equivalent vehicle performance in urban and suburban traffic.

TABLE 3. Comparison of VW (Passat) TDI Vehicles - Variable Nozzle Turbocharger (VNT) vs. Conventional Wastegate Turbocharger [5]

	Wastegated Turbo	VNT Turbo
Displacement	1896 cm^3	1896 cm^3
Nominal Power	66 kW	81 kW
Nominal Speed	4000 r/min	4150 r/min
Specific Power	34.8 kW/L	42.7 kW/L
Maximum Torque	202 Nm @ 1900 r/min	235 Nm @ 1900 r/min
Maximum BMEP	1.34 Mpa	1.56 MPa
Euromix* Fuel Consumption	5.4L/100 km	5.2L/100 km
0-100 km/h	14.4 s	12.1 s
Maximum Speed	173 km/h	188 km/h

*Euromix fuel consumption is the average of that for Euro city cycle, that for a steady 90 km/h and that for a steady 120 km/h, all equally weighted.

One enabler for high power density is a variable geometry turbocharger, which is now being used on production HSDI diesel vehicles. VW and Audi both offer versions of their TDI engines with conventional wastegate turbochargers and also versions with VNT's (variable nozzle turbochargers). Relative performance data for the VW 1.9 TDI engines is shown in Table 3 [5].

FUEL EFFICIENCY

It is clear from Table 3 that the VNT provides not only improved performance, but also reduced vehicle fuel consumption. As would be expected, the VNT is more costly than the conventional turbocharger. Nonetheless, it is likely that future vehicles will make greater use of this technology, particularly in premium vehicle packages.

From an efficiency point of view, HSDI diesels currently offer around 35% lower volumetric fuel consumption than equivalent performance SI engines as shown in Table 4 below.

Table 4. Fuel Consumption and Performance of Opel Vectra Vehicles with Gasoline and DI Diesel Engines.

Engine	Gasoline x16 XEL	Diesel x20 DTL
Top Speed (km/h)	175	178
Acceleration, sec. (0-100 km/h)	15.5	15.5
Fuel Cons. (L/100km)		
City	9.0	6.0
90 km/h	4.8	3.8
120 km/h	6.1	5.0
Euromix	6.6	4.9

From Table 4 it can be seen that an Opel Vectra vehicle with a gasoline engine consumes 35% more fuel than one with a 2.0-L low charge (i.e., non-intercooled) ECOTEC diesel on the Euromix test cycle for equivalent performance. On an energy basis, the consumption increase is about 23% for the gasoline vehicle with an equal increase in CO_2 emissions. These differences in consumption and CO_2 emissions are caused by the roughly 12% volumetric energy difference between diesel and gasoline fuels.

Fuel efficiency for future HSDI diesels may not improve significantly due to combustion modifications that will be required to meet future exhaust emissions standards. Nonetheless, as was noted earlier in comparing HSDI diesel technologies with competing technologies, HSDI diesel engines offer the highest fuel econ-omy alternative among piston engine candi-dates and the highest demonstrated fuel econ-omy among the other engine alternatives.

HYBRIDIZATION

Currently, a number of development efforts are being pursued jointly by the Big 3 U.S.' automakers through USCAR and the Federal Department of Energy to exploit the potential of hybrid drive trains [6]. Many options exist to configure hybrid power trains. These range from a power boost parallel hybrid with a small amount of hybridization at one end of the spectrum to a series hybrid at the other end. Various power sources can be used among these choices ranging from near-term piston engines, such as the HSDI diesel, to mid-term technologies, such as gas turbine and Stirling engines, to long-range technologies, such as fuel cells.

Since the focus of this paper is on the HSDI diesel engine as an option, it is worth considering what hybrid configurations make the most sense with HSDI diesel engines. The efficient operation over broad load and speed ranges, which is characteristic of the HSDI diesel, makes it well suited for parallel hybrid applications, although series configurations are also an option. One such parallel hybrid approach would involve a flywheel alternator-starter placed between the engine and transmission, which would facilitate packaging a parallel hybrid with relatively small modifications to current powertrain packaging layouts. The potential efficiency gains that might be achievable from an HSDI diesel hybrid would depend on the details of the powertrain configuration, operational strategy, and the efficiency of the system for energy storage and withdrawal [6].

EMISSIONS CHALLENGES AND OPPORTUNITIES

PROJECTIONS BASED ON VEHICLE DATA

Currently, HSDI diesel-powered passenger cars have been certified for emissions compliance in both the U.S. and Europe. However, as was noted earlier in Table 2, future emissions standards are tightening. The next tightening comes with the Euro Stage III standards in the year 2000, which require a 36% reduction in CO, a 38% reduction in NOx + H2, and a 50% reduction in PM from Stage II levels. In addition to compliance with the legislated exhaust constituents shown in Table 2, there is a strong interest in Europe for reducing CO_2 emissions to address global climate change concerns. The European Union (EU) has proposed an emissions target for the 2005/2010 time frame of 120 g/km CO_2- In addition, there is a German target involving tax incentives to encourage CO_2 emissions below 90 g/km. In recognition of the low fuel consumption of diesel engines and their corresponding low CO_2 emissions. European emissions targets in Stage III and Stage IV are less stringent for diesel engines than for gasoline engines. This approach reflects the European view that local pollution issues must be balanced with global issues like climate change and energy conservation.

Current production vehicles in Europe have some margin relative to Stage II compliance, which takes them toward Stage III compliance. For example, the ECOTEC diesel vehicle egisZions are compared to the standards in Table 5 [3].

TABLE 5. ECOTEC Diesel Vectra Emissions, g/km

	CO	NOx	HC+NOx	PM
ECOTEC Diesel	0.37		0.71	0.039
Stage II Stds	1.0	0.5	0.9	0.1
Stage III Stds	0.64	0.25	0.56	0.05

However, in spite of already achieving future CO and PM emissions levels with the ECOTEC engine, considerable additional effort is required to meet the H2+NO_x goals. Auto manufacturers, independent engine manufacturers, and engine consulting firms are working on roughly the same set of technologies to achieve NOx control. For example, vehicles with advanced prototype four-valve HSDI diesel engines in vehicles of different inertia weights have achieved emissions below the Stage III values as shown in Figure 6. In this case, the values for the 1400 kg vehicle (7] were achieved with a 2.2-L four-cylinder engine, while the results for 1590 kg and 1810 kg were achieved with a 3.0-L V6 [8]. Both of these engines were equipped with advanced high-pressure electronic radial distributor pump type injection systems with two-stage injection nozzles. The 2.2-L engine was also equipped with a VNT and a fresh passive (i.e., no extra fuel added to the exhaust stream) lean NOx catalyst.

The more challenging Euro Stage IV values proposed for NOx/PM in 2005 are also shown in Figure 6. The NOx/PM values are one half

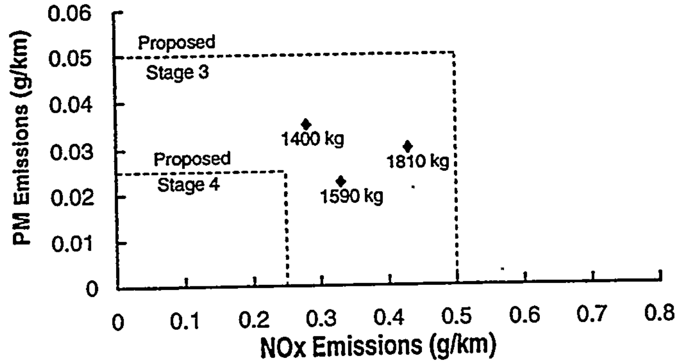

Figure 6. Development Status Toward Achiev ing Euro Stage III Emissions for 1400 kg [7], 1590 kg [81 and 1810 kg [8] Vehicles.

those for Stage III. The final values enacted will be based on progress demonstrated before the end of 1998. At the present time, some of the technology development for Stage IV tar-gets is being conducted on single-cylinder en-gines. For example, we have been trying to develop combustion system technologies to achieve the future emissions standards such as U.S. Tier 2 and ULEV as well as the Euro Stage IV values. Some of these efforts are described next.

SINGLE-CYLINDER ENGINE STUDIES AND PROJECTIONS OF FUTURE HSDI ENGINES

Single-cylinder engine studies are convenient tools in efforts to improve combustion systems. They yield the data on which projections of full-scale engine performance can be made, thereby providing the information required to decide on the extent of a developmental program. In this section, after describing our single-cylinder engine studies, we will use results of these studies to assess the future potential of HSDI engines.

Engine and Injection System Description

Small-bore, high speed HSDI diesel engine combustion was studied using a single-cylinder engine with a high-pressure common-rail fuel injection system.

Figure 7 shows the significant features of the combustion system. The cylinder head had four vertical valves actuated by dual overhead camshafts. The fuel injector was vertical and centrally located. The combustion chamber was formed by the re-entrant bowl in the piston and received the multiple sprays from the fuel injector. Swirl was promoted by the use of one tangential and one helical intake port. Cylinder heads having two different swirl levels were available (i.e., values of 1.1 and 1.7). Swirl was further increased for some cases by closing the inlet to either intake port. The cylinder liner and head were mounted on an AVL 528 single-cylinder research engine.

To meet power and emission requirements, turbocharging and/or supercharging to boost intake manifold pressure above atmospheric pressure is required for HSDI diesels at high load conditions. For the single-cylinder engine, such boosting was simulated by raising inlet and exhaust pressures to appropriate values.

Exhaust gas recirculation (EGR) was accomplished by drawing exhaust gas from the exhaust plenum and introducing it into the intake plenum through an external pipe, where it mixed with the inlet airflow regulated by a critical airflow system. Temperature of the inlet charge was regulated independently by heating the inlet air.

Cylinder pressure was measured and recorded using a conventional water-cooled piezoelectric pressure transducer surface mounted in the combustion chamber.

Table 6 contains an abbreviated list of key engine specifications:

An experimental high-pressure common-rail system consisting of an electronically controlled high-pressure fuel rail, continuous high pressure pump, and a rateshape controllable injector from Ganser-Hydromag of Zürich [9] was used. Fuel was spilled from the high pressure common-rail to obtain the desired rail

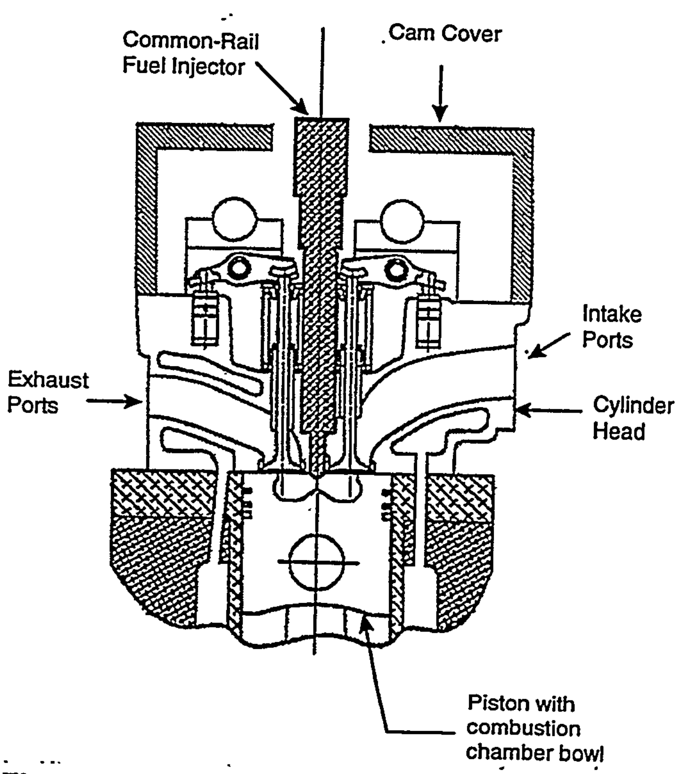

Figure 7. Single-Cylinder Engine Combustion System.

TABLE 6. Single-Cylinder Engine Specifications

Bore	76	mm
Stroke	94	mm
Compression Ratio	18.7	--
Squish Distance	0.7	mm
Bowl Depth	11	mm
Bowl Entrance Diameter	40	mm
Maximum Bowl Diameter	45	mm
Diameter of Intake Valves	24	mm
Diameter of Exhaust Valves	22	mm
Swirl Ratios	1.1, 1.7	--

The engine was run on Amoco Premium low sulfur diesel fuel with key properties listed in Table 7.

TABLE 7. Diesel Fuel Properties for Single-Cylinder Experiments

Specific gravity	0.83	-
Hydrogen/Carbon ratio	1.93	-
Stoichiometric mass A/F	14.6	-
Lower Heating Value	42.8	MJ/kg
Cetane Number	53	-
Aromatics	22	%
T95	321	C
Sulfur	<0.05	% wt

pressure. That pressure was limited to a maximum of 120.0 MPa (1200 bar) with the current hardware. The injector was actuated with single or multiple control pulses, the latter allowing pilot or post injection to be investigated. A needle lift sensor gave information on actual opening and closing timings.

A multi-hole valve-covered-orifice (VCO) spray tip was used. For the tests reported, nozzle tips were used having five spray holes ranging in diameter from 0.16 to 0.19 mm. Spray angles of 135° to 160° were used. Protrusion of the injector tip into the chamber was controlled with shims at the seat and was set to give maxi-mum smoke-limited power.

Fuel injection rate was controlled primarily by selection of rail pressure, as injection rate varies in proportion to rail pressure. The injector also has an internal adjustment allowing different slopes of needle lift opening that additionally influence the overall rate of injection for a given control pulse and rail pressure. Needle closing rate, however, is independent of rail pressure or opening rate settings. Closing occurs rapidly, but without needle bounce.

More information on the engine test procedures used can be found in the Appendix.

Combustion System Parameter Effects

Examples of tests exploring some of the important parameters controlling combustion and emissions are discussed in this section.

Rail Pressure and Pilot Injection

Rail pressure was varied from 500 to 1000 bar and tests were run with and without pilot injection. Engine speed was 1200 r/min, IMEP 575 kPa, and injection pulse timing was set to 3.50 btdc. The pilot pulse of 0.4ms was at 7.5° btdc, introducing about 10% of the fuel in advance of main injection. These tests were run without EGR, and naturally aspirated (NA) inlet conditions were maintained.

Rail pressure and pilot injection each affect average injection rate -- the injected fuel quantity divided by injection duration as observed from needle lift measurements. As one would imagine, injection rate increases with increasing rail pressure and decreases with pilot injection as noted in Figure 8.

In the following discussion, exhaust emission results are reported as emissions index, which is defined as ratio of the mass of emissions in grams to the engine fuel mass in kilograms. Smoke emissions are reported as smoke number, which is proportional to the concentration of carbon particles in the exhaust. A smoke number of two is considered the visibility limit.

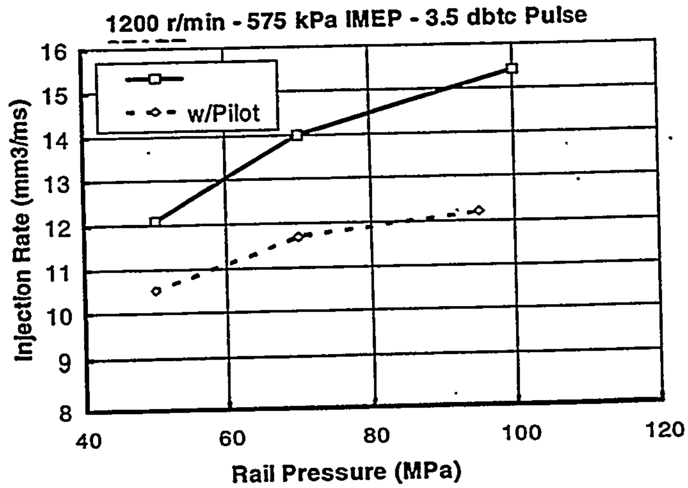

Figure 8. Effects of Rail Pessure and Pilot on Injection Rate.

NOx emission index (EI) increases with increasing rail pressure; and slightly higher values were reached with pilot injection (Figure 9). NOx typically will increase with faster burning rates and/or more advanced combustion timings. Pilot injection reduces injection rate by extending the time over which fuel is injected, which tends to lower NOx. For this example, however, the advanced timing of the pilot offset the rate effect, resulting in higher NOx.

Smoke emissions decreased with increasing rail pressure but levels were significantly higher with pilot injection (Figure 10). Increased injection rate, with its attendant increase in burning rate, is expected to reduce smoke emissions. The increase in smoke with pilot injection is typical and likely due to such details as shifting to more diffusion burning and reduced pre-mixed burning. Also, the main injected fuel will overtake that from pilot injection, so combustion

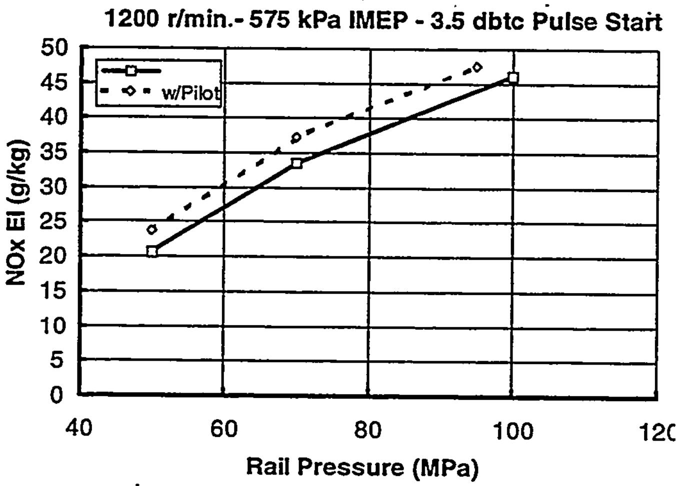

Figure 9. Effects of Rail Pressure and Pilot Injection on NOx Emissions.

could occur in areas already depleted of oxygen from combustion of the pilot.

The response of smoke to rail pressure noted in Figure 10 is a key feature of common-rail fuel injection that has great significance for both emissions and maximum torque at low engine speed operation, since in principle, high injection pressures can be achieved independent of engine speed. However, these advantages must be weighed against the disadvantages of common-rail systems. This topic will be discussed in more detail later under Enabling Technologies, Fuel Injection.

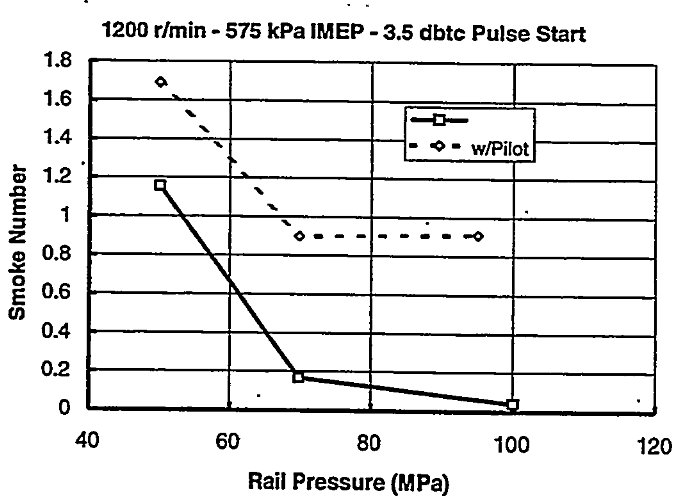

Figure 10. Effects of Rail Pressure and Pilot Injection on Smoke Emissions.

The high injection rates at the high rail pressures increased the maximum burn rate. Figure 11, which increased the rates of cylinder pressure rise (Figure 12). Pilot injection is shown to significantly reduce maximum burn rate and pressure rise rate since less fuel is injected prior to ignition, limiting the amount of rapid, pre-mixed burning. Because maximum cylinder pressure rise is a primary indicator of combustion generated noise, techniques such as pilot injection are desirable for reducing combustion noise.

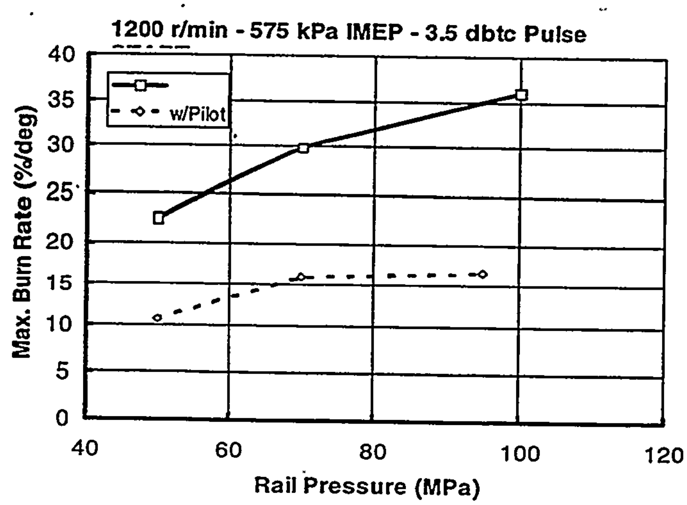

Figure 11. Effects of Rail Pressure and Pilot Injection on Maximum Burn Rate.

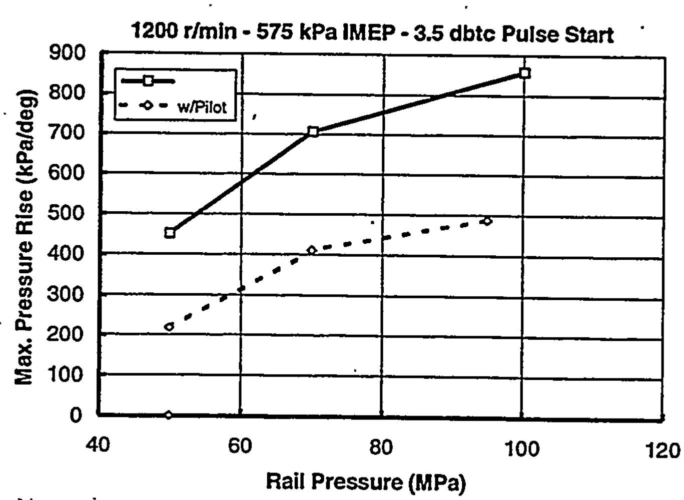

Figure 12. Effects of Rail Pressure and Pilot Injection on Maximum Rate of Pressure Rise.

EGR and Injection Timing

At a rail pressure of 700 bar and with pilot injection, EGR was increased to near the maximum amount for stable combustion. Injection timing was then progressively retarded at the high EGR condition by delaying the pilot and main pulse timings. As EGR increases, Figure 13, NOx decreases substantially, while smoke increases. However, a small amount of injection retard (4.50) lowered smoke to the no-EGR levels, with little effect on NOx. The effect of SOI (start of injection) timing on smoke is believed due to subtle changes in the details of the spray and bowl interactions to which smoke can show high sensitivity, such as in this example. Supporting evidence for this supposition was found when a small additional injection timing retard of only 0.50 caused a five-fold increase in smoke compared the level at 4.5° of retard.

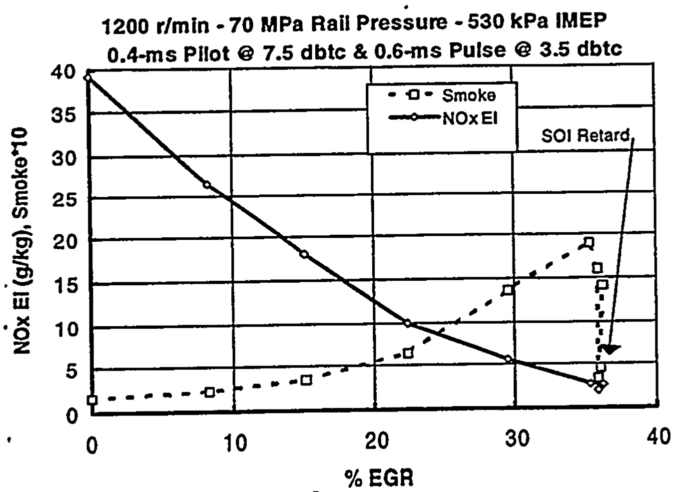

Figure 13. Variation of NOx and Smoke Emissions with Exhaust Gas Recirculation.

The air-fuel ratio for this particular test is shown in Figure 14. As EGR increases, air-fuel ratio decreases since exhaust gas is simply displacing inlet air. For this example, the air-fuel ratio decreases

The amount of EGR that may be applied to decrease NOx is dictated by the magnitude of the air-fuel ratio with no EGR and the minimum acceptable air-fuel ratio with maximum EGR. At light loads, such as idle where air-fuel ratio may be in the order of 100:1, more EGR can be used than for a heavy load condition, where the air-fuel ratio without EGR may be on the order of 25:1. At heavy loads, however, boosting intake pressure, as with a turbocharger, increases air-fuel ratio, also allowing more EGR to be used. In most cases, the minimum acceptable air-fuel ratio at heavy loads will be determined by the allowable smoke limit. At light loads, combustion instability or an abrupt increase in gaseous emissions or particulates will be the likely limitations.

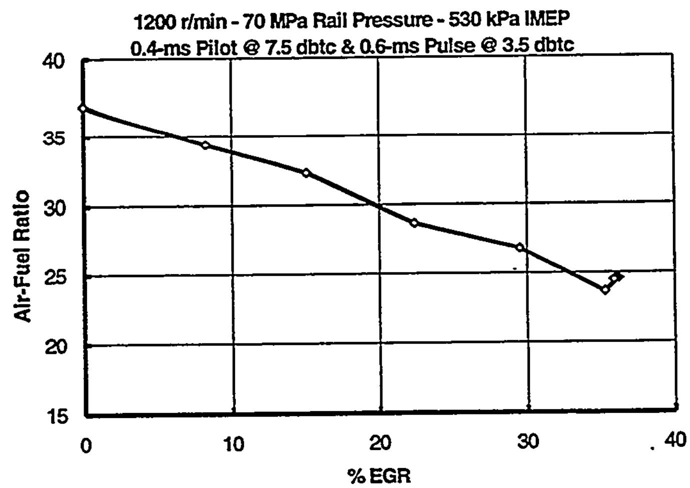

Figure 14. Variation of Air-Fuel-Ratio With Exhaust Gas Recirculation.

Port Throttling (Increased Swirl)

Inlet port throttling tests were run at 2400 r/min, 600 kPa IMEP, 700 bar rail pressure using pilot injection while closing either the throttle to the tangential port or to the helical port. Naturally aspirated inlet conditions were maintained and no EGR was used. With both port throttles open, the computed swirl ratio is 1.1 (determined by using experimental port-flow data and an engine simulation computer program). Closing the tangential port forces all the airflow through the helical port, resulting in a modest increase in swirl ratio to about 1.4. For this particular set of ports, closing the helical port and forcing all the airflow through the tangential port results in a larger calculated swirl ratio of 1.75. These tests were repeated with a high swirl head which had a base swirl ratio of 1.7, increasing to 2.2 and 2.8 as either the tangential or the helical port, respectively, was fully closed. Data also are included for the case without pilot injection for the high-swirl head.

As shown in Figure 15, the effect of swirl on smoke depended upon which port was throttled. At this particular operating condition, throttling the helical port increased smoke, whereas the opposite effect occurred with throttling the tangential port. The high swirl head produced lower levels of smoke as did the elimination of pilot injection. Obviously, the in-cylinder air motion affects smoke, but this effect cannot be correlated by a single parameter as simple as the swirl number.

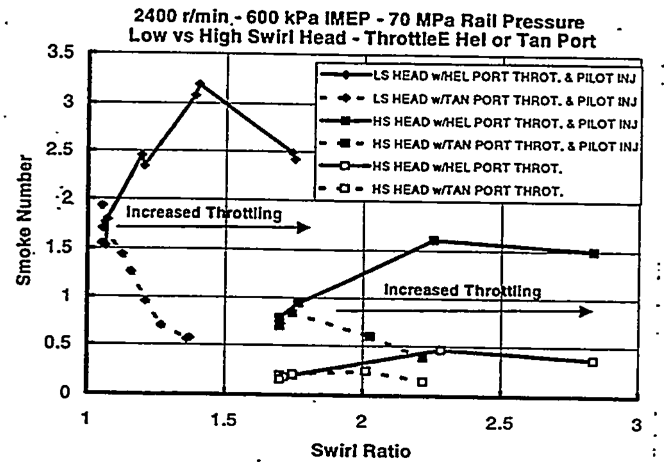

Figure 15. Effects of Intake Swirl on Smoke Emissions. LS and HS Denote Low and High Swirl, Respectively.

Increased swirl in general accelerates combustion as evidenced by reduced combustion durations, increased peak burn rates, and increased maximum cylinder pressure-rise rates. As shown in Figure 16, combustion duration decreased or increased swirl by throttling either port for both cylinder heads with and without pilot injection.

The influences of fuel injection rail pressure, use of pilot injection, amount of EGR, and intake flow swirl ratio, whose individual effects on emissions indices at one fixed operating condition for a single-cylinder engine have just been discussed, represent some of the major parameter effects that are used to optimize HSDI diesel combustion. Now, with these influences as a guide to the choice of the best values for these parameters, we will estimate the overall emissions performance of a six cylinder version of the single-cylinder engine

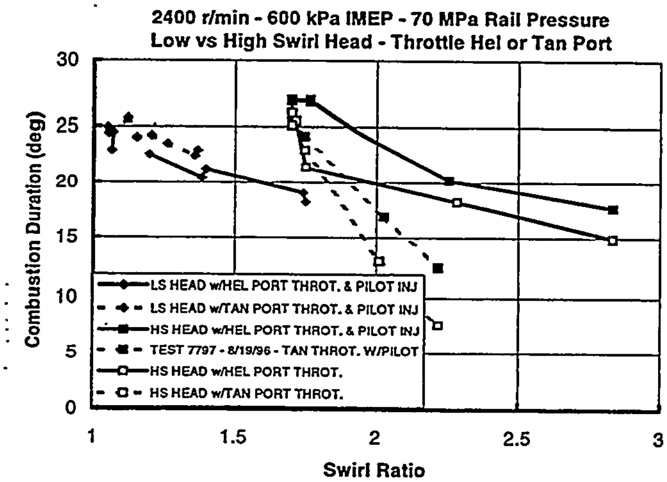

Figure 16. Effects of Intake Swirl on the Combustion Duration. LS and HS Denote Low and High Swirl, Respectively.

operating in a vehicle over the FTP driving cycle. The results will be expressed not as emissions indices, but as vehicle emissions in grams per mile.

Projections for NOx and PM During the FTP

Steady-state tests were run for multiple speed-load conditions to simulate operation on the U.S. FTP test schedule for the 2.56-L engine powering a 1588-kg test weight vehicle. Although such a simulation neglects warm-up and transient effects, those effects are generally less significant for an engine using direct fuel injection (diesel or gasoline). The emissions results are indicative of engine-out emissions as no catalyst or aftertreatment was used. The qualitative trend data generated by such tests provide valuable insights, and the quantitative information shows if one is "in the ball park".

To obtain the test points, first a vehicle simulation computer program called "GPSIM" [10] is run for the desired powertrain and specified transient test cycle. Multiple steady-state test points (e.g., seven) are then chosen after analysis of the resulting engine speed and brake torque histogram. Motoring data from a multi-cylinder HSDI engine having an engine-driven rotary injection pump, expressed as MMEP (motoring mean effective pressure) are then used to convert BMEP values derived from the speed and torque histogram to IMEP (i.e., IMEP = BMEP + MMEP). The single-cylinder engine is then run at these seven settings of speed and IMEP.

Combined Effects: Based on the parameter effects described earlier, a combination of measures can be chosen that demonstrates the NOx/PM emissions capability of this particular combustion system. To demonstrate this, a strategy was chosen that employs: high rail pressure to decrease smoke and particulate emissions; high levels of EGR to decrease NOx; and significant (simulated) turbocharger boost at high loads to decrease smoke and particulate emissions while enabling even higher EGR levels. Specifically, the highest available rail pressure of 1200 bar was used, except at light loads and idle, where rail pressure was reduced to 900 bar. Pilot injection was used at light loads for reductions in gaseous emissions with little particulate penalty and for a noise benefit. Boost levels were the maximum consistent with state-of-the-art turbochargers for light-duty vehicle size engines. The high swirl head was used for these tests, but without port throttling. With this strategy, the single-cylinder engine was then operated at each of the seven test points with EGR and injection timing adjusted to drive NOx and PM emissions as close to the Tier 2 standards as possible.

Figure 17 gives total and dry PM versus NOx results as EGR is increased. In this case, the total particulate and dry particulate mass values were determined from filter measurements.

After weighing for total particulates, the filters were vacuum baked to drive off the volatile material and re-weighed to determine the dry quantity. The dry particulate indicates the minimum particulates possible assuming an ideal oxidation catalyst that removes all the volatile material from the particulate.

U.S. Federal emissions standards for particulates and NOx from passenger car diesel vehicles are shown as boxes in Figure 17. Results for the maximum EGR level (54% averaged over 7 test points) suggest that it might be possible to achieve Tier 2 emissions

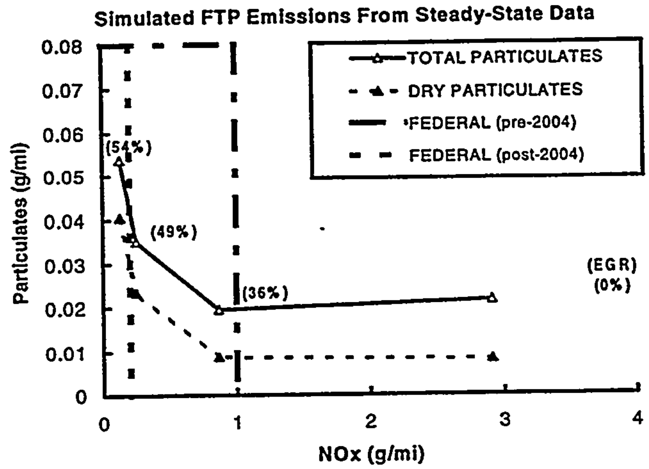

Figure 17. Potential for Meeting Future Standards of the Combustion System Evaluated in the Single Cylinder Tests Based on FTP Simulation.

levels with this combustion system. This suggestion is somewhat misleading since it would not be possible to calibrate a multi-cylinder engine using this combustion system and the high EGR levels for each of the seven test points that make up each data point on Figure 17: practical EGR systems would not provide the uniform cylinder-to-cylinder EGR distribution required. If one cylinder exceeded the EGR level used for these single-cylinder experiments, then that cylinder would contribute more PM emissions than the other cylinders, possibly pushing the engine emissions too high. Similarly, during an engine transient, the EGR level called for by the engine control system may be exceeded by the actual amount of EGR received by the cylinder due to any number of reasons (e.g., engine manifold pressure wave dynamics, EGR valve transient response, turbocharger and/or wastegate response, etc.). Thus, to avoid exceeding the chosen EGR limit for the combustion system, the multi-cylinder engine would have to be calibrated with lower levels of EGR, thereby producing higher levels of NOx than those obtained in the single-cylinder engine tests.

ENABLING TECHNOLOGIES FOR EMISSIONS CONTROL

Based on the single-cylinder work described above as well as on other HSDI work going on throughout the industry [4,6-7,11-13], it is clear that a number of enabling technologies will be required to provide the best opportunity for achieving the most stringent emissions standards. The following list identifies the major technologies.

COMBUSTION SYSTEM

In order to maximize fuel economy thereby, minimizing CO_2 emissions, a direct-injection combustion system will be required. Four valves per cylinder should be used to allow placing the fuel injector in a central, vertical position as shown in Figure 18. This positioning improves the symmetry of the combustion chamber and spray, thereby reducing smoke and PM emissions [11]. However, the use of four valves complicates the choice of intake and exhaust port arrangement since both helical and tangential port types can be arranged in different combinations and in different locations. These port choices all influence and are influenced by valve train preferences as well. Figure 19 shows the intake port arrangement of the single-cylinder engine used for the work reported above, which combined a helical and a tangential port. Figure 20 illustrates an alternative arrangement of two tangential ports, which is similar to that used on the ECOTEC engine. These two port arrangements require different valve placement as shown in Figure 21.

Two intake ports enable the use of port throttling to vary inlet swirl (Figure 22). Inlet swirl is an important parameter for matching flows with fuel spray characteristics over the engine speed range. An example of how the port flow coefficients and in-cylinder swirl parameters trade off is shown in Figure 23. Effects of port throttling on combustion were evaluated with the single-cylinder tests. As was noted, swirl level alone is not an adequate descriptor of the inlet and in-cylinder flow.

Ultimately, the interaction between the flowfield in the piston bowl combustion chamber near top dead center and the fuel spray must be optimized. This optimization involves the fuel spray characteristics, the inlet flow, and the piston bowl shape. A number of piston bowls

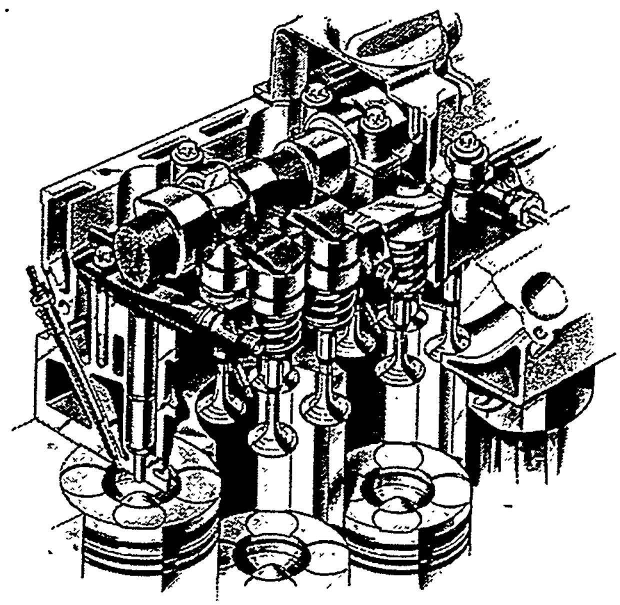

Figure 18. Cutaway of the ECOTEC 2.0 Cylinder Head [3].

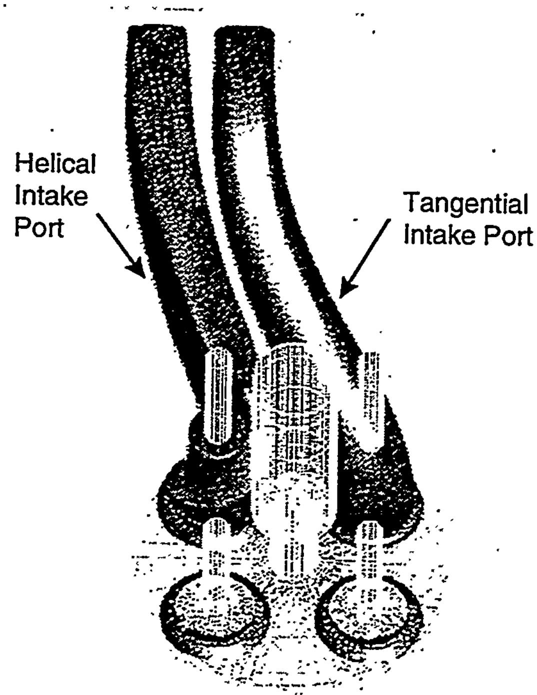

Figure 19. Intake Port Arrangement for the Single-Cylinder Test Engine.

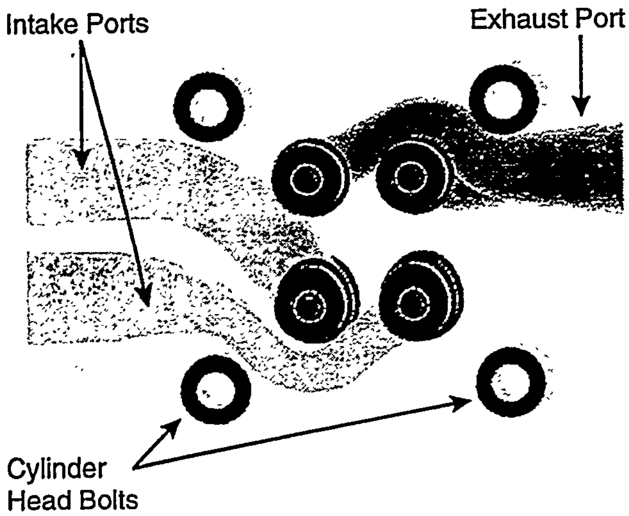

Figure 20. Alternative Intake Port Arrangement.

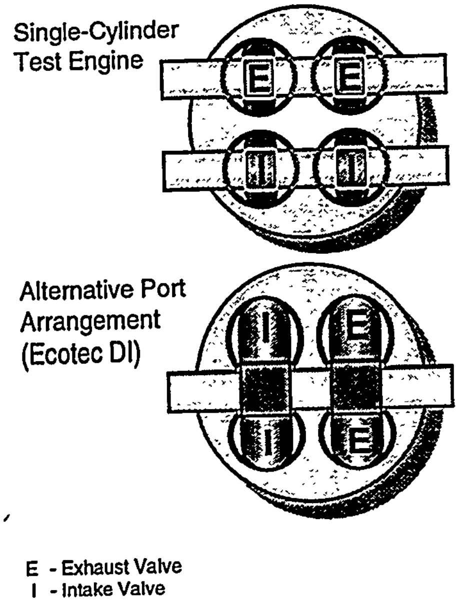

Figure 21. Two Valve Placement Alternatives

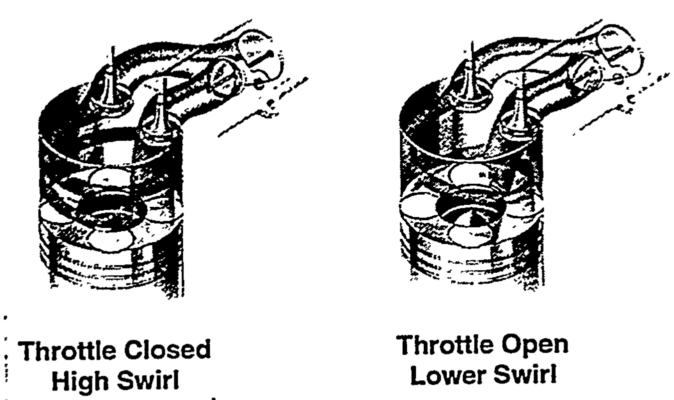

Figure 22. Port Throttling Arrangement Used on the ECOTEC DI Engine [3].

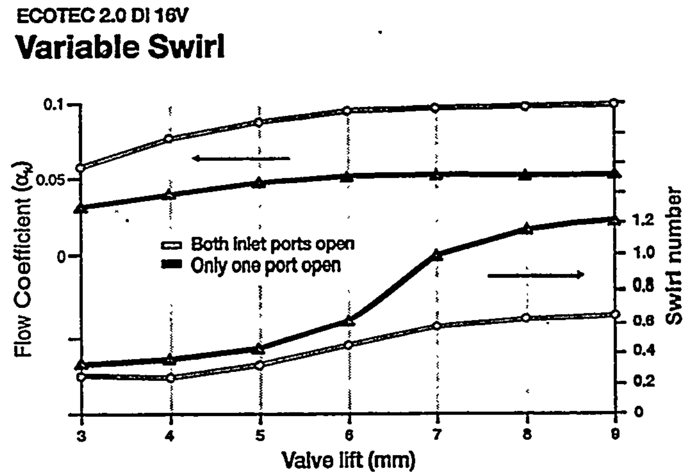

Figure 23. Effect of Port Throttling on Port Flow Coefficients and Swirl at Various Intake Valve Lifts [3].

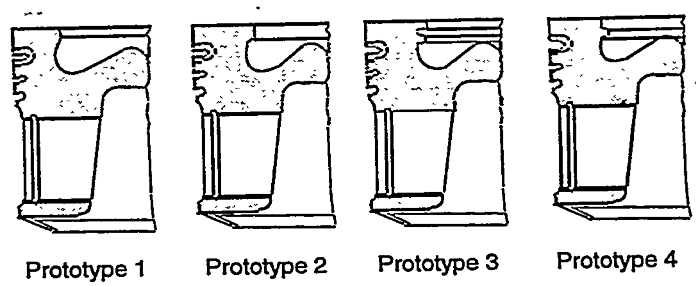

Figure 24. Combustion Chamber Design for the ECOTEC 2.0 Engine [3].

studied during the development of the ECOTEC engine are shown in Figure 24. The critical parameters in piston bowl design are the squish ratio (i.e., ratio of bowl throat area to cylinder bore area), the re-entrancy (i.e., ratio of bowl throat diameter to maximum bowl diameter), bowl lip shape, bowl depth, and bowl interior shape. However, influence of piston bowl geometry on combustion must be balanced with its influence on piston temperature distribution to assure long-term durability of the piston.

EGR SYSTEM

It is clear from the single-cylinder results shown earlier that EGR is a major enabler for NOx emissions control. Figure 25 shows the EGR map for the ECOTEC engine as it is currently calibrated for Euro Stage II emissions standards where EGR levels up to 30% are used. Considerably higher EGR levels will be required to achieve future emissions standards as seen from Figure 17. Configuring an EGR system for a multi-cylinder HSDI diesel engine to deliver such high levels involves consideration of numerous issues. First, there are tradeoffs associated with obtaining good cylinder-to-cylinder distribution versus good transient response. For example, if the EGR is delivered to the engine close to the inlet ports, transient response is enhanced, but obtaining good cylinder-to-cylinder distribution is more challenging. On the other hand, if EGR is delivered far upstream of the inlet manifold, mixture uniformity (and, therefore, cylinder-to-cylinder uniformity) is enhanced, but transient response is more difficult to maximize because EGR must be cleared out of a larger volume before a full air charge can be delivered to the cylinders.

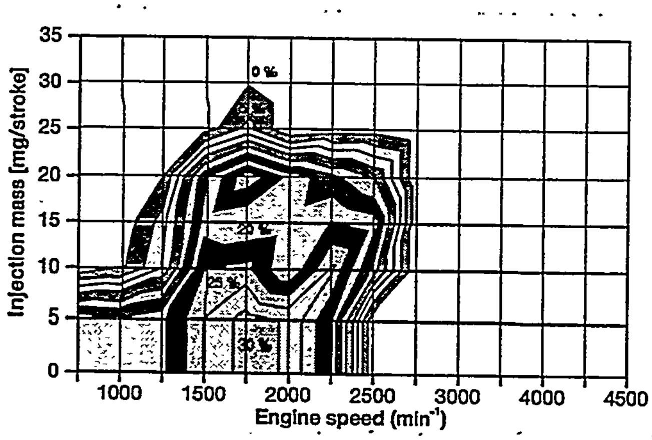

Figure 25. EGR Map as a Function of Fueling Level and Engine Speed for ECOTEC 2.0 DI Engine [3].

A second issue with EGR is cooling. Cooled EGR reduces NOx emissions by lowering charge temperature. However, depending on the system routing used, providing cooling for EGR can add complexity to the engine plumbing and create packaging challenges. One solution to this dilemma is to deliver the EGR using cooled passages internal to the cylinder head. Such an approach was used on the ECOTEC engine and is shown in Figure 26. During cold start, this EGR routing also enhances coolant warm-up and, therefore, passenger compartment heating, which is more of a challenge with HSDI diesel engines because of their low heat rejection.

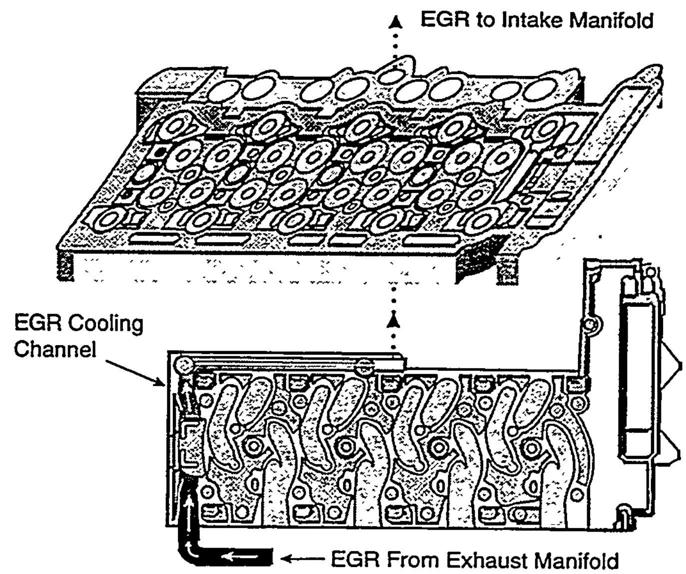

Figure 26. Exhaust Gas Recirculation Channel for the ECOTEC 2.0 Engine [3]. (shown are two horizontal slices thorough cylinder head. Lower slice shows routing of internal EGR passage through cylinder head from exhaust side to intake side. Upper slice shows exit of EGR passage after EGR passes through upper slice)

To reach minimum NOx emissions within these system constraints, EGR flow rates must be provided by the EGR system at levels that are as near to maximum combustion tolerance as cylinder-to-cylinder variation and transient response error allow.

With a typical HSDI diesel engine system, EGR is taken from the exhaust upstream of the turbocharger turbine, and it is delivered downstream of the turbocharger compressor and intercooler to avoid deposit build up in the compressor and intercooler. Depending on the turbocharger and engine characteristics, insufficient pressure difference may exist between exhaust and intake to provide sufficiently high EGR rates. One solution to this problem is to utilize an air inlet throttle, which adds some throttling losses to the engine, but may be a more practical approach than routing EGR upstream of the compressor with its system fouling and transient response shortcomings. Another approach may be possible if a VNT is used. By controlling the turbine nozzle area, exhaust back pressure could be controlled, thus providing additional means of controlling

EGR flow rate.

Needless to say, integrating the control requirements of all these functions to achieve optimum engine system performance will require extensive effort, but it will be essential to achieving emissions targets.

FUEL INJECTION SYSTEMS

The fuel injection system (FIS) is the most critical element of the combustion system of HSDI diesels, and it is a major enabler for combustion modifications to achieve optimum emissions control. For future HSDI diesel engines, the following key criteria must be met by a FIS to be a candidate for selection:

- Sufficiently high injection pressure
- Sufficiently flexible injection rate shape
- Small enough to be packaged in small-bore, four-valve cylinder head designs
- Cost competitive with electronic distributor pump systems
- Parasitic losses equal to or below those of electronic distributor pump systems
- Acceptable durability, reliability, etc.

First let us consider injection pressure. There is a well known link between smoke and particulate emissions and injection pressure (see Figure 10). In recognition of this relationship, injection pressures have been rising as illustrated by Figure 27 where maximum injection pressures or pressure ranges versus engine speed are shown for three injection systems: an older distributor-pump-based injection system (Bosch VP-37); a new distributor-pump-based system (Bosch VP-44); and a future Bosch common-rail system [14]. The expected pressure range potential of common-rail systems is also shown. The VP-37 system generates about 90 MPa at the nozzle. The Bosch VP-44 system, which was first introduced on the GM ECOTEC DI, generates about 130 MPa at the nozzle in its current form, although higher pressure versions are expected in the future.

The common-rail pressure characteristic curve shown is for a suggested calibration [14] for a

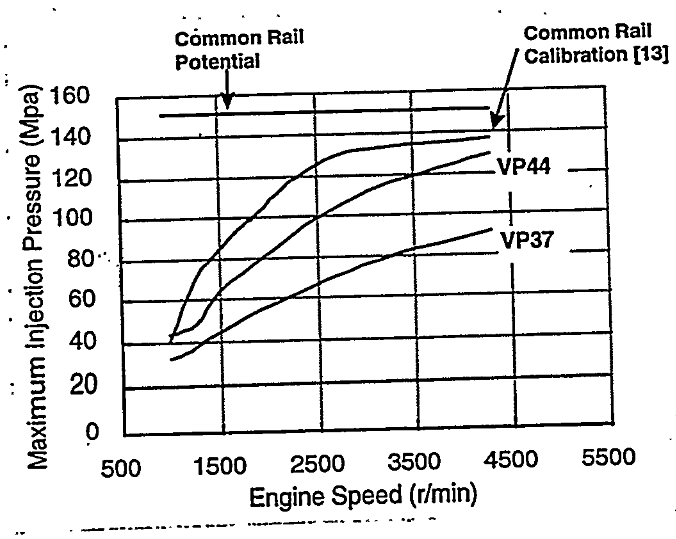

Figure 27. Injection Pressure Calibration Curves for the VP37, VP44 [3], and Common Rail Injection System [14].

future Bosch common rail system expected to be in production soon. However, as shown by the "common-rail potential" range, common-rail systems can generate higher pressures at low engine speeds and there are two potential benefits from this capability. First, high injection pressures allow increased smoke-limited peak torque due to the beneficial effect of such pressures on particulate formation. Second, we saw earlier from the single-cylinder results that higher injection pressures can be beneficial for minimizing NOx/PM emissions at low-speed part-load conditions. However, the emissions benefit of higher pressures at low engine speeds and part loads must be balanced against higher parasitic losses from increased fuel-injection pump work, which increases fuel consumption. In any event, to meet future needs for small HSDI's, a FIS should be capable of 150 MPA maximum injection pressure at as low an engine speed as possible.

Next, let us consider rate shape flexibility. Future injection systems should provide the capability for multiple injection events per combustion event. Figure 28 illustrates the needle lift versus time characteristics which may be required to optimize emissions control. The pilot pulse must reliably inject small quantities of fuel (e.g., 1-5 mm^3) at any crank angle right up to the main injection pulse. This pilot capability is important for control of emissions and noise as shown earlier with the single-cylinder results and as shown by others [15]. But pilot capability is also important for improved starting and emissions performance at low ambient temperatures [16]. When the pilot event occurs just before the main pulse, this capability is similar to that provided by dual-spring injectors, which are used with good effect on current passenger HSDI diesels with distributor pump FISs. The influence of dual-spring nozzles on the lift curve is illustrated in Figure 28 by the "boot toe" portion of the needle-lift curve.

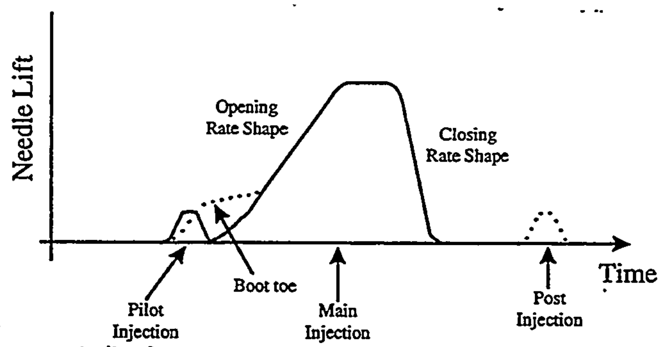

Figure 28. Flexible Needle Lift Profiles Needed for Future Injection Systems.

Accurate control of the opening rate shape is important for controlling combustion noise and NOx emissions. Injection systems such as the Ganser system used in our studies provide a means for adjusting the opening rate as shown in Figure 29. For ultimate injection control, the main injection event should have an electronically controllable opening rise rate with a rapid and repeatable closing event with no needle bounce or after injections. Finally, the FIS should be capable of injecting pilot-like quantities (i.e., as small as 1-2 mm³) after the main injection (post injection) for two possible benefits. First, it has been shown that carefully scheduled post injections can improve the NOx/PM tradeoff in heavy duty engines [17] and reduce smoke in HSDI's [18]. Second, there are potential benefits from injection either during expansion or exhaust to provide additional exhaust hydrocarbons for use with a lean NOx catalyst.

In addition to functional requirements, there are also packaging constraints on fuel injectors

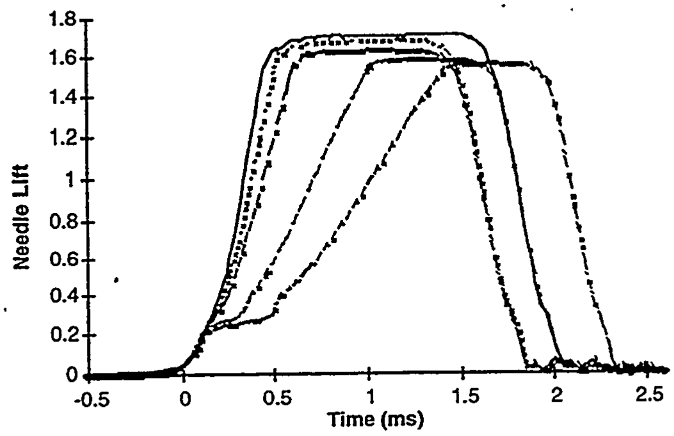

Figure 29. Needle Lift Profiles with Different Opening Rate Settings.

imposed by valve train components. These components can interfere with the top of the fuel injectors and can constrain the size of the bottom end of injectors by the need to maximize valve size while still maintaining adequate coolant flow within the cylinder head to cool exhaust ports, fuel injectors, fire deck, etc. This problem is particularly difficult for smaller bore engines (i.e., less than 80 mm bore), which are being studied for the European market [191 as well as for application to PNGV (Partnership for a New Generation of Vehicles) in the U.S.[20]

New developments in some other fuel injection system aspects could also be beneficial. Examples of such developments are the capability to manufacture smaller injector hole sizes or injector designs with variable nozzle hole area. At the present time, the practical lower bound for production nozzle hole sizes is in the range of 0.16 to 0.18 mm. This lower bound is determined by manufacturing constraints. Recent engine experiments with micro-hole nozzles that were 0.06 mm in size have demonstrated significantly shorter ignition delays and lower particulate emissions than with larger holes [21,22]. In practice, smaller holes are preferred at low load and speed to control engine noise and particulates. There is some evidence that small holes limit the maximum penetration of the liquid core of the spray jets independent of injection pressure [19]. Whereas small holes are beneficial to low load and speed conditions, more holes or larger holes are required at full load and speed conditions to deliver sufficient fuel in a short enough time. The normal increase in injection pressure between low and high speed helps, but even more flexibility is desirable.

Variable flow area injector nozzles have long captured the imagination of DI injector designers. One approach to variable area nozzle tips is the poppet-covered orifice design which incorporates two sets of holes that are progressively uncovered with increasing needle lift [23]. An alternate approach relies on the alignment of a control surface and the nozzle hole to control the effective nozzle flow area [24]. Performance of both of these injector designs has shown promise, but neither concept has been put into production.

FUEL CHARACTERISTICS

Another potential enabler for minimizing emissions is modification of the fuel characteristics.

Reformulated Diesel Fuel

Beginning in October 1993, a low-sulfur reformulated diesel fuel was required for all on-highway use in the 49 states excluding California. Specifically, this fuel has the following limits: 0.05 weight percent sulfur maximum, 40 cetane number minimum, and 35 volume percent aromatics maximum. At the same time, California required all vehicular diesel fuel to meet the same sulfur requirement and an aromatics limit of only 10 volume per-cent maximum. Alternatively, in California, diesel fuel demonstrating equivalent or lower emissions as a reference fuel in a specific heavy-duty diesel engine test procedure could be sold [25].

The California diesel fuels are generally blended to achieve the equivalent emissions option [26,27]. Aromatic content of commercial diesel fuels in California may exceed 27% [27], but this, combined with very low fuel sulfur and high cetane number, still meets the emissions requirements. Such fuels can be expected to produce significant HC and CO reductions along with some particulate emissions advantage. However, the effect on NOx emission is minimal [26]. NOx has generally proven the most difficult of all the regulated emissions to alter by fuel reformulation.

The recently completed European Auto/Oil study (EPEFE) considered the effects of fuel properties (i.e., density, polyaromatics, cetane number, and T95 distillation point) on the emissions from 19 light-duty diesel vehicles. These vehicles included 17 passenger cars and two trucks. Fourteen of them used indirect injection (1131) combustion systems and the remaining five used direct injection (DI). All of them were fitted with oxidation catalysts [28]. The vehicles were tested using the European MEG test cycle which combines urban and extra-urban test cycles. Linear regression equations were developed to show the effects of fuel properties on the exhaust emissions averaged over the test fleet [29]. Table 8 shows the reductions in exhaust emissions predicted when all four fuel properties were changed simultaneously as indicated.

TABLE 8. Predicted Emission Reductions (%) As Fuel Properties Are Changed [29]

HC	CO	NOx	PM
34	42	-3	24

Based on the following simultaneous changes in fuel properties:

Density, kg/m3	Polyaromatics, %mass	Cetane no.	T95, °C
855 → 826	8 → 1	50 → 58	370 → 325

Interestingly, the NOx emissions increased slightly. These results were dominated by the IDI vehicles, which generally showed an increase in NOx as cetane number increased. On the other hand, the DI vehicles generally emitted lower NOx as cetane number increased. Other studies on reformulated diesel fuels with reduced density, polyaromatic, and T95 (and increased cetane number) have reported NOx reductions (e.g., [30]). In any case, the effect of moderate diesel fuel reformulation on NOx emissions appears to be small. However, in those cases where NOx is the critical emission in achieving emissions compliance, it may be found that reduction of other regulated emissions (i.e., HC, CO, and PM) via fuel composition changes will provide an opportunity to recalibrate the vehicle's emission control system for reduced NOx without increasing the other emissions above their original values. Further, substantial reductions in unregulated emissions (e.g., the mutagenicity of particulate soluble organic fraction and formaldehyde emissions) have been reported with reformulated diesel fuels [31].

Studies are currently under way to determine the emissions effects with more aggressively reformulated diesel fuels. Perhaps, the ultimate possible reformulated hydrocarbon-based diesel fuel is represented, not by a petroleum-derived fuel, but, rather, by fuels now being produced in pilot plant operations using the Fischer-Tropsch (F-T) process with natural gas as the starting material [32]. For example, Sasol has reported that diesel fuels with over 70 cetane number, less than 0.1% aromatics, and less than 10 ppm sulfur can be produced using their "Slurry Phase Distillate" version of the F-T process [33].

Beyond hydrocarbon-based reformulated diesel fuel, oxygenated hydrocarbons have been the most commonly suggested blending components for the purpose of reformulation. Alcohols have been used for this purpose, but their high octane numbers and correspondingly low cetane numbers have generally made them more attractive for use in reformulated gasoline. Oxygenated diesel fuel produced by the addition of modified vegetable oils (e.g., esterified rapeseed and soy) to diesel fuel appears to be a better choice, since the vegetable oils generally exhibit relatively high cetane numbers [34]. Their high viscosities, relatively poor low temperature properties (e.g., pour point) and decreased stability dictate that these oils be used as blending components (e.g., 20%) in diesel fuel rather than neat. An EMA (Engine Manufacturers Association) paper on the use of biofuels [34] concluded that they may be beneficial for reducing PM. Although engines fueled with these blends demonstrated reduced particulate and CO emissions, they also tended toward increased NOx emissions [34]. In addition, these fuels have typically been quite expensive compared to conventional diesel fuel, and it is not clear that they will be cost competitive for some time into the future.

Regardless of which advanced processes ultimately provide reformulated diesel fuels with the greatest value, it seems clear that the trend toward extremely low sulfur will continue. When sulfur is reduced, the possibility of catalyst poisoning is also reduced, and both oxidation and lean NOx exhaust converters become possible. Volkswagen was first to market an HSDI diesel passenger car that included a lean NOx converter, but others are likely to follow in an effort to take advantage of the outstanding fuel economy of the light-duty HSDI diesel engine while still achieving stringent emissions reductions. It remains to be shown how fuel reformulation will affect the performance of these converters.

<u>Alternative Fuels</u>

Many alternative fuels have been proposed for diesel engines over the years, including pulverized coal, neat alcohols, diesel fuel-water emulsions, and, more recently, "designer fuels" - materials specifically synthesized for use as diesel fuels. For the purposes of this discussion, emulsified fuels and dimethyl ether (one of the designer fuels) will be discussed, as they are currently of the most interest. A wealth of literature exists on the various other alternative diesel fuels, but a review of these other alternatives is beyond the scope of this paper.

Diesel/water emulsions have been suggested repeatedly as a practical means to simultaneously reduce particulate and NOx emissions. The addition of water or any diluent (e.g., EGR) is expected to reduce the combustion temperature and, thereby, decrease the formation of NOx [35]. The presence of water also decreases overall particulate emission in most cases. Indeed, studies of the effect of emulsified diesel/water fuels (typically 10 to 50% water) have shown that it is possible to simultaneously reduce NOx and smoke compared to a baseline diesel fuel [36,37]. Recently, Caterpillar participated in a joint venture to develop proprietary diesel/water emulsions for reduced emissions. Although the joint venture has been dissolved, Caterpillar's work continues, and significant benefits in both particulate and NOx emissions have been reported [38] along with some disadvantages (e.g., higher HC emission). Further reports on emulsified diesel fuel performance from Caterpillar and other companies are expected. Even if successful in reducing emissions, these fuels will need to overcome major barriers to their widespread use in the form of higher cost, inadequate infrastructure development, and fuel system incompatibilities.

Dimethyl ether (DME) consists of two methyl groups chemically bonded to a central oxygen atom. It is currently produced from natural gas, and it is used as a propellant for cosmetics aerosols sold in Europe. As with methanol, since the fuel is partially oxidized and it contains no carbon-carbon bonds, the possibility of forming carbonaceous particulate emissions during combustion of DME is essentially eliminated. However, unlike methanol, DME has a high enough cetane number - >55 [391 - to perform well as a compression-ignition fuel. Also, unlike methanol, DME is a gas at ambient temperature and pressure, so it must be stored under pressure as a liquid similar to LPG (liquefied petroleum gas). Its physical properties (e.g., density, viscosity, lubricity, etc.) are so different from diesel fuel that the entire fuel system must be redesigned [40]. Preliminary studies have concluded that it should be possible to just achieve ULEV emissions using a properly designed DME-based fuel injection system with an HSDI engine and an oxidizing catalytic converter in a passenger car [39].

It seems clear that DME and, perhaps, other fuels will be able to produce much larger emissions reductions than will be possible with diesel fuel for a specific level of complexity of the fuel storage and injection system. However, it is not clear that this apparently inherent emissions advantage can offset the fuel's lack of established supply infrastructure and the need to distribute and carry a pressurized fuel onboard the vehicle. Before DME or other alternative fuels become used extensively in compression ignition engines, the tradeoff between the complexity of engine and fuel system development with reformulated (liquid) fuels and their emission performance must be established. If emissions targets can be achieved with liquid fuels that can be distributed and handled by the existing diesel fuel infrastructure, it is unlikely that alternative compression ignition fuels will play a significant role.

EXHAUST AFTERTREATMENT

As noted earlier, two exhaust constituents pose the greatest challenge in meeting future diesel emission standards: NOx and particulates. Given the interrelationship that exists between the engine-out levels of these two, any aftertreatment that reduces the emissions of one will allow the other to be reduced by engine calibration. The approaches currently offering the greatest hope of reducing exhaust NOx concentrations are the selective reduction of the NOx by either a catalyst or by a catalyst working in conjunction with a plasma discharge in the exhaust. The approaches currently offering the greatest hope of reducing exhaust particulate mass are particulate filters, an exhaust plasma (which is reported to reduce NOx as well), and catalysts to oxidize the heavy hydrocarbons before they condense on the soot to become the soluble oil fraction (SOF).

NOx Reduction

Lean NOx catalysts have received a great deal of attention in the 1990's. These catalysts are designed to react HC's present in the exhaust selectively with NOx rather than with oxygen. Figure 30 shows the HC and NOx conversions as a function of temperature for one such catalyst developed by the USCAR Lean NOx CRADA [41]. Typical of all such catalysts, NOx conversion is poor at low temperature, the conversion rises to a maximum at an intermediate temperature, and then it falls as the temperature increases further. The maximum NOx conversion generally takes place close to the temperature at which the HC lights off. The temperature at which the maximum NOx conversion occurs is a function of the active metals used, with the maximum for Pt-containing catalysts in the 200-250 C range, while the maximum for Cu-containing catalysts is in the 450-500 C range. The activity-temperature characteristics of the Pt-based catalyst are most compatible with diesel exhaust, the temperature of which ranges between 150 C and 400 C in the Euro or U.S. test cycles. The major problems faced by lean NOx catalysts are: the narrowness of the temperature window; the limited NOx conversion, even at the optimal temperature; and the need for HC's for the selective reduction of NOx. The inherently low level of HC's in the exhaust from direct-injection diesels will require that the exhaust HC's be enhanced by the addition of fuel during either the expansion or exhaust strokes, as was suggested above in the Fuel Injection discussion. The Pt catalysts face the additional shortcomings of being highly active for sulfate formation and for forming N_2O, a principal NOx reduction product, which is a potent greenhouse gas.

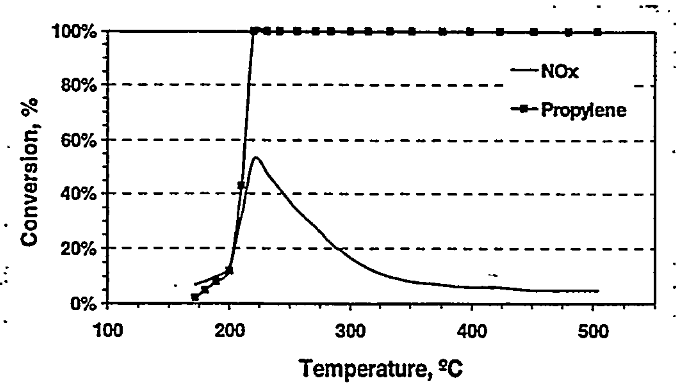

Figure 30. HC and NOx Conversions with a Pt-Containing Lean-NOx Catalyst.

The concept of using a nonthermal plasma to remove NOx has been discussed for a long time but has generally been dismissed as impractical. Penetrante et al [42] have shown that the minimum electrical energy to remove 100 ppm of NO from a gas by plasma-induced gas-phase reactions is 15 Joules per liter of exhaust treated (6.5 ppm NOx/Joule/L,), a value too high to be of interest for automotive applications. Recently, though, coupling the plasma approach with a catalyst to reduce the energy requirement of the system has been considered. Most of the work in this area is proprietary, and while many claims abound, little information exists in the technical literature. NOxTECH, a firm that plans to commercialize the plasma-catalyst system for NOx removal, has reported that they obtain greater than 70% removal of both NOx and particulates from diesel exhaust with less than a 7% fuel economy penalty [43].

Particulate Reduction

The amount of particulates in diesel engine exhaust can be reduced by filtering, and this approach has been investigated extensively. The primary difficulty is not the filtration (or trapping), but the regeneration of the trap by burning off the collected soot. The combustion temperature of soot can vary but generally begins at about 500° C [44], a temperature that is not encountered often in the exhaust of a light-duty direct-injection diesel. The ignition of the soot can be initiated by a burner or by electrical heat. The ignition temperature of the soot can also be lowered by the addition of iron, copper, or cerium additives to the fuel. Pattas et al. [45] were able to induce trap regeneration at 300° C by doping the fuel with 100 ppm of cerium, but they found that the increased back pressure due to the soot accumulation could result in as much as a 15% fuel economy penalty.

As mentioned previously, the nonthermal plasmas that are reported to reduce NOx are also reported to reduce particulates, but such claims have not been verified. Measuring the impact of catalysts on particulates can be difficult, since catalyst beds can retain particulates for extended periods of time.

Heavy HC's that are present in diesel exhaust will condense on the particulates, thereby increasing the mass of the particulates. Oxidation catalysts have been used for a number of years to oxidize some of the HC's in diesel exhaust, reducing both the HC and particulate emissions. However, catalysts that are highly active for HC oxidation also tend to be active for oxidizing SO_2 to SO_3, which collects on particulates, also increasing the mass of the particulates. Catalysts can be designed to offer reasonable HC oxidation performance without excessive levels Of SO_2 oxidation activity [46]. Hammerle et a[. [47] recently showed an 18% reduction in the particulate emissions from an HSDI diesel engine during an FTP test using such an oxidation catalyst.

Exhaust aftertreatment for diesel engines is a critical technological area to the ability of diesel engines to meet future emissions standards.

Despite much progress in this area, many challenges remain that offer opportunities for progress.

ENGINE CONTROLS

Another important enabler for emissions control is highly capable engine controls.

The high EGR tolerance and low NOx/PM trade-off observed in the single-cylinder engine tests cannot be exploited in a multi-cylinder engine without a capable emissions control system. With the enabling technologies listed above, the emissions control system must manage control of the fuel injection process, which, for a common-rail system, involves numerous parameters such as pilot injection, rail pressure, injection rate, and post injection during the expansion or exhaust strokes to provide HC for a lean NOx catalyst. In addition to fuel injection, the control system must manage the actuation and modulation of port throttles (if used), EGR valve, inlet throttle (if used), turbocharger (either wastegate or VNT control), and possibly a separate catalyst fuel system, if a common-rail injection system is not used. One capability that would be useful for meeting future emissions standards is closed-loop control of EGR, which would not only enable more aggressive EGR calibrations, but also compensate for changes in engine and EGR system components over the life of the vehicle.

HYBRIDIZATION TO ENABLE EMISSIONS CONTROL

Although hybridization was discussed earlier with regard to performance and efficiency, hybridization also offers opportunities as an emissions control enabler to help achieve Tier 2, ULEV, and Euro Stage IV emissions standards with HSDI diesel engines. For example, emissions occurring during engine transients could possibly be reduced through hybridization by applying stored energy to shave transient change rates. Also, during more steady-state operation, there may be opportunities to modify exhaust temperature by modulating energy storage system charging or withdrawal rates to enable more optimum exhaust gas after-treatment. However, the extent to which such opportunities exist or can be effectively exploited remains to be determined.

CLOSURE

Today's HSDI diesel engine offers demonstrated fuel economy advantages unmatched by any other engine. HSDI diesels are also proven and, in their latest implementations, offer similar pleasability to that of the SI engine. However, to remain a contender in the future, HSDI diesels must meet the challenges posed by tighter emissions standards and tighter competition for product value (i.e., "value" standards). Because HSDI diesels are more costly to produce than SI engines, future value standards will be most easily met by HSDI diesels for customers for whom fuel costs are an important part of the cost of vehicle ownership, thus allowing recovery of the higher initial purchase price. Stringent value competition will continue to be provided by technologies based on the SI engine, particularly for customers with lower fuel costs. In addition, evolutions of the conventional SI engine are expected to meet the future emissions standards with much less difficulty than for HSDI diesels. Hybrid drivetrains offer an opportunity and a challenge for HSDI diesels -- an opportunity for the use of diesels as part of such drivetrains and competition for diesels when other fuel-driven power sources are used.

Thus, HSDI diesels are one option for future personal transportation powertrains whose attributes must be weighed against the different attribute sets of other technologies as we configure powertrains and vehicles for the next century for differing markets throughout the world.

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APPENDIX - SINGLE-CYLINDER ENGINE INSTRUMENTATION AND TEST PROCEDURES

A brief description of test measurements and procedures is given in this appendix for the single cylinder engine-dynamometer studies reported in this paper.

GASEOUS EXHAUST EMISSION MEASUREMENTS

Gaseous exhaust emissions measurements were made as listed in Table A-1. An additional CO_2 analyzer was used to sample CO_2 concentrations in the intake plenum to calculate EGR or in the dilution mini-tunnel to calculate dilution ratio for calculation of particulate mass.

TABLE A-1

Component	Units	Analyzer Type
HC	ppm C_3	Heated Flame Ionization Detector
CO	ppm	Infrared Analyzer
CO_2	%	Infrared Analyzer
O_2	%	Magneto-Pneumatic Analyzer
NOx	ppm	Chemiluminescence Analyzer using NO_2 to NO Converter

EXHAUST PARTICULATES

Actual particulate mass measurements were made on a more selective basis due to the time and effort required for collection and analysis. Particulate emissions were measured on a filter using a dilution mini-tunnel [48]. A small fraction of the exhaust is extracted and diluted within the tunnel. A portion of the diluted exhaust is then drawn though the filter.

Prior to test, new filters (47-mm diameter Pallflex Fiberfilm T60A20) were baked in a vacuum oven for 4 hours at 210°C while maintaining a vacuum of 88 kPa. Following stabilization for at least one hour in a temperature- and humidity-controlled room, the filters were weighed using a micro-balance before and after collection to determine the total particulate mass. Then, to drive off the volatile particulate material; the filters were baked in the vacuum oven again, using the same procedure as for new filters. Re-weighing the baked filters after stabilization gave the dry particulate mass.

During collection, a filter temperature of 50° C and a face velocity of 0.7 m/s were maintained. Dilution ratio was controlled within the range of 5 to 8:1.

OTHER MEASUREMENTS AND CALCULATIONS

The data for each test point were acquired every half second and averaged over a 90-s period. Fuel injected quantities and timing were specified with control pulses to the fuel injector. Fuel rail pressure was controlled independently to alter injection rate. The resulting fuel flow was measured with a constant volume flow meter that outputs 110 pulses per cm^3 of fuel. Airflow was controlled and measured by varying the upstream pressure to four critical flow nozzles of different sizes that can be selected in any combination. Measured air-fuel ratio was checked with that from exhaust gas analysis using both carbon-balance and oxygen-balance calculations. Exhaust gas analysis also was used to convert raw emission concentration measurements to emission index (mass emissions in grams per kg of fuel). By controlling airflow and exhaust pressure, inlet and exhaust pressure levels were set to simulate naturally aspirated or turbocharged operating conditions.

Cylinder pressure was measured with an AVL QC32C water-cooled piezoelectric pressure transducer mounted flush in the cylinder head. The common rail fuel injector included a needle-lift sensor. For routine testing, cylinder pressure and needle lift were digitized every 1.0 crank angle degree using an engine-driven shaft encoder and digital oscilloscope, producing a data file averaging 64 non-consecutive cycles. Post processing included pressure analysis and a simple heat release analysis of that average file. More detailed analysis, when warranted, was accomplished by digitizing 64 consecutive cycles of cylinder pressure and needle lift every 0.2° crank angle using a dedicated high-speed data acquisition system and post-processing the individual cy-cles with a rigorous pressure and heat release analysis.

MULTIPLE-POINT OPERATING CONDITIONS FOR SIMULATION OF FTP

Table A-2 gives the seven steady-state test conditions used in this paper to project emissions on the urban test cycle (FTP) from single-cylinder engine data. The modeled powertrain includes a 2.56-L DI diesel engine with a 5-speed manual transmission powering a 1588-kg test weight vehicle. The single-cylinder engine is run at each speed and IMEP (dynamometer indicated, not cylinder-pressure based) listed in Table A-2. That IMEP gives the required BMEP when using motoring mean effective pressures (MMEP) typical of multi-cylinder engines. The equation for MMEP given in Table A-2 is a correlation of passenger car size DI diesel engines having rotary injection pumps.

For each test point, the fuel rate times the ratio of the multi-cylinder to single-cylinder engine displacement (2.56/0.426) gives the projected multi-cylinder fuel rate. The product of that fuel rate (kg/s), the pollutant emission index (g/kg), and time (s) gives the pollutant mass emissions (g) for each test point. Then the sum of pollutant mass for all test points divided by the FTP cycle distance (7.44 miles) gives the projected g/mi values.

TABLE A-2

Point #	Schedule	Engine Speed (rpm)	Brake Torque (Nm)	BMEP (kPa)	IMEP* (kPa)	Time (s)	Energy (kWh)
7-POINT FTP							
1	FTP	800	2.0	10	181	363.5	0.3117
2	FTP	1714	122.9	604	793	62.2	0.5007
3	FTP	2296	85.3	419	633	67.8	0.5835
4	FTP	2215	24.4	120	330	212.0	0.9177
5	FTP	1731	7.33	36	226	279.9	0.6485
6	FTP	1706	57.6	283	472	216.5	1.032
7	FTP	1027	18.9	93	266	62.2	0.1006
					SUM =	1264.1	
					(TOT. FTP =	1372.1)	

* MMEP = 176.4 -1.880E-2*(rpm) +1.526E-5*$(r/min)^2$