SwRI 4820

DEVELOPMENT OF BASELINE AND CONTROLLED EXHAUST EMISSION RATES FOR OFF-HIGHWAY VEHICLE ENGINES

FINAL REPORT

Contract No. A198-076

Prepared for

California Air Resources Board
Mobile Source Division
9528 Telstar Avenue
El Monte, California 91731

By

Southwest Research Institute
Automotive Products and Emissions Research Division
6220 Culebra Road
P.O. Drawer 28510
San Antonio, Texas 78228-0510

July 1993

SOUTHWEST RESEARCH INSTITUTE
SAN ANTONIO HOUSTON
DETROIT WASHINGTON, DC

DEVELOPMENT OF BASELINE AND CONTROLLED EXHAUST EMISSION RATES FOR OFF-HIGHWAY VEHICLE ENGINES

Prepared by

Jeff J. White

FINAL REPORT

Prepared for

California Air Resources Board
Mobile Source Division
9528 Telstar Avenue
El Monte, California 91731

July 1993

Approved:

Charles T. Hare

Charles T. Hare, Director
Department of Emissions Research
Automotive Products and
Emissions Research Division

ABSTRACT

Southwest Research Institute performed this project for ARB to develop baseline and controlled exhaust emission rates for off-highway vehicle engines. Existing emission data for this category were reviewed and engines were recommended for which baseline emission testing was most needed. Test procedures were evaluated and specific test cycles and analytical methods were recommended. Ten off-highway vehicle category engines were tested to determine baseline emissions. Emission control technologies applicable to this category were evaluated. The more promising technologies were engine tested to demonstrate the potential emissions reductions achievable. Spark-ignited engine technologies evaluated included alternate fuels (LPG and CNG), exhaust gas recirculation (EGR), and three-way catalyst, in conjunction with closed-loop control. Diesel engine technologies evaluated included retarded injection timing and turbocharging. This information will be used by ARB to develop exhaust emission standards and regulations for the off-highway vehicle category.

ACKNOWLEDGEMENTS

This project was performed by the Department of Emissions Research in the Automotive Products and Emissions Research Division at Southwest Research Institute, San Antonio, Texas. Jeff J. White, Manager of the Certification, Audit, and Compliance Section, was the Principal Investigator and prepared this report. James N. Carroll, Research Engineer, directed laboratory testing activities and analyzed and compiled the engine emission data. Rick D. Frierson and Daryl W. Krenmueller performed the emissions testing. Sharon A. Tondre performed the word processing for this report.

Southwest Research Institute was assisted by Booz·Allen & Hamilton, Inc. in performance of tasks 1 and 4. Booz·Allen provided engine and equipment inventory information, and technology analysis. Robert M. Kreeb directed program activities performed by Booz·Allen & Hamilton.

This report was submitted in fulfillment of ARB Contract No. A198-076, "Development of Baseline and Controlled Exhaust Emission Rates for Off-Highway Vehicle Engines," by Southwest Research Institute under the sponsorship of the California Air Resources Board. Work was completed as of February 26, 1993.

DISCLAIMER

The statements and conclusions in this report are those of the contractor and not necessarily those of the California Air Resources Board. The mention of commercial products, their source or their use in connection with material reported herein is not to be construed as either an actual or implied endorsement of such products.

TABLE OF CONTENTS

TABLE OF CONTENTS (CONT'D)

TABLE OF CONTENTS (CONT'D)

APPENDICES

LIST OF FIGURES

LIST OF TABLES

LIST OF TABLES (Cont'd)

LIST OF TABLES (Cont'd)

I. SUMMARY AND CONCLUSIONS

The objective of this project was to develop baseline exhaust emission rates for midrange or off-highway vehicle (OHV) engines in order to provide reliable data for the development of exhaust emission standards and regulations for the off-highway vehicle category. Major activities were:

- Summarize existing category emission data
- Analyze the relative emissions contribution of category equipment according to engine technology and end-use application
- Recommend, and then conduct engine emission testing to supplement existing data
- Recommend exhaust emissions test procedures for category equipment
- Evaluate emission control technologies applicable to OHV engines
- Emission test feasible emission control technologies to demonstrate effectiveness.

Available emission test data were compiled and analyzed to identify deficiencies in off-highway vehicle (OHV) category emissions data. While much of the existing information is quite old, some recently developed data on diesel forklift engines and diesel nonroad engines greater than 100 hp was found. Category equipment population was analyzed to determine the proportion of total emissions contributed by application and engine technology type. Based on this information, specific recommendations for an engine test matrix were formulated. Engine manufacturers were contacted, and test engines were solicited in exchange for test data.

Emission test procedures were evaluated for this category. In its expanded form, the OHV category contains a very diverse group of equipment, which for various technical reasons, can probably not all be evaluated using the same test procedure. The following recommendations are made:

Equipment	Test Procedure
OHV Diesel, 25 - <175 hp	ISO 8178-C1 cycle
OHV Otto-cycle, ≥25 hp	ISO 8178-G1 or C2 cycle
Off-road motorcycles	On-road motorcycle FTP
ATVs	On-road motorcycle FTP
Snowmobiles	ISO 8178-G1 cycle

The ISO 8178-C1 cycle is well suited for OHV category diesel engines. California recently adopted this procedure for regulation of new 1996 and later heavy-duty off-road diesel cycle engines (≥175 hp class). This procedure has found acceptance with ARB and EPA, as well as engine manufacturers. The selection of a test cycle for OHV SI engines is more difficult. ARB has favored the ISO 8178-G1 cycle (6-mode J-1088 type) due to its application in regulations for utility and lawn and garden equipment, less than 25 hp. In the course of this program, an ISO 8178 "revised C2" cycle was proposed by the Engine Manufacturers Association (EMA) and the Industrial Truck Association (ITA). This cycle is based primarily on lift truck duty cycles which may or may not be representative of all SI engines in this category. What little data have been generated comparing the two cycles suggests the revised C2 cycle may yield higher emissions results than the G1 cycle. In any event, this cycle has the strong support of both EMA and ITA, and is therefore recommended for consideration by ARB for OHV SI engines.

Ten off-highway vehicle engines were tested to determine baseline emissions. Test procedure ISO-8178 cycle C1 was used for diesel engines, and cycle G1 was used for spark-ignited engines, except the golf car engine which used the J1088 test cycle. Baseline emissions are shown below:

Engine ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
	THC	CH_4	RHC	CO	NO_x	PM	
Diesel Engines (ISO 8178-C1)							
76 HP DI Diesel	0.89	NA	NA	1.54	6.08	0.38	0.38
75 HP DI Diesel	0.64	NA	NA	3.51	7.24	0.59	0.38
30 HP Air-Cooled	2.72	0.03	2.69	5.50	8.50	1.40	0.48
35 HP Water-Cooled	1.10	0.03	1.07	2.64	3.24	0.51	0.51
50 HP Water-Cooled	1.74	0.02	1.72	5.61	7.27	0.58	0.38
Spark-Ignited Engines (ISO 8178-G1)							
8.5 HP Golf Car*	7.53	0.40	7.14	109	8.32	0.16	0.78
40 HP Lift Truck	0.90	0.03	0.87	2.87	16.9	0.02	0.50
60 HP SI Utility	2.02	0.34	1.68	126	8.30	0.02	0.61
60 HP CLC/TWC Utility	0.09	0.01	0.08	2.72	0.53	0.01	0.52
100 HP SI Utility	2.79	0.24	2.55	93.9	9.44	0.01	0.58
50 HP Snowmobile	96.7	1.13	95.5	347	0.33	3.11	1.01
*J1088 test cycle							

Emission reduction technologies were applied to selected engines and evaluated for emission benefits. Spark-ignited engine technologies evaluated included alternate fuels (LPG and CNG), exhaust gas recirculation (EGR), and three-way catalyst, in conjunction with closed-loop control. Diesel engine technologies evaluated included retarded injection timing and turbocharging. A number of technologies were found to be very effective in reducing emissions.

- The 60 HP spark-ignited utility engine was converted to LPG, and then CNG operation to evaluate its performance using these two alternate fuels. Comparisons with gasoline-fueled baseline emissions were dominated by the much leaner calibrations employed with these gaseous fuels. HC and CO emissions were considerably reduced, but large increases were observed in NO_X emissions. In addition to emission effects, there was a reduction in available power in the LPG and CNG configurations.

- Exhaust gas recirculation (EGR) was very effective at reducing NO_X emissions from the 40 HP LPG lift truck engine, although at the expense of increased HC and CO emissions and fuel consumption.

- The 60 HP spark-ignited utility engine was refitted with an LPG closed-loop control system and a three-way catalyst. With these state-of-the-art emission control technologies, emissions from this engine were reduced to very low levels. THC and NO_X emissions were reduced by 84 and 96 percent, respectively, compared to open-loop, non-catalyst, LPG-fueled operation.

- Retarding diesel engine injection timing was found to be effective in reducing NO_X emissions. NO_X reductions of 36 percent, 9 percent, and 26 percent were observed for the 75 HP DI, 35 HP water-cooled, and the 50 HP water-cooled diesel engines, respectively. However, these NO_X reductions were achieved at the expense of sharply increased PM and HC emissions. Particulate emissions increased by 33 to 49 percent, as a result of these timing changes.

- Turbocharging, in conjunction with retarded injection timing, reduced CO, NO_X, and PM emissions from the 75 HP DI diesel engine by 70, 23, and 21 percent, respectively, compared to baseline emissions. Turbocharging, however, constitutes a major engine design change, with attendant, substantial development costs. The applicability of turbocharging, or other "on-road" diesel engine emission reduction technologies to OHV engines is subject to both technical and economic considerations.

While substantial emissions reductions were achieved using these technologies, care must be taken in their application. For example, while retarding injection timing was found to be effective in reducing diesel NO_X emissions, it did so at the expense of higher particulate emissions and fuel consumption. The effects of engine design changes on emissions, fuel consumption, performance, and durability are highly interrelated. A careful, systems approach must be taken to design modification and emission reduction technology application to achieve the desired benefits.

II. RECOMMENDATIONS

Very little emissions data was found for:

- Off-highway SI utility engines
- Off-road motorcycles
- ATVs
- Snowmobiles.

More data is needed for each of these types of equipment to adequately define their baseline emissions.

Recreational equipment was found to be highly specialized in design. While off-road motorcycles and ATVs may be able to take advantage of on-road technology, no such analog exists for snowmobiles. A demonstration program to evaluate the feasibility of emissions reduction is recommended, prior to establishing regulations for snowmobiles.

Additional work is recommended to determine whether the revised C2 cycle, which was proposed by EMA and ITA, is appropriate for all SI engines in this diverse category. The advantages of numerical consistency of standards in the utility, OHV, and off-road heavy-duty diesel categories must be taken into consideration in the resolution of this test cycle question. For example, if the C2 cycle <u>does</u> tend to give higher emission results than the G1 cycle, this "offset" would need to be accounted for in establishing C2 cycle based OHV standards that are consistent with utility engine standards.

For OHV SI engines, a closed-loop control system, in conjunction with a three-way catalyst, is recommended as the most advanced technology for consideration for future demonstration programs. This type of system achieved extremely low emissions when applied to the 60 HP SI utility engine in this program.

III. BACKGROUND - CATEGORY DEFINITION

A project kickoff meeting was held at ARB's offices in El Monte, CA on January 14, 1992. Representatives from ARB, SwRI, and Booz-Allen and Hamilton were in attendance. The off-highway vehicle (OHV) category was extensively redefined since our proposal was submitted. Originally, the OHV category was defined to include gasoline- and diesel-fueled engines from 25-40 hp, plus snowmobiles. Since then, the category was redefined to include three separate subcategories:

- Preempted farm and construction equipment, which consists of equipment engines between 25 and 175 hp that are primarily used for commercial, farm, and construction applications. This equipment is preempted from state and local emission regulation by the Federal Clean Air Act.

- Off-highway utility equipment which utilizes gasoline and alternatively-fueled Otto-cycle engines, 25 hp and greater, and diesel engines from 25 to less than 175 hp.

- Off-highway recreational equipment, which includes off-road motorcycles, all-terrain vehicles (ATVs), snowmobiles, golf cars, and similar equipment.

While final regulatory directions are not yet decided, it was agreed at the kickoff meeting to focus the attention of this study on the originally defined, lower-power end of this category, consisting of engines of roughly 25-50 hp. No fixed upper limit was chosen to avoid cutting off engines representative of the lower end of this category which may be just above the 50 hp level. Other studies have been done which address emissions issues for off-road equipment with 50 and greater horsepower. Energy and Environmental Analysis, Inc. (EEA) performed a study entitled, "Feasibility of Controlling Emissions from Off-Road Heavy-Duty Construction Equipment"(1)* for ARB in 1988 and 1989. This study addressed emissions, regulatory, and technology issues regarding equipment 50 hp and greater. More recently, a study entitled, "Off-Road Mobile Equipment Emission Inventory Estimate,"(2) was performed by Booz-Allen and Hamilton, Inc. for ARB, summarizing emission inventory estimates for selected California off-road mobile sources. Additionally, new emission data is being developed by EPA for higher-horsepower engines. Four heavy-duty diesel engines were tested by Southwest Research Institute (SwRI) for EPA in 1991, and two more were tested in 1992. Consistent with the above, SwRI has supported ARB's needs for information to the fullest extent possible within the scope of our proposal.

*Numbers in parentheses designate references at end of report.

IV. TASK 1 - TEST ENGINE RECOMMENDATIONS

Task 1 called for the development of recommendations for an engine testing matrix to provide baseline and controlled exhaust emission data for OHV engines. The focus of this task was to identify those engine models which represent a relatively high proportion of the population of this class for which little emission test data exists. Two subtasks were defined to support recommendations for the engine testing matrix:

Subtask 1.1 - Establish a listing of all available emission test data.
Subtask 1.2 - Identify the proportion of total emissions contributed by various end products and associated engine types.

A. Subtask 1.1 - Listing of Available Emission Test Data

Much of the available emission data for OHV engines is quite old and of very limited usefulness. These data were summarized in two reports prepared for EPA by SwRI, entitled; "Nonroad Emission Factors,"(3) by M.N. Ingalls, February 1991; and "Nonroad Source Emissions - Reference Information,"(4) by Charles M. Urban, August 1991. Emission factors were initially developed in a series of studies conducted by SwRI for the EPA in 1972 and 1973. Since that time, numerous studies have analyzed or redeveloped emission factors, occasionally without incorporating any additional original emissions data. Primary sources of emissions data for the OHV category are listed in Table 1. These data have been summarized according to source and application and are presented in Tables 2 through 14.

Many manufacturers have shared emission data with ARB and EPA in private meetings and workshops, in support of company or industry positions on regulation. These data are confidential and not available to SwRI for summary in this report, but do constitute one of the best sources of current emission data on OHV engines.

In 1980, the Farm and Industrial Equipment Institute, Engine Manufacturers Association, and Construction Industry Manufacturers Association canvassed their members for emissions data. These data were compiled and averaged, and reported to ARB by EMA as emission factors. These data, as well as emission factors from EPA document 460/3-81-005 (AP-42) are reported and further analyzed in, "Feasibility, Cost, and Air Quality Impact of Potential Emission Control Requirements on Farm, Construction, and Industrial Equipment in California,"(5) Document PA 841, Environmental Research and Technology, Inc., May 1987, (CAL/ERT). Few new emission measurements on farm equipment have been found since the factors reported in the CAL/ERT report. These factors are summarized in Table 2 on a brake specific (g/bhp-hr) basis. The CAL/ERT report also estimated individual equipment usage factors (expressed as percent of total category energy output) for the purpose of generating a single, usage-weighted emission factor for this category.

Emission factor data for construction, mining, and forestry equipment are summarized in Table 3. As was the case for agricultural equipment, few new emissions measurements were found since the CAL/ERT study. The data presented in Table 3 are from the fourth edition of AP-42(6) as updated with new data from the CAL/ERT study. The data are again presented as both individual equipment brake specific factors, and as a category usage (energy output) weighted average emission factor.

TABLE 1. SOURCES OF OHV ENGINE EMISSION FACTORS AND DATA

OHV Engines - Agricultural, Construction, and Utility Equipment
"Farm And Industrial Tractors--Emission Trends and Their Impact," SAE Paper 730829, G.C. Hardwick and C.R. Hudson, September 1973.
"Emission Characteristics of Small Industrial Engines," SAE Paper 730857, J.B. O'Sullivan, W.A. Summerson, and J.A. Russell, September 1973.
"Exhaust Emissions from Uncontrolled Vehicles and Related Equipment Using Internal Combustion Engines, Final Report-Part 5, Heavy-Duty Farm, Construction, and Industrial Engines," C.T. Hare and K.J. Springer, October 1973.
"Compilation of Air Pollutant Emission Factors," AP-42, Volume II, Part II, Environmental Protection Agency, January 1975.
"Exhaust Emissions from Farm, Construction, and Industrial Engines and Their Impact," SAE Paper 750788, C.T. Hare, et al, September 1975.
"Feasibility, Cost, and Air Quality Impact of Potential Emission Control Requirements on Farm, Construction, and Industrial Equipment in California," Document PA841, Environmental Research & Technology, May 1982.
"Clean-Burning Diesel Engines," Interim Report AFLRL No. 169, Southwest Research Institute, H.E. Dietzmann, August 1983.
"Clean-Burning Diesel Engines--Phase II," Interim Report AFLRL No. 178, Southwest Research Institute, H.E. Dietzmann, December 1984.
"Clean-Burning Diesel Engines--Phase III," Interim Report BFLRF No. 215, Southwest Research Institute, H.E. Dietzmann and L.R. Smith, March 1986.
"Feasibility of Controlling Emissions From Off-Road Heavy-Duty Construction Equipment," Energy and Environmental Analysis, Inc., December 1988.
"Nonroad Engine and Vehicle Emissions Study - Report," US EPA, November 1991.
Recreational Equipment Engines
"Small Engine Exhaust Emissions and Air Quality in the United States," SAE Paper 720198, J.A. Donahue, G.C. Hardwick, H.K. Newhall, K.S. Sanvordenker, and N.C. Woelffer, 1972.
"Exhaust Emissions from Uncontrolled Vehicles and Related Equipment Using Internal Combustion Engines, Final Report-Part 3, Motorcycles," Southwest Research Institute, C.T. Hare and K.J. Springer, May 1973.
"Exhaust Emissions from Uncontrolled Vehicles and Related Equipment Using Internal Combustion Engines, Final Report-Part 7, Snowmobiles," Southwest Research Institute, C.T. Hare and K.J. Springer, April 1974.
"Motorcycle Emission Control Demonstration," Report EPA-460/3-77-020 by T.L. Ullman and C.T. Hare, December 1977.
"Particulate Measurement Motorcycle Test Results," EPA Report MC78-01 by E. Danielson, February 1978.

TABLE 2. AGRICULTURAL EQUIPMENT EMISSION FACTORS

	% Energy Output	g/bhp-hr			Weighted, g/bhp-hr		
		HC	CO	NO_x	HC	CO	NO_x
Diesel							
Tractor, 2WD 100+ hp	33.0	1.84	4.23	11.59	0.607	1.40	3.82
Tractor, 4WD	29.5	0.89	3.28	10.98	0.263	0.97	3.24
Tractor, 2WD, 20-90 hp	22.0	2.16	6.42	10.94	0.475	1.41	2.41
Combines, Self-Propelled	5.8	1.90	3.25	13.36	0.110	0.19	0.77
Windrower, Self-Propelled	4.3	2.21	6.85	10.50	0.095	0.29	0.45
Forage Harvester Sweet Corn Harvester	2.0	0.96	2.84	9.98	0.019	0.06	0.20
Balers, Self-Propelled Cotton Pickers Cotton Strippers Orchard Sprayers	1.7	2.23	3.78	7.78	0.038	0.06	0.13
Mower Conditioner Compact Loaders	1.7	1.13	4.29	9.69	0.019	0.07	0.16
Average Diesel					1.63	4.4	11.2
Gasoline							
Average					2.8	163	7.8
Source: CAL/ERT Report							

TABLE 3. CONSTRUCTION, MINING, AND FORESTRY EMISSION FACTORS

	% Energy Output	g/bhp-hr			Weighted, g/bhp-hr		
		HC	CO	NO_x	HC	CO	NO_x
Diesel							
Track Type Tractor, 90+ hp	21.82	0.37	1.65	6.60	0.081	0.360	1.440
Wheel Load 2-1/2 cu. yd.	13.32	0.60	2.07	8.31	0.080	0.276	1.107
Ind. Wheel Tractor	11.37	1.76	7.34	11.91	0.200	0.835	1.354
Wheel Tractor Scraper	10.79	0.55	2.45	7.46	0.059	0.264	0.805
Log Skidder	6.31	0.61	3.18	9.82	0.038	0.201	0.620
Off-Highway Trucks	5.84	0.37	2.28	8.15	0.022	0.133	0.476
Motor Graders	5.59	0.36	1.54	7.14	0.020	0.086	0.399
Hydraulic Excavator, Crawler	4.94	1.22	3.18	11.01	0.060	0.157	0.544
Trencher, Concrete Paver, Compact Loader	3.89	1.10	4.57	10.02	0.043	0.178	0.390
Wheel Loader 2-1/2 cu. yd	3.60	1.29	3.26	9.24	0.046	0.117	0.333
Track Type Loader, 90+ hp	2.81	0.47	1.56	7.76	0.013	0.044	0.218
Track Type Tractor 20-89 hp	2.21	1.33	2.91	9.63	0.029	0.064	0.213
Track Type Loader, 20-89 hp	1.97	1.80	3.02	10.97	0.035	0.059	0.216
Roller Compactor, Static	1.24	0.88	5.33	11.84	0.011	0.066	0.147
Crane Lattice Boom, Wheel & Crawler	1.04	0.59	4.99	12.45	0.006	0.052	0.129
Crane, Hydraulic, Wheel, One Station	0.97	0.80	7.80	14.69	0.008	0.076	0.142
Hydraulic Excavator, Wheel	0.96	1.22	3.18	11.01	0.012	0.031	0.106
Roller Compactor, Vibratory	0.68	1.06	6.72	14.27	0.007	0.046	0.097
Crane, Hyd. Wheel, Multi-Station	0.41	0.68	3.71	12.47	0.003	0.015	0.051
Bituminous Paver	0.24	0.99	5.19	11.18	0.002	0.012	0.027
Total Diesel					0.78	3.07	8.81
Average Gasoline					2.8	163	7.8
Source: AP-42							

A more recent study, "Feasibility of Controlling Emissions from Off-Road, Heavy-Duty Construction Equipment,"(1) by Energy and Environmental Analysis, Inc., reports emission data provided by engine manufacturers. Most of these data are from on-highway heavy-duty diesel engines which are probably <u>not</u> representative of OHV category engine emissions. A list of emission factors by model year for heavy-duty construction equipment engines was generated by EEA and is presented in Table 4. The methodology used in deriving these factors is not explained.

A three phase study(7,8,9) completed by SwRI in March 1986 for the U.S. Army Belvoir Research and Development Center determined emissions from seven diesel forklift engines using the 13-mode test procedure. These data are summarized in Table 5. Several different fuels were used in this study, and emission rates were determined for a number of unregulated species in addition to HC, CO, and NO_x. Particulate emission rates were determined at six of the 13 modes. These are presented in Table 6. This study is a good source of reasonably current data on diesel forklift emissions.

Emission data on snowmobiles are primarily from the 1974 SwRI report prepared for the EPA entitled, "Exhaust Emissions from Uncontrolled Vehicles and Related Equipment Using Internal Combustion Engines - Part 7, Snowmobiles,"(10) by C.T. Hare and K.J. Springer. This study mapped emissions from four different snowmobile engines using steady-state conditions. Cycle composite brake specific emissions, based on 19-22 steady-state modes and engine-specific weighting factors, are presented in Table 7. These data are supplemented with an emission factor(11) provided to ARB by ISIA which was determined using the same test procedure as SwRI. This factor is an average derived from tests of eight 1986 MY snowmobile engines, two each from the four snowmobile engine manufacturers. There are considerable differences between the SwRI and the ISIA data. The ISIA data seem more credible in light of other recently generated two-stroke engine emissions data.

Very little recent emissions data exists on off-road motorcycles. Emission results from seven motorcycles were reported by SwRI in a 1973 study for the EPA entitled, "Exhaust Emissions from Uncontrolled Vehicles and Related Equipment Using Internal Combustion Engines - Part 3, Motorcycles."(12) These results are summarized in Table 8.

A study performed by SwRI for the EPA in 1977 entitled, "Motorcycle Emission Control Demonstration"(13) determined baseline emissions on 10 motorcycles including 2-stroke, 4-stroke, and rotary engine types. Two FTPs and two FETs were run with each motorcycle. Standard motorcycle FTP traces were used for Class II and III motorcycles. The Class I motorcycle trace was used for Class I motorcycles. The standard FET was used for all motorcycles. These data are presented in Table 9.

A study entitled, "Particulate Measurement Motorcycle Test Results"(14) was performed by E. Danielson for the EPA in 1978. These data are presented in Table 10. Gaseous and particulate emission results were reported for two motorcycles. The most recent motorcycle emission factors are those published by ARB in its, "Proposal to Control Emissions from Off-Road Motorcycles," (Mail-Out #90-58).(15) These are presented in Table 11 and appear to have been derived from a 1974 study performed by SwRI for the EPA entitled, "Methodology for Estimating Emissions from Off-Highway Mobile Sources for the RAPS Program."(16)

TABLE 4. HEAVY-DUTY CONSTRUCTION EQUIPMENT ENGINE EMISSION FACTORS BY MODEL YEAR (g/bhp-hr)

Model Year	HC	NO_x	Particulate
Pre-1970	1.8	14.0	0.7
1970-1974	1.5	13.0	0.6
1975-1979	1.4	12.0	0.5
1980-1984	1.3	11.0	0.5
1985-1990	1.2	11.0	0.5
(No control) 1991+	1.1	11.0	0.5
Source: EEA			

TABLE 5. SUMMARY OF 13-MODE EMISSION RESULTS DIESEL FORKLIFT ENGINES

Engine Description	Rated HP	Fuel Type	Emission Rate, g/hp-hr		
			HC	CO	NO_x
Deutz F3L 912W	48	MiL. Ref.	0.454	1.627	4.445
Deutz F3L 912W	48	Mil. Hi S	0.335	1.472	4.684
Deutz F3L 912W	48	EPA Cert.	0.385	1.851	4.679
Perkins 4.2032	59	Mil. Ref.	3.205	5.438	5.361
Perkins 4.2032	59	EPA Cert.	2.740	4.879	6.406
Deutz F4L 912W	59	EPA Cert.	0.409	1.691	4.302
Perkins 4.2482	80	EPA Cert.	1.060	2.396	8.647
Isuzu C-240	43	EPA Cert.	0.506	3.769	3.167
Teledyne TMD-20	38	EPA Cert.	0.380	2.948	2.966
Peugeot XD3P	48	EPA Cert.	0.527	2.240	1.856
Source: SwRI/BFLRF report					

TABLE 6. SUMMARY OF PARTICULATE EMISSION RATES FROM DIESEL FORKLIFT ENGINES OPERATING ON EPA CERT. FUEL (g/hp-hr)

Engine Speed, rpm	Engine Load, %	Deutz F3L 912W	Perkins 4.2032	Perkins 4.2482	Deutz F4L 912W	Isuzu C-240	Teledyne TMD-20	Peugeot XD3P
Idle	2	*	*	*	*	NA	NA	NA
Peak Torque	2	5.47	16.09	6.79	8.98	7.96	7.90	18.6
Peak Torque	25	0.60	1.00	0.51	0.72	0.81	0.51	0.95
Rated	2	19.26	23.37	4.89	12.30	11.49	15.1	29.2
Rated	50	0.53	0.88	0.24	0.34	0.48	0.23	0.44
Rated	100	0.11	0.62	0.18	0.12	1.22	1.86	0.51

* Not reported due to low horsepower at idle.
Source: SwRI/BFLRF Report.

TABLE 7. SNOWMOBILE ENGINE EMISSIONS DATA

Snowmobile Engine	Emissions, g/hp-hr		
	HC	CO	NO_x
Arctic 440	89	142	1.43
Arctic 440 (rich)	110	270	0.97
Polaris 335	118	235	1.81
Rotax 248	196	63	3.36
OMC 528 Rotary	21	356	3.01
Source: SwRI			

ISIA Composite Snowmobile Emission Factor

Emissions, g/hp-hr	
$HC+NO_x$	CO
152	395

TABLE 8. SUMMARY OF MOTORCYCLE EMISSION RESULTS USING 1972 FTP

Motorcycle	HC, g/mi	CO, g/mi	NO_x as NO_2, g/mi
Harley-Davidson FLH	5.55	76.9	0.125
Honda CL350K3	4.04	46.5	0.0523
Honda SL100	2.04	22.1	0.331
Kawasaki 125F-6	9.78	7.72	0.163
Suzuki T250	20.7	34.8	0.0380
Triumph T120R	5.42	46.1	0.107
Yamaha DT1-E	16.5	26.3	0.0445
Source: SwRI			

TABLE 9. SUMMARY OF MOTORCYCLE BASELINE EMISSION TEST RESULTS

Motorcycle	Actual Displ., cm3	1978 HC Standard, g/km	Run	FTP Emissions, g/km HC	FTP CO	FTP CO_2	FTP NO_x	FTP Fuel km/l	FET Emissions, g/km HC	FET CO	FET CO_2	FET NO_x	FET Fuel km/l	WOT Accel. 0-100 kph, seconds
Suzuki GT-750	738	14.	1	14.08	12.10	71.29	0.03	17.4	7.58	8.27	67.63	0.04	22.25	6.4
3 cyl. 2-s			2	14.16	12.04	73.45	0.03	17.1	8.55	9.58	66.37	0.03	21.46	
Kawasaki KH-500	498	10.	1	19.03	25.95	56.95	0.02	14.9	9.85	25.40	50.45	0.02	19.15	6.7
3 cyl. 2-s			2	19.49	26.54	54.83	0.02	14.9	10.96	27.29	49.86	0.02	18.26	
Yamaha RD-400C	398	8.5	1	10.21	12.64	59.29	0.02	21.1	4.95	12.15	54.83	0.03	25.99	7.6
2 cyl. 2-s			2	10.42	12.60	57.17	0.03	21.4	4.63	9.29	54.44	0.05	27.81	
Kawasaki KE-175	174	5.1	1	7.44	23.89	35.69	0.02	24.4	6.71	35.73	34.42	0.03	20.82	16.0
1 cyl. 2-s			2	7.51	24.42	36.31	0.02	24.0	6.25	33.75	32.50	0.03	22.10	
Suzuki TS-100	98	5.0	1	7.15	13.96	37.87	0.03	28.6	6.84	24.45	37.52	0.04	23.85	21.4[a]
1 cyl. 2-s			2	7.03	12.41	37.65	0.03	29.7	8.04	23.96	36.14	0.04	23.45	
Honda GL-1000	999	14.	1	2.73	12.20	118.45	0.39	16.1	0.64	4.06	87.92	0.55	24.17	5.6
4 cyl. 4-s			2	2.84	10.26	108.38	0.36	17.6	0.64	4.35	86.89	0.55	24.31	
Kawasaki KZ-900	903	14.	1	3.35	29.47	77.27	0.13	17.6	0.83	20.21	60.63	0.28	24.50	5.5
4 cyl. 4-s			2	3.20	27.10	69.02	0.16	19.4	0.84	20.03	56.35	0.25	25.73	
Honda CB-360T	356	7.9	1	2.34	29.75	45.58	0.09	23.7	0.73	22.54	43.86	0.17	28.54	9.0
2 cyl. 4-s			2	2.81	29.21	37.30	0.08	25.7	0.70	22.26	40.25	0.16	30.06	
Honda XL-125	124	5.0	1	0.71	10.60	37.03	0.13	42.2	0.44	8.58	48.13	0.96	36.96	32.2
1 cyl. 4-s			2	0.85	12.61	33.27	0.13	42.3	0.47	7.71	48.06	0.93	37.75	10.2[a]
Suzuki RE-5	497	10.	1	6.37	21.97	142.04	0.22	12.0	2.05	6.82	107.94	0.36	18.60	6.8
rotary			2	6.31	23.46	134.53	0.19	12.3	2.06	6.94	103.03	0.38	19.33	
1978 standards				b	17.	-----	----							
1980 standards				5.0	12.	-----	----							
Statutory standards (LDV)				0.25	2.1	-----	0.62							

[a] WOT acceleration time for 0-80 kph
[b] See third column of table
Source: SwRI/EPA Report

TABLE 10. SUMMARY OF MOTORCYCLE PARTICULATE TEST RESULTS

	HC, g/km	CO, g/km	Particulate, g/km		
			Weighted	Cold Start	Hot Start
KAWASAKI KE-100					
EPA #1 Test	5.5	20	0.320	0.314	0.325
EPA #2 Test	5.6	21	0.353	0.365	0.344
YAMAHA DT-100					
EPA #1 Test	3.6	10	0.068	0.088	0.053
EPA #2 Test	4.0	11	0.095	0.108	0.086
Source: Danielson/EPA Report					

TABLE 11. ARB MOTORCYCLE EMISSION FACTORS (g/mile)

Vehicle Type	Engine Type	Emission Type		
		HC	CO	NO_x
On-Road Motorcycles	4-stroke	2.12	13.0	1.06
Off-Road Motorcycles	2-stroke	24.0	32.0	0.06
Off-Road Motorcycles	4-stroke	4.0	39.0	0.36
Source: ARB Mail Out #90-58				

The most recent compilation of emission factor information is reported in EPA's, "Nonroad Engine and Vehicle Emissions Study - Report,"[17] dated November 1991. This report contains a very extensive listing of emission factors and supporting information which is incorporated by reference. EPA selected emission factors from the following sources:

Agricultural and Construction Equipment	
- Diesel	CAL/ERT and EMA
- Gasoline	AP-42
Industrial Equipment	AP-42
Light Commercial Equipment <50 hp	
- Diesel	Radian refrigeration
- Gasoline	CARB Utility and Lawn & Garden TSD
Off-Road Motorcycles	CARB proposal
Snowmobiles	AP-42 and ISIA

An additional list of emission factors for diesel construction and agricultural equipment was submitted to EPA by the Engine Manufacturers Association (EMA). They are presented in Table 12. These factors are population weighted based on data obtained by individual manufacturers using the 8-mode version of the ISO 8178 test procedure. Unfortunately, emission factors for particulate matter were not reported. A comparison of EMA emission factors with those from AP-42 (CAL/ERT) is presented in Table 13.

The EPA report also presents test results from a recent joint EPA/Industry program conducted to assess test cycles for nonroad equipment. These data are presented in Table 14. It is noted that the particulate emission rates from these four 1991 diesel nonroad engines are considerably lower than the emission factors reported in AP-42.

The EPA report noted deficiencies in emission factor information in:

- Recent particulate matter data for diesel engines powering agricultural and construction equipment
- Emissions data on gasoline engines powering construction and agricultural equipment
- Emissions data for LPG-powered industrial equipment
- Emissions data on gasoline- and LPG-powered light commercial equipment (<50 hp)
- Emissions data on snowmobiles.

TABLE 12. ENGINE MANUFACTURERS ASSOCIATION NONROAD DIESEL-POWERED EQUIPMENT EMISSION FACTORS

Equipment Category	Engine Population Weighted Emissions, g/bhp-hr		
	NO_x	HC	CO
Crawler Tractor	10.3	0.9	2.4
Crawler Loader	10.0	0.6	2.4
Wheel Loader	10.3	0.6	2.4
Scraper	8.7	0.5	2.5
Motor Grader	9.6	1.1	1.9
Dumper	8.1	0.6	1.4
Crawler Excavator	10.5	0.6	2.5
Wheel Excavator	11.0	0.4	2.8
Backhoe Loader	10.1	1.0	3.4
Skid Steer Loader	9.6	1.5	4.5
Log Skidder	11.3	0.6	2.6
Crane	10.3	0.9	2.1
Roller and Compactor	9.3	0.8	3.1
Paver	10.3	0.6	3.2
Farm Tractor	10.5	0.7	3.2
Grain Combine	11.5	0.9	2.1
Cotton Picker	12.0	0.5	2.2

Note: All manufacturers' data collected using the 8-Mode Emissions Test Cycle and Weighting Factors (ISO-8178).
Source: EMA/EPA NR Report

TABLE 13. COMPARISON OF AP-42 (CAL/ERT) AND EMA CONSTRUCTION EQUIPMENT DIESEL EMISSION FACTORS (g/hp-hr)

AP-42	EMA	HC		CO		NO_x	
		AP-42	EMA	AP-42	EMA	AP-42	EMA
Tracked tractors	Crawler tractor	0.75	0.9	2.15	2.4	7.81	10.3
Tracked loaders	Crawler loader	1.11	0.6	2.26	2.4	9.3	10
Motor graders	Motor grader	0.36	1.1	1.54	1.9	7.14	9.6
Scrapers	Scraper	0.55	0.5	2.45	2.5	7.46	8.7
Off-highway trucks Pavement cold planers Wheel dozers	Dumper	0.37	0.6	2.28	1.4	8.15	9.6
Wheeled loaders	Wheel loader	0.97	0.6	2.71	2.4	8.81	10.3
Wheeled tractors		1.76		7.34		11.91	
Rollers	Roller & compactor	0.97	0.8	6.03	3.1	13.05	9.3
Wheeled dozers		0.37		2.28		8.15	
Miscellaneous		1.01		4.6		11.01	
Log skidders	Log skidders	0.61	0.6	3.18	2.6	9.82	11.3
Hyd. excav./crawlers	Crawler excavator	1.22	0.6	3.18	2.5	11.01	10.5
Trenchers		1.1		4.57		10.02	
Concrete pavers		1.1		4.57		10.02	
Compact loaders	Backhoe loaders	1.1	1	4.57	3.4	10.02	10.1
	Skid steer loader		1.5		4.5		9.6
Crane lattice booms		0.59		4.99		12.45	
Cranes	Crane	0.8	0.9	7.8	2.1	14.69	10.3
Hyd. excav. wheels	Wheel excavator	1.22	0.4	3.18	2.8	11.01	11
Bituminous pavers	Paver	0.99	0.6	5.19	3.2	11.18	10.3

Source: EPA NR Report

TABLE 14. RESULTS OF EPA/INDUSTRY TEST CYCLE EVALUATION PROGRAM
1991 NONROAD VERSION ENGINES

Engine	HC, g/hp-hr		CO, g/hp-hr		NO_x, g/hp-hr		PM, g/hp-hr	
	FTP	8-Mode	FTP	8-Mode	FTP	8-Mode	FTP	8-Mode
100 hp	1.08	0.8	2.7	2.2	12.14	11.1	0.59	0.41
139 hp	0.86	0.48	3.61	3.07	10.81	11.67	0.4	0.44
285 hp	1.81	1.21	5.06	1.49	6.55	6.5	0.58	0.2
450 hp	0.38	0.36	3.81	0.8	11.18	12.1	0.26	0.12
Avg.	1.03	0.71	3.79	1.89	10.17	10.34	0.46	0.29
Avg. FTP/ Avg. 8-Mode		1.4		2.0		1.0		1.6
Source: EPA NR Report								

B. Subtask 1.2 - Identify Emissions Contribution by Engine Type and Application

As a first step in the analysis of OHV equipment, work was done to identify manufacturers and engines in this category. Attention was directed primarily toward equipment in the 25-50 hp range since equipment of higher horsepower ratings has been addressed in other studies. Information about engine models and typical applications was compiled from product literature, and from telephone interviews with manufacturer associations (EMA, ISIA, and ITA) and manufacturer representatives. More than 30 manufacturers of engines in the 25-50 hp range were identified. This compilation is included in Appendix A. Information was collected into three groups:

- General engine manufacturers
- Lift truck manufacturers and engines
- Recreational equipment and engines.

A large number of engine models exists powering equipment in construction, agricultural, industrial, mining, utility, and recreational applications. Please note that this is not a complete listing of <u>all</u> engines and manufacturers in this category, but that it is a partial listing which represents the diversity of the category.

Engines in this category can broadly be broken down in the following categories:

- Agricultural
- Construction
- Industrial
- Recreational

For the Agricultural, Construction, and Industrial segments, there does not appear to be any clear engine size or horsepower at which engine/equipment sales drop off and then pick back up at some higher horsepower, i.e., there is a continuum of product offerings and engine power ratings from about 15 hp up to about 500 hp for these categories. There does, however, appear to be a gradual shift in engine technology for diesel engines at about 70 to 80 horsepower. Starting at this horsepower range, diesel engines become available in turbocharged versions, and injection systems are more sophisticated and can include variable-advance timing mechanisms. Injection pressures are also generally higher, resulting in improved fuel efficiency.

Diesel engines below 50 hp used in construction and agricultural equipment are overwhelmingly dominated by Japanese manufacturers. Kubota and Yanmar appear to be the clear market leaders--particularly in the "wheeled" equipment segment. Mitsubishi and Perkins also offer engines in this hp range but sell fewer engines. Diesel engines used in "wheeled" or self-propelled equipment are dominated by water cooled, direct injected engines. Indirect injection engines are generally more prevalent in smaller equipment, closer to 25 hp than to 50 hp. (Air-cooled engines dominate the market under 25 hp).

The engines used for powering "transportable" equipment such as gensets, welders, compressors, pumps, and refrigeration units appear to have a somewhat different technology profile and a different engine supplier base than "wheeled" or self-propelled equipment (although there is considerable overlap). These applications have a higher share of IDI engines as well as a higher share of air-cooled (DI) engines than self-propelled equipment

Kubota is a major supplier of engines for stationary equipment in this horsepower range (as they are for self-propelled equipment), but are in competition with a different set of manufacturers than in the self-propelled market. Major suppliers for various "transportable" equipment include Lister-Petter, Deutz, Honda, Perkins, Teledyne, and others. Lister-Petter also offers a range of air-cooled engines with the majority of their market closer to 25 hp than to 50 hp. There are very few models (and sales) of indirect injection, air-cooled engines due to the rather severe cooling requirements for IDI versus DI engines.

Forklift truck applications are dominated by LPG engines with about 60 to 70 percent of the total market. Major suppliers include Toyota, Hercules, Ford, and Isuzu. Ground support equipment in the 25 to 50 hp range is split between diesel- and gasoline-powered engines. Ford is by far the market leader in gasoline engines for airport ground support equipment. Diesel equipment engine sales are split between Hercules, Continental, Perkins, and others.

Recreational equipment is exclusively gasoline-powered. Snowmobiles are virtually 100 percent two-strokes. ATVs are approximately 20 percent two-strokes, 80 percent four-stroke; and off-road motorcycles are about 30 percent two-stroke and 70 percent four-stroke.

In our kickoff meeting, ARB expressed interest in learning more about engines powering golf cars. In discussions with manufacturers, it was learned that Club Car, EZ-Go, and Yamaha (the "big three") dominate the golf car market. All of the major manufacturers have an awareness of the environmental considerations related to their products and are utilizing four-stroke, OHV engines. These engines are manufactured by:

Kawasaki HI	-	Club Car
Fuji HI	-	EZ-Go
Yamaha	-	Yamaha

These engines are all air-cooled utility-type engines rated at 8-9 hp.

Based on our interviews with engine and equipment manufacturers, and on data supplied by PSR, our preliminary estimates for emissions by equipment category and by fuel type are shown in Figures 1 through 4. Based on this inventory estimate, and an understanding of the types of engine technologies used in the various applications, our recommendations for an engine test matrix are shown in Table 15.

TABLE 15. RECOMMENDED ENGINE TEST MATRIX

Engine	Fuel	Cooling	Type	HP	Application
1	G	Air	2-stroke	~50	Snowmobile
2	G	Air	4-stroke, OHV	~9	Golf car
3	LPG	Water	4-stroke, OHV	Med.	Forklift
4	G	Water	4-stroke, OHV	Med.	Miscellaneous
5	G	Water	4-stroke, OHV	High	Industrial
6	G	Water	FBC w/cat.	Med.	High-tech Forklift
7	D	Air	DI	Low	Utility
8	D	Water	DI	Med.	Miscellaneous
9	D	Water	IDI	Med.	Miscellaneous
10	D	Water	DI	High	Miscellaneous

The snowmobile engine is recommended due to its high hydrocarbon contribution and due to the lack of emission data for this type of equipment. The golf car engine is included because ARB was directed to address this type of equipment in the OHV category. Four each of gasoline/LPG engines and diesel engines are recommended. While diesel engines dominate this category in population and usage, some recent emissions data is available on off-road diesel engines. Very little data exists for gasoline/LPG engines. Four SI engines, including an advanced technology engine and a higher horsepower industrial engine are therefore recommended. Air-cooled diesel engines are more typical of lower horsepower applications and are almost always direct-injected. An IDI vs. DI engine pair is recommended at a medium power output to compare these two designs. One more higher power DI diesel engine is recommended (possibly turbocharged) in view of the popularity of the fuel efficient DI design.

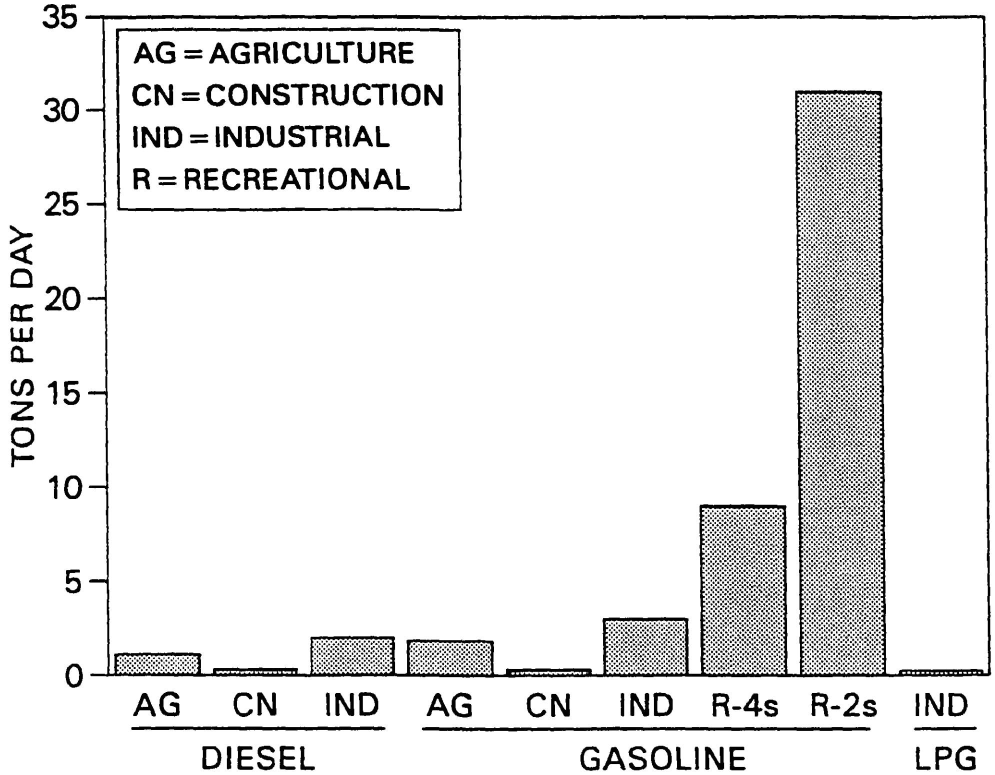


FIGURE 1. ANNUAL HC EMISSION INVENTORY
(TONS/DAY: 25 TO 40 HP RANGE)

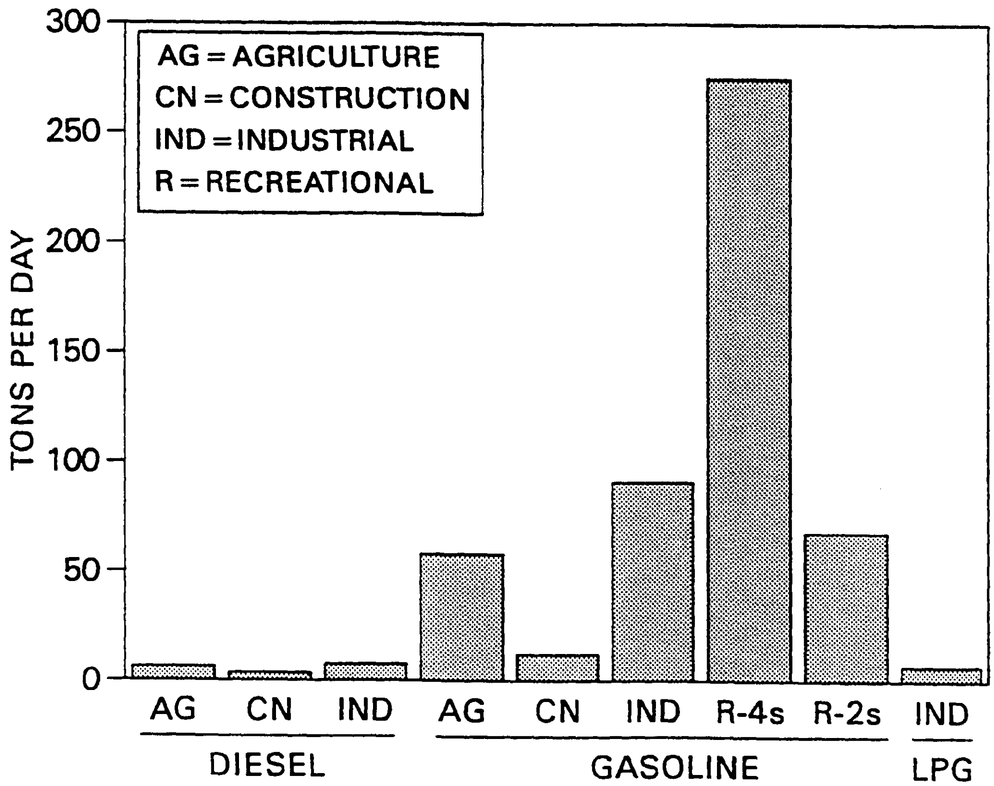


FIGURE 2. ANNUAL CO EMISSION INVENTORY
(TONS/DAY: 25 TO 40 HP RANGE)

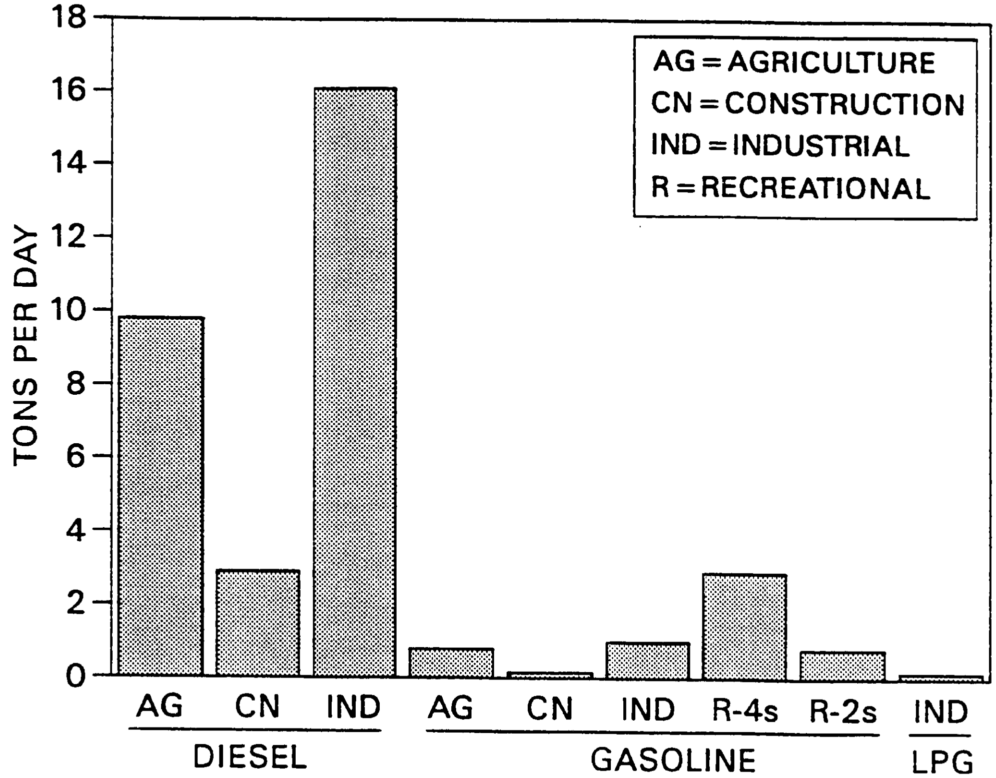


FIGURE 3. ANNUAL NOX EMISSION INVENTORY
(TONS/DAY: 25 TO 40 HP RANGE

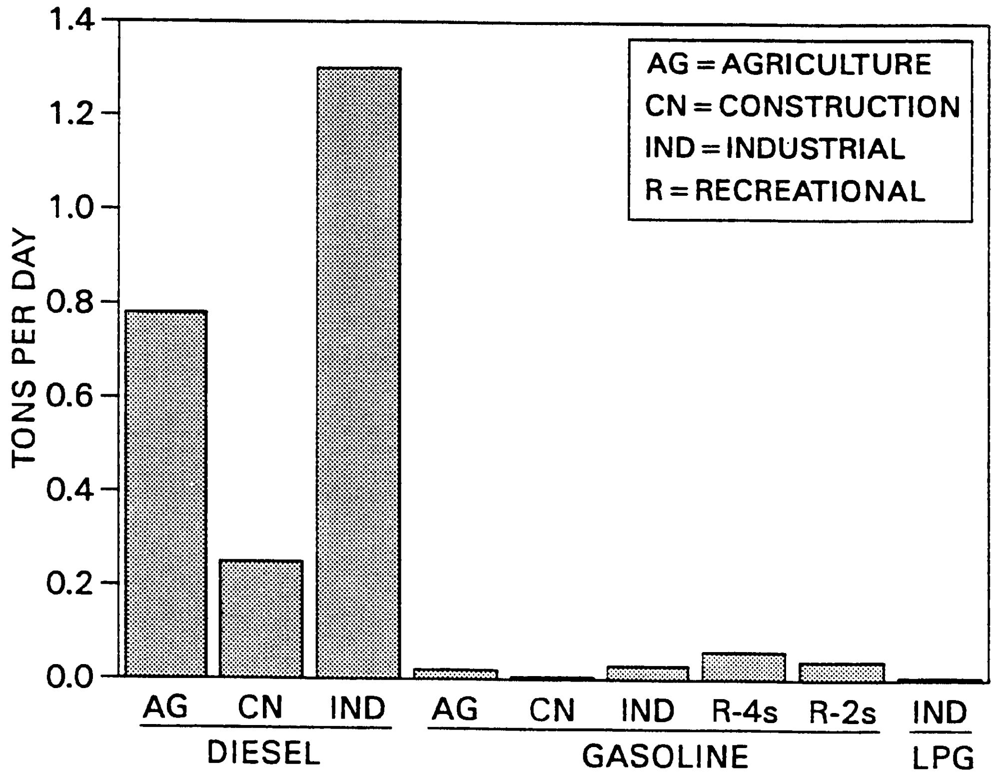


FIGURE 4. ANNUAL PM EMISSION INVENTORY
(TONS/DAY: 25 TO 40 HP RANGE)

Manufacturers were approached regarding their interest in loaning SwRI engines for this test program in exchange for test data. Several manufacturers expressed interest in this arrangement. Candidate engines approved by ARB for the test matrix are listed in Table 16. Unfortunately, Nissan and Mitsubishi later withdrew their engine offers. These were replaced by a Teledyne TM27 engine, operated in a baseline configuration, and then modified to a three-way catalyst/closed-loop control configuration.

TABLE 16. ARB-APPROVED CANDIDATE TEST ENGINES

Matrix No.	Manufacturer	Model	Cylinders	Type	Cooling	Power Output	Displacement
1	Rotax	503	2	2-stroke	Fan	50 hp @ 6800 rpm	497 cc
2	E Z Go/Fuji	FE290	2	OHV	Air	8.5 hp @ 3600 rpm	295 cc
3	Toyota	4Y-LP	4	OHV	Water	42@2100/53@2500	2.2 L
4	Nissan	H2O	4	OHV	Water	51 hp @ 2300 rpm	2.0 L
5	Ford	4.9L-G/LP/NG	6	OHV	Water	100 hp @ 2800 rpm	4.9L
6	Mitsubishi	4G64B	4		Water		
7	Lister-Petter	TS3	3	DI	Air	31 hp @ 2500 rpm	1.9L
8	Kubota	V2203DI-B	4	DI	Water	40 hp @ 2800 rpm	2197 cc
9	Lister-Petter	LPWS4	4	IDI	Water	38 hp @ 3600 rpm	1860 cc
10	John Deere	4045	4	DI/NA	Water	71 hp @ 2500 rpm	4.5 L

Nissan and Mitsubishi engine offers withdrawn.
Replaced by Teledyne TM27.

V. TASK 2 - EVALUATION AND DEVELOPMENT OF EXHAUST EMISSION TEST PROCEDURES

Emission test procedures were evaluated for this category. In its expanded form, the OHV category contains a very diverse group of equipment which, for various technical reasons, can probably not all be evaluated using the same test procedure. The following recommendations are made:

Equipment	Test Procedure
OHV Diesel, 25 - <175 hp	ISO 8178-C1 cycle
OHV Otto-cycle, ≥25 hp	ISO 8178-G1 or C2 cycle
Off-road motorcycles	On-road motorcycle FTP
ATVs	On-road motorcycle FTP
Snowmobiles	ISO 8178-G1 cycle

California recently adopted the 8-mode ISO 8178-C1 test procedure for new 1996 and later heavy-duty off-road diesel cycle engines. This procedure is derived from the old-13-mode certification test procedure and has found acceptance with ARB and EPA, as well as engine manufacturers. Diesel engines with less than 175 hp have essentially the same design features and operating characteristics as engines with greater than 175 hp. This procedure is clearly appropriate for OHV diesel engines. Although utility class (<25 hp) diesel engines will be required to use the ARB 6-mode J1088 type procedure, small diesel engine manufacturers are familiar with both procedures and occasionally use an SAE version (J1003) of the 13-mode test procedure. The 8-mode procedure is currently being revised by an EPA/Industry task force. ARB may wish to incorporate these revisions in its version of this procedure.

Numerous steady-state test cycles have been applied to spark-ignited (SI) engines over the years. Most of these are old and no longer in use, or they are highly application specific and not appropriate to a broad range of engine applications. Although it is an obvious candidate, the 8-mode ISO 8178-C1 procedure is not recommended for OHV SI (Otto cycle) engines. These engines more typically operate up through ~85 percent of their rated speed with momentary operation in the 85-100 percent speed range for peak loads. Gasoline-fueled SI engines are typically calibrated rich at rated speed to provide maximum power, and for cooler operation to promote engine durability. Engines with these types of calibrations generate very high levels of HC and especially CO, which are not representative of normal operation, when run at the rated speed modes of the 8-mode C1 cycle. The Federal heavy-duty transient test procedure is also not recommended. While it does do the best job of measuring transient emissions, the on-road transient cycle is not representative of off-road and industrial equipment operation. Additionally, the use of a transient cycle would impose a facilities cost burden on the many manufacturers of smaller OHV engines who do not have transient test facilities.

The six-mode procedure which ARB adopted for utility engines uses five modes at 85 percent of rated speed plus an idle mode, providing a load traverse at 85 percent of rated engine speed. While no single procedure is likely to model the many and diverse applications of OHV engines, a load traverse at a moderately high engine speed provides a good overall picture of SI engine emissions. ISO has incorporated this mode schedule in the 8178 procedure as test cycle G1 for "utility applications <25 hp." While some have expressed the concern that this is a "small engine" procedure, there is nothing limiting the use of these test modes to lower power engines. The use of a cycle defined under the ISO 8178 procedure has the additional advantages of providing calibration procedures, sampling options, and other desirable features. It also provides a measure of consistency between the procedure for SI engines and the 8-mode, C1 cycle procedure for diesel engines.

During this program, an ISO 8178 "revised C2" cycle was proposed by the Industrial Truck Association (ITA) and the Engine Manufacturers Association for application to OHV SI engines. This cycle was accepted by ISO as a replacement for the out-dated C2, "Light-Duty Industrial Engine Test Cycle." This cycle is based primarily on lift-truck type duty cycles, which may not be representative of some of the more highly loaded SI engine applications, such as pumps or generators. Program SI engines were emission tested using the 6-mode, G1 cycle, as directed by ARB. ARB has indicated a preference for the G1 (J1088 type) cycle due to its higher composite load factor, and its application to regulations covering utility and lawn and garden equipment, less than 25 hp. One program engine was tested using both this revised C2 cycle and the G1 cycle. These results are discussed in Section V of this report. A separate program sponsored by ITA compared emissions using these two cycles on three different lift truck engines. Results do vary depending upon the cycle used, however, more data would be required to fully address the questions of which cycle is more representative and appropriate for this category. In any event, this cycle has the strong support of both EMA and ITA, and is therefore recommended for consideration by ARB for OHV category SI engines.

It is reasonable to apply the on-road motorcycle procedure to off-road motorcycles, although off-road motorcycles are typically operated at higher acceleration and deceleration rates. ATVs can also be tested using this procedure, although dynamometer modifications may be required to enable ATVs to fit on motorcycle dynamometer rolls.

Snowmobiles are another interesting category of equipment. The test procedure used by SwRI in 1974 was a multi-mode mapping procedure which is not practical for use in certification. A number of the arguments made relative to the application of the 6-mode, G1 cycle procedure for gasoline engines apply with snowmobiles as well. The International Snowmobile Industry Association has indicated their acceptance of a 6-mode procedure derived from J1088 providing that a mixing chamber is not used and that dilute (CVS) sampling is allowed as an alternate. SwRI agrees with these provisions and recommends a 6-mode ISO 8178-G1 type procedure be used for snowmobiles.

Recommendations for sampling and analysis methods are as follows:

Total hydrocarbons (THC)

- heated flame ionization detector (HFID)

Reactive hydrocarbons (RHC)
- determine as (THC - methane)

Methane
- gas chromatograph with FID (SAE J1151)

Carbon monoxide
- non-dispersive infrared analyzer (NDIR)

Carbon dioxide
- non-dispersive infrared analyzer (NDIR)

Oxides of nitrogen
- chemiluminescence analyzer

Particulate matter
- dilute exhaust gas sampling and particulate filter weight gain (40 CFR 86 Subpart N)

Formaldehyde
- dinitrophenylhydrazone derivative method using liquid chromatograph with UV detector

Total engine effluent may be determine using measured engine intake air and fuel flows or using a carbon-base calculation based on fuel flow, depending on the provisions of the test procedure selected for use. Since most particulate matter is smaller than 1 micron in diameter, it is reasonable to assume that PM_{10} equals total PM. Particulate matter fractions at smaller diameters may be determined using a cascade impactor.

VI. TASKS 3 AND 5 - ENGINE BASELINE AND ADVANCED TECHNOLOGY EMISSIONS RESULTS

Baseline emissions of ten engines were determined, as required by Task 3, Determination of Baseline Emission Factors. To minimize test time, emission reduction technology evaluations required by Task 5 - Advanced Engine Testing, were performed in parallel with baseline evaluations. Engines and associated configurations tested are summarized in Table 17. Baseline emission results are summarized in Table 18.

Diesel engines were tested following the ISO 8178 test procedure using the C1 cycle. This procedure and cycle are widely used by diesel engine manufacturers, and have been adopted by California for regulation of new 1996 and later heavy-duty off-road diesel cycle engines and equipment engines. Spark-ignited engines were tested using the ISO 8178 G1 cycle. This is a six-mode procedure based on the SAE J1088 recommended practice, and was preferred by CARB due to its application in California regulations covering utility and lawn and garden equipment engines less than 25 hp. During this program, an ISO-8178 "revised C2" cycle was proposed by the Industrial Truck Association (ITA) and the Engine Manufacturers Association (EMA) for application to SI engines. This cycle was evaluated on one of the program engines for comparison with the G1 cycle. Program test cycles are described in Table 19.

Baseline and advanced technology results are discussed below according to individual engines.

A. Emission Test Results

1. <u>75 and 76 HP NA DI Diesel Engines</u>

Baseline emission testing of the two John Deere engines was performed under an EPA project. The objective of that project was to compare transient cycle and eight-mode cycle emissions from these engines. Both engines are naturally-aspirated DI diesel engines. The 4.5 L engine has been modified by Deere to reduce NO_X emissions and is considered the low emissions version of the two. Eight-mode emissions results (ISO 8178-C1) from these two engines are summarized in Table 20. Methane and RHC emissions were not determined for these engines under the EPA program. Results are generically labeled to provide confidentiality to the engine manufacturer. For ARB internal identification, the 75 HP DI Diesel is the 3.9 L John Deere engine, and the 76 HP DI Diesel is the 4.5 L John Deere engine. A complete summary of transient and eight-mode results, including comparisons between the two test procedures, is included in Appendix B. It is interesting to compare the relative improvement in emissions of the 4.5 L engine as compared to the 3.9 L engine. The eight-mode results show a 36 percent and a 16 percent reduction in PM and NO_X, respectively. The heavy-duty transient cycle is less sensitive to these improvements, showing only a 5 percent reduction in both PM and NO_X. While the 4.5 L engine is definitely cleaner than the 3.9 L engine, the magnitude of the improvement is highly dependent upon the test procedure.

TABLE 17. ENGINES AND CONFIGURATIONS TESTED

Engine Model	Type	Engine ID	Configurations Tested
John Deere 4045	4.5L NA DI Diesel	76 HP DI Diesel	Baseline
John Deere 4039	3.9L NA DI Diesel	75 HP DI Diesel	Baseline Retarded Timing Retard + Turbocharge
Lister-Petter TS3	1.9L AC DI Diesel	30 HP Air-Cooled Diesel	Baseline
Lister-Petter LPWS4	1.9L IDI Diesel	35 HP Water-Cooled Diesel	Baseline Retarded Timing
Kubota V2203DI-B	2.2L DI Diesel	50 HP Water-Cooled Diesel	Baseline Retarded Timing
Fuji FE290	0.3L AC OHV SI	8.5 HP Golf Car	Baseline
Toyota 4Y	2.2L LPG SI	40 HP LPG Lift Truck	Baseline EGR Baseline-EMA Cycle
Teledyne TM27	2.7L SI	60 HP Spark-Ignited Utility	Baseline (Gasoline) LPG-Fueled CNG-Fueled LPG/CLC/TWC
Ford	4.9L SI	100 HP Spark-Ignited Utility	Baseline
Rotax 503	0.5L 2-Stroke SI	50 HP Snowmobile	Baseline

TABLE 18. ENGINE BASELINE EMISSIONS

Engine ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
	THC	CH_4	RHC	CO	NO_x	PM	
Diesel Engines (ISO 8178-C1)							
76 HP DI Diesel	0.89	NA	NA	1.54	6.08	0.38	0.38
75 HP DI Diesel	0.64	NA	NA	3.51	7.24	0.59	0.38
30 HP Air-Cooled	2.72	0.03	2.69	5.50	8.50	1.40	0.48
35 HP Water-Cooled	1.10	0.03	1.07	2.64	3.24	0.51	0.51
50 HP Water-Cooled	1.74	0.02	1.72	5.61	7.27	0.58	0.38
Spark-Ignited Engines (ISO 8178-G1)							
8.5 HP Golf Car*	7.53	0.40	7.14	109	8.32	0.16	0.78
40 HP Lift Truck	0.90	0.03	0.87	2.87	16.9	0.02	0.50
60 HP SI Utility	2.02	0.34	1.68	126	8.30	0.02	0.61
60 HP CLC/TWC Utility	0.09	0.01	0.08	2.72	0.53	0.01	0.52
100 HP SI Utility	2.79	0.24	2.55	93.9	9.44	0.01	0.58
50 HP Snowmobile	96.7	1.13	95.5	347	0.33	3.11	1.01

*J1088 test cycle

TABLE 19. PROGRAM TEST CYCLES

Speed	Load, %	Mode Weighting Factors: ISO 8178 C1 8-Mode	ISO 8178 G1 Utility	ISO 8178 EMA/ITA Proposed C2	J1088
Rated (100%)	100	0.15			
Rated (100%)	75	0.15			
Rated (100%)	50	0.15			
Rated (100%)	25			0.06	
Rated (100%)	10	0.1			
Rated (100%)	2				
Int. (85%)	100		0.09		0.09
Int. (85%)	75		0.2		0.2
Int. (85%)	50		0.29		0.29
Int. (85%)	25		0.3		0.3
Int. (85%)	10		0.07		
Int. (85%)	min.				0.07
Int. (50-70%)	100	0.1		0.02	
Int. (50-70%)	75	0.1		0.05	
Int. (50-70%)	50	0.1		0.32	
Int. (50-70%)	25			0.3	
Int. (50-70%)	10			0.1	
Int. (50-70%)	2				
Idle	0	0.15	0.05	0.15	0.05
Composite Load Factor		57%	47%	32%	46%

TABLE 20. 75 HP AND 76 HP NA DI DIESEL ENGINES BASELINE ISO 8178-C1 EMISSION RESULTS

Test Date	Test No.	Max HP	Emissions (g/hp-hr)				BSFC lb/hp-hr
			BSHC	BSCO	$BSNO_X$	PM	
75 HP DI Diesel							
5/6/92	8-Mode #1	75	0.62	3.47	7.17	0.584	0.376
5/7/92	8-Mode #2	74	0.65	3.54	7.30	0.592	0.384
8-MODE AVERAGE			0.64	3.51	7.24	0.588	0.380
76 HP DI Diesel							
5/26/92	8-Mode #1	75	0.91	1.63	6.05	0.397	0.382
5/26/92	8-Mode #2	76	0.86	1.44	6.11	0.359	0.373
8-MODE AVERAGE			0.89	1.54	6.08	0.378	0.378
% Change, 76 HP vs 75 HP Engine			+39	-56	-16	-36	-1

Emission reduction technologies were applied to the 3.9 L John Deere engine for comparison with engine baseline emissions. First, injection timing was retarded 4.5°, and then a turbocharger was installed. Duplicate eight-mode emission tests were performed in each of these configurations (retarded timing and retarded timing with turbocharging). Eight-mode results and comparative differences between the three configurations are summarized in Table 21. Retarding the injection timing 4.5° reduced NO_X emissions by 36 percent relative to baseline emissions. As expected, HC and PM emissions increased following the timing change, by 66 percent and 49 percent, respectively. Application of the turbocharger in conjunction with retarded timing reduced HC, CO, and PM emissions by 38, 71, and 47 percent, respectively, and increased NO_X emissions by 21 percent, relative to emissions with retarded timing alone. The combination of retarded timing plus turbocharging reduced CO, NO_X, and PM emissions by 70, 23, and 21 percent, respectively, compared to baseline emissions.

Directionally, these results are consistent with other data on the effects of retarded timing and turbocharging on diesel engine emissions. The magnitude of specific reductions (or increases) is dependent on a number of factors and could vary considerably with different engine models, baseline configurations, and specific settings.

TABLE 21. 75 HP NA DI DIESEL ENGINE COMPARISON OF ISO 8178-C1 EMISSION RESULTS IN THREE CONFIGURATIONS

Test Date	Test No.	Max HP	Emissions (g/hp-hr)				BSFC lb/hp-hr
			HC	CO	NO_x	PM	
BASELINE (14.5° NO T/C)							
5/6/92	8-Mode #1	75	0.62	3.47	7.17	0.584	0.376
5/7/92	8-Mode #2	74	0.65	3.54	7.30	0.592	0.384
8-MODE AVERAGE			0.64	3.51	7.24	0.588	0.380
RETARD TIMING (10° NO T/C)							
6/3/92	8-Mode #3	72	1.07	3.84	4.73	0.881	0.376
6/4/92	8-Mode #4	74	1.05	3.39	4.54	0.871	0.372
8-MODE AVERAGE			1.06	3.62	4.64	0.876	0.374
% Change, Timing Change vs Baseline			+66	+3	-36	+49	-2
RETARD TIMING & TURBOCHARGE (10° + T/C)							
6/9/92	8-Mode #5	73	0.64	1.07	5.62	0.450	0.379
6/10/92	8-Mode #6	74	0.67	1.05	5.55	0.475	0.368
8-MODE AVERAGE			0.66	1.06	5.59	0.463	0.374
% Change, Timing Change & T/C vs Timing Change			-38	-71	+21	-47	0
% Change, Timing Change & T/C vs Baseline			+3	-70	-23	-21	-2

For purposes of test cycle comparison, two hot-transient tests were also run (at SwRI expense) in each of the three configurations. Transient cycle results are summarized in Table 22. It is significant that hot-transient results present a picture of technology effectiveness similar to that observed with eight-mode results. Although the specific results generated with the two procedures vary (more for HC and PM, less for CO and NO_X), the direction, and even the magnitude of the changes in emissions observed using the two test procedures are similar. Interestingly, changes in NO_X emissions resulting from the application of the two emission reduction technologies are observed to a greater degree in eight-mode results than in transient cycle results. The transient cycle procedure is typically assumed to be the more sensitive of the two test procedures. Detailed summaries plus copies of individual test results are included in Appendix B.

2. <u>30 HP Air-Cooled Diesel Engine</u>

Baseline emissions of the Lister-Petter TS3 air-cooled DI diesel engine were measured using the eight-mode ISO 8178-C1 cycle. Results are summarized in Table 23. Both hydrocarbon and particulate emissions are higher for this engine than other diesel engines in this category. This may be associated with its air-cooled design, which may employ larger cylinder clearances to accommodate the wider range of operating temperatures expected using air-cooling. Detailed results are included in Appendix C.

3. <u>35 HP Water-Cooled Diesel Engine</u>

Emissions of the Lister-Petter LPWS4 water-cooled diesel engine were also measured using the eight-mode ISO 8178-C1 cycle. Following baseline testing, engine injection timing was retarded 4° to reduce NO_X emissions and the engine was retested. Results are summarized in Table 24. Methane and aldehyde emissions were determined for the baseline configuration only.

Baseline emissions from the 35 HP water-cooled, IDI Lister-Petter diesel engine are considerably lower than from the air-cooled Lister-Petter engine. Both NO_X and PM emissions were well controlled at 3.24 and 0.51 g/hp-hr, respectively, which is consistent with Lister-Petter's comment that this is one of their low emissions designs. The minor NO_X reduction (9%) achieved by retarding the engine's timing suggests this engine has already been optimized for lower NO_X. As expected, retarded injection timing resulted in detrimental changes to other emissions, increasing THC by 100 percent and PM by 37 percent. Maximum engine power was reduced 10 to 15 percent by the timing change. Detailed results are included in Appendix D.

4. <u>50 HP Water-Cooled Diesel Engine</u>

Emissions of the Kubota V2203DI-B water-cooled diesel engine were measured using the eight-mode ISO 8178-C1 cycle. Following baseline testing, engine injection timing was retarded 6° (manufacturer's recommendation) to reduce NO_X emissions and the engine was retested. As with prior testing of other engines, methane and aldehyde analyses were performed for baseline configurations, only. Results are summarized in Table 25.

TABLE 22. 75 HP NA DI DIESEL ENGINE COMPARISON OF HOT TRANSIENT EMISSION RESULTS IN THREE CONFIGURATIONS

Test Date	Test No.	Max HP	Emissions (g/hp-hr)				BSFC lb/hp-hr
			HC	CO	NO_x	PM	
BASELINE (14.5° NO T/C)							
5/13/92	TH10	75	1.36	3.25	7.32	0.589	0.459
5/13/92	TH11	75	1.37	2.76	7.33	0.530	0.446
HOT-START AVERAGE			1.37	3.01	7.33	0.560	0.453
RETARD TIMING (10° NO T/C)							
6/3/92	TH13	74	2.17	3.07	5.15	0.692	0.453
6/4/92	TH14	74	2.17	3.08	5.09	0.675	0.458
HOT-START AVERAGE			2.17	3.08	5.12	0.684	0.456
% Change, Timing Change vs Baseline			+58	+2	-30	+22	+1
RETARD TIMING & TURBOCHARGE (10° + T/C)							
6/9/92	TH15	74	1.26	1.57	5.93	0.489	0.464
6/10/92	TH16	74	1.30	1.51	5.84	0.486	0.457
HOT-START AVERAGE			1.28	1.54	5.89	0.488	0.461
% Change, Timing Change & T/C vs Timing Change			-41	-50	+15	-29	+1
% Change, Timing Change & T/C vs Baseline			-7	-49	-20	-13	+2

TABLE 23. 30 HP AIR-COOLED DIESEL ENGINE ISO 8178-C1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
		THC	CH_4	RHC	CO	NO_x	PM	
7-29-92	ACD-1	2.59	0.03	2.57	5.37	8.40	1.36	0.47
7-29-92	ACD-2	2.85	0.03	2.83	5.63	8.59	1.43	0.48
	Mean	2.72	0.03	2.69	5.50	8.50	1.40	0.48

TABLE 24. 35 HP WATER-COOLED DIESEL ENGINE BASELINE AND 4° RETARDED TIMING CONFIGURATIONS ISO 8178-C1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
		THC	CH_4	RHC	CO	NO_x	PM	
Baseline								
8-11-92	WCD-1	1.16	0.03	1.13	2.57	3.23	0.54	0.51
8-13-92	WCD-2	1.03	0.03	1.00	2.70	3.24	0.46	0.50
Mean Baseline		1.10	0.03	1.07	2.64	3.24	0.51	0.51
4° Retarded Timing								
8-17-92	WCDR-1	2.39			2.83	2.99	0.81	0.53
8-18-92	WCDR-2	2.00			2.77	2.92	0.59	0.53
Mean Retarded		2.20			2.80	2.96	0.70	0.53
% Change, Timing Change vs. Baseline		+100			+6	-9	+37	+4

TABLE 25. 50 HP WATER-COOLED DIESEL ENGINE BASELINE AND 6° RETARDED TIMING CONFIGURATIONS ISO 8178-C1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
		THC	CH_4	RHC	CO	NO_x	PM	
Baseline								
8-31-92	50HPDI-1	1.81	0.02	1.79	5.49	7.50	0.57	0.38
9-1-92	50HPDI-2	1.66	0.02	1.64	5.73	7.04	0.59	0.38
Mean Baseline		1.74	0.02	1.72	5.61	7.27	0.58	0.38
6° Retarded Timing								
9-9-92	46HPDIR-1	1.94			5.65	5.44	0.77	0.40
9-10-92	46HPDIR-2	2.00			5.68	5.27	0.76	0.40
Mean Retarded		1.97			5.67	5.36	0.77	0.40
% Change, Timing Change vs. Baseline		+13			+1	-26	+33	+5

Baseline emissions from the 50 HP water-cooled, DI Kubota diesel engine are higher than from the 35 HP water-cooled IDI Lister-Petter engine. Kubota engine baseline NO_x was 7.27 g/hp-hr, which is over twice that of the IDI Lister-Petter engine. This is consistent with the higher peak combustion temperature associated with the DI design which produces more NO_x than the IDI design. The Kubota engine also emitted ~14 percent more PM than the IDI Lister-Petter engine, however, particulate formation is influenced by many factors and may not always be lower with the IDI design. The Kubota engine was much more fuel efficient than the Lister-Petter engine. Kubota engine BSFC was 0.38 g/hp-hr, which is 25 percent less than the IDI Lister-Petter engine. The DI design is well known for its fuel efficiency and typically preferred over the IDI design where fuel cost or consumption is a primary consideration.

Retarding the timing of the Kubota engine produced changes in emissions similar to those observed with the Lister-Petter engine. Retarded timing reduced NO_x by 26 percent to 5.36 g/hp-hr, however, PM increased by 33 percent to 0.77 g/hp-hr. BSFC also increased 5 percent and maximum engine power was reduced by the timing change. Thus, retarding injection timing is observed to provide a simple means of reducing NO_x emissions, but at the cost of increased particulate emissions and fuel consumption, and reduced engine power. Detailed results are included in Appendix E.

5. <u>8.5 HP Golf Car Engine</u>

The 8.5 hp air-cooled, OHV, 2-cylinder Fuji engine supplied by EZ-GO was installed on the 30 hp horizontal shaft dynamometer. Duplicate J1088 tests were run on the Fuji engine. The J1088 procedure was specified by CARB for this engine to enable a direct comparison with other utility engines less than 25 hp. Results are summarized in Table 26.

TABLE 26. 8.5 HP GOLF CAR ENGINE J1088 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
		THC	CH_4	RHC	CO	NO_X	PM	
6-10-92	Golf 1	7.62	0.40	7.22	109	8.37	0.16	0.78
6-11-92	Golf 2	7.44	0.39	7.05	109	8.27	0.15	0.77
	Mean	7.53	0.40	7.14	109	8.32	0.16	0.78

This engine is calibrated somewhat leaner than other >225 cc displacement utility engines previously tested, as evidenced by its lower CO and higher NO_X emissions. The engine's HC + NO_X emissions of 15.85 exceed ARB's 1994 utility engine standard of 10 g/hp-hr HC+NO_X. Oxides of nitrogen emissions could be reduced with the application of EGR. Hydrocarbon emissions could be reduced with a catalyst, although catalyst effectiveness would be limited by the on-off, short-cycle operation which is typical of golf cars. Copies of individual golf car engine test results are included in Appendix F.

6. <u>40 HP LPG Lift Truck Engine</u>

The 2.2 L, LPG-fueled Toyota lift truck engine was installed in the cell and coupled to the 175 hp dynamometer. Duplicate ISO 8178-G1 cycle tests were run on the engine. Results are summarized in Table 27.

TABLE 27. 40 HP LPG LIFT TRUCK ENGINE ISO 8178-G1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
		THC	CH_4	RHC	CO	NO_X	PM	
6-23-92	Lift 1	0.93	0.03	0.90	2.84	16.0	0.02	0.49
6-26-92	Lift 2	0.87	0.03	0.84	2.89	17.8	0.01	0.51
	Mean	0.90	0.03	0.87	2.87	16.9	0.02	0.50

The Toyota engine is calibrated for lean operation to minimize emissions and fuel consumption. Test mode air/fuel ratios ranged from 14.7 at idle to over 17, with values of 15 - 16 being typical. This, coupled with the good mixing qualities of a gaseous fuel, helps provide the low HC and CO emissions observed. Oxides of nitrogen emissions were correspondingly high, as expected for operation at these lean conditions. While a catalyst would certainly be effective at converting HC and CO in this exhaust gas environment, further reduction is hardly necessary since the HC and CO levels are already so low.

Following baseline testing, EGR was installed on the engine to reduce NO_X emissions. EGR was provided using a 3/8 in. diameter line connected between the exhaust system and the intake system below the carburetor. EGR level was manually set at each test mode to provide a moderate NO_X reduction with no more than a 10 percent loss in power. Results are summarized in Table 28. EGR reduced the lift truck engine's weighted total NO_X emissions by 71 percent to 4.9 g/hp-hr. HC and CO emissions increased by 77 percent and 46 percent, respectively, and BSFC increased by eight percent due to the comparatively high levels of EGR employed. These results should be viewed as an upper limit to the application of EGR, which is more typically calibrated to provide a 40 – 50 percent reduction in NO_X with commensurately smaller adverse increases in HC and CO emissions. Copies of these test results are included in Appendix G.

TABLE 28. 40 HP LPG LIFT TRUCK ENGINE ISO 8178-G1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)			BSFC lb/hp-hr
		THC	CO	NO_X	
Baseline					
06-23-92	Lift 1	0.93	2.84	16.0	0.49
06-26-92	Lift 2	0.87	2.89	17.8	0.51
	Mean	0.90	2.87	16.9	0.50
With EGR					
07-06-92	Lift 1 - EGR	1.59	4.47	5.0	0.54
07-09-92	Lift 2 - EGR	1.59	3.92	4.7	0.53
	Mean	1.59	4.20	4.9	0.54
% Change, EGR vs Baseline		+77	+46	-71	+8

Lift truck engine baseline emissions were also evaluated using the EMA/ITA proposed "Industrial SI Cycle" (7-2-92 version). This cycle is described in Table 29. Results are summarized in Table 30. Detailed results are included in Appendix G.

TABLE 29. EMA/ITA PROPOSED INDUSTRIAL SI CYCLE JULY 2, 1992 VERSION

Engine Speed	Torque, %	Weight Factor, %
Idle	0	15
Rated	25	6
Intermediate	100	2
Intermediate	75	5
Intermediate	50	32
Intermediate	25	30
Intermediate	10	10

TABLE 30. 40 HP LPG LIFT TRUCK ENGINE EMA/ITA PROPOSED INDUSTRIAL SI CYCLE EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)			BSFC lb/hp-hr
		THC	CO	NO_X	
ISO 8178-G1 Cycle					
8178-G1 Mean		0.90	2.87	16.9	0.50
EMA/ITA Proposed Cycle					
07-05-92	Lift 1 - EMA	1.30	6.38	14.4	0.59
07-05-92	Lift 2 - EMA	1.03	5.10	13.2	0.51
EMA/ITA Mean		1.17	5.74	13.8	0.55
% Difference, EMA vs G1		+30	+100	-18	+10

Brake specific HC emissions were observed to increase at lighter engine loads, which accounts for the higher HC emissions measured with the EMA/ITA cycle. Higher CO emissions are associated with the higher weighting factor assigned to the idle mode in the EMA/ITA cycle, plus the fact that this engine runs at an idle setting which is richer than its other operating modes. Lower NO_X emissions with the EMA/ITA cycle are most likely associated with its lower load factor relative to the G1 cycle (32% EMA/ITA vs 47% G1). It should be noted that this is a single-engine comparison between these two test cycles, which does not represent the general case. Engines with different calibrations would definitely exhibit different results. The higher weighting factor assigned to the idle mode may or may not be representative of all the SI engines in the OHV category. One would also question the importance to a cycle of a mode assigned a two percent weighting factor.

7. <u>60 HP Spark-Ignited Utility Engine</u>

The 2.7L Teledyne TM27 engine was tested in its baseline (gasoline fueled) configuration using the ISO 8178-G1 cycle. Results are summarized in Table 31. In its gasoline-fueled configuration, the Teledyne engine is calibrated rich (~13:1 A/F), presumably for best power. It has higher HC and CO and lower NO_X emissions than the 2.2L LPG-fueled Toyota lift truck engine tested earlier. These trends are consistent with the differences in calibration for the two engines. The LPG Toyota engine was calibrated for lean operation, resulting in low HC and CO emissions, but high NO_X emissions.

TABLE 31. 60 HP SPARK-IGNITED ENGINE ISO 8178-G1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
		THC	CH_4	RHC	CO	NO_x	PM	
9-21-92	60HPSI-1	2.05	0.34	1.71	130	8.33	0.01	0.62
9-22-92	60HPSI-2	1.98	0.34	1.64	121	8.26	0.02	0.59
Mean		2.02	0.34	1.68	126	8.30	0.02	0.61

Following gasoline-fueled baseline testing, the Teledyne engine was converted to LPG, and then CNG operation to evaluate its performance using these two alternate fuels. Results are summarized in Table 32. The engine was calibrated considerably leaner in both LPG and CNG configurations. Cycle weighted air/fuel ratios were 13.0 for gasoline, 19.0 for LPG, and 20.2 for CNG. The stoichiometric air/fuel ratios for these three fuels were 14.56, 15.68, and 17.15, respectively, and thus cycle weighted lambdas ($air/fuel_{actual} \div air/fuel_{stoichiometric}$) were 0.89 for gasoline, 1.21 for LPG, and 1.18 for CNG. These leaner calibrations yield very large reductions in HC and CO, and large increases in NO_x, compared to gasoline operation. THC+NO_x emissions actually increased for LPG- and CNG-fueled configurations due to the large NO_x increases. In addition to emission effects, there was a reduction in available power in the LPG and CNG configurations. Due to the leaner calibrations, and to a lesser degree the increased mixture volume occupied by these lower molecular weight gaseous fuels, mode 6 power (maximum at 85% of rated speed) was reduced by 14 percent and 22 percent in the LPG and CNG configurations, respectively. Detailed results are included in Appendix H.

8. <u>60 HP Spark-Ignited Utility Engine - Closed Loop Control with Three-Way Catalyst</u>

The 2.7L Teledyne TM27 engine was refitted with an Impco Alternate Fuel Management, LPG closed-loop control system. This is a production system designed to control engine operation at stoichiometry. The system and its primary components are described in Table 33.

TABLE 32. 60 HP SPARK-IGNITED ENGINE GASOLINE, LPG, AND CNG-FUELED CONFIGURATIONS ISO 8178-G1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)				BSFC lb/hp-hr	Mode 6 hp
		THC	CO	NO_x	THC+NO_x		
Gasoline Baseline							
9-21-92	60HPSI-1	2.05	130	8.33	10.38	0.62	60.7
9-22-92	60HPSI-2	1.98	121	8.26	10.24	0.59	60.6
Mean		2.02	126	8.30	10.31	0.61	60.7
LPG							
9-24-92	60HPSI LPG1	0.59	2.38	13.8	14.39	0.49	52.0
9-24-92	60HPSI LPG2	0.51	2.33	15.5	16.01	0.53	52.0
Mean		0.55	2.36	14.7	15.20	0.51	52.0
% Change, LPG vs. Gasoline		-73%	-98%	+77%	+47%	-16%	-14%
CNG							
10-7-92	60HPSI CNG1	0.94	3.52	10.9	11.84	0.51	47.3
10-7-92	60HPSI CNG2	0.83	3.46	11.4	12.23	0.52	47.2
Mean		0.89	3.49	11.2	12.04	0.52	47.3
% Change, CNG vs. Gasoline		-56%	-97%	+35%	+17%	-15%	-22%

TABLE 33. IMPCO ALTERNATE FUEL MANAGEMENT SYSTEM

Impco Carburetor with Feedback Gas Valve
Impco Converter
Impco VFF30 Fuelock
AFCP-1 Fuel Control Processor
Exhaust Gas Oxygen Sensor
Barometric Pressure Sensor
Manifold Absolute Pressure Sensor
Throttle Position Sensor
Engine Coolant and Air Charge Temperature Sensors
Knock Sensor
Crankshaft Position Sensors

A 126 in.^{3} three-way catalyst (5.66 inch diameter x 5.00 inch long) supplied by Engelhard Industries was installed in the exhaust system approximately 3 feet from the exhaust manifold. The engine was tested using the ISO 8178-G1 cycle using the production (analog) control system described above. The second test was run using a prototype, digital control system supplied by Impco. Results are summarized and compared to engine open-loop results in Table 34.

TABLE 34. 60 HP SPARK-IGNITED ENGINE CLOSED-LOOP CONTROL SYSTEM WITH THREE-WAY CATALYST ISO 8178-G1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr	Mode 6 hp
		THC	CH_4	RHC	CO	NO_x	PM		
LPG Closed-Loop Control and Three-Way Catalyst									
10-1-92	60HPSI CL-1	0.08	0.00	0.07	0.64	0.95	0.01	0.51	58.5
10-2-92	60HPSI CL-2	0.10	0.01	0.09	4.80	0.10	0.00	0.52	58.0
Mean		0.09	0.01	0.08	2.72	0.53	0.01	0.52	58.3
Gasoline Baseline		2.02	0.34	1.68	126	8.30	0.02	0.61	60.7
% Change, CLC vs. Gasoline		-96%	-97%	-95%	-98%	-94%	-50%	-15%	-4%
LPG Open-Loop		0.55			2.36	14.7		0.51	52.0
% Change, CLC vs. LPG		-84%			+15%	-96%		+2%	+12%
CNG Open-Loop		0.89			3.49	11.2		0.52	47.3
% Change, CLC vs. CNG		-90%			-22%	-95%		0%	+23%

With state-of-the-art closed-loop control at stoichiometry and a three-way catalyst, emissions from this engine were reduced to very low levels. THC and NO_x levels were reduced from 84 to 96 percent compared to open-loop, non-catalyst operation with gasoline, LPG, or CNG. There were slight differences in operation between the analog (CL1) and the digital (CL2) controller. CO and NO_x levels suggest slightly richer operation occurred with the prototype digital controller. The stoichiometric calibration also restores much of the power that was lost with the leaner, open-loop LPG and CNG configurations. Mode six power in closed-loop, stoichiometric operation was 58.3 hp, which was only 4 percent less than the gasoline-fueled configuration. Clearly, very low emissions are achievable by S.I. engines in this category with the application of these technologies. Detailed results are included in Appendix I.

9. <u>100 HP Spark-Ignited Utility Engine</u>

The 4.9L, six-cylinder Ford engine was tested in gasoline configuration using the ISO-8178-G1 cycle. Results are summarized in Table 35. This is a carbureted engine which is calibrated rich for power like the gasoline version of the Teledyne TM27. Emissions levels are very similar to those measured with the TM27 engine. Detailed results are included in Appendix J.

TABLE 35. 100 HP SPARK-IGNITED ENGINE ISO 8178-G1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
		THC	CH_4	RHC	CO	NO_x	PM	
10-23-92	100HP Utility 3	2.83	0.24	2.59	97.9	9.19	0.01	0.58
10-26-92	100HP Utility 4	2.74	0.23	2.51	89.9	9.69	0.01	0.57
Mean		2.79	0.24	2.55	93.9	9.44	0.01	0.58

10. <u>50 HP Snowmobile Engine</u>

The 50 hp Rotax 503 snowmobile engine was tested in its baseline configuration using the ISO 8178-G1 test procedure. This is a two-cylinder, fan-cooled, two-stroke engine of 497 cc displacement. It is rated at 50 hp at 6800 rpm by the manufacturer. The engine is fitted with two Mikuni carburetors, and employs oil injection for cylinder lubrication. The engine was "hard mounted" and tested using the 50 hp dynamometer, but would not idle satisfactorily connected to the dynamometer. Therefore, idle mode emissions were obtained with the engine decoupled from the dynamometer. Only one ISO 8178-G1 test was obtained, because the high impact power pulses of this engine exhausted our supply of coupling inserts. Results are summarized in Table 36.

TABLE 36. 50 HP SNOWMOBILE ENGINE ISO 8178-G1 EMISSION TEST RESULTS

Test Date	Test ID	Emissions (g/hp-hr)						BSFC lb/hp-hr
		THC	CH_4	RHC	CO	NO_x	PM	
2-12-93	Snowmobile	96.7	1.13	95.5	347	0.33	3.11	1.01

Emission results are similar to those observed with two-stroke engines in other programs. Two-stroke engines have high hydrocarbon and particulate emissions due to the crankcase scavenging process. A certain fraction of the intake charge is untrapped in the combustion chamber and passes directly through the engine and out the exhaust port. THC emissions from this engine (96.7 g/hp-hr) are comparable to results reported by SwRI in 1974 in a study performed for the EPA. Composite HC + NO_x emissions (97.0 g/hp-hr) are lower than an International Snowmobile Industry Association (ISIA)- reported composite snowmobile emission factor (152 g/hp-hr); but it should be noted that the current SwRI data represent a single engine, and that a wide range of THC emission levels would be expected for engines of this type. It should also be noted that this engine does not conform to current OEM specifications, and may represent an updated model.

The snowmobile engine's CO emissions (347 g/hp-hr) are also very high. This is a result of the rich calibration which is typically employed to provide smooth engine response, and a measure of "fuel cooling." NO_x emissions are low (0.33 g/hp-hr) due to the rich mixture and the dilution of the intake charge with residual exhaust gases. Particulate emissions are very high (3.11 g/hp-hr), composed primarily of an aerosol of uncombusted lubricant entrained by the engine exhaust. Aldehyde emissions (0.90 g/hp-hr) were also very high for this engine. Detailed results are included in Appendix K.

B. Aldehyde Results

Speciated aldehyde emissions were determined for program engines in baseline configurations using the DNPH-LC procedure to provide information for ARB's toxic air contaminant studies. Aldehyde emissions were not determined for the John Deere engines under the EPA program. Aldehyde results are summarized in Table 37. Detailed results for individual engines are presented in Tables 38-46.

Formaldehyde emissions were the highest of the individual aldehyde and ketone species, comprising from 44 percent to 79 percent of the total of all species. The 40 HP Lift Truck engine formaldehyde emission percentage (79%) was significantly higher than the other engines, perhaps due to its use of LPG fuel (90%+ propane). Aldehyde emission levels are similar to those observed with similarly calibrated SI engines. As a percentage of total hydrocarbon emissions, total aldehyde emissions ranged from ~1 percent (50 HP Snowmobile) to ~13 percent (40 HP Lift Truck). Lower aldehyde/hydrocarbon percentages were observed with SI engines with rich calibrations. The highest aldehyde/hydrocarbon percentage was observed with the LPG-fueled Lift Truck engine, which had a lean calibration. The three-way catalyst equipped Teledyne engine (60 HP CLC/TWC Utility) had almost no aldehyde emissions. The two-stroke snowmobile engine had extremely high aldehyde emissions. Detailed modal speciation results are included in Appendix L.

TABLE 37. ALDEHYDE EMISSIONS SUMMARY

Engine ID	Emissions (mg/hp-hr)			Formaldehyde / Total Ald.	Total Ald. / THC
	Formaldehyde	Acetaldehyde	Total Ald.		
Diesel Engines (ISO 8178-C1)					
30 HP Air-Cooled	139	37	234	59%	8.6%
35 HP Water-Cooled	59	15	102	58%	9.3%
50 HP Water-Cooled	49	14	85	58%	4.9%
Spark-Ignited Engines (ISO 8178-G1)					
8.5 HP Golf Car*	192	35	330	58%	4.4%
40 HP Lift Truck	91	16	115	79%	12.8%
60 HP SI Utility	36	7	65	55%	3.2%
60 HP CLC/TWC Utility	0	0	0	---	---
100 HP SI Utility	49	8	90	54%	3.2%
50 HP Snowmobile	397	93	898	44%	0.9%
*J1088 test cycle					

TABLE 38

Aldehyde and Ketone Emissions from a 30 HP Air–Cooled Diesel Engine Weighted Average Results Using ISO–8178–C1 Test Cycle

Test Number	ACD–1	ACD–2	Average ISO–8178–C1
Formaldehyde, mg/hr	2302	2112	2207
Formaldehyde, mg/hp–hr	143	135	139
Acetaldehyde, mg/hr	613	559	586
Acetaldehyde, mg/hp–hr	38	36	37
Acrolein, mg/hr	348	296	322
Acrolein, mg/hp–hr	22	19	20
Acetone, mg/hr	197	176	186
Acetone, mg/hp–hr	12	11	12
Propionaldehyde, mg/hr	135	120	128
Propionaldehyde, mg/hp–hr	8	8	8
Crotonaldehyde, mg/hr	108	101	105
Crotonaldehyde, mg/hp–hr	7	6	7
Isobutyraldehyde/MEK, mg/hr	68	85	77
Isobutyraldehyde/MEK, mg/hp–hr	4	5	5
Benzaldehyde, mg/hr	87	78	83
Benzaldehyde, mg/hp–hr	5	5	5
Hexanaldehyde, mg/hr	25	23	24
Hexanaldehyde, mg/hp–hr	2	1	2
Total Aldehydes, mg/hr	3882	3552	3717
Total Aldehydes, mg/hp–hr	242	227	234

TABLE 39

Aldehyde and Ketone Emissions from a 35 HP Water–Cooled Diesel Engine Weighted Average Results Using ISO–8178–C1 Test Cycle

Test Number	WCD–1	WCD–2	Average ISO–8178–C1
Formaldehyde, mg/hr	1071	1044	**1057**
Formaldehyde, mg/hp–hr	60	58	**59**
Acetaldehyde, mg/hr	301	245	**273**
Acetaldehyde, mg/hp–hr	17	14	**15**
Acrolein, mg/hr	173	161	**167**
Acrolein, mg/hp–hr	10	9	**9**
Acetone, mg/hr	135	84	**110**
Acetone, mg/hp–hr	8	5	**6**
Propionaldehyde, mg/hr	63	57	**60**
Propionaldehyde, mg/hp–hr	4	3	**3**
Crotonaldehyde, mg/hr	57	56	**56**
Crotonaldehyde, mg/hp–hr	3	3	**3**
Isobutyraldehyde/MEK, mg/hr	30	29	**29**
Isobutyraldehyde/MEK, mg/hp–hr	2	2	**2**
Benzaldehyde, mg/hr	51	50	**50**
Benzaldehyde, mg/hp–hr	3	3	**3**
Hexanaldehyde, mg/hr	14	26	**20**
Hexanaldehyde, mg/hp–hr	1	1	**1**
Total Aldehydes, mg/hr	1894	1752	**1823**
Total Aldehydes, mg/hp–hr	106	98	**102**

TABLE 40

Aldehyde and Ketone Emissions From A 50 HP DI Diesel Utility Engine Weighted Average Results Using ISO−8178−C1 Test Cycle

Test Number	50 HP DI−1	50 HP DI−2	Average ISO−8178−C1
Formaldehyde, mg/hr	1309	985	1147
Formaldehyde, mg/hp−hr	56	42	49
Acetaldehyde, mg/hr	389	286	338
Acetaldehyde, mg/hp−hr	17	12	14
Acrolein, mg/hr	76	48	62
Acrolein, mg/hp−hr	3	2	3
Acetone, mg/hr	159	48	104
Acetone, mg/hp−hr	7	2	4
Propionaldehyde, mg/hr	72	57	65
Propionaldehyde, mg/hp−hr	3	2	3
Crotonaldehyde, mg/hr	47	43	45
Crotonaldehyde, mg/hp−hr	2	2	2
Isobutyraldehyde/MEK, mg/hr	71	71	71
Isobutyraldehyde/MEK, mg/hp−hr	3	3	3
Benzaldehyde, mg/hr	79	156	117
Benzaldehyde, mg/hp−hr	3	7	5
Hexanaldehyde, mg/hr	36	50	43
Hexanaldehyde, mg/hp−hr	2	2	2
Total Aldehydes, mg/hr	2238	1745	1991
Total Aldehydes, mg/hp−hr	95	74	85

TABLE 41

Aldehyde and Ketone Emissions From An 8.5 hp Golf Car Engine Weighted Average Results Using ISO–8178–G1 Test Cycle

Test Number	GOLF–1	GOLF–2	Average ISO–8178–G1
Formaldehyde, mg/hr	452	442	**447**
Formaldehyde, mg/hp–hr	194	190	**192**
Acetaldehyde, mg/hr	84	80	**82**
Acetaldehyde, mg/hp–hr	36	34	**35**
Acrolein, mg/hr	20	18	**19**
Acrolein, mg/hp–hr	8	8	**8**
Acetone, mg/hr	34	38	**36**
Acetone, mg/hp–hr	15	16	**15**
Propionaldehyde, mg/hr	20	22	**21**
Propionaldehyde, mg/hp–hr	8	10	**9**
Crotonaldehyde, mg/hr	13	12	**13**
Crotonaldehyde, mg/hp–hr	6	5	**6**
Isobutyraldehyde/MEK, mg/hr	12	11	**12**
Isobutyraldehyde/MEK, mg/hp–hr	5	5	**5**
Benzaldehyde, mg/hr	141	136	**139**
Benzaldehyde, mg/hp–hr	61	59	**60**
Hexanaldehyde, mg/hr	0	0	**0**
Hexanaldehyde, mg/hp–hr	0	0	**0**
Total Aldehydes, mg/hr	776	761	**768**
Total Aldehydes, mg/hp–hr	333	327	**330**

TABLE 42

Aldehyde and Ketone Emissions From An LPG Lift Truck Engine Weighted Average Results Using ISO–8178–G1 Test Cycle

Test Number	LIFT–1	LIFT–2	Average ISO–8178–G1
Formaldehyde, mg/hr	1627	1543	**1585**
Formaldehyde, mg/hp–hr	93	89	**91**
Acetaldehyde, mg/hr	282	285	**284**
Acetaldehyde, mg/hp–hr	16	16	**16**
Acrolein, mg/hr	65	63	**64**
Acrolein, mg/hp–hr	4	4	**4**
Acetone, mg/hr	63	29	**46**
Acetone, mg/hp–hr	4	2	**3**
Propionaldehyde, mg/hr	27	24	**26**
Propionaldehyde, mg/hp–hr	2	1	**1**
Crotonaldehyde, mg/hr	3	2	**2**
Crotonaldehyde, mg/hp–hr	0	0	**0**
Isobutyraldehyde/MEK, mg/hr	1	3	**2**
Isobutyraldehyde/MEK, mg/hp–hr	0	0	**0**
Benzaldehyde, mg/hr	2	1	**1**
Benzaldehyde, mg/hp–hr	0	0	**0**
Hexanaldehyde, mg/hr	0	0	**0**
Hexanaldehyde, mg/hp–hr	0	0	**0**
Total Aldehydes, mg/hr	2069	1950	**2009**
Total Aldehydes, mg/hp–hr	119	112	**115**

TABLE 43

Aldehyde and Ketone Emissions From A 60 HP SI Utility Engine on Gasoline Weighted Average Results Using ISO−8178−G1 Test Cycle

Test Number	60 HP SI−1	60 HP SI−2	Average ISO−8178−G1
Formaldehyde, mg/hr	966	1069	**1017**
Formaldehyde, mg/hp−hr	34	37	**36**
Acetaldehyde, mg/hr	184	203	**194**
Acetaldehyde, mg/hp−hr	6	7	**7**
Acrolein, mg/hr	34	45	**40**
Acrolein, mg/hp−hr	1	2	**1**
Acetone, mg/hr	0	0	**0**
Acetone, mg/hp−hr	0	0	**0**
Propionaldehyde, mg/hr	45	39	**42**
Propionaldehyde, mg/hp−hr	2	1	**1**
Crotonaldehyde, mg/hr	14	21	**17**
Crotonaldehyde, mg/hp−hr	0	1	**1**
Isobutyraldehyde/MEK, mg/hr	32	43	**37**
Isobutyraldehyde/MEK, mg/hp−hr	1	2	**1**
Benzaldehyde, mg/hr	441	536	**489**
Benzaldehyde, mg/hp−hr	15	19	**17**
Hexanaldehyde, mg/hr	22	31	**26**
Hexanaldehyde, mg/hp−hr	1	1	**1**
Total Aldehydes, mg/hr	1737	1987	**1862**
Total Aldehydes, mg/hp−hr	61	70	**65**

TABLE 44

Aldehyde and Ketone Emissions From A 60 HP SI Utility Engine on LPG with Closed Loop Control and Three-Way Catalyst Weighted Average Results Using ISO-8178-G1 Test Cycle

Test Number	Analog 60 HP CL-1	Digital 60 HP CL-2	Average ISO-8178-G1
Formaldehyde, mg/hr	2	1	**1**
Formaldehyde, mg/hp-hr	0	0	**0**
Acetaldehyde, mg/hr	4	3	**4**
Acetaldehyde, mg/hp-hr	0	0	**0**
Acrolein, mg/hr	0	0	**0**
Acrolein, mg/hp-hr	0	0	**0**
Acetone, mg/hr	0	0	**0**
Acetone, mg/hp-hr	0	0	**0**
Propionaldehyde, mg/hr	0	2	**1**
Propionaldehyde, mg/hp-hr	0	0	**0**
Crotonaldehyde, mg/hr	0	0	**0**
Crotonaldehyde, mg/hp-hr	0	0	**0**
Isobutyraldehyde/MEK, mg/hr	0	0	**0**
Isobutyraldehyde/MEK, mg/hp-hr	0	0	**0**
Benzaldehyde, mg/hr	0	0	**0**
Benzaldehyde, mg/hp-hr	0	0	**0**
Hexanaldehyde, mg/hr	0	0	**0**
Hexanaldehyde, mg/hp-hr	0	0	**0**
Total Aldehydes, mg/hr	6	6	**6**
Total Aldehydes, mg/hp-hr	0	0	**0**

TABLE 45

Aldehyde and Ketone Emissions From A 100 HP Utility Engine On Gasoline Weighted Average Results Using ISO–8178–G1 Test Cycle

Test Number	100 HP UTILITY–3	100 HP UTILITY–4	Average ISO–8178–G1
Formaldehyde, mg/hr	2137	2283	**2210**
Formaldehyde, mg/hp–hr	47	50	**49**
Acetaldehyde, mg/hr	350	364	**357**
Acetaldehyde, mg/hp–hr	8	8	**8**
Acrolein, mg/hr	187	247	**217**
Acrolein, mg/hp–hr	4	5	**5**
Acetone, mg/hr	163	199	**181**
Acetone, mg/hp–hr	4	4	**4**
Propionaldehyde, mg/hr	40	60	**50**
Propionaldehyde, mg/hp–hr	1	1	**1**
Crotonaldehyde, mg/hr	59	41	**50**
Crotonaldehyde, mg/hp–hr	1	1	**1**
Isobutyraldehyde/MEK, mg/hr	53	75	**64**
Isobutyraldehyde/MEK, mg/hp–hr	1	2	**1**
Benzaldehyde, mg/hr	924	939	**932**
Benzaldehyde, mg/hp–hr	20	21	**21**
Hexanaldehyde, mg/hr	0	0	**0**
Hexanaldehyde, mg/hp–hr	0	0	**0**
Total Aldehydes, mg/hr	3912	4210	**4061**
Total Aldehydes, mg/hp–hr	87	93	**90**

TABLE 46

Aldehyde and Ketone Emissions From A 50 HP Snowmobile Engine On Gasoline
Weighted Average Results Using ISO–8178–G1 Test Cycle

Test Number	50 HP Snowmobile
Formaldehyde, mg/hr	7,244
Formaldehyde, mg/hp–hr	397
Acetaldehyde, mg/hr	1,706
Acetaldehyde, mg/hp–hr	93
Acrolein, mg/hr	709
Acrolein, mg/hp–hr	39
Acetone, mg/hr	1,537
Acetone, mg/hp–hr	84
Propionaldehyde, mg/hr	359
Propionaldehyde, mg/hp–hr	20
Crotonaldehyde, mg/hr	235
Crotonaldehyde, mg/hp–hr	13
Isobutyraldehyde/MEK, mg/hr	685
Isobutyraldehyde/MEK, mg/hp–hr	38
Benzaldehyde, mg/hr	3,740
Benzaldehyde, mg/hp–hr	205
Hexanaldehyde, mg/hr	170
Hexanaldehyde, mg/hp–hr	9
Total Aldehydes, mg/hr	16,385
Total Aldehydes, mg/hp–hr	898

VII. TASK 4 - EVALUATION OF EMISSION CONTROL TECHNOLOGY FOR OFF-HIGHWAY VEHICLE ENGINES

To date, the equipment in these product categories has been largely unregulated with regard to exhaust emissions. However, the engines used in equipment in this category must meet a very broad range of operational and economic requirements. Engine designs are therefore quite diverse with regard to level of technical sophistication, design life, reliability requirements, fuel consumption characteristics, maintenance requirements and product cost. In general, engines used in the construction, farm, and industrial equipment categories are designed for higher average load factors than an automobile engine. Passenger car engines typically operate at less than a 30 percent load factor, while load factors for industrial equipment average approximately 50 percent. Such operating characteristics in turn impact cooling system design, structural load requirements and other engine design parameters. The typical duty cycle for an industrial engine also tends to be less transient than for an automobile engine, thus influencing design complexity and requirements for fuel delivery systems, injection and ignition systems, and the exhaust system. Because of the diversity of applications, the engines used in these off-road equipment categories (farm, construction, industrial, and recreational) can range from extremely complex and costly (approaching on-highway automotive technology levels) to very simple, inexpensive designs. For example, selected European forklift manufacturers currently offer engines equipped with 3-way catalytic converters, high energy ignition systems and exhaust gas recirculation. Such equipment is often used in a closed environment (such as inside a factory) where emission levels must be kept extremely low. The cost of such equipment, however, is quite high compared to "standard" models.

While a small percentage of the engines in these equipment categories have been designed for low emissions, most have been optimized for maximum performance, reliability, durability, and to a lesser extent fuel economy. Another unique characteristic of the mobile industrial equipment market is the fairly widespread use of multifuel engines capable of operating on gasoline, LPG or natural gas with only relatively minor engine modifications. Such designs increase the versatility of the equipment, reduce emissions, and help to keep costs down by using a common engine design for all three fuels.

A. Emission Control Technologies for Spark-Ignited Engines

1. Background

As emissions from engines in off-highway equipment are completely unregulated, many of the engine designs in this category are reminiscent of those used in passenger cars in the early 1960s. These engines use overhead valves actuated by pushrods, with two valves per cylinder. Fuel metering is accomplished by simple carburetors. Fuel mixture is ordinarily very rich at idle, lean at low part throttle, and rich at full load. Lean operation at part throttle enhances fuel economy, compared to that of rich mixtures. Transition to rich mixtures at high loads is required to control combustion knock, and to maximize power. Several major technological differences between most current engines in off-highway equipment and automotive engines of the early 1960s are:

- The incorporation of electronic ignition systems, in place of point and coil systems
- Design modifications for unleaded gasoline
- Use of positive crankcase ventilation
- Widespread use of LPG.

2. Passenger Car Emission Control Experience

Since many off-highway equipment engines are so similar to passenger car engines of the early 1960s, as a guide to applicable emission control techniques, it may be useful to review the experience of emission control of passenger cars. Unburned hydrocarbons in blow-by gases escaping from the crankcase constituted about 25 percent of of the HC emissions from an uncontrolled passenger car, and were identified early as an easy target for control. By the mid-1960s, crankcase ventilation systems that routed all blow-by gases back to the induction system were required on California automobiles. The first exhaust emission standards were established in California in 1966, limiting HC and CO on the basis of a 7 mode steady-state test procedure. These fairly lenient standards were met by techniques such as leaner operation, heated air intakes, and in some models, injection of air into the exhaust manifold.

Evaporative Emission Control. Evaporative emissions are the source of about 20 percent of the hydrocarbons from the uncontrolled automobile. In response to the imposition of evaporative emission standards in 1971, manufacturers began installing evaporative emission control systems. These systems rely on a canister of activated charcoal to adsorb vapors from the fuel tank and carburetor float bowls formed during hot soaks and diurnal heating. When the vehicle is restarted, a purge control valve clears the canister by allowing the vapor to desorb and flow into the intake system at a controlled rate.

Early NOx controls. In 1970, the Federal Clean Air Act was signed into law, mandating a 90 percent reduction in automotive emission rates by 1975. The Environmental Protection Agency (EPA) was also established at this time. The EPA developed a new emission test procedure, the CVS-72, which exercised the vehicle over a duty cycle based on actual driving patterns recorded in Los Angeles (LA-4 cycle), that was considerably more rigorous than earlier steady-state cycles. Emission standards, incorporating meaningful NOx limits as of model year 1972 in California and 1973 nationally, became increasingly stringent during the 1970s. To reduce NOx emissions, manufacturers began using exhaust gas recirculation, sometimes in combination with retarded timing and reduced compression ratios. Already-lean operation combined with these techniques tended to result in poor driveability and greatly reduced power.

In the early 1970s, manufacturers realized that major advances in emission control technology would be needed to achieve the level of control required by the Clean Air Act. Two widely divergent approaches were investigated:

1) Perfecting engine designs capable of low engine-out emissions; and

2) Exhaust aftertreatment, either by thermal or catalytic means.

Lean Burn Engines. Ford initially preferred the first approach and investigated the potential of extremely lean burning stratified charge engines. The PROCO ("programmed combustion") research engine used direct fuel injection to create a mixture locally rich near the spark plug, but increasingly lean with distance from the spark. This arrangement results in easy initiation of a flame kernel and rapid flame propagation. As the flame moves into successively leaner mixtures, it cools, so that NOx formation is suppressed. By the time the flame reaches the quench zone, very little fuel is present, so formation of unburned hydrocarbons is minimal. Combustion at a very lean average mixture minimizes formation of CO. Accordingly, engine-out emissions of all criteria pollutants are low.

Honda utilized a similar strategy in its "Controlled Vortex Combustion Chamber" engine. Rather than using direct fuel injection, Honda used two intake systems, one operating richer than the rich misfire limit, and the other, leaner than the lean misfire limit. Most of the intake charge was admitted through the lean system which supplied the intake manifold, flowed through the intake valve, and filled the main combustion chamber. The rich mixture was admitted into a small prechamber through a second intake valve. During the compression stroke, gas flow from the main chamber to the prechamber diluted the prechamber mixture to slightly rich of stoichiometric. The spark plug was located in the prechamber. Ignition caused the gas in the prechamber to form a hot jet that flowed out of the prechamber through an orifice, into the main combustion chamber. Powerful turbulent mixing resulted in rapid, fairly complete combustion. This engine was clean enough so that Honda did not switch to catalytic converters until 1978.

Oxidation Catalysts. General Motors concentrated its effort on exhaust aftertreatment, with the goal of developing a practical catalytic converter. Catalysts had been used for decades in the process industry to initiate reactions at lower temperatures than would otherwise be possible. Gasoline engine exhaust gas temperatures vary from 300°C to 400°C at idle to as much as 900°C at full power. Catalytic oxidation of HC and CO can be accomplished at temperatures as low as 250°C. The problem was to design a catalyst that would survive long enough in the exhaust stream to be practical. Thermal stresses and chemical poisons in the exhaust environment tend to deactivate catalysts. In particular, lead is a powerful catalyst poison, so catalytic converters would have to be used with unleaded gasoline. The associated loss of octane dictated that compression ratios be lowered from about 9.5 to about 8.5, resulting in less power and lower fuel efficiency.

Early catalytic converters were based on designs adapted from the chemical process industry, and used platinum and palladium impregnated on a substrate of pelletized alumina. They were designed for oxidation only. NOx control continued to be achieved through EGR and retarded timing at part throttle. With the use of oxidation catalysts, richer calibration was allowed, since unburned hydrocarbons and CO could be oxidized in the catalyst by air injected into the exhaust at some point upstream of the catalyst. This allowed manufacturers to recover some of the driveability and performance lost because of earlier in-cylinder emission control techniques.

Spark plug misfiring results in large amounts of air and unburned fuel flowing into the exhaust stream. If a catalyst is present, it will oxidize this fuel, and be heated to temperatures high enough to cause thermal deactivation via sintering or even melting. To prevent misfiring, General Motors installed high-energy electronic ignition systems on all catalyst equipped cars. This is one of many examples whereby emission standards have actually tended to increase the durability and reliability of automobiles.

Dual Bed Three-Way Catalytic Converters. By 1978, General Motors had developed three-way catalytic converters, that would reduce NOx to N_2 and O_2, as well as oxidize hydrocarbons and CO. For Model-Year 1980, the California NOx standard was tightened from 1.5 to 1.0 g/mi. Three-way catalysts were considered necessary to meet this standard. A separate catalyst with rhodium as the active material was initially used for NOx reduction. This process cannot take place in an oxidizing environment since excess fuel must be supplied to utilize essentially all of the oxygen present. The main mechanism for NOx reduction is the oxidation of CO to CO_2, so an adequate concentration of CO is necessary for the process to work efficiently.

By this time, the industry had largely shifted from pelletized catalyst substrates to ceramic monoliths, formed as a bundle of thin-walled tubes. This construction results in 1) lower thermal mass, and consequently, faster light-off; and 2) lower back pressure. In early GM three-way converters, the oxidation catalyst was located downstream from the reduction catalyst, so it was called a dual bed system. Oxidation was accomplished by injecting air into the exhaust stream at a point between the two catalysts. Tight control of equivalence ratio is necessary to assure both adequate NOx reduction and to avoid excessive temperatures in the oxidation catalyst. An oxygen sensor working in conjunction with microprocessor control maintained the equivalence ratio at about 0.1% excess fuel, at which point 95 percent NOx reduction can occur (Figure 5). Early American three-way systems relied on feedback-controlled carburetors for this purpose.

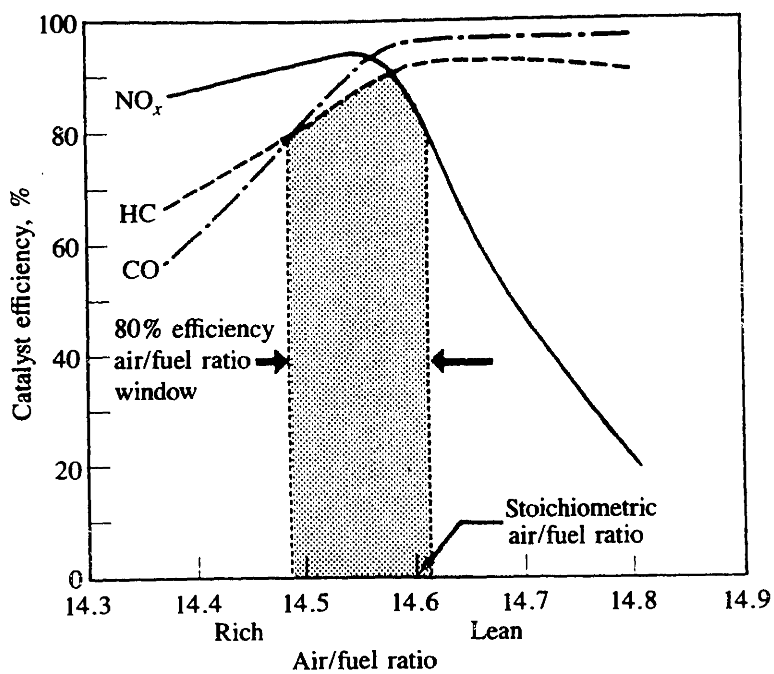


FIGURE 5. THREE-WAY CATALYST CONVERSION EFFICIENCY AS A FUNCTION OF EXHAUST GAS AIR/FUEL RATIO

Source: J. T. Kummer: "Catalysts for Automobile Emission Control", *Progress in Energy Combustion Science,* V. 6, pp. 177-199, 1981

Single Bed Three-Way Catalytic Converters. In 1977, Volvo introduced models equipped with the Lambda Sond emission control system. It combined reduction and oxidation functions in a single three-way catalyst. An oxygen sensor, mass air flow sensor and microprocessor controlled fuel injection were used to rapidly cycle the mixture in a very narrow region from slightly rich of stoichiometric to slightly lean. Storage of CO and oxygen on the active sites during rich and lean excursions, respectively, enabled very high overall conversion efficiency. Greater NOx control was achieved than in dual bed systems, since the reoxidation of N_2 to NO in the oxidation catalyst is avoided. With the Lambda Sond system, no air injection was required, and the engine's stoichiometric, rather than rich, calibration improved fuel economy. Port fuel injection allowed better mixture control than carburetors could attain, and required less cold start enrichment. Microprocessor controlled injection systems can also cut off fuel flow during motoring, resulting in improved emissions and additional fuel economy improvement.

By the mid 1980s, the design features of the Lambda Sond System had been adopted by almost every manufacturer offering cars for sale in the United States. Development of integrated microprocessor control systems for fuel delivery, ignition timing and EGR (such as the Bosch Motronic system) allowed further improvements in emissions and fuel economy. Continuing improvements in catalyst formulation and substrate design have dramatically increased catalyst durability compared to that of the late 1970s.

3. Emerging Approaches to Passenger Car Emission Control

Four-valve heads with centrally located sparkplug. Recently, passenger car engine designers have been increasingly incorporating dual overhead cams actuating four valves per cylinder. This design naturally allows for a hemispherical combustion chamber with a centrally located sparkplug. Although intended primarily to produce high specific output by minimizing intake pressure loss at high engine speeds, this design also benefits emissions:

- Since the flame front is initiated at the center of a hemisphere, combustion is rapidly completed, minimizing the time available for NOx formation

- Combustion chamber surface area and crevice volume are low, which tends to minimize formation of unburned hydrocarbons.

Lean oxygen sensors. Because fuel burns most completely with slightly excess air, lean operation results in better fuel utilization than with stoichiometric mixtures. Since three-way catalytic converters require stoichiometric mixtures, concern with improving fuel economy has led automobile manufacturers to revisit other emission control techniques that allow lean engine calibration. This has been made more practical by the recent development of lean oxygen sensors. Conventional oxygen sensors are very sensitive to the *transition* between lean and rich mixtures, but have very poor sensitivity in mixtures that are significantly richer or leaner than stoichiometric (Figure 6). Lacking an adequate oxygen sensor, designers were unable to use feedback to control lean mixtures, and achieve the high control precision that it allows.

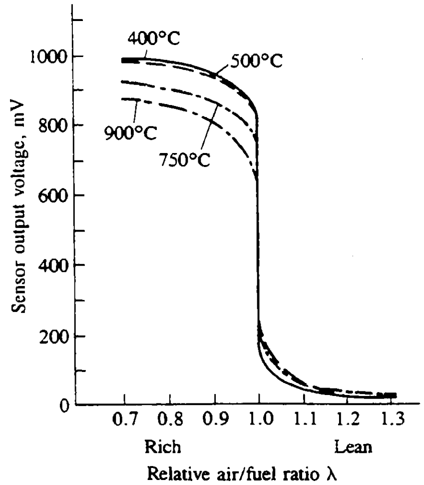


FIGURE 6. TYPICAL RESPONSE CURVE OF A CONVENTIONAL OXYGEN SENSOR

Source: E. Hamann et. al: "Lambda Sensor with YxO3-Stabilized ZrO2 Ceramic for Application in Automotive Emission Control Systems", SAE 770401 (1977).

Honda has recently incorporated a lean oxygen sensor in the engine of one model in its Civic line. In the new Honda VTEC-E engine, used in the Civic VX, NOx is controlled by producing extremely lean mixtures at part throttle, in addition to EGR. Lean operation is achieved by sequential electronic port fuel injection, 4 valves/cylinder, and clever intake port geometry that produces a swirl-stratified charge. This 1.5L engine is rated at 92 BHP. In the Civic VX, its EPA fuel economy rating is 48 MPG City, and 55 MPG Highway. The NOx control achievable by this technology is not as great as that achievable by a three-way catalytic converter, however, it is less costly to implement. This is evidenced by the fact that lean operation is used in the Federal version of this engine, but a three-way catalyst is used in the California version (Federal and California NOx standards are 0.7 g/mi and 0.4 g/mi, respectively). NOx reduction from an uncontrolled spark ignited engine is estimated to be 70 to 80 percent using lean burn technology versus 85 to 95 percent reduction using a three-way catalytic converter system. Lean burn technology, however, could offer designers of off-road industrial engines a means of substantially reducing NOx without the need for a three-way converter. Perhaps more importantly, the lean burn technology would improve the fuel economy of the baseline engine and add less cost to the engine than a three-way catalytic converter system.

4. Emission Control Approaches for Spark-Ignited Off-Highway Engines

Experience with passenger car emission control technology may be used as a rough guide to control approaches applicable to off-highway equipment. Generally, control effectiveness and cost have increased with each successive generation of emission control technology. Accordingly, several different emission control packages may be identified for off-highway engines, and differentiated on the bases of cost and control effectiveness. These are summarized and arranged in order of increasing effectiveness in Table 47.

Package 1 would entail minimal modifications to the existing engine design. A carbon canister and purge valve would be added for control of evaporative emissions. The purge valve would open to a fixed opening once pressure in the canister reaches the release point, and purge into the intake system. NOx would be controlled by EGR and retarded spark advance. EGR rate would be fixed, with the EGR valve actuated by intake manifold vacuum. Spark advance could be retarded by a simple retarding of the static timing. An air pump would be installed to inject air into the exhaust manifold, where it would burn HC and CO.

TABLE 47. EMISSION CONTROL PACKAGES FOR OFF-HIGHWAY ENGINES

Package	Evaporative	NOx	HC & CO
1	Carbon canister & purge valve	Fixed EGR & spark timing retard	Air injection
2	Carbon canister & purge valve. Elimination of carburetor float bowl.	Variable EGR & ignition timing	Open loop port fuel injection
3	Carbon canister & purge valve w/ electronic purge control. Elimination of carburetor float bowl.	Variable EGR & ignition timing	Open loop port fuel injection, oxidation catalyst
4	Carbon canister & purge valve w/ electronic purge control. Elimination of carburetor float bowl.	Variable EGR & ignition timing. Extremely lean calibration at part load, stratified charge	Closed loop fuel injection controlled by lean oxygen sensor, oxidation catalyst.
5	Carbon canister & purge valve w/ electronic purge control. Elimination of carburetor float bowl.	Variable EGR & ignition timing. Stoichiometric calibration	Closed loop fuel injection controlled by conventional oxygen sensor, plus 3-way catalytic converter

Package 2 would be an advance over Package 1 primarily by incorporating open-loop electronic port fuel injection. This system would require the addition of:

1) Electric fuel pump with constant delivery pressure
2) Mass air flow sensor or speed/density sensor
3) Intake air temperature sensor
4) Coolant temperature sensor
5) Engine speed sensor
6) Throttle position sensor
7) Electronic control module.

The cost of electronic engine controls, fuel injectors and the associated sensors has fallen dramatically in recent years as production volumes have increased to meet the requirements of on-highway vehicles, making this technology more feasible for off-highway engines. There are several advantages of electronic fuel injection over carburetors:

- A major source of evaporative emissions — the carburetor float bowl — is eliminated.

- Injection at the port greatly reduces the cylinder-to-cylinder mixture variation that occurs in carbureted multicylinder engines. Better fuel atomization is realized by injection through a nozzle at 10-20 psig. These features allow leaner overall calibration and less cold start enrichment.

- Electronic control allows arbitrary variation of mixture as a function of engine speed and load, with the potential for much better optimization for emissions than with a carburetor. Mixture control is provided by a look-up table.

- Using the sensors needed for electronic fuel injection, and with the addition of an electrically actuated EGR valve, it is fairly straightforward for the electronic control unit to vary EGR rate according to speed and load. Spark advance can be similarly controlled. Better control of EGR and spark advance allows for greater NOx reductions, while avoiding the driveability degradation that can be caused by fixed EGR rates and retardation of ignition timing.

An example of the emission benefits of applying port electronic fuel injection to a one-cylinder overhead valve utility engine is shown in Figure 7.

In Package 3, an oxidation catalyst is added, while retaining all of the components in Package 2. Since the oxidation catalyst will not function without free oxygen in the exhaust gas, some recalibration of the mixture control map to increase the lean operating range might be required. Alternatively, the addition of an air pump might be required. Details of the control strategy for EGR, ignition timing and mixture will depend on the emission targets for NOx, HC, and CO. Generally, lean fuel system calibration will likely increase combustion chamber temperatures over those in the original engine enough to require modification of the cooling system. For example, cast iron heads may have to be replaced by aluminum to take advantage of aluminum's higher thermal conductivity.

In Package 4, the use of lean combustion is optimized for maximum NOx control. The open-loop mixture control system used in Package 3 is replaced by a lean oxygen sensor and feedback control. This allows calibration that is closer to the lean misfire limit, and tighter mixture control in transient operation. Combustion chamber modifications to promote a stratified charge could also be part of this package. An oxidation catalyst is retained for aftertreatment of HC and CO. Lean operation may reduce peak power by 15 percent, compared to the non-emission controlled rating. However, fuel economy should be improved.

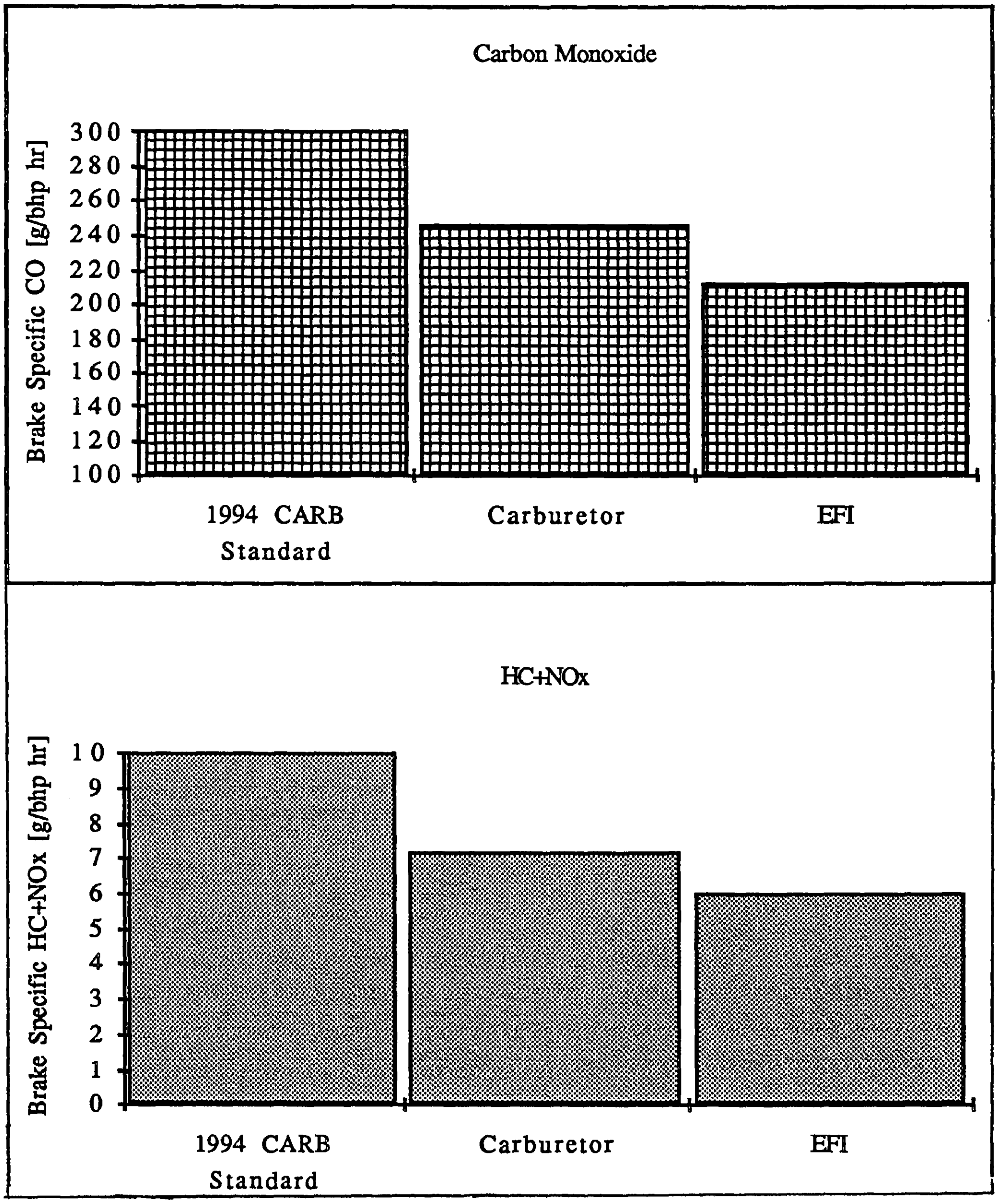


FIGURE 7. EFFECT OF ELECTRONIC FUEL INJECTION IN A 0.4L UTILITY ENGINE

Note: Test engine is a single cylinder 0.4L overhead valve utility engine. Engine was emission tested using SAE procedure J1088.

Source: Mark Swanson: *An Emission Comparison Between a Carburetor and an Electronic Fuel Injection System For Utility Engines*, SAE 911806

Package 5 uses current automotive emission control technology: a three-way catalytic convertor, oxygen sensor, and electronic fuel injection calibrated for stoichiometric operation. This approach gives greater NOx reductions than Package 4, and similar HC and CO levels. Emission control system cost will be somewhat higher than for Package 4. This results from the use of a three-way catalytic converter, which is about 15 percent more expensive than a two-way convertor. However, since the fuel system calibration is stoichiometric, thermal loads may be lower than with the lean calibrations used in Packages 3 and 4, resulting in less need to modify the cooling system. Package 5 would be preferred in applications requiring high power density, and Package 4, in applications where high fuel efficiency is more important than power density. As noted earlier, selected manufacturers of off-road industrial equipment have already adapted on-highway automotive technology to their industrial engines. Products incorporating such technology, however, are considered extremely "high end" and are available in only limited production.

5. Alternative Fuels in Off-Highway Engines

Of the three major alternative automotive fuels — methanol, compressed natural gas (CNG), and liquefied petroleum gas (LPG), LPG is by far the most suitable for use in off- highway vehicles. Many of these engines are used in the field, far away from natural gas pipelines and commercial refueling facilities. LPG is the only portable alternative fuel with an established supply infrastructure.

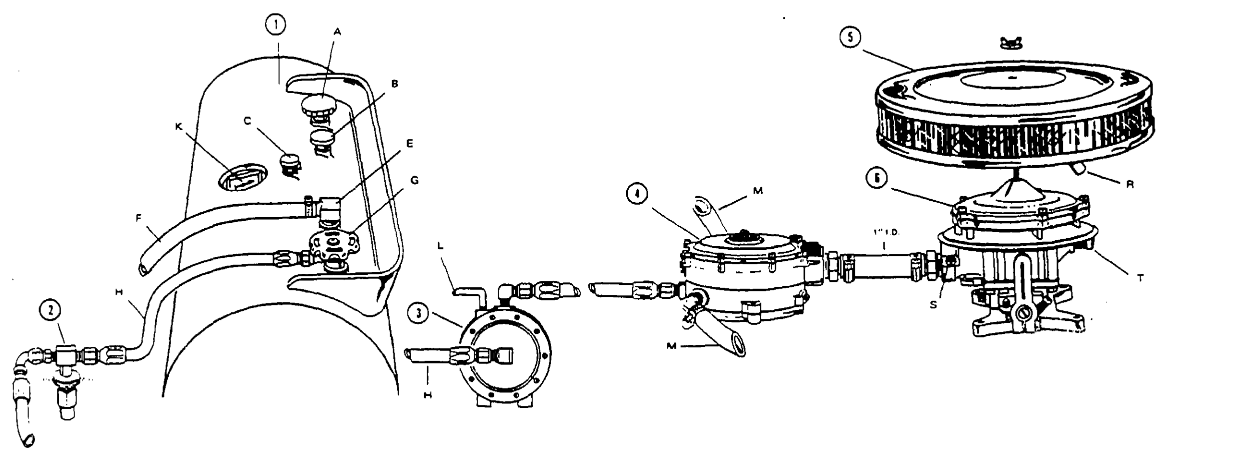
Methanol requires use of corrosion resistant tanks and fuel systems, and suffers from limited availability. Methanol's toxicity dictates that great care be exercised in storage and handling. Since methanol absorbs water, water contamination in storage tanks readily occurs. M85 must be used in spark ignited engines to assure adequate cold-start performance. The emission benefits of M85, relative to gasoline, are minimal — in fact, reformulated gasoline (Stage II CARB specification) may offer lower ozone reactivity.

CNG storage tanks are extremely bulky and heavy, compared to those for liquid fuels, and its use is very constrained by the need for compressor stations capable of compression to 3000 psi.

LPG, in contrast, has a well established refueling infrastructure, and is already widely used in off-highway vehicles, such as lift trucks. Emission testing by SwRI (Aug. 4, 1992 progress report to ARB) and others indicates that LPG in non-emission-controlled engines generally achieves much lower emissions of HC and CO than gasoline (at a given equivalence ratio), but at the cost of considerably higher NOx. LPG is gasified before being mixed with air in the intake system (Figure 8). This results in excellent fuel/air mixing and uniform cylinder-to-cylinder mixture consistency, even with simple mechanical mixer/carburetors. Accordingly, lean operation with good driveability is easily achieved. Testing by SwRI (2.2L 4-cyl LPG Toyota Forklift) indicates that NOx may be reduced by as much as two-thirds by introducing EGR.

Flame speed and octane rating of LPG are similar enough to gasoline that gasoline engines may be converted to LPG with very little mechanical modification. Thus, a comparatively low cost approach to emission control in this engine category would be to operate on LPG, using lean calibration for controlling HC and CO, and use moderate amounts of EGR for controlling NOx. In converting to LPG from gasoline, the engine will be derated about 15%, due to both LPG's lower volumetric efficiency (gaseous fuels displace more air than liquid fuels), and the effect of lean calibration.

An often overlooked benefit of gaseous fuels is that the fuel system is sealed. LPG vehicles have zero evaporative emissions, and minimal refueling emissions. Thus, converting a gasoline vehicle to LPG would almost totally eliminate these emission sources, without having to install and maintain an evaporative emission control system. In the case of refueling emissions, this is especially beneficial; since many of these engines are refueled in the field, and in-field refueling is not easily amenable to existing techniques for capturing refueling emissions. (Stage II vapor recovery in the field is impractical/difficult).



1 **LP-GAS MOTOR FUEL TANK**
(A) Filter valve (B) Vapor return valve (C) 100% Outage valve (E) Relief valve (F) Vent line to outside of vehicle (G) LP-gas valve (H) LP-gas high pressure hose line (K) Fuel gauge

2 **SS802 HYDROSTATIC RELIEF VALVE**
Vented to outside of vehicle

3 **VFF30 VACUUM FUELOCK AND FILTER**
Prevents flow of fuel when engine stops with ignition switch on. Mount in a fire protected location (H) LP-gas inlet line. (L) Connects to air valve vacuum port in mixer.

4 **IMPCO MODEL EB CONVERTER**
Two stage regulator and vaporizer (M) Water inlet or outlet from engine. Must be brass fittings.

5 **IMPCO AF1-7 AIR CLEANER**
Low profile design, dri-type PCV Connection

6 **IMPCO MODEL *CA225 CARBURETOR**
*Accepted by California Air Resources Board — *Resolution No. 70-9A (T) Idle adjustment screw* (S) Balance line connection

7 **INSTALLATION**
Installations must meet all state and local requirements for location of filling and venting devices.

FIGURE 8. TYPICAL LPG FUEL SYSTEM

Source: Impco Corporation

Safety concerns of using LPG in an industrial environment have largely been addressed by manufacturers of forklift trucks where LPG is widely used. LPG is generally considered safer than gasoline, since it is non-toxic and is stored in a very strong sealed container of proven design.

B. Diesel Engine Emission Control Technologies

Diesel engines generally exhibit high NOx and PM emissions compared to Otto-cycle engines. The diesel combustion process is characterized by high temperatures and pressures which cause relatively high NOx formation. Also, diesel fuel itself is comprised of a variety of heavy hydrocarbon chains. PM emissions form due to the incomplete combustion of the heavy hydrocarbon molecules. Partially burned lubricating oil also contributes to PM emissions. There are a variety of techniques which may be used to reduce the emissions from diesel engines. They can generally be grouped into the following major categories.

- Air induction system modifications
- Fuel injection system modifications
- Exhaust gas recirculation
- Exhaust aftertreatment devices
- Alternative fuels.

1. Air Induction System Modifications

The combustion process in a diesel engine can be made more efficient by improving the airflow into the combustion chamber and increasing the amount of oxygen present in the charge air. Also, the fuel/air mixing process can be enhanced by inducing swirl as the air enters the cylinders. A variety of engine design options/modifications are available to diesel engine manufacturers in the 25 to less than 175 HP range to improve air flow.

a. Intake Manifold and Port Design

Advanced design diesel engines take advantage of the intake manifold and port geometry to induce swirl and tumble as the air enters the combustion chamber. This improves the atomization of the fuel and creates a more well mixed charge, which increases combustion efficiency. The increased swirl also prevents fuel from impinging on the relatively cool cylinder walls, a problem which leads to incomplete combustion and an increase in PM emissions. The design of the intake manifold and port must be matched with other engine design parameters such as injection pressure and spray pattern to effect proper fuel atomization. The diesel engines used in off-road mobile equipment from 25 to less than 175 HP represent a broad range of engine design vintages. Some engine families are relatively old, while others represent state-of-the-art designs that have incorporated advanced airflow techniques. There would appear to be at least some room for improvement in intake manifold and port design for many engine families.

b. Combustion Chamber Modifications

Piston design can have a significant effect on the combustion process. Modern engine designs utilize a variety of innovative piston bowl shapes to improve the air/fuel mixing process. The compression ratio can also be used as a tool for controlling emissions. Higher compression ratios increase BMEP and generally lead to improved efficiency. Higher ratios also reduce the ignition delay (the time between the start of injection and the start of combustion). This effectively allows the designer to retard the timing of the engine without really affecting the start of combustion. The retarded timing will in turn help to lower NOx by limiting the total duration of combustion (and therefore the available time for NOx to form). Finally, several engine manufacturers are continuing to focus on controlling oil consumption through improved piston ring designs and new materials. This will help to reduce particulate emissions. These combustion chamber modifications, however, are not without their drawbacks. Increased compression ratio also increases the mechanical stresses in the crankshaft, connecting rod, piston pin and bearings. Thermal stresses are also increased. In addition, design changes to control oil consumption generally tend to increase engine wear and reduce ring life. Modifications to the diesel engine in these areas of design must be made very carefully to take full advantage of the emission benefits without compromising engine reliability and durability. Again, the diesel engines in this category represent a continuum of development stages in these areas of product design. We suspect that several engine manufacturers will be able to further reduce both NOx and PM emission through continued work in combustion chamber design. The degree of improvement will be highly dependent on specific engine design and technological sophistication of the engine.

c. Turbocharging

Turbocharging has long been recognized as a means of increasing power in diesel engines. Turbocharging, in conjunction with other design changes, can provide lower PM emissions. Turbochargers are more efficient than superchargers since they effectively use the heat energy in the exhaust (that would otherwise be wasted) to power the turbine. Turbochargers can be used together with retarded timing to control both NOx and PM emissions.

SwRI has evaluated the effects of turbocharging by adding a turbocharger to an existing naturally-aspirated diesel engine. The engine was emission tested before and after turbocharging using both an 8-mode ISO 8178 test schedule and a hot transient test cycle. The engine tested was a 3.9 liter, direct injection, water-cooled, in-line 4 cylinder diesel rated at 75 HP. The manufacturer of this engine offers a turbocharger "add-on" package that is used primarily for altitude compensation. Calibration of the turbocharger package is therefore aimed at increasing power output at high altitudes. The manufacturer has indicated that considerable development work would be required to optimize the turbocharger package for performance, emissions and durability. As shown in Figure 9, turbocharging the engine reduced PM emissions by nearly 50 percent in the eight-mode cycle, while causing a slight increase in NOx. HC and CO levels were also reduced by roughly 50 percent.

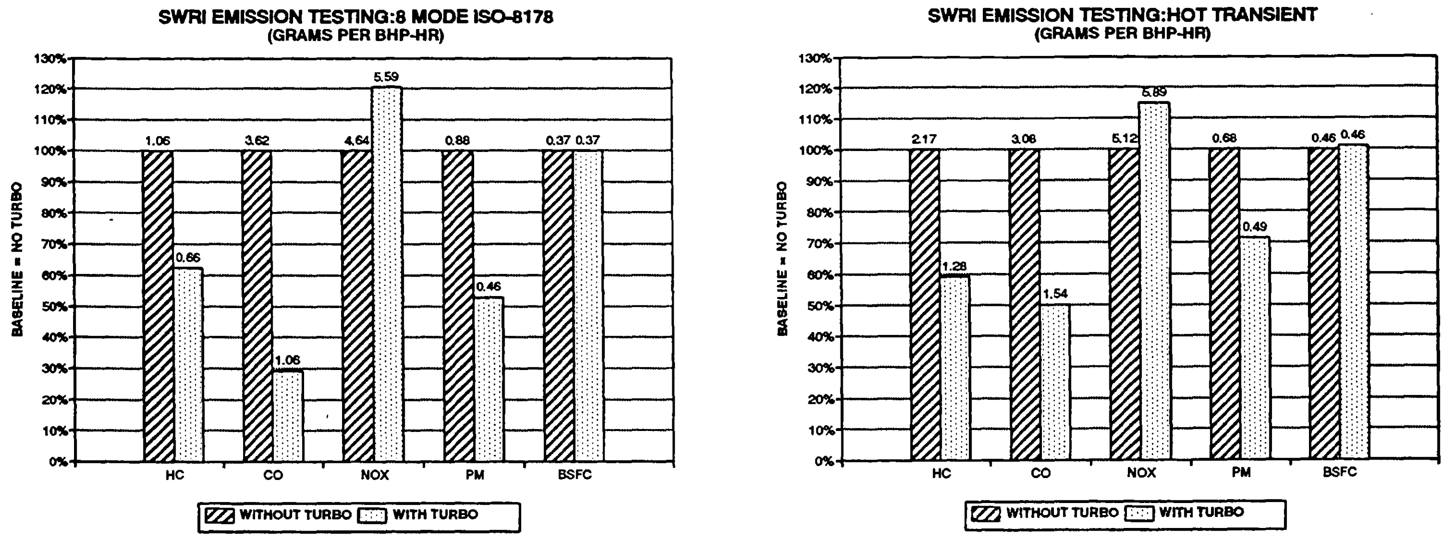


FIGURE 9. EMISSION IMPACTS FROM TURBOCHARGING A DIESEL ENGINE

The adaptation of an existing naturally-aspirated diesel engine to turbocharging, however, may require significant engine modifications. For example, compression ratio and intake port design should change to match the new airflow characteristics. The piston bowl design and fuel injection spray pattern and pressures could also ideally be modified to complement the effects of turbocharging. Turbocharging will increase the cooling requirements of the engine so that a new water pump, radiator and/or cylinder head design may be necessary. Finally, adapting an air-cooled and/or indirect injection diesel engine with a turbocharger may be extremely difficult due to the added thermal stresses imposed by the turbocharger. Most air-cooled and IDI engine designs are already at the limits of acceptable operating temperature for good durability and reliability.

d. <u>Charge-Air Cooling</u>

Charge-air cooling is also a commonly used design technique for on-road diesel engines to reduce NOx and increase fuel efficiency. Air coming out of the turbocharger is quite hot after being compressed and is less dense than cooler air. The oxygen mass is therefore lower on a volumetric basis. An "intercooler" or radiator-type device can be positioned between the turbo outlet and the intake manifold to cool the charge air and increase its density. Heavy-duty diesel engine manufacturers utilize both water-to-air and air-to-air aftercooling devices to cool the charge air. The air-to-air type aftercooler can yield lower charge air temperatures, since ambient air is much cooler than engine coolant. However, because the heat transfer coefficient is lower for air-to-air than for water (or coolant)-to-air designs, the size of the aftercooler must be larger for air-to-air aftercooler units to effectively cool the charge air, which may exacerbate packaging problems.

Air-to-air aftercooling may not be easily implemented on much of the equipment in the below-100 HP off-road mobile source category. Much of the equipment is not turbocharged and will therefore not need an aftercooler, and packaging constraints may limit its application.

2. Fuel Injection Systems

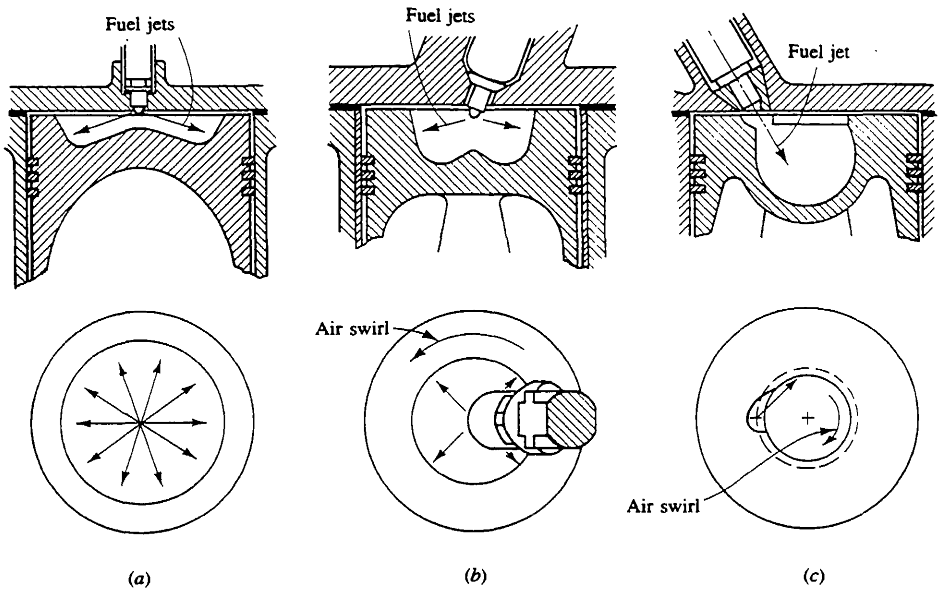
The task of the fuel injection system is to meter the appropriate quantity of fuel to each cylinder for a given engine speed and load. Also, the fuel must be injected at the appropriate time in the cycle, at the desired rate, and the correct spray configuration required for the particular combustion chamber employed. It is important that injection begin and end cleanly, avoiding any secondary injections. To accomplish this task, fuel is usually drawn from the fuel tank by a supply pump, and forced through a filter to the injection pump. The injection pump sends fuel under pressure to the injection lines, which carry fuel to the injector nozzles located in each cylinder head. Excess fuel, circulated through the system for cooling, goes back to the fuel tank.

Engine designers can use a variety of modifications to the diesel fuel injection system to help reduce NOx and PM emissions. These modifications focus on injection timing control, injection pressures, and injection spray pattern and rate shaping.

a. Indirect Versus Direct Injection Diesel Engines

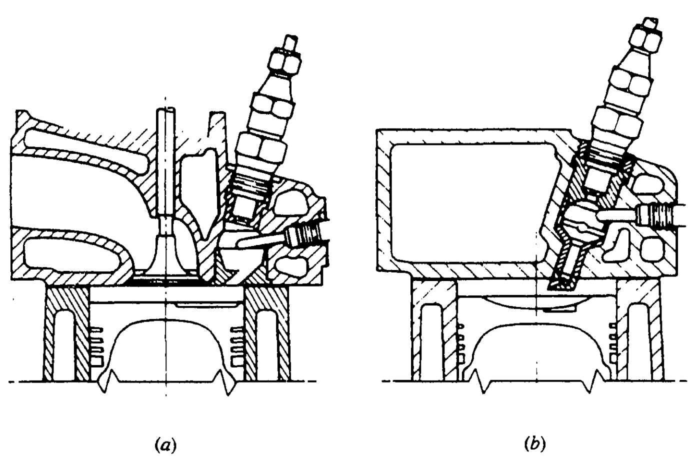
Diesel engines are divided into two basic categories according to their combustion chamber design: (1) *direct-injection (DI) engines*, which have a single open combustion chamber into which fuel is injected directly; and (2) *indirect-injection (IDI) engines*, where the chamber is divided into two regions and the fuel is injected into the "prechamber" which is connected to the main chamber (situated above the piston crown) via a nozzle, or one or more orifices. Typical configurations of combustion chamber and cylinder head designs are shown in Figure 10 for direct injection engines, and in Figure 11 for indirect injection engines. The important characteristics of the most common types of both DI and IDI engine designs are presented in Table 48.

The diesel engines in the 25 to less than 175 HP category are dominated by the so-called "high swirl" DI and IDI designs, as indicated in Table 48. In this HP range and product category, the DI engines are by far the most popular, due primarily to improved fuel efficiency compared with similarly designed IDI engines. Direct injection engines also tend to consume less oil, and operate at lower temperatures so that cooling requirements are less demanding than for IDI engines. An excellent comparison between DI and IDI performance characteristics can be demonstrated based on Lister-Petter's LPW series engines which are offered in both DI and IDI versions for identically sized 2,3 and 4 cylinder models. All engine models are essentially the same except for the DI versus IDI difference. Table 49 presents a comparison of fuel consumption for the different engines at a variety of operating conditions. Output power at each test point was the same for the two engines. Overall, the DI versions (designated as LPW models) are 10 to 15 percent more efficient than the IDI versions (designated as LPWS models).



Common types of direct-injection compression-ignition or diesel engine combustion systems: (*a*) quiescent chamber with multihole nozzle typical of larger engines; (*b*) bowl-in-piston chamber with swirl and multihole nozzle; (*c*) bowl-in-piston chamber with swirl and single-hole nozzle. (*b*) and (*c*) used in medium to small DI engine size range.

FIGURE 10. TYPICAL DIRECT INJECTION COMBUSTION CHAMBER DESIGNS



Two common types of small indirect-injection diesel engine combustion system: (*a*) swirl prechamber; (*b*) turbulent prechamber.

FIGURE 11. TYPICAL INDIRECT INJECTION COMBUSTION CHAMBER DESIGNS

TABLE 48. CHARACTERISTICS OF COMMON DIESEL COMBUSTION SYSTEMS

System	Direct injection				Indirect injection	
	Quiescent	Medium swirl	High swirl "M"	High swirl multispray	Swirl chamber	Pre-chamber
Size	Largest	Medium	Medium—smaller	Medium—small	Smallest	Smallest
Cycle	2-/4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke
Turbocharged/supercharged/naturally aspirated	TC/S	TC/NA	TC/NA	NA/TC	NA/TC	NA/TC
Maximum speed, rev/min	120–2100	1800–3500	2500–5000	3500–4300	3600–4800	4500
Bore, mm	900–150	150–100	130–80	100–80	95–70	95–70
Stroke/bore	3.5–1.2	1.3–1.0	1.2–0.9	1.1–0.9	1.1–0.9	1.1–0.9
Compression ratio	12–15	15–16	16–18	16–22	20–24	22–24
Chamber	Open or shallow dish	Bowl-in-piston	Deep bowl-in-piston	Deep bowl-in-piston	Swirl pre-chamber	Single/multi-orifice pre-chamber
Air-flow pattern	Quiescent	Medium swirl	High swirl	Highest swirl	Very high swirl in pre-chamber	Very turbulent in pre-chamber
Number of nozzle holes	Multi	Multi	Single	Multi	Single	Single
Injection pressure	Very high	High	Medium	High	Lowest	Lowest

TABLE 49. FUEL ECONOMY COMPARISON OF DI VERSUS IDI ENGINES

No. of Cylinders	Engine Design	Fuel Consumption @ rpm (gal/hr: 75% load)					Average Increase from DI Baseline
		1500	1800	2000	2500	3000	
2	DI	0.39	0.47	0.53	0.66	0.82	
	IDI	0.43	0.53	0.61	0.74	0.92	
	% difference	10%	13%	15%	12%	12%	12.4%
3	DI	0.59	0.71	0.79	0.99	1.22	
	IDI	0.64	0.78	0.91	1.11	1.37	
	% difference	8%	10%	14%	12%	12%	11.2%
4	DI	0.79	0.95	1.05	1.32	1.63	
	IDI	0.85	1.04	1.22	1.48	1.83	
	% difference	8%	9%	16%	12%	12%	11.4%

While fuel economy and performance are advantages of the DI design, IDI engines offer considerably lower emissions of both NOx and particulates, at least for naturally aspirated diesel engines. Again, the Lister-Petter engines referred to in Table 49 offer an excellent comparison of the emission characteristics typical of IDI versus DI engines. Lister-Petter has tested these engine families for NOx emissions using a European test cycle (ECE R49-01). NOx emissions are reported in grams per kWh. Smoke levels are reported as a "K" factor, with lower numbers indicating less particulates in the exhaust. Test results are presented in Table 50 (LPW models are DI, and LPWS models are IDI).

TABLE 50. EMISSION TEST RESULTS FOR SIMILARLY DESIGNED IDI VERSUS DI ENGINES (LISTER-PETTER LPW/LPWS SERIES ENGINES)

	Gaseous Emissions					Smoke Emissions	
	Emissions, g/kWh			Regulation ECE R49-01	Directive EEC 88/77	Free Acceleration "K"	Regulation ECE R24-03
	NO_x	HC	CO				
LPW2	7.19	2.06	4.23	E11 49R 011200	UK 1200	0.81	E11 24R 03625
LPW3	7.18	1.74	4.71	E11 49R 011201	UK 1201	0.94	E11 24R 03626
LPW4	7.66	2.03	4.76	E11 49R 011202	UK 1202	0.83	E11 24R 03627
LPWS2	4.29	2.04	2.07	E11 49R 011197	UK 1197	0.69	E11 24R 03628
LPWS3	4.39	0.57	1.40	E11 49R 011198	UK 1198	0.44	E11 24R 03629
LPWS4	4.46	1.08	1.67	E11 49R 011199	UK 1199	0.53	E11 24R 03630

Based on this testing, the IDI engine design reduced NOx by 40 to 45 percent, and smoke by an average of about 35 percent, compared to the DI engine. It should be understood that these results are for a single specific engine and may not necessarily hold true for engine designs from other manufacturers. In the IDI engine, the high rate of combustion and heat release in the prechamber typically produces high levels of particulates. Particulates undergo increased burning as the combustion process moves into the main chamber since additional oxygen (and time) are available. NOx emissions in the IDI engine are typically reduced due to the richer conditions in the prechamber, and the expansion and cooling of the combustion gases in the main chamber. Peak temperatures in the main chamber are reduced compared to DI engines, thereby lowering NOx. PM emissions from an IDI engine can be either lower or higher than a DI engine depending on injection pressures, combustion chamber geometry, compression ratio and many other factors. IDI engines are used extensively in mining applications, which demand minimum emission levels. Engine manufacturers could convert to IDI engine designs to lower emissions, but customers would suffer a 10 to 15 percent fuel economy penalty. Product cost is also about 10 percent higher for IDI engines compared to DI designs, all else being equal.

The Lister-Petter LPWS4 engine listed in Table 50 was also tested by SwRI using the 8-mode ISO 8178 test procedure. Results of that testing are presented in Table 51. The NO_x emission rate of 4.34 grams per kWh compares closely with the NO_x emission rate of 4.46 grams per kWh for the LPWS4 listed in Table 50.

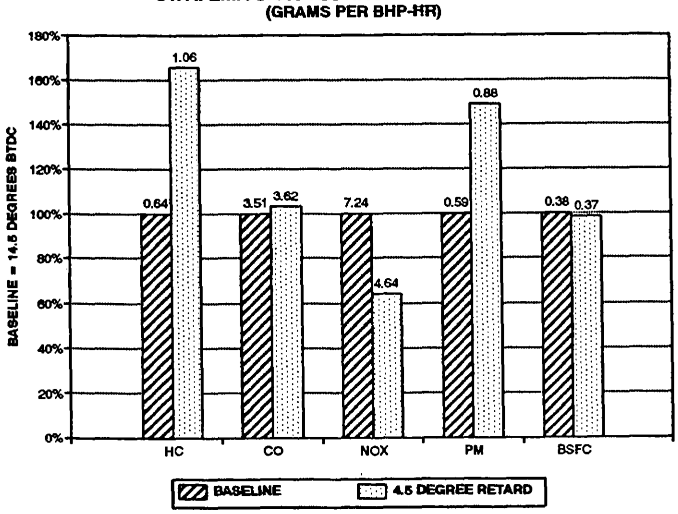
TABLE 51. SWRI EMISSION TEST RESULTS OF LISTER-PETTER LPWS4 IDI ENGINE (ISO 8178 TEST CYCLE)

Test Date	Test I.D.	Emissions (g/kWh)				BSFC, kg/kWh
		THC	CO	NOx	PM	
8-11-92	WCD1	1.56	3.45	4.33	0.72	0.31
8-13-92	WCD2	1.38	3.62	4.34	0.62	0.30
Mean Baseline		1.48	3.54	4.34	0.68	0.31
Results converted to metric units						

b. Injection Timing

Retarding the injection timing can have a profound effect on reducing NOx emissions from diesel engines in this product and HP category. By retarding the injection timing, the peak combustion chamber temperature and pressure moves beyond top-dead-center, thereby lowering the peak combustion chamber temperature and reducing NOx. On the downside, retarding the timing will tend to increase PM emissions since the combustion process may not be fully complete when the exhaust valve opens, and fuel efficiency will deteriorate since the BMEP is reduced. Also, the exhaust valve temperatures will tend to increase since combustion is essentially occurring later in the cycle (and closer to exhaust valve opening). To evaluate the effects of timing on diesel engines in this HP range, SwRI performed emission testing on a 3.9L, naturally aspirated direct injected diesel engine rated at 75 HP. Testing was performed at baseline timing and at 4.5° retarded timing. The engine was tested using both the 8-mode ISO 8178 test cycle and a hot transient test cycle. The results of the testing are shown in Figure 12.

As shown in Figure 12, NOx emissions were reduced by about 36 percent based on the 8-mode testing. Particulate emissions, however, increased by about 50 percent, as did HC. It is interesting to note that the transient emission testing showed similar directional results with regard to NOx and PM emissions, but the effects were not as pronounced.



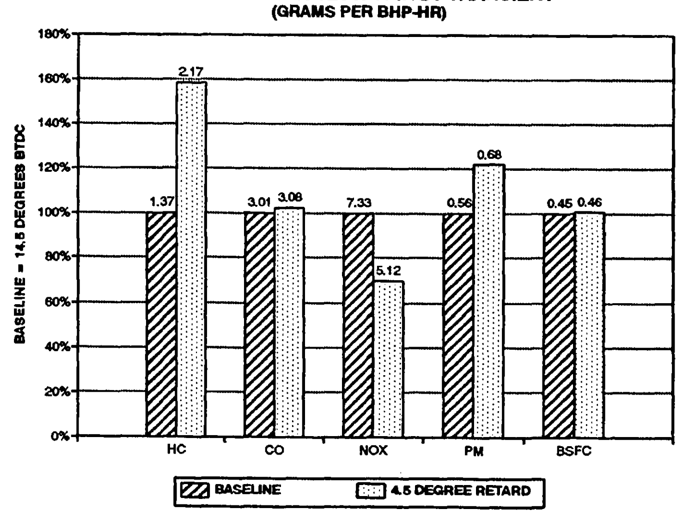


FIGURE 12. EFFECTS OF TIMING RETARDATION ON DIESEL ENGINE EMISSIONS

The diesel engines in this HP and product category have not been optimized for low emissions, and injection timing has likely been set for optimum fuel economy and power. Injection timing could probably be retarded slightly to yield NOx emission reductions of 10 to 20 percent with only a moderate increase in particulate emissions. Additional engine design modifications could then be employed to offset the increase in PM emissions including: improved airflow management as previously discussed (i.e., intake/port design, turbochargers, aftercooling), combustion chamber design, oil control, higher injection pressures, and improved injection nozzle design. Improved diesel fuel formulations (low sulfur, low aromatics, high cetane) will also help to limit increases in PM emission from retarded timing. Retarded timing is therefore a technique that can be employed by most of the diesel engines in this product category to reduce NOx. Indirect injection and air-cooled engines will likely be able to tolerate less timing retardation due to the increased operating temperatures (at the exhaust valve and seat area) caused by extending the combustion process later in the cycle. It also should be noted that without some of the offsetting engine design changes previously mentioned (to address increased PM emissions), retarding the timing can cause slight increases in fuel consumption (perhaps less than 2 percent).

c. Injection Pressure

Increasing the injection pressure in a direct injected diesel engine effectively serves to improve the atomization of the fuel, resulting in smaller fuel droplets, more turbulent mixing of the air and fuel, and a more efficient combustion process. Particulate emissions will therefore be lowered, which in turn will permit further timing retardation to control NOx. Increasing the injection pressure can also help facilitate a reduction in the duration of the injection event without a loss in injection quality. This can be accomplished by modifying the cam in the injection pump that controls the shape and rate of the fuel injection event, and by redesign of the fuel injector itself. Higher injection pressures will permit more fuel to flow through the nozzle tip without increasing the size or number of spray holes, which could deteriorate the spray pattern. Reducing the duration of the fuel injection event makes possible further timing retardation without excessive loss of power or increased smoke. However, excessively short injection periods can have the negative effect of increasing peak pressures and limiting the complete combustion of the fuel. Also, increasing injection pressures is not desirable for indirect injection engines, since high velocity spray patterns can cause fuel to spray outside of the prechamber, thus deteriorating combustion quality.

Two major types of fuel injection pumps are commonly employed for delivering and metering the fuel; rotary distributor-type pumps, and in-line injection pumps.

Distributor-type rotary injection pumps are generally used on smaller DI engines (less than 75 HP), and on the full range of IDI engines. Current design rotary injection pumps produce no more than about 5000 to 7000 PSI line pressure, and are available from a limited number of manufacturers including Standadyne, Lucas, and Robert Bosch. These pumps have only one plunger and barrel. A rotating cam plate acts to both stroke the plunger and accurately meter fuel to each injection nozzle. The unit, therefore, combines the functions of a pump and distributor. A schematic of a rotary type pump is shown in Figure 13.

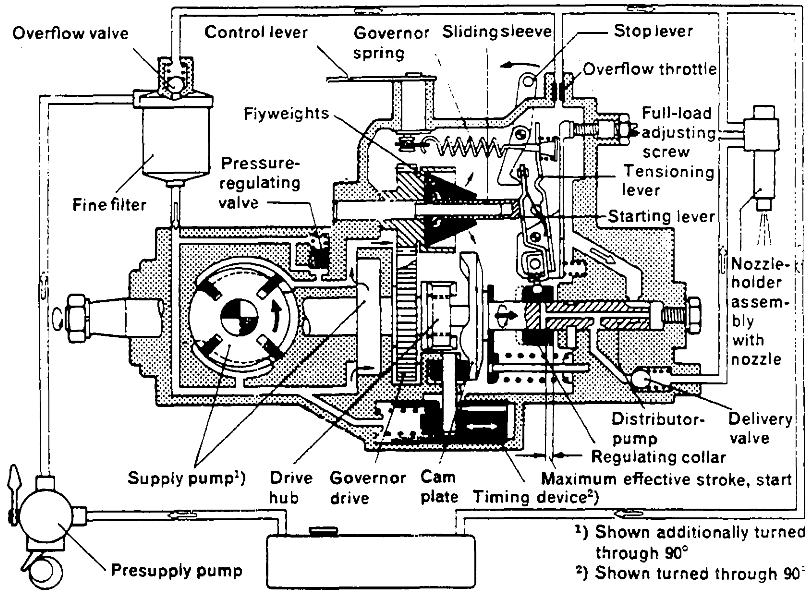
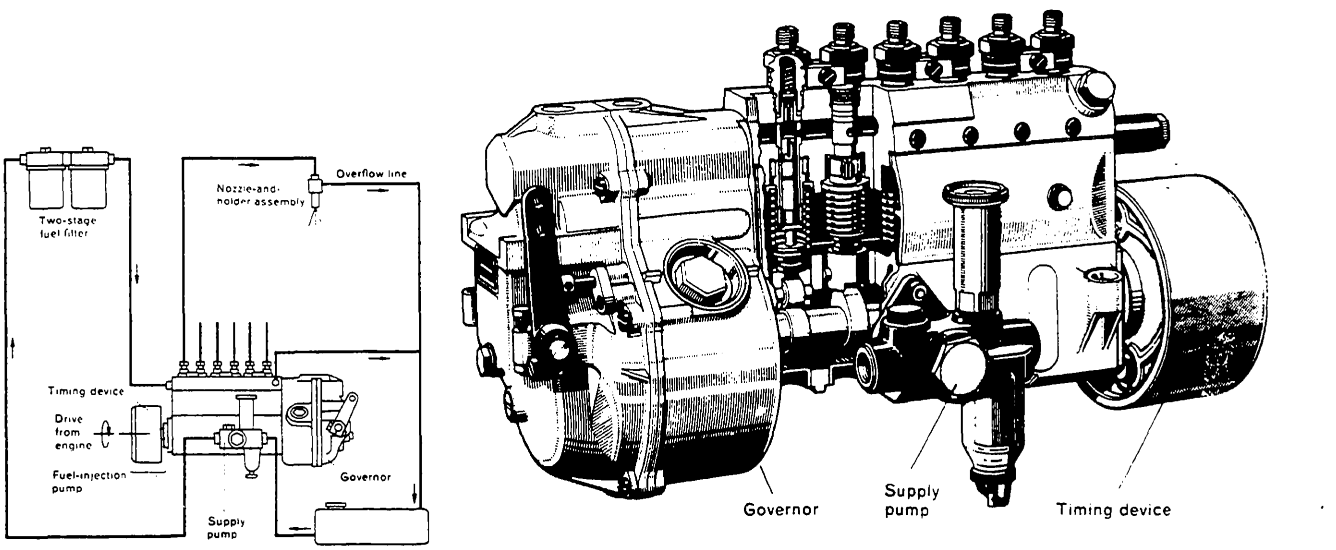


FIGURE 13. TYPICAL ROTARY TYPE DIESEL FUEL INJECTION PUMP

These units are usually driven by a power take-off from the timing gear. Rotary injection pumps are generally more compact and less expensive than in-line type pumps, but cannot achieve as high injection pressures. Rotary pumps are about 50 to 60 percent of the cost of a similarly sized in-line pump. An important feature of the rotary pump is the ability to rather easily adjust injection timing based on speed and load of the engine. In other words, the injection timing can be modified (albeit to a fixed degree) with a change in engine load or speed. This feature can effectively be used to advance timing during starting and low load conditions (to improve startability and idle quality), and then retard timing at high load conditions to reduce NOx. Injection system manufacturers are experimenting with very high pressure design rotary injection pumps, but cost of the pumps will likely be very high (higher than in-line pumps) due to apparent limited demand.

In-line injection pumps are predominantly used in medium to large direct injection diesel engines, but are also used in some relatively small engines. In-line injection pumps used on engines in the 25 to 100 HP range are capable of pressures in the 7,000 to 12,000 PSI range. The very latest design in-line injection pumps are capable of even higher pressures, perhaps up to 20,000 PSI. An in-line pump contains a plunger and barrel assembly for each cylinder. Each plunger is raised by a cam on the pump camshaft and is forced back by the plunger return spring. The amount of fuel delivered is altered by varying the effective plunger stroke. This is achieved by means of a control rod or rack, which moves in the pump housing and rotates the plunger via a ring gear or linkage lever on the control sleeve. A typical in-line injection pump is shown in Figure 14.



Diesel fuel system with in-line fuel-injection pump (type PE).[12] *(Courtesy Robert Bosch GmbH.)*

FIGURE 14. TYPICAL IN-LINE DIESEL INJECTION PUMP

In-line injection pumps are also driven off the timing gear. The drawback of in-line injection pumps (in addition to being much more costly than rotary pumps) is that they cannot easily accommodate timing changes based on engine operating conditions. Therefore, timing is fixed and must represent a compromise setting over the entire speed/load range. (However, this is often not a major drawback since diesel engines in this product category do in fact generally operate over a relatively narrow speed/load range). Over the past few years, injection pump manufacturers have begun to offer electronically controlled versions of their in-line injection systems which are capable of complete timing control as well as control of other injection parameters. However, these pumps are expensive, often costing nearly as much as the engines they are used on.

Also, a variant of the in-line injection pump system is being offered by selected manufacturers on recently re-designed engines in the medium size diesel product category. This system utilizes an individual injection pump positioned at each cylinder and driven off of the same camshaft that operates the valves. The output of each injection pump is delivered to the fuel injector via a short high pressure fuel line. This design improves servicing and maintenance of the fuel injection system and should increase reliability. Such a system is offered on Lister-Petter's recently introduced "Alpha Series" engine line. The injection pressure capability for this type of system is similar to standard in-line injection pumps.

Diesel engine manufacturers who currently use rotary type injection pumps could convert to in-line pumps to increase injection pressures and thereby lower NOx and PM emissions. Both types of pumps are driven in a similar manner (off the timing gear), and we suspect that packaging of the in-line pump would not be a problem for most manufacturers and equipment applications. Higher cost will be the major drawback for this emission control technique. Some engine manufacturers who are already using an in-line injection pump may be able to convert to even higher pressure in-line pumps, but at some cost penalty. The benefits of relatively small increases in injection pressure will be highly dependent on specific engine designs. Finally, advanced design high pressure electronically controlled in-line injection pumps could be installed on many of the diesel engines in this HP range to offer both an increase in injection pressure and full timing control. Significant reductions in NOx and PM could likely be achieved - - probably in the range of 20 to 40 percent for both effluents. As noted, however, this type of injection pump is very expensive.

d. Injector Nozzle Design

Fuel injection spray pattern characteristics can have a significant impact on combustion quality in direct injected diesel engines. The injector spray pattern must be designed to compliment the combustion chamber design and airflow characteristics of the engine. Considerable advances in fuel injector design have occurred over the last several years primarily as a result of on-road diesel engine emissions research. Fuel injectors exhibit a tendency to leak slightly just before and just after the actual beginning and end of injection. This phenomenon can lead to increased smoke and PM emissions. The most recent fuel injector designs incorporate features that significantly reduce or eliminate this leakage. Research focused on optimizing spray tip hole diameter and length, orientation, and number of holes has also led to improved combustion and emission characteristics. The injector nozzles used in diesel engines in the 25 to 100 HP off-road category have generally been optimized for performance and fuel economy. The latest design engine offerings from selected manufacturers likely incorporate state-of-the-art injector designs, but many engine families could benefit from improved injector design and research. Incorporating features of injector nozzles used in state-of-the-art on-road diesel engines, and applied research in this area, should offer designers of small/medium size off-road diesel engines some opportunity for reducing emissions.

3. Exhaust Gas Recirculation

Exhaust gas recirculation (EGR) has long been recognized as a means for reducing NOx emissions in diesel engines. A typical EGR system taps into the exhaust gas flow directly at the output of the exhaust manifold and redirects a portion of the exhaust back into the intake. This effectively dilutes the intake charge with gases (CO_2 and N_2) that have a higher heat capacity than air. The peak combustion temperature is thereby reduced, thus lowering NOx formation. At light load operating conditions, EGR can have a beneficial effect on emissions *and* combustion quality. The hot exhaust gas can reduce ignition delay, thus reducing NOx at light loads. At higher loads the EGR tends to reduce NOx by simply reducing peak combustion chamber temperatures. Moderate levels of EGR can reduce NOx emissions by 20 to 30 percent in uncontrolled engines. The major drawbacks of EGR, however, are increased smoke and PM emissions, and a severe increase in engine wear due to contamination of the lube oil with carbonaceous material. Several engine manufacturers have experimented with EGR in the past and have generally found that engine durability and reliability are compromised with increasing use of EGR. Recent research efforts in the industry to improve EGR systems have focused on precisely controlling the flow rate and temperature of the recirculated exhaust gas at varying engine operating conditions. Such systems would presumably utilize modulated EGR flow valves controlled by an electronic processor. The processor would control the position of the valves based on engine speed, load, temperature and other operating conditions. In addition, research has focused on improving the wear characteristics of combustion chamber components in the presence of EGR. This research is directed at improved materials and lubricating oils. It should be noted that EGR is considerably more difficult to apply in turbocharged engines.

It would certainly appear feasible to implement simple, non-modulated EGR systems on small to medium size (25-100 hp) diesel engines to reduce NOx emissions in the 10 to 30 percent range. The real development challenge, however, will be to limit increases in PM and smoke, and have acceptable engine durability. Engine durability testing must address the variety of applications and operating conditions in which a particular engine model will be required to perform. Also, the EGR development program must be coordinated with other engine modifications targeted at reducing emissions. Such programs require careful planning, and can be protracted in the absence of materials, lubricating oil and design "breakthroughs." On the positive side, the reformulated diesel fuel to be introduced into California in 1993 should help to considerably reduce durability problems associated with EGR. Several studies have suggested that increased engine wear associated with EGR is directly linked to the sulfur content in the fuel and the formation of sulfuric acid. The very low sulfur content in California diesel fuel should reduce these concerns and make EGR more practical.

4. Aftertreatment Devices

Exhaust aftertreatment devices are the focus of extensive research efforts for on-highway heavy-duty diesel engines. Aftertreatment devices could be applied to small and medium size diesel engines as well, but cost and packaging issues will be especially acute concerns for much of the equipment used in the off-highway category. Aftertreatment devices can be classified by the type of effluent(s) targeted for reduction:

- Particulate Reduction:
  - Particulate Traps
  - Oxidation Catalysts

- NOx Reduction
  - Selective Catalytic Reduction (SCR) using ammonia as the reducing agent
  - DE-NOx catalysts which use hydrocarbons in the exhaust as the reducing agent.

NOx reduction aftertreatment devices are in the early stages of development. Preliminary testing indicates that SCR systems are capable of NOx reductions up to 80 percent while reduction efficiency of the DE-NOx hydrocarbon catalysts is somewhat less. These systems are relatively complex and expensive. Most industry experts agree that catalytic diesel NOx reduction systems are 7 to 10 years away from being commercially viable for heavy-duty on-road applications. Commercial viability for off-road small to medium size diesel engines is likely even further away due to the previously mentioned cost and packaging constraints.

Particulate emissions in diesel engine exhaust can be reduced through the use of either particulate traps, or oxidation type catalysts. Particulate traps are by far the more tested technology while oxidation catalysts have been the focus of intensive research in recent years as a means of achieving significant reductions in PM emissions without the complexity of particulate traps. These two systems are discussed below.

a. Particulate Traps

Particulate traps are capable of reducing PM emissions by 70 to 90 percent. There are several design approaches for particulate traps, but most use a ceramic monolith element to filter the particulate material in the exhaust. The key to successful operation of traps is the ability to efficiently burn off the particulate material that has accumulated in the ceramic element. This process is referred to as "regeneration," and must be periodically initiated to prevent the filter from clogging and increasing exhaust backpressure. Particulate traps are approaching commercial viability for on-highway truck applications. The most popular model in use today is the *Donaldson Electrically Regenerated* wall flow trap. This trap is being used by DDC on their 6V-92 diesel engine for transit bus applications. This system, while effective in reducing emissions, is also quite bulky. A schematic of the system is shown in Figure 15.

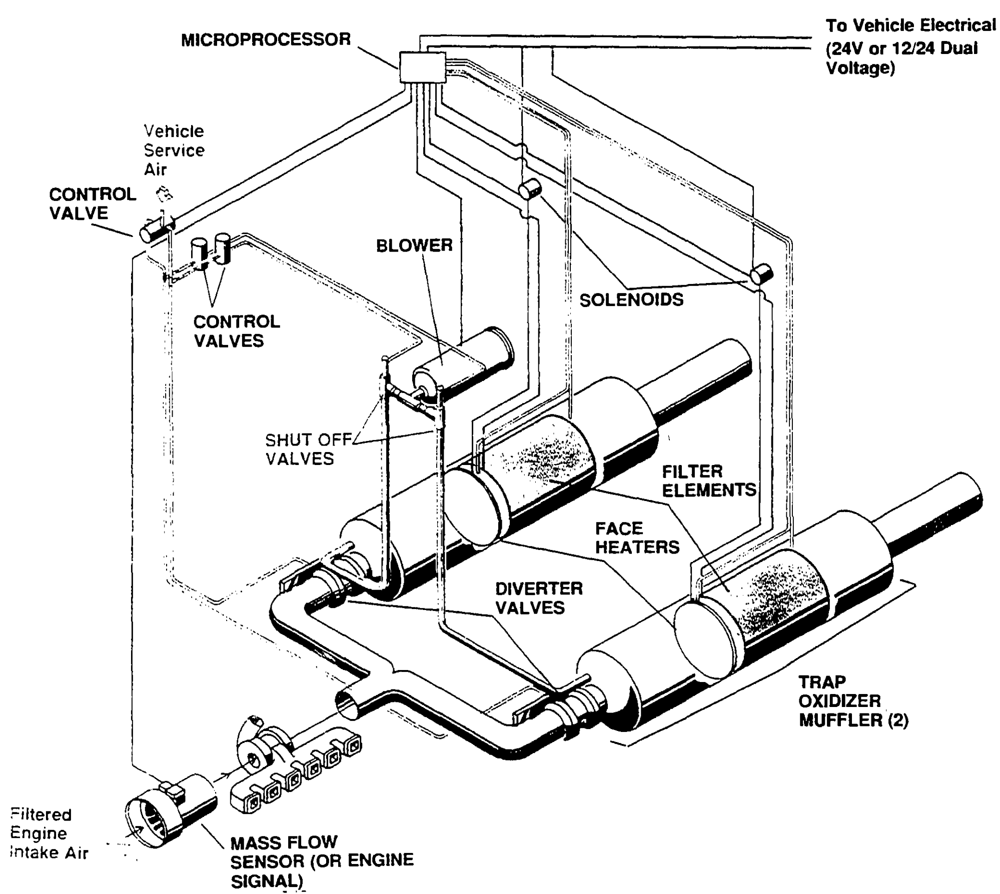


FIGURE 15. DONALDSON DUAL TRAP SYSTEM

We believe it would be quite difficult to adapt such a particulate trap system to small to medium size (25-100 hp) diesels. Again, packaging of the system would be difficult for much of the equipment/vehicles in the off-highway mid-size equipment category, and the cost of the system to the end user would be quite high. For example, transit buses recently purchased by NYCTA equipped with the Donaldson trap system cost $8,000 more than a standard bus. These costs reflect the additional engineering and systems costs of this pilot program. These systems would cost less in mass production.

b. Diesel Oxidation Catalysts

Diesel particulates are composed of both insoluble and soluble matter. Insoluble particulates include carbonaceous matter, ash and sulfates, while soluble particulates are made up of partially burned lube oil and fuel. Different engines exhibit varying percentages of these two types of particulate material in their exhaust. However, the soluble organic fraction (SOF) of total particulates is in the 15 to 50 percent range for most engines. Diesel oxidation catalysts are similar in design to those used with gasoline engines. However, the diesel catalyst must be effective at the lower temperatures characteristic of diesel exhaust. Catalytic converters are effective in oxidizing organic compounds in the gaseous phase, but will not oxidize solid carbonaceous particles. Therefore, a catalyst reduces diesel PM by oxidizing the SOF, but allows the insoluble solid particulate to escape. Recent testing of oxidation catalysts shows that they are capable of reducing total PM emissions by as much as 25 to 40 percent. This is accomplished by oxidizing 50 to 80 percent of the SOF present. Oxidation catalysts are most effective with engines producing "wet" particulate having a high proportion of SOF. "Dry" engines with excellent oil control, for example, have largely solid particulate emissions, and so will be less suitable for catalytic PM aftertreatment than engines with poorer oil control. Oxidation catalysts are quite simple compared to particulate traps and require no regeneration or auxiliary controls. Very little price information is available for oxidation catalysts, but cost should be only a fraction of that of a particulate trap system. Oxidation catalysts, however, have not yet achieved commercial viability. Sulfur and phosphorus (an oil additive) tend to deactivate catalytic converters. Diesel engine oil additive packages may have to be reformulated for catalytic converters. Low sulfur fuel (less than 0.05% S) will clearly be needed with diesel catalysts. A major design challenge remaining in the development of diesel oxidation catalysts is to achieve acceptable durability, even at the sulfur concentrations (0.02 – 0.04%) characteristic of low sulfur fuel.

Sulfur presents another design challenge as well. Sulfur is present in diesel exhaust as SO_2. If the catalyst oxidizes the SO_2 to SO_3, the SO_3 easily combines with water to form sulfuric acid (H_2SO_4). The sulfuric acid then collects on the particulate filter during emission testing and counts as particulate material. Thus, the diesel catalytic converter must selectively oxidize exhaust hydrocarbons without oxidizing SO_2. Since hydrocarbons efficiently oxidize at lower temperatures than SO_2, one approach to controlling SO_3 formation is to locate the catalytic converter in the exhaust system to optimize operating temperature. In any event, additional research is needed to develop selective SOF conversion catalyst technologies.

Oxidation catalysts are approaching commercial viability. Because of their relatively simple packaging and low cost (compared to a particulate trap), they could offer a cost-effective approach for controlling PM emissions on small to medium size (25-100 hp) diesel engines.

5. Alternative Fuels for Diesel Engines

The conversion of diesel engines to alternative fuels has been extensively demonstrated in medium- and heavy-duty on-road trucks and buses. Technology for converting diesel engines to methanol, natural gas and LPG operation is the focus of considerable research and development both in the U.S. and other countries. Feasibility has clearly been demonstrated, as have the emissions benefits. Development work on these engine conversion technologies is currently focused on improving reliability, durability, and fuel economy, as well as further reductions in emissions. Essentially, all of the technological developments and principles which make possible the use of alternative fuels in on-road diesel engines can also be applied to the smaller off-road diesel engines. Similar emission reduction benefits can be expected in the small to medium size diesel engines as were realized in the slightly larger diesel truck and bus engines. NOx emissions could be reduced by as much as 50 to 75 percent from diesel levels, and PM emissions could be cut by about this same amount by converting to methanol or natural gas operation. However, as with the larger on-road diesel engines, major engine development programs would be required to optimize the conversion of these small to medium size diesel engines to alternative fuels. Also, for some of the equipment in the off-road category, the packaging constraints associated with larger and heavier fuel systems (i.e., CNG and methanol) could be a significant application engineering challenge. Finally, refueling logistics for selected off-road mobile source equipment would be particularly troublesome. Unlike large, centrally fueled fleets, much of the equipment is operated in remote sites with only a few pieces of equipment at each site. CNG refueling would be especially difficult. The use of alternative fuels for reducing emissions in off-road small to medium size diesel equipment would appear to be a long term strategy that could be exercised after other more conventional approaches had been implemented.

6. Summary of Emission Reduction Strategies for Diesel Powered Equipment

Several means for reducing NOx and particulate emissions from diesel engines have been identified in this section. The various approaches for reducing emissions from diesel engines can broadly be defined based on the difficulty and/or time required for implementing the control strategy. We have categorized the emission reduction techniques described in this chapter in Table 52.

TABLE 52. DIESEL ENGINE EMISSION REDUCTION STRAGEGIES

Near Term	Little or no design modifications	• Injection timing • IDI engines (in place of DI engines) • Improved diesel fuel formulations
Medium Term	Adaptation of existing technologies; moderate design modifications	• Increased injection pressures • Improved injector nozzles • Use of EGR • Particulate traps • Oxidation catalysts • Charge air cooling
Long Term	Major engine modifications, or engine redesign	• Intake manifold and port design • Combustion chamber modifications • Turbocharging • Alternative fuels

The likely range of emission reduction effectiveness for each control strategy is described in the previous sections along with a discussion of associated operational implications. Essentially all of the control strategies described have been implemented, or are being developed for on-highway diesel engines. Some of the control strategies can be readily adapted to off-road industrial engines. Such strategies include retarded injection timing, increased injection pressures and improved injector nozzle design, and the use of EGR. Other strategies, such as the use of aftertreatment devices or charge air cooling could be particularly cumbersome to implement due to packaging constraints on selected types of equipment in the industrial category. The use of alternative fuels for off-highway diesel industrial equipment would require significant engine redesign, and would be difficult to implement due to the limited refueling infrastructure for alternative fuels.

It is also important to note that many of the engine design modifications described in this section do not necessarily scale well (from a cost perspective) for small and medium size (25-100 hp) engines. Turbochargers, high pressure injection pumps, electronic unit injectors, engine electronics and various types of sensor units were all originally developed for comparatively high horsepower, high volume truck engines. Unfortunately, while a 100 HP engine may be 50 percent of the price of a 250 HP engine, the turbochargers needed for each engine would cost nearly the same. Similar dis-economies of scale also apply to sophisticated injection pumps and electronic controls. The major portion of the costs of such components is attributable to high tolerance machining, precise assembly techniques, and engineering and design costs. Basic material costs (which do vary more directly with engine size) represent a rather small portion of the cost of such emission control equipment.

Finally, it is important to recognize that to maintain product reliability and integrity, a systems design approach to reducing emissions must be taken. Design modifications such as combustion chamber geometry, intake manifold and port design, compression ratio, injection pressure, etc. are all highly interrelated with regard to their effects on emissions, fuel economy, performance, durability, and reliability. A single, simple design modification, while reducing one effluent, will almost invariably increase another effluent, or will compromise some other aspect of engine operation.