United States
Environmental Protection
Agency

EPA-600/7-81-127
July 1981

EPA

Research and Development

ENVIRONMENTAL ASSESSMENT OF COMBUSTION MODIFICATION CONTROLS FOR STATIONARY INTERNAL COMBUSTION ENGINES

Prepared for

Office of Air Quality Planning and Standards

Prepared by

Industrial Environmental Research
Laboratory
Research Triangle Park NC 27711

RESEARCH REPORTING SERIES

Research reports of the Office of Research and Development, U.S. Environmental Protection Agency, have been grouped into nine series. These nine broad categories were established to facilitate further development and application of environmental technology. Elimination of traditional grouping was consciously planned to foster technology transfer and a maximum interface in related fields. The nine series are:

1. Environmental Health Effects Research
2. Environmental Protection Technology
3. Ecological Research
4. Environmental Monitoring
5. Socioeconomic Environmental Studies
6. Scientific and Technical Assessment Reports (STAR)
7. Interagency Energy-Environment Research and Development
8. "Special" Reports
9. Miscellaneous Reports

This report has been assigned to the INTERAGENCY ENERGY-ENVIRONMENT RESEARCH AND DEVELOPMENT series. Reports in this series result from the effort funded under the 17-agency Federal Energy/Environment Research and Development Program. These studies relate to EPA's mission to protect the public health and welfare from adverse effects of pollutants associated with energy systems. The goal of the Program is to assure the rapid development of domestic energy supplies in an environmentally-compatible manner by providing the necessary environmental data and control technology. Investigations include analyses of the transport of energy-related pollutants and their health and ecological effects; assessments of, and development of, control technologies for energy systems; and integrated assessments of a wide range of energy-related environmental issues.

EPA REVIEW NOTICE

EPA-600/7-81-127
July 1981

ENVIRONMENTAL ASSESSMENT OF COMBUSTION MODIFICATION CONTROLS FOR STATIONARY INTERNAL COMBUSTION ENGINES

by

H. I. Lips, J. A. Gotterba, and K. J. Lim
Acurex Corporation
Energy & Environmental Division
485 Clyde Avenue
Mountain View, California 94042

Contract No. 68-02-2160
Program Element No. EHE 624A

Prepared for

EPA Project Officer -- J. S. Bowen
EPA Deputy Project Officer -- R. E. Hall

Industrial Environmental Research Laboratory
Office of Environmental Engineering and Technology
Research Triangle Park
North Carolina 27711

U.S. ENVIRONMENTAL PROTECTION AGENCY
OFFICE OF RESEARCH AND DEVELOPMENT
WASHINGTON, DC 20460

DISCLAIMER

This Final Report was furnished to the U.S. Environmental Protection Agency by Acurex Corporation, Energy & Environmental Division, Mountain View, California 94042, in partial fulfillment of Contract No. 68-02-2160. The opinions, findings, and conclusions expressed are those of the authors and not necessarily those of the Environmental Protection Agency or of cooperating agencies. Mention of company or product names is not to be considered as an endorsement by the Environmental Protection Agency.

PREFACE

This is the fourth in a series of five special reports to be documented in the "Environmental Assessment of Stationary Source NO_x Combustion Modification Technologies" (NO_x EA). Specifically, this report documents the environmental assessment of combustion modification controls applied to stationary reciprocating internal combustion engines. The NO_x EA, a 36-month program which began in July 1976, is sponsored by the Combustion Research Branch of the Industrial and Environmental Research Laboratory of EPA (IERL-RTP). The program has two main objectives: (1) to identify the multimedia environmental impact of stationary combustion sources and combustion modification controls applied to these sources, and (2) to identify the most cost-effective, environmentally sound combustion modification controls for attaining and maintaining current and projected NO_2 air quality standards to the year 2000.

The NO_x EA will assess the following combination of process parameters and environmental impacts:

- Major fuel combustion stationary NO_x sources: utility boilers, industrial boilers, gas turbines, internal combustion (IC) engines, and commercial and residential warm air furnaces. Other sources (including mobile and noncombustion) will be considered only to the extent that they are needed to determine the NO_x contribution from stationary combustion sources.
- Conventional and alternate gaseous, liquid and solid fuels
- Combustion modification controls with potential for implementation to the year 2000; other controls (flue gas cleaning, mobile controls) will be considered only to estimate the future need for combustion modifications.
- Source effluent streams potentially affected by NO_x controls.

- Primary and secondary gaseous, liquid and solid pollutants potentially affected by NO_x controls.
- Pollutant impacts on human health and terrestrial or aquatic ecology.

To achieve the objectives discussed above, the NO_x EA program approach is structured as shown schematically in Figure P-1. The two major tasks are: Environmental Assessment and Process Engineering (Task B5), and Systems Analysis (Task C). Each of these tasks is designed to achieve one of the overall objectives of the NO_x EA program cited earlier. In Task B5, of which this report is a part, the environmental, economic, and operational impacts of specific source/control combinations will be evaluated. On the basis of this assessment, the incremental multimedia impacts from the use of combustion modification controls will be identified and ranked. Task C will in turn use the results of Task B5 to identify and rank the most effective source/control combinations to comply, on a local basis, with the current NO_2 air quality standards and projected NO_2 related standards.

As shown in Figure P-1, the key tasks supporting Tasks B5 and C are Baseline Emissions Characterization (Task B1), Evaluation of Emission Impacts and Standards (Task B2), and Experimental Testing (Task B3). The arrows in Figure P-1 show the sequence of subtasks and the major interactions among the tasks. The oval symbols identify the major outputs of each task. The subtasks under each main task are shown on the figure from the top to the bottom of the page in roughly the same order in which they will be carried out.

As indicated above, this report is a part of the Process Engineering and Environmental Assessment Task. The goal of this task is to generate process evaluations and environmental assessments for specific source/control combinations. These studies will be done in order of descending priority. In the first year of the NO_x EA, all the sources and controls involved in current and planned NO_x control implementation programs were investigated. The "Preliminary Environmental Assessment of Combustion Modification Techniques" (Reference P-1) documented this effort and established a priority rankings based on source emission impact and potential for effective NO_x control, to be used in the current ongoing detailed evaluation.

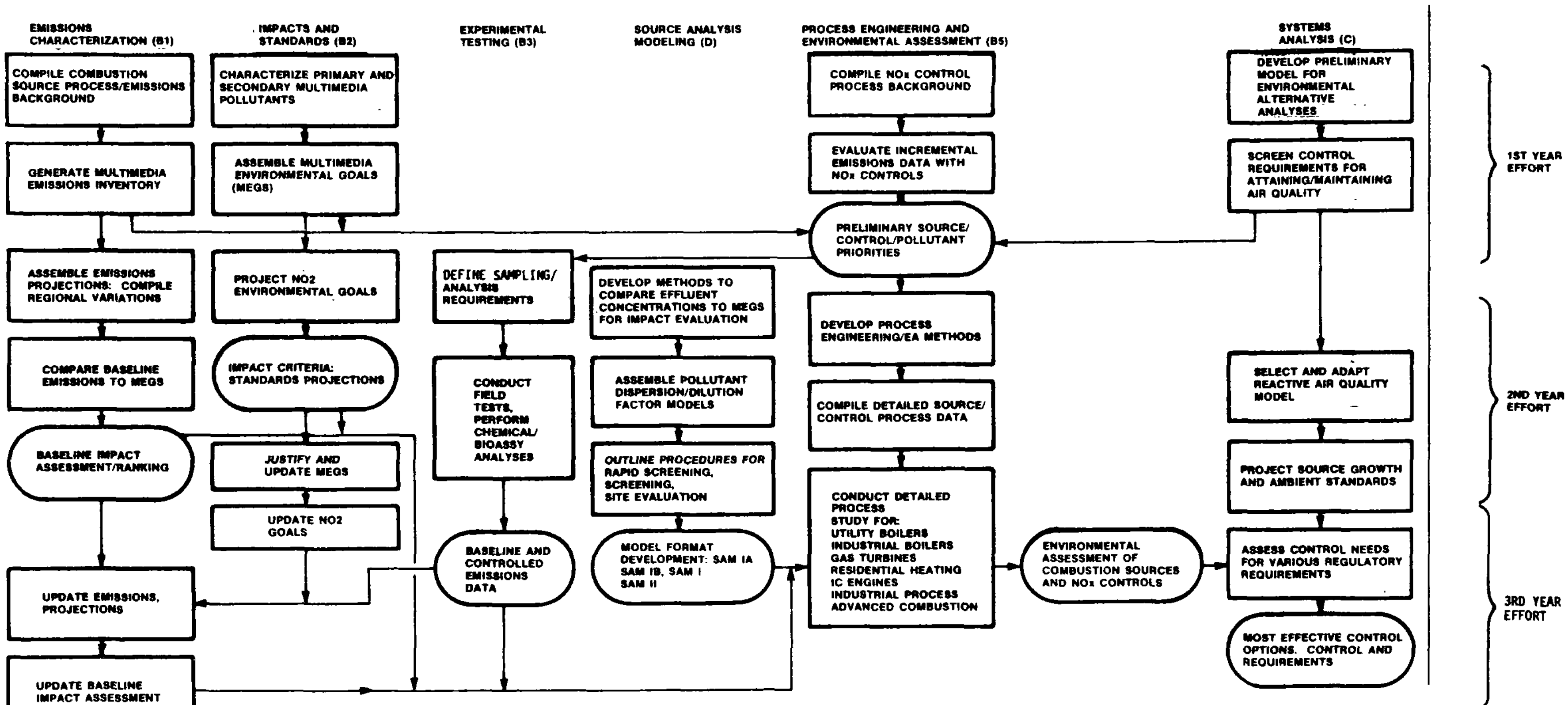


Figure P-1. NO_x EA program approach.

This report presents the assessment of combustion modification controls for the fourth source category to be treated, internal combustion engines. Other environmental assessment reports documented are:

- Environmental Assessment of Utility Boiler Combustion Modification NO_x Controls (Reference P-2)
- Environmental Assessment of Industrial Boiler Combustion Modification NO_x Controls (Reference P-3)
- Environmental Assessment of Combustion Modification Controls for Stationary Gas Turbines (Reference P-4)
- Environmental Assessment of Combustion Modification Controls for Residential and Commercial Heating Systems (Reference P-5)

REFERENCE FOR PREFACE

P-1. Mason, H. B., et al., "Preliminary Environmental Assessment of Combustion Modification Techniques: Volume II. Technical Results," EPA-600/7-77-119b, NTIS-PB 276 681/AS, October 1977.

P-2. Lim, K. J., et al., "Environmental Assessment of Utility Boiler Combustion Modification NO_x Controls: Volume I. Technical Results; Volume II, Appendices," EPA-600/7-80-075a, b, April 1980.

P-3. Lim, K. J., et al., "Industrial Boiler Combustion Modification NO_x Controls: Volume I. Environmental Assessment," EPA-600/7-81-126a, July 1981.

P-4. Larkin, R., et al., "Combustion Modification Controls for Stationary Gas Turbines: Volume I. Environmental Assessment," EPA-600/7-81-122a, July 1981.

P-5. Castaldini, et al., "Combustion Modification Controls for Residential and Commercial Heating Systems: Volume I. Environmental Assessment," EPA-600/7-81-123a, July 1981.

CONTENTS

CONTENTS (continued)

CONTENTS (Concluded)

FIGURES

FIGURES (Concluded)

TABLES

TABLES (Concluded)

SECTION 1

EXECUTIVE SUMMARY

This is the fourth in a series of five special reports to be documented in the "Environmental Assessment of Stationary Source NO_x Combustion Modification Technologies" (NO_x EA). Specifically, this report documents the environmental assessment of stationary reciprocating internal combustion (IC) engines with primary emphasis on NO_x combustion controls. The program has two main objectives: (1) to identify the multimedia environmental impact of stationary combustion sources and combustion modification controls applied to these sources, and (2) to identify the most cost-effective, environmentally sound combustion modification controls for attaining and maintaining current and projected NO_2 air quality standards to the year 2000.

With more NO_x controls being implemented in the field and expanded control development anticipated for the future, there is currently a need to: (1) ensure that the current and emerging control techniques are technically and environmentally sound and compatible with efficient and economical operation of systems to which they are applied, and (2) ensure that the scope and timing of new control development programs are adequate to allow stationary sources of NO_x to comply with potential air quality standards. The environmental assessment of stationary reciprocating internal combustion (IC) engines helps to address these needs by evaluating the operational, economic and environmental impacts from applying combustion modification controls.

Internal combustion engines are the second largest contributors of NO_x emissions from stationary anthropogenic sources in the U.S. -- constituting an 18.9 percent share (Reference 1-1). Because of this high NO_x level and their potential for control, stationary IC engines have been selected as one of the major source categories to be treated under the NO_x EA program.

1.1 OVERVIEW OF STATIONARY RECIPROCATING INTERNAL COMBUSTION ENGINES

This report concentrates on large bore engines since the smaller engines tend to be adaptations of mobile engines and could thus conceivably use the same control techniques. Also the largest amount of NO_x emitted from stationary IC engines comes from large bore engines. These engines are used mainly as electrical generators, pumps, and compressors. As electrical generators they are used by small public utilities for baseload power and by some larger utilities as peaking units, and at remote industrial sites and sites where a large amount of power is needed. Their major use as pumps and compressors is in pipeline service and oil and gas production.

The large bore engines (75 kW/cylinder) tend toward lower speeds, usually less than 1000 RPM and burn three major types of fuel: diesel, natural gas, and dual fuels (mixtures of diesel and gas). The natural gas engines are spark ignited, while the diesel and dual fuel engines are compression ignited. Some engines can burn residual oil and some use sewage gas. Both two- and four-stroke models can be found in this size range and the engine may be turbocharged, which usually increases efficiency. Typical heat rates are 9 to 11 MJ/kWh (8500 to 10,500 Btu/kWh). Smaller engines are not usually as efficient but have a lower initial capital cost. Also the larger engines tend to be operated unattended and at constant speed and are in use almost continuously.

1.2 WASTE STREAMS AND POLLUTANTS OF MAJOR CONCERN

The major waste streams are emitted through the exhaust pipe. Hydrocarbons (HC) can be emitted from the fuel before combustion, especially from natural gas-fired engines, but these emissions are minor. There may also be some emissions from the crankcase caused by blowby but this is a minor source. Also, the cooling system may release minor water pollutant emissions. Liquid wastes in the form of used crankcase oil may be another pollutant, but is not a major one.

Some average values for exhaust pipe emissions are listed in Table 1-1 (References 1-2 and 1-3). The HC emissions listed are total hydrocarbons; in the case of natural gas engines, these are mainly methane. Although the table lists factors for all engine sizes, this report focuses on the larger engines. Note that NO_x is the major pollutant for the large engines. Diesel engines may also emit polycyclic

TABLE 1-1. EMISSIONS FACTORS FOR IC ENGINES, g/kWh[a] (References 1-2, 1-3)

Fuel		NO_x	CO	HC
Gasoline	>15 kW	11.9	137	11.2
	<15 kW	7.5	395	27.5
Diesel	>375 kW[b]	17.3	2.4	0.6
	<375 kW[c]	16.6	6.0	2.8
Natural gas		15.4	3.8	6.5
Dual Fuel		11.0	2.7	4.1

[a]Emission factors for gasoline and diesel engines are modal averages; those for natural gas and dual fuel are for rated conditions. Modal averages mean that some of the NO_x numbers are taken from the constant power out portion of mobile tests.
[b]Based on an average of rated condition levels from engines considered
[c]Weighted average of two- and four-stroke engines. Weighting factors = 2/3 for four-stroke and 1/3 for two-stroke

organic matter (POM). EPA is actively researching this area of concern (Reference 1-4).

1.3 STATUS OF ENVIRONMENTAL PROTECTION ALTERNATIVES

There are three major types of pollution controls for IC engines: operational adjustment, catalytic exhaust gas treatment, and combustion chamber redesign. Operational adjustment techniques are essentially proven and can be applied now. Some combustion chamber redesign techniques have been tested but most are unproven. Using catalysts to reduce NO_x emissions has only been laboratory tested and not tested on an actual engine under lean burning conditions. NO_x control catalysts have been used on engines operating under rich conditions. Catalysts have also been used as CO and HC control devices.

The operational adjustment techniques are derate, ignition or injection retard, air-to-fuel ratio change, reduced manifold air temperature, and exhaust gas recirculation. Table 1-2 summarizes the expected NO_x reduction along with the change in the brake specific fuel consumption (BSFC). Notice that most techniques have major effects on BSFC.

Derating controls NO_x by lowering combustion temperatures, but could require larger engines and increase carbon monoxide (CO) and HC emissions. No major operation impacts are expected except for increased fuel consumption.

For a lean burning engine, changes in the air-to-fuel ratio control NO_x emissions by adding more air per unit of fuel and lowering the combustion temperature. Although altering this ratio may increase smoke and HC emissions, no major problems could be expected unless the ratio is adjusted to allow misfiring. The onset of engine misfiring is used as the maximum limit of this control technique.

Increased manifold air cooling also lowers the combustion temperature and may allow a higher air-to-fuel ratio. The manifold air cooling system may require more maintenance since it would be important that the cooling system operate at maximum efficiency for this control technique to be applied. Also a larger cooling system could be required.

Ignition retard causes combustion to occur later in the power stroke, thereby lowering both the peak combustion temperature and time

TABLE 1-2. NO_x REDUCTION AND FUEL CONSUMPTION PENALTIES FOR DIESEL, DUAL-FUEL, AND GAS ENGINES

Control Approach		Engine Fuel Type					
		Diesel		Dual Fuel		Natural Gas	
		% NO_x Reduction	ΔBSFC, %[a]	% NO_x Reduction	ΔBSFC, %[a]	% NO_x Reduction	ΔBSFC, %[a]
Derate (D)	3%	--	--	--	--	<20	2
	6%	--	--	--	--	<40	3
	10%	--	--	<20	4	--	--
	20%	<20	4	--	--	--	--
	25%	5-23	1-5	1-33	1-7	5-90	2-12
Retard (R)	2°	<20	4	<20	3	--	--
	4°	<40	4	<40	1	<20	3
	8°	28-45	2-8	50-73	3-5	8-40	2-7
Air-to-Fuel (A)	2%	--	--	--	--	<20	2
	3%	--	--	<20	0	--	--
	5%	--	--	--	--	<40	7
	±10%	7-8	3	25-40	1-3	20-80	5-12
Manifold Air Temperature (M')	311K(100°F)	7-15	0-2	18-37	0-1	28	0
	315K(107°F)	--	--	--	--	<20	0
	318K(113°F)	--	--	<20	1	--	--
Internal EGR		5	2	--	--	<20	5
		--	--	--	--	5-35	0-8
External EGR	10%	<20	1	20	1	<20	0
		33	1	--	--	33	0
Retard and Manifold Air Temperature		<20	1	<20	1	<20	3
		10-24	0-1	25	2	30-40	5-6
Retard & Air-to-Fuel		<20	8	<20	1	<20	4
		<40	16	<40	2	<40	8
		35-65	5-26	56	2	17-52	4-11
Retard and Manifold Air Temperature and Air-to-Fuel		20	0	<20	2	<20	2
		--	--	40	3	<40	4
		--	--	--	--	40-65	6-7
Air-to-Fuel and Manifold Air Temperature		<20	2	--	--	--	--
		20-30	3	--	--	--	--
Water Injection (H_2O/fuel ratio)	50%	25-35	2-4	--	--	25-35	1-2
	100%			--	--	60-75	2-5
Catalytic Reduction (Projected)		50-80	0	50-80	0	50-80	0
Combustion Chamber Modifications (Projected)	Increased Mixing	10-30	<5	20-40	<5	20-40	<5
	Staged Combustion	10-30	0	10-30	0-7	10-30	0-2

[a]ΔBSFC is increase in brake specific fuel consumption

spent at high temperatures, but increases exhaust temperature. This control method can reduce the life of exhaust valves and exhaust system.

Exhaust gas recirculation (EGR) reduces NO_x by lowering the combustion temperature. However, EGR increases maintenance problems, especially for diesel engines. The exhaust gas recirculation system in addition to the turbocharger can require periodic cleaning because of the particulate loading in diesel exhaust units.

Although combustion chamber redesign techniques have not been commercially demonstrated, they potentially have the least operational impact. The goal of the combustion chamber redesign program is to design an engine that will have a smaller fuel impact. Also it is hoped that the tuning of the new engine will not be as critical.

Because large engines are usually lean burning, catalytic reduction of NO_x will most likely require a reducing agent such as ammonia. Thus there are potential operational problems associated with ammonia injection. There will be an additional system to maintain as well as the potential for additional harmful emissions such as NH_3 and HCN.

The effects of these controls on NO_x, HC, and CO are summarized in Tables 1-3 through 1-10. Estimated costs for certain operational adjustment techniques are also presented. The percentage cost increases are based on engine cost only (e.g., the generator costs are not included). The incremental initial capital cost of the available controls range from zero to 5 percent of an uncontrolled engine. However, the total annualized cost to control can increase the cost of power from an engine by 1 to 14 percent, a significant impact because of additional fuel and maintenance requirements.

Best Available Control Technology (BACT)

Currently the best available control technologies to control NO_x emissions from large bore engines are: (1) retarded ignition or fuel injection, (2) air-to-fuel ratio changes, (3) increased manifold air cooling, and (4) derating. These techniques can either be employed alone or in combinations with each other. The best combination will be very engine dependent. But in general retard is best for diesel-fueled engines, air-to-fuel ratio changes for natural gas, and either control for dual fuel. A 40 percent reduction in NO_x can usually be achieved without causing any major operational problems except for increased fuel

TABLE 1-3. ABBREVIATIONS FOR ENGINE TYPE AND EMISSION CONTROL TECHNOLOGY USED IN FOLLOWING TABLES

Abbreviation	Explanation
Air Charging	
BS	Blower scavenged
NA	Naturally aspirated
TC	Turbocharged (and intercooled)
Control Technology	
D	Derating
R	Retard
TC	Turbocharged (and intercooled)
A/F	Increased air-to-fuel ratio
MAT	Decreased inlet manifold air temperature
EGR(I)	Exhaust gas recirculation -- internal
EGR(E)	Exhaust gas recirculation -- external
INJ	Modified injectors
H_2O	Water induction
CCR	Combustion chamber redesign

TABLE 1-4. EFFECTS OF CONTROLS ON ENGINES LARGER THAN 5.7 × $10^{-3} m^3$/cyl: NO_x EMISSIONS

Fuel	Diesel				Dual Fuel				Natural Gas			
Strokes/Cycle	Two		Four		Two		Four		Two		Four	
Control \ Air Charging	BS	TC	NA	TC	BS	TC	NA	TC	BS	TC	NA	TC
D	↓			↓				↓	↓	↓	↓	↓
R	↓	↓		↓		↓		↓	↓	↓	↓	↓
A/F				↓		↓		↓	↓	↓	↓	↓
TC	↓	—		—		—		—	↓	—	↑	—
MAT	↓	↑		↓		↓		↓	↓	↓		↓
INJ	↓	↑↓										
EGR(I)				↑				↑	↓		↓	
EGR(E)		↓				↑						
H_2O	↓			↓				↓			↓	↓
CCR-7			↓	↓								

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

TABLE 1-5. EFFECTS OF CONTROLS ON ENGINES LARGER THAN $5.7 \times 10^{-3} m^3$/cyl: CO EMISSIONS

Fuel	Diesel				Dual Fuel				Natural Gas			
Strokes/Cycle	Two		Four		Two		Four		Two		Four	
Control \ Air Charging	BS	TC	NA	TC	BS	TC	NA	TC	BS	TC	NA	TC
D	↑			↓				↑↓	↑	↑	↑	↑
R	↑	↑		↑		↑		↑	↑	↑		↓
A/F				↑↓		↑		↓		↑	—	
TC	↑								↓		↓	
MAT	↑	↓		↑↓		↑		↑↓	↑	↑		↑
INJ	↓	↑↓										
EGR(I)				↓				↓	↑		—	
EGR(E)		↑				↑						
H_2O	↓			↑↓				↓			↑	↑
CCR												

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

TABLE 1-6. EFFECTS OF CONTROLS ON ENGINES LARGER THAN 5.7 x $10^{-3} m^3$/cyl: HC EMISSIONS

Fuel	Diesel				Dual Fuel				Natural Gas			
Strokes/Cycle	Two		Four		Two		Four		Two		Four	
Control \ Air Charging	BS	TC	NA	TC	BS	TC	NA	TC	BS	TC	NA	TC
D				↑				↑	↑	↑	↑	↑
R	—	↑↓		↑↓		↑		↓	↑	↑	↑	↑
A/F				↑↓		↑		↑		↑		↑
TC	↓								↓			
MAT	↑	↑↓		↑↓		↑		↑↓	↑	↑		↑
INJ	↓	↓	↓									
EGR(I)				↓				↑	↑			
EGR(E)		—				↓						
H_2O	↑			↑↓				↑			↑	↑
CCR												

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

TABLE 1-7. ESTIMATED INCREMENTAL COST OF AIR-TO-FUEL INCREASE (1978)

	Engine/Fuel Type			
			Natural Gas	
	Diesel (Electrical Generation)	Dual Fuel (Electrical Generation)	(Oil and Gas Pipeline)	(Oil and Gas Production)
Capital ($/kW)	0	0	0	0
Maintenance ($/kWh)	0.0002	0.0002	0.0002	0.0002
Fuel ($/kWh)	0.0031	0.0004	0.0004	0.0004
Total Annualized ($/kWh)[a]	0.003	0.001	0.001	0.001
Percent increase in initial cost	0	0	0	0
Percent increase in annual cost	7.0	3.0	3.0	3.0
Percent reduction in NO_x	20	40	40	40

[a]Assumes 8000 hours operating year.

TABLE 1-8. ESTIMATED INCREMENTAL COST OF EXTERNAL EXHAUST GAS RECIRCULATION (1978)

	Engine/Fuel Type			
			Natural Gas	
	Diesel (Electrical Generation)	Dual Fuel (Electrical Generation)	(Oil and Gas Pipeline)	(Oil and Gas Production)
Capital ($/kW)	12	12	12	4
Maintenance ($/kWh)	0.002	0.004	0.004	0.004
Fuel ($/kWh)	0	0	0	0
Total Annualized ($/kWh)[a]	0.0023	0.0043	0.0043	0.0041
Percent increase in initial cost	5	5	5	5
Percent increase in annual cost	5.3	14	14	13
Percent reduction in NO_x	20	20	20	20

[a]Assumes 8000 hours operating year.

TABLE 1-9. ESTIMATED INCREMENTAL COST DUE TO RETARD (1978)

	Engine/Fuel Type			
			Natural Gas	
	Diesel (Electrical Generation)	Dual Fuel (Electrical Generation)	(Oil and Gas Pipeline)	(Oil and Gas Production)
Capital ($/kW)	0	0	0	0
Maintenance ($/kWh)	0.0016	0.0016	0.0016	0.0016
Fuel ($/kWh)	0.0012	0.0007	0.0008	0.0009
Total Annualized ($/kWh)[a]	0.003	0.002	0.002	0.002
Percent increase in initial cost	0	0	0	0
Percent increase in annual cost	7.0	6.4	6.1	6.2
Percent reduction in NO_x	20-30	20-30	20-30	20-30

[a]Assumes 8000 hours operating year.

TABLE 1-10. ESTIMATED INCREMENTAL COST OF COMBINED CONTROLS FOR LARGE BORE DIESEL ENGINES (1978)

	Control Technique		
	Retard	Air-to-Fuel Changes and Manifold Air Cooling	Air-to-Fuel and Retard
Capital ($/kW)	0	3.6	0
Maintenance ($/kWh)	0.0016	0.001	0.0018
Fuel ($/kWH)	0.0024	0.0015	0.0015
Total Annualized ($/kWh)[a]	0.0040	0.0026	0.0033
Percent increase in capital cost	0	1.5	0
Percent increase in annual cost	9.3	6.0	7.7
Percent reduction in NO_x	40	30-40	40

[a]Assumes 8000 hours operating year.

consumption and possible increased maintenance. Since the baseline emissions of these engines vary greatly, the actual control level achievable also varies greatly. For dual fuel and gas engines a 40 percent NO_x reduction causes an increase in the brake specific fuel consumption by about 2 to 7 percent while for diesel engines the increased fuel consumption would be 4 to 8 percent based on the large bore engines tested. HC and CO emissions may also increase by up to 50 percent. On a mass basis these increases would be much less than the amount of NO_x reduced.

Best Promising Techniques

The best promising future control techniques are combustion chamber redesign and catalytic exhaust gas treatment. Combustion chamber redesign techniques have the potential of giving the same NO_x emission reduction as the BACT technique but at a lower cost and smaller fuel penalty. If very low NO_x emissions are required, catalytic exhaust gas treatment is the only technique with that potential. For lean burning engines NH_3 would probably have to be used as a NO_x reducing agent. The engine would require greater operator attention because of the ammonia injection equipment, and handling and storage facilities. Also the use of NH_3 reflects an energy consumption in itself since NH_3 is currently produced from natural gas. Thus catalytic exhaust gas treatment of NO_x could have a significant fuel impact.

1.4 DATA NEEDS AND RECOMMENDATIONS

There are two major weaknesses in the amount of available data for combustion modification controls on IC engines. The information on operational effects and long term durability of these control techniques is very limited, especially combustion chamber redesign data and catalytic exhaust gas treatment. More information on combining these controls to achieve the optimal low emissions/high efficiency is needed. Also the available data on emissions other than NO_x, CO, and total hydrocarbons are very limited. The amount and type of organics emitted from these large bore engines are not very well understood. The potential mutagenicity of organic emissions in diesel exhaust is of major concern.

Research is needed on designing a high efficiency low NO_x emitting engine. Even with the best available controls applied, the large bore stationary reciprocating internal combustion engine is the highest NO_x emitter on a heat input basis of all the major combustion sources.

EPA is currently sponsoring several programs in the health effects area as well as new engine designs for low NO_x and high efficiency. These programs should help resolve many of the major data gaps in the operational and environmental impacts of NO_x controls. One of these programs is being conducted by A. D. Little under contract to the EPA. They are presently testing new combustion chamber modification techniques on large bore one cyclinder engines in the laboratory and plan to test these techniques on full size engines in 1981 (Reference 1-5).

REFERENCES FOR SECTION 1

1-1. Waterland, L.R., *et al.*, "Environmental Assessment of Stationary Source NO_x Control Technologies -- Final Report," Acurex Draft Report FR-80-57/EE, EPA Contract 68-02-2160, Acurex Corp., Mountain View, CA, April 1980.

1-2. Youngblood, S. B., and G. R. Offen, "Emissions Inventory of Currently Installed Stationary Reciprocating IC Engines," Acurex Internal Communication, Acurex Corporation, Mountain View, CA, September 23, 1975.

1-3. "Stationary Internal Combustion Engines. Background Information: Proposed Standards," EPA-450/3-78-125a, April 1979.

1-4. Barth, D. S., and S. M. Blacker, "The EPA Program to Assess the Public Health Significance of Diesel Emissions," JAPCA, Vol. 28, No. 8, 1978, pg. 769.

1-5. Personal communication with Jack Wasser, EPA, Research Triangle Park, North Carolina, April 30, 1980.

SECTION 2
INTRODUCTION

This report assesses the operational, economic, and environmental impacts from applying combustion modification controls to stationary internal combustion engines. With more NO_x controls being implemented in the field and expanded control development anticipated for the future, there is currently a need to: (1) ensure that the current and emerging control techniques are technically and environmentally sound, and compatible with efficient and economical operation of systems to which they are applied, and (2) ensure that the scope and timing of new control development programs are adequate to allow stationary sources of NO_x to comply with potential air quality standards. The NO_x EA program addresses these needs by (1) identifying the incremental multimedia environmental impact of combustion modification controls, and (2) identifying the most cost-effective source/control combinations to achieve ambient NO_2 standards.

2.1 BACKGROUND

The 1970 Clean Air Act Amendments designated oxides of nitrogen (NO_x) as one of the criteria pollutants requiring regulatory controls to prevent potential widespread adverse health and welfare effects. Accordingly, in 1971, EPA set a primary and secondary National Ambient Air Quality Standard (NAAQS) for NO_2 of 100 $\mu g/m^3$ (annual average). To attain and maintain the standard, the Clean Air Act mandated control of new mobile and stationary NO_x sources, each of which currently emits approximately half of the manmade NO_x nationwide. Emissions from light duty vehicles (the most significant mobile source) were to be reduced by 90 percent to a level of 0.25 g NO_2/km (0.4 g/mile) by 1976. Stationary sources were to be regulated by EPA Standards of Performance for New Stationary Sources (NSPS), which are set as control technology becomes available. Additional standards required to attain air quality in the Air

Quality Control Regions (AQCRs) could be set for new or existing sources through the State Implementation Plans (SIPs).

Since the Clean Air Act, techniques have been developed and implemented that reduce NO_x emissions by a moderate amount (30 to 60 percent) for a variety of source/fuel combinations. In 1971, EPA set NSPS for large steam generators burning gas, oil, and coal (except lignite). Recently, more stringent standards for utility boilers burning all gaseous liquid and solid fuels have been promulgated, along with standards for lignite fired utility boilers. In addition, NSPS have been promulgated for stationary gas turbines and are currently being considered for stationary internal combustion engines and intermediate size (industrial) steam generators. Local standards also have been set, primarily for new and existing large steam generators and gas turbines, as parts of State Implementation Plans in several areas with NO_x problems. This regulatory activity has resulted in reducing NO_x emissions from individual controlled sources by 30 to 60 percent. The number of controlled sources is increasing as new units are installed with factory equipped NO_x controls.

Emissions have been reduced comparably for light duty vehicles. Although the goal of 90 percent reduction (0.25 g NO_2/km) by 1976 has not been achieved, emissions were reduced by about 25 percent (1.9 g/km) for the 1974 to 1976 model years and in 1979 were reduced to 50 percent to 1.25 g/km. Achieving the 0.25 g/km goal has been deferred indefinitely because of technical difficulties and fuel penalties. Initially, the 1974 Energy Supply and Environmental Coordination Act deferred compliance to 1978. The Clean Air Act Amendments of 1977 defined the 0.25 g/km emission level as a research goal and set the standard of 0.62 g/km (1 g/mile) for the 1981 model year and beyond.

Because the mobile source emission regulations have been relaxed, stationary source NO_x control has become more important for maintaining air quality. Several air quality planning studies have evaluated the need for stationary source NO_x control in the 1980's and 1990's in view of recent developments (References 2-1 through 2-10). These studies all conclude that relaxing mobile standards, coupled with the continuing growth rate of stationary sources, will require more stringent stationary source controls than current and impending NSPS provide. This conclusion has been reinforced by projected increases in the use of coal in stationary sources. The studies also conclude that the most cost-effective way to achieve these reductions is by using combustion modification NO_x controls in new sources.

It is also possible that separate NO_x control requirements will be needed to attain and/or maintain additional NO_2 related standards. Data on the health effects of NO_2 suggest that the current NAAQS should be supplemented by limiting short term exposure (References 2-4 and 2-11 through 2-14). In fact, the Clean Air Act Amendments of 1977 require EPA to set a short term NO_2 standard for a period not to exceed 3 hours unless it can be shown that such a standard is not needed. The need for a short-term standard is currently under review by EPA.

EPA is continuing to evaluate the long range need for additional NO_x regulation as part of strategies to control oxidants or pollutants for which NO_x is a precursor, e.g., nitrates and nitrosamines (References 2-4, 2-11, and 2-15 through 2-18). These regulations could be source emission controls or additional ambient air quality standards. In either case, additional stationary source control technology could be required to assure compliance.

In summary, since the Clean Air Act, near term trends in NO_x control are toward reducing stationary source emissions by a moderate amount; hardware modifications in existing units or new units of conventional design will be stressed. For the far term, air quality projections show that more stringent controls than originally anticipated will be needed. To meet these standards, the preferred approach is to control new sources by using low NO_x redesigns.

2.2 ROLE OF STATIONARY INTERNAL COMBUSTION ENGINES

Internal combustion (IC) engines are the second largest contributor of stationary source NO_x emissions in the U.S. Figure 2-1 shows that IC engines were the origin of 18.9 percent of all stationary source NO_x

emissions for the year 1977 (Reference 2-19). In fact, NO_x emissions are the principal pollutants from the larger IC engines. Furthermore, total stationary source NO_x emissions are projected to increase unless adequate controls are developed (Reference 2-20).

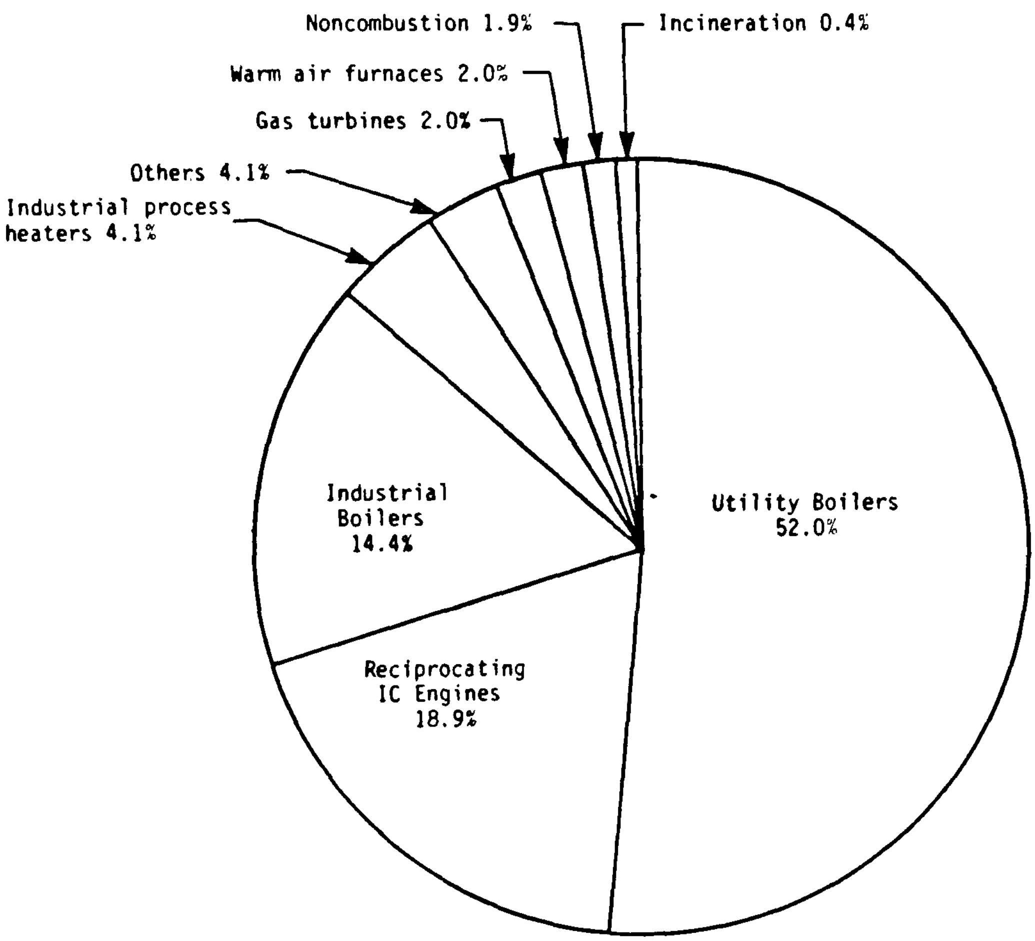
Given this background and their potential NO_x control, IC engines were selected as the third source category to be treated under the NO_x EA program. The "Preliminary Environmental Assessment of Combustion Modification Techniques" (Reference 2-8) concluded that modifying combustion process conditions is the most effective and widely used technique for achieving 20 to 60 percent reduction in oxides of nitrogen. Nearly all current NO_x control applications use combustion modifications.

2.3 OBJECTIVE OF THIS REPORT

This report provides comprehensive, objective, and realistic evaluations and comparisons of the important aspects of the available combustion control techniques, using a common and uniform basis for comparison. The objective is to perform an environmental assessment of combustion modification techniques for IC engines to:

- Determine their impact on the achievement of selected environmental goals, based on a comprehensive analysis from a multimedia consideration
- Ascertain the effect of their application on engine performance and identify potential problem areas
- Estimate the economics of their operation
- Estimate the limits of control achievable by combustion modification
- Identify further research and development and/or testing required to optimize combustion modification techniques and to upgrade their assessments.

Since larger engines are the primary emitters of NO_x, while the smaller engines mainly emit CO, this report concentrates on the large bore engines. Furthermore, medium to small engines are modified mobile engines and could use the same emission control techniques as mobile engines. Since controls for mobile engines have been discussed extensively elsewhere (e.g., Reference 2-21) they will not be treated in detail here.



Total: 10.5 Tg/yr (11.6×10^6 tons/yr)

Figure 2-1. Distribution of stationary anthropogenic NO_x emissions for the year 1977 (Reference 2-19).

2.4 ORGANIZATION OF THIS REPORT

Evaluating the effectiveness and impacts of NO_x combustion controls applied to IC engines requires assessing their effects on both controlled source performance, especially as translated into changes in operating costs and energy consumption, and on incremental emissions of other pollutants as well as NO_x. To perform such an evaluation, it is necessary to:

- Characterize the source category with regard to equipment, fuels, and emissions (Sections 3, 4)
- Identify NO_x formation mechanisms and relate fuels to their emissions potential (Section 4)
- Evaluate the performance of current and potential control techniques available for implementation (Section 5)
- Assess the operational and cost impacts, including energy impacts, of implementing controls (Section 5)
- Evaluate the environmental impact of controls through the analysis of incremental emissions (Section 6)
- Make an overall assessment and ranking of control techniques for the major equipment/fuel categories (Section 1)
- Identify research and development needs (Section 7)

REFERENCES FOR SECTION 2

2-1. Crenshaw, J. and A. Basala, "Analysis of Control Strategies to Attain the National Ambient Air Quality Standard for Nitrogen Dioxide," presented at the Washington Operation Research Council's Third Cost Effectiveness Seminar, Gaithersburg, MD, March 1974.

2-2. "Air Quality, Noise and Health -- Report of a Panel of the Interagency Task Force on Motor Vehicle Goals Beyond 1980," Department of Transportation, March 1976.

2-3. McCutchen, G. D., "NO_x Emissions Trends and Federal Regulation," presented at the AIChE 69th Annual Meeting, Chicago, December 1976.

2-4. "Air Program Strategy for Attainment and Maintenance of Ambient Air Quality Standards and Control of Other Pollutants," Draft Report, U.S. EPA, Washington, D.C., October 1976.

2-5. "Annual Environmental Analysis Report, Volume 1 Technical Summary," The MITRE Corporation, MTR-7626, September 1977.

2-6. Personal communication with R. Bauman, Strategies and Air Standards Division, Office of Air Quality Planning and Standards, U.S. EPA, October 1977.

2-7. "An Analysis of Alternative Motor Vehicle Emission Standards," U.S. Department of Transportation, U.S. EPA, U.S. FEA, May 1977.

2-8. Mason, H. B., et al., "Preliminary Environmental Assessment of Combustion Modification Techniques: Volume II, Technical Results," EPA-600/7-77-119b, NTIS-PB 276 681/AS, October 1977.

2-9. Greenfield, S. M., et al., "A Preliminary Evaluation of Potential NO_x Control Strategies for the Electric Power Industry," EPRI TR-13300, April 1977.

2-10. Waterland, L. R., et al., Environmental Assessment of Stationary Source NO_x Control Technologies -- Second Annual Report," EPA-600/7-79-147, NTIS-PB 300 469, June 1979.

2-11. French, J. G., "Health Effects from Exposure of Oxides of Nitrogen," presented at the AIChE 69th Annual Meeting, Chicago, November 1976.

2-12. "Scientific and Technical Data Base for Criteria and Hazardous Pollutants -- 1975 EPA/RTP Review," EPA-600/1-76-023, NTIS-PB 253 942/AS, January 1976.

2-13. Shy, C. M., "The Health Implications of an Non-Attainment Policy, Mandated Auto Emission Standards, and a Non-Significant D terioration Policy," presented to Committee on Environmental and Public Works, Serial 95-H7, February 1977.

2-14. "Report on Air Quality Criteria for Nitrogen Oxides," AP-84, Science Advisory Board, U.S. EPA, June 1976.

2-15. "Report on Air Quality Criteria: General Comments and Recommendations," Report to the U.S. EPA by the National Air Quality Advisory Committee of the Science Advisory Board, June 1976.

2-16. "Air Quality Criteria Document for Oxides of Nitrogen: Availability of External Review Draft," Federal Register, Vol. 43, pp. 55, 117, December 12, 1978.

2-17. Personal communication with M. Jones, Strategies and Air Standards Division, Office of Air Quality Planning and Standards, U.S. EPA, September 1976.

2-18. "Control of Photochemical Oxidants -- Technical Basis and Implications of Recent Findings," EPA-450/2-75-005, July 1975.

2-19. Waterland, L. R., et al., "Environmental Assessment of Stationary Source NO_x Control Technologies -- Final Report," Acurex Final Report FR-80-57/EE, EPA Contract 68-02-2160, Acurex Corporation, Mountain View, CA, April 1980.

2-20. Mason, H. B., et al., "Utility Boiler NO_x Emission Characterization," in Proceedings: Second NO_x Control Technology Seminar, EPRI FP-1109-SR, July 1979.

2-21. Stork, E. O., "Environmental Impact Statement and Economic Impact Analysis Revised Heavy-Duty Engine Regulations for 1979 and later Model Years," EPA Mobile Source Air Pollution Control, August 4, 1977.

2-22. Cadle, S. H., and G. J. Nebel, "Control of Automotive Emissions," October 1978, G M Research Laboratories.

2-23. Klein, H. I., and R. W. Sbuschning, "Automotive Emissions and Fuel Economy - A Primer," 1977 Ford Engineering Forum, June 1977.

SECTION 3
SOURCE CHARACTERIZATION

Stationary reciprocating internal combustion (IC) engines are found in a variety of applications where there is a requirement for mechanical work in the form of shaft power. Installations vary greatly, ranging from within large urban centers to remote areas. Because of their versatility, IC engines range in size from 1 kW to over 10 MW power output.

Although reciprocating IC engines are manufactured in almost every size and serve industry throughout, a given industry in general will not use all engine sizes. Small and very small engines are typically used as consumer products around the home and small farm. At the other extreme are the large engines, almost always found operating continuously in utility applications such as electric power generation or pipeline pumping (Reference 3-1). This report concentrates on the larger size engines since they produce three quarters of all NO_x emissions from installed stationary engines (see Section 2), and their population, operation, and emission characteristics can be quantified.

3.1 ENGINE DESIGN TYPES

All reciprocating internal combustion engines operate by the same basic process. A combustible mixture is first compressed in a small volume between the head of a piston and its surrounding cylinder. The mixture is then ignited, and the resulting high pressure products of combustion push the piston through the cylinder. This movement is converted from linear to rotary motion by a crankshaft. The piston returns, pushing out exhaust gases, and the cycle is repeated.

Although all reciprocating IC engines follow the same basic process, there are variations which classify engine types. Engines are generally classified by their: (1) fuel burned, (2) method of ignition, (3) combustion cycle, and (4) charging method.

3.1.1 Fuel Type

The three primary fuels for stationary reciprocating internal combustion engines are: gasoline, diesel (No. 2) oil, and natural gas. Gasoline is used primarily for mobile and portable engines. Construction sites, farms, and households typically use converted mobile engines for stationary application because their cost is often less than an engine designed specifically for stationary purposes (Reference 3-2). In addition, mobile engine parts and service are readily available, and gasoline is easily transported to the site. Thus gasoline is an essential fuel for small and medium size stationary engines.

Diesel fuel oil is also easily transported, and therefore is used in small and medium sized engines. Also, the generally higher efficiencies exhibited by diesel engines makes diesel oil an ideal fuel for large engines where operating costs must be minimized. Diesel is thus the most versatile fuel for stationary reciprocating engines.

Natural gas is used more than any other fuel for large stationary IC engines (Reference 3-3), typically operating pumps or compressors on gas pipelines. This fuel may see decreasing usage if gas supplies decrease.

Other fuels are also burned in stationary IC engines, but their use is limited. Some engines are burning heavy fuel oils, and a few burn almost any other liquid fuel (Reference 3-3). Gaseous fuels such as sewer gas are sometimes used at wastewater treatment plants where the gas is available (Reference 3-4). Stationary IC engines can be modified to burn almost any liquid or gaseous fuel if the engine is properly designed and adjusted.

3.1.2 Method of Ignition

Ignition is the means of initiating combustion in the engine cycle. There are two methods used for stationary reciprocating IC engines: compression ignition (CI) and spark ignition (SI).

In compression ignition engines, combustion air is first compression heated in the cylinder, and diesel fuel oil is then injected into the hot air. Ignition is spontaneous as the air is above the autoignition temperature of the fuel. Spark ignition engines initiate combustion by the spark of an electrical discharge. Usually the fuel is mixed with the air in a carburetor (for gasoline) or at the intake valve (for natural gas), but occasionally the fuel is injected into the compressed air in the cylinder. Although all diesel fueled engines are compression ignited and all gasoline and gas fueled engines are spark ignited, gas can be used in a compression ignition engine if a small amount of diesel fuel is injected into the compressed gas/air mixture to initiate burning. Such dual fuel (DF) engines are usually designed to burn any mixture ratio of gas and diesel oil, from 6- to 100-percent oil (based on heating value).

CI engines usually operate at a higher compression ratio (ratio of cylinder volume when the piston is at the bottom of its stroke to volume when it is at the top) than SI engines because fuel is not present during compression; hence there is no danger of premature autoignition. Since engine thermal efficiency rises with increasing pressure ratio (and pressure ratio varies directly with compression ratio), CI engines are more efficient than SI engines. This increased efficiency is gained at the expense of poorer response to load changes and a heavier structure to withstand the higher pressures.

3.1.3 Combustion Cycle

As previously mentioned, the combustion process for stationary reciprocating internal combustion engines consists of compressing a combustible mixture by a piston, igniting it, and allowing the high pressures generated to push the piston back. This process may be accomplished in either four strokes or two strokes of the piston.

In the four-stroke cycle, the sequence of events may be summarized as follows (see Figure 3-1):

1. Intake stroke -- suction of the air or air and fuel mixture into the cylinder by the downward motion of the piston through the cylinder.
2. Compression stroke -- compression of the air or air and fuel mixture, thereby raising its temperature and reducing its volume.
3. Ignition and power (expansion) stroke -- combustion and consequent downward movement of the piston by pressure from the expanding gases with energy transfer to the crankshaft.
4. Exhaust stroke -- expulsion of the exhaust gases from the cylinder by the upward movement of the piston.

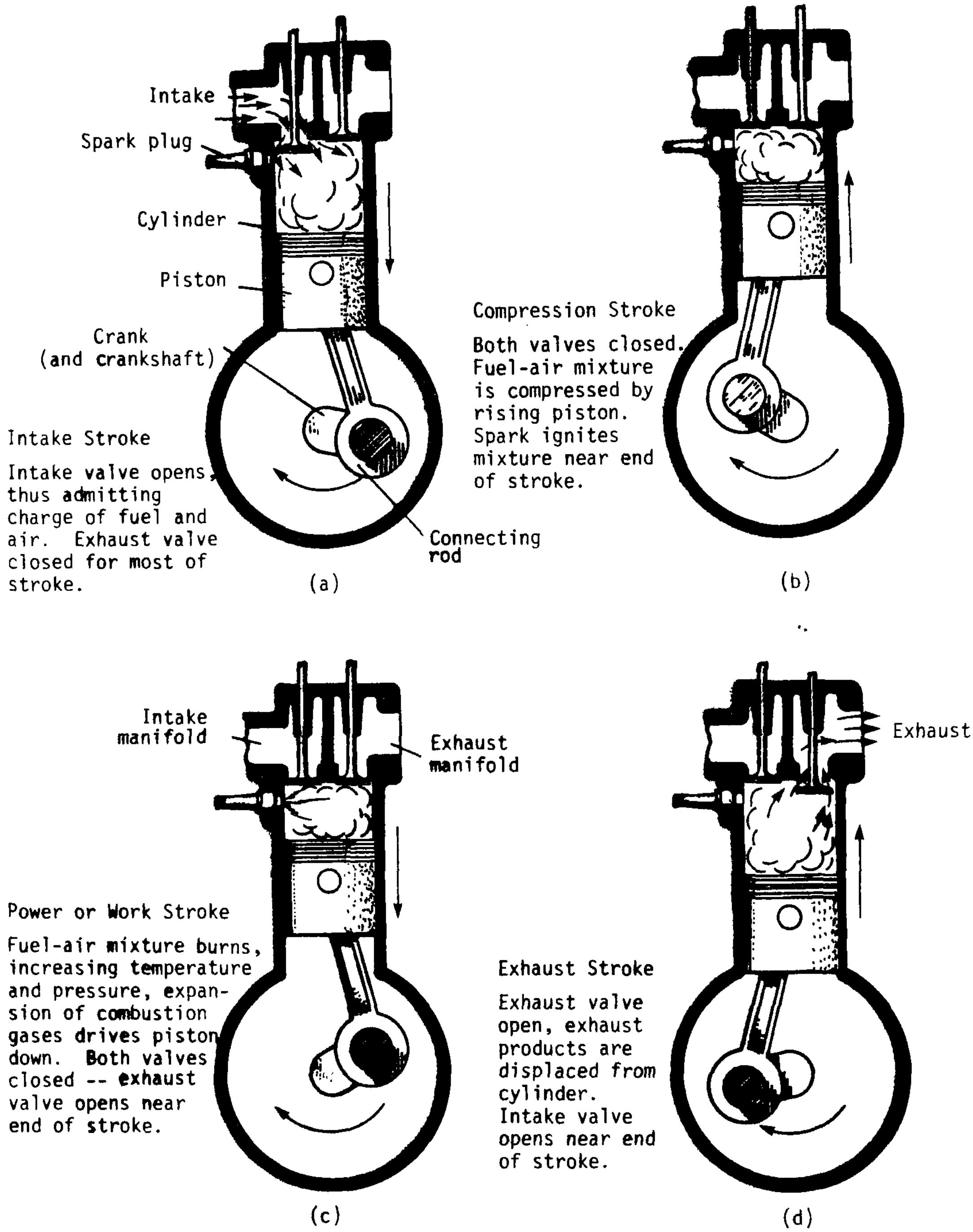


Figure 3-1. The four-stroke, spark-ignition (SI) cycle. Four strokes of 180° of crankshaft rotation each, or 720° of crankshaft rotation per cycle (Reference 3-4).

A two-stroke cycle completes the power cycle in one revolution of the crankshaft as compared to two revolutions for the four-stroke cycle (see Figure 3-2). As the piston moves to the top of the cylinder, air or an air and fuel mixture is compressed for ignition. Following ignition and combustion, the piston delivers power as it moves down through the cylinder. Eventually it uncovers the exhaust ports (or exhaust valves open). As the piston begins the next cycle, exhaust gas continues to be purged from the cylinder, partially by the upward motion of the piston and partially by the scavenging action of the incoming fresh air. Finally, all ports are covered (and/or valves closed), and the fresh charge of air or air and fuel is again compressed for the next cycle.

Two-stroke engines have the advantage of higher horsepower-to-weight ratio compared to four-stroke engines when both operate at the same speed. In addition, if ports are used instead of valves, the mechanical design of the engine is simplified. However, combustion can be better controlled in a four-stroke engine and excess air is not needed to purge the cylinder. Therefore, four-stroke engines tend to be slightly more efficient, and may emit less pollutants (primarily unburned hydrocarbons) than two-stroke engines (Reference 3-1).

3.1.4 Charging Method

Charging is the method of introducing air or the air and fuel mixture into the cylinder. Three methods are commonly used: natural aspiration, turbocharging and blower scavenged.

A naturally aspirated engine uses the vacuum created behind the moving piston during the intake stroke to suck in the fresh air charge. This process tends to be somewhat inefficient, however, since the actual amount of air drawn into the cylinder is only about 50 to 75 percent of the displaced volume (Reference 3-5). A more efficient method of charging is to pressurize the air (or air and fuel) and force it into the cylinder. This may be done with either a turbocharger or a supercharger. The turbocharger is powered by a turbine that is driven by the energy in the relatively hot exhaust gases, while a supercharger is driven off the engine crankshaft. Air pressurization increases the power density, or power output per unit weight (or volume) of the engine, since more air mass can be introduced into the cylinder. As air pressure increases, its temperature also rises because of the action of the compressor on the

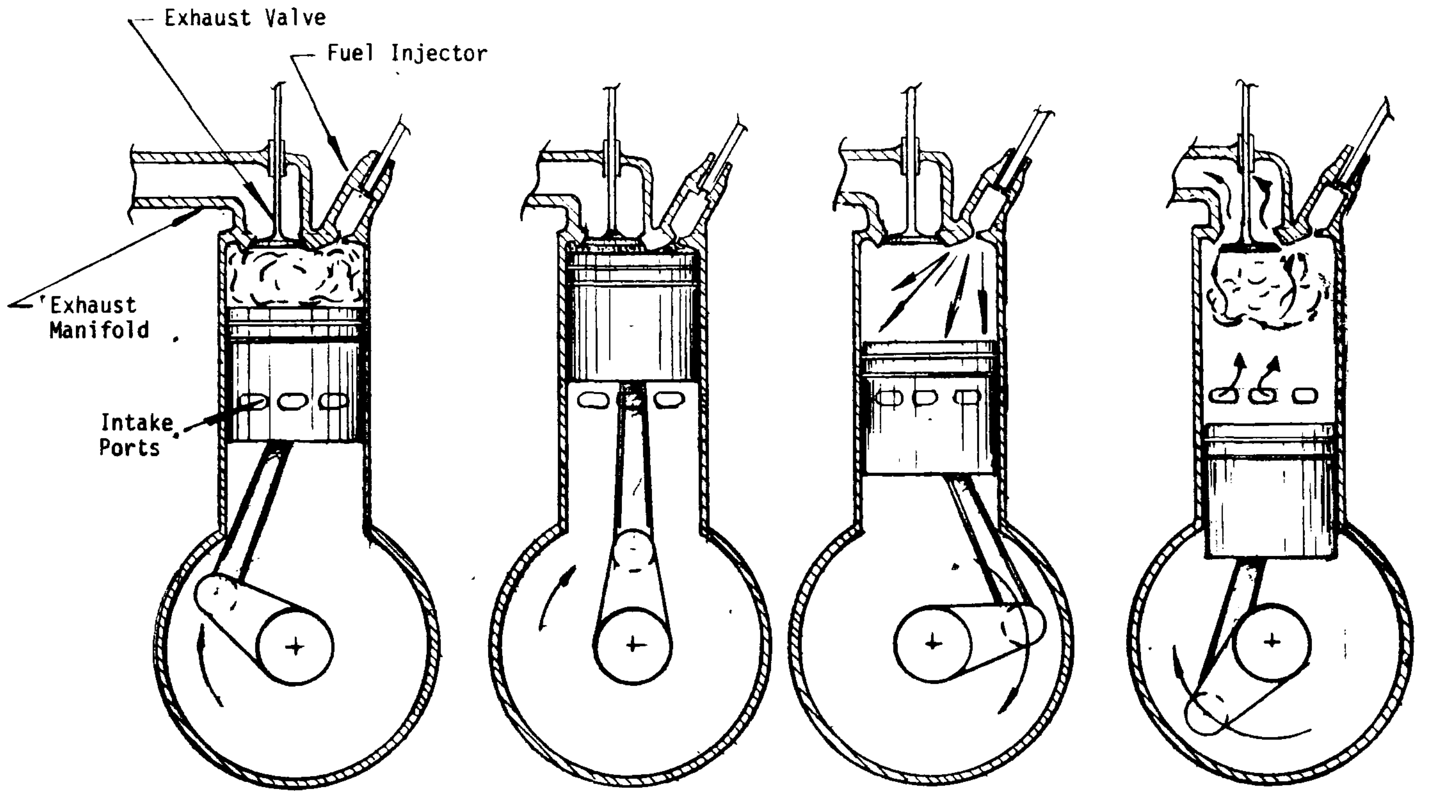


Figure 3-2. Cylinder events for a two-stroke blower-scavenged IC engine (Reference 3-4).

air. Therefore, the pressurized air is often cooled before entering the cylinder to further increase power by allowing more air mass to be introduced into the cylinder. This process is called intercooling or aftercooling.

Two-stroke engines are often air-charged by a blower, which also aids in purging the exhaust gases. Such systems are called blower-scavenged. This method is less efficient than turbocharging because the blower produces less pressure than a turbine. However, high volumetric flow rates are achieved, effectively purging the cylinder of exhaust gases.

In a CI engine, fuel is injected into the cylinder near the end of the compression stroke; whereas, in an SI engine, the fuel is usually added to the air downstream of the turbocharger if any is used, and before the mixture enters the cylinder. This is done with a carburetor. However, some SI engines (particularly large natural gas fueled ones) inject the fuel into the intake manifold just ahead of the valves, or into the cylinder as done with CI engines.

Two methods of injection are commonly used. Direct injection places the fuel directly into the cylinder and the principal combustion chamber. These units are also called open chamber engines because combustion takes place in the open volume between the top of the piston and the cylinder. In contrast is indirect injection, where combustion begins in a fuel rich (oxygen deficient) atmosphere in a smaller antechamber and then expands into the cooler, excess air region of the main chamber. These latter engines are also called divided or precombustion chamber systems.

3.2 APPLICATIONS AND RANK

Stationary reciprocating IC engines can be classified into four characteristic size ranges, with common applications within each range.

3.2.1 Large Bore, High Power, Low- and Medium-Speed Engines

The large bore, high power (>75 kW/cylinder and less than 1000 rpm) engines characteristically are four-cycle, compression ignition engines designed to operate on either diesel oil or a mixture of oil and natural gas (dual fuel) (Reference 3-1). Two cycle are also common. The remainder of the engines are either spark ignited natural gas engines, or engines which operate as either spark ignited or compression ignited (dual fuel or oil) and can easily be switched in the field in response to fuel availability. These engines, therefore, typically find uses in industries which seek best economy, and usually have sufficiently large installations to provide several fuel types. Typical industries where these large bore engines may be found are municipal electric power generation, oil and gas pipeline transmission, and oil and gas production. In these industries, the engine is run continuously. Based on 1976 data, only about 1000 to 2000 of these engines are sold per year, with a total production value of $80 to $150 million (1976 dollars). Sales have generally been declining, although sales of diesel engines for electric power generation are up (References 3-2 and 3-4).

3.2.2 Medium Power, High Speed Engines

This size (7.5 to 75 kW/cyl and greater than 1000 rpm) class of engines has the greatest variety, with some larger units equaling the power of the large engine class. There are basic differences, however, which separate the two groups. Large bore engines produce high power output at low speeds due to their large displacement and consequent high power per cylinder. Medium bore engines, in contrast, have lower power per cylinder (and therefore more cylinders for the same engine horsepower). They achieve high outputs by utilizing higher rotative speeds. Thus, for the same power rating, high speed engines are smaller, less expensive, and capable of running at a wider range of speeds. The smaller engines within this size range are produced by truck and tractor engine manufacturers and therefore can be considered mobile engines modified for stationary installation and constant speed operation (Reference 3-2). Fuels burned are typical mobile fuels, either diesel oil or gasoline, although there are a very few (usually modified) natural gas engines within this size range.

These engines are used in miscellaneous industrial, commercial, nonpropulsive marine, and agricultural applications where shaft power is needed and electric motors cannot be used. Unlike the large engines which are typically sold directly to the user, these engines are most often sold to Original Equipment Manufacturers who purchase engines for their manufactured generators, compressors or pumps, or other equipment, and sell the complete package to the user (References 3-1, 3-6, 3-7).

Precise production data are unavailable because the number of mobile engines modified for stationary use is unknown. However, in 1976, sales of diesel medium power, high speed engines for stationary applications have been estimated in the range of 60,000 to 80,000 units/year over recent years, with a total value of $150 to $200 million per year (1976 dollars, FOB plant) (Reference 3-1). Annual sales for gasoline medium power, high speed engines have been approximately 100,000 with a total value of $50 million (FOB plant) (1976). Sales of these engines have been erratic from one year to the next, but the general trend seems to be upward. The largest sales gains have been in the 1000 kW to 3500 kW range, for OEM generator packages (Reference 3-8).

3.2.3 Small Engines and Very Small Engines

Small engines are distinguished from the other engines in that they are mostly one- and two-cylinder engines of less than 40 horsepower. These engines are mostly diesel and gasoline, one- and two-cylinder models, with some four-cylinder models. Almost all have four strokes per power cycle and usually are air cooled.

Small engines are typically used in generator sets, small pumps and blowers, off-the-road vehicles, and refrigeration compressors for trucks and railroad cars. Very small engines find additional uses for lawn and garden equipment, chain saws, and recreational vehicle generators. Sales of small engines are in the range of $400 to $500 million per year (1976). Diesel engines used for drip irrigation systems have been showing the largest gain recently (Reference 3-9).

3.2.4 Engine Users

On the basis of installed capacity, the principal stationary applications of IC engines are: oil and gas pipelines, oil and gas production, general industrial (including construction), electrical power generation, and agriculture.

Pipeline installations are concentrated in the oil and gas producing states on the Gulf Coast and in the Midwest. Electric generation includes base load generation, principally for municipalities in the plains and Midwest, peaking power, emergency standby for vital public services and large buildings in urban areas, and remote generation (for mines and homes) in rural areas. Agricultural applications are primarily irrigation pumping, concentrated in those areas with irrigated farm lands (such as central and southern California, Arizona, and northwestern Texas). Additional agricultural applications include frost control, harvesting (auxiliary engines), and some remote electric generation. Construction applications include portable compressors, welders, pumps, electric generation, and material handling equipment.

3.3 TYPICAL ENGINES IN EACH SIZE CLASS

The previous section identified three engine size ranges. This section summarizes those mechanical aspects considered typical of each size category.

3.3.1 Large Bore, High Power Engines

These engines are almost always designed for permanent installation and rarely sold as a package. They typically are multicylinder in "V" or inline configurations, with large displacements per cylinder and correspondingly high cylinder power (>75 kW/cyl). Rotational speeds are less than 1000 rpm. Fuels for these engines are either diesel oil or natural gas, with some engines capable of burning either or both fuels (requiring at least 5 percent diesel oil). Engines can also be modified to run on other fuels if the cost of the alternate fuel is less.

Large bore engines are always designed to give best economy (except for emergency standby use) and therefore are usually turbocharged and aftercooled to increase the air charge to the cylinders. Efficiencies (energy out/energy in) are typically 35 to 40 percent based on the lower heating value of the fuel (Reference 3-1).

3.3.2 Medium Bore, Medium Power Engines

Most medium power stationary IC engines are modified heavy duty mobile engines, although some manufacturers' units are only for stationary use (Reference 3-2). Because sales are typically to small industries which do not have extensive fuel supplies or process facilities or sales for use in remote areas, most installations are packaged self-contained units. This allows these engines to be semi-portable. Like the large bore engines, these engines are multicylinder in "V" or inline configurations, but have much smaller displacements and power per cylinder (7.5 kW to 75 kW). Engine speed is usually greater than 1000 rpm.

Fuels are typical of mobile engines, either diesel oil or gasoline. Engines are often turbocharged to increase thermodynamic efficiency to as high as 38 percent. Naturally aspirated engines in this size range have typical efficiencies of 30 percent (References 3-1, 3-2, 3-6).

3.3.3 Small Bore Engines

Small engines are typically one- or two-cylinder models and are usually air cooled. They are sold in packaged units or separately, and rarely are permanently installed. Displacements per cylinder are small, with total engine output less than 75 kW. Depending on use, engine speed ranges from 500 rpm to over 3000 rpm.

Most of these engines are naturally aspirated. The prime fuel is gasoline, although there are several manufacturers of diesel engines in this size range. Efficiencies of these engines are generally low, ranging from 25 percent to 35 percent (References 3-6 and 3-10).

REFERENCES FOR SECTION 3

3-1. "Stationary Internal Combustion Engines. Background Information: Proposed Standards," EPA 450/3-78-125a, April 1979.

3-2. Roessler, W. U., et al., "Assessment of the Applicability of Automotive Emission Control Technology to Stationary Engines," EPA-650/2-74-051, NTIS-PB 237 115, July 1974.

3-3. Salvesen, K. G., et al., "Emissions Characterization of Stationary NO_x Sources: Volume I. Results," EPA-600/7-78-120a, NTIS PB 284 520, June 1978.

3-4. Obert, E. F., "Internal Combustion Engines," Intext Educational Publishers, New York, NY, 1973.

3-5. Baumeister, T., ed., Marks' Standard Handbook for Mechanical Engineers, Seventh Edition, McGraw-Hill, New York, NY, 1976, pp. 9-144.

3-6. Letter from C. I. Taggart, White Motor Corporation, Eastlake, OH, to R. L. Seiffert, U.S. Environmental Protection Agency, Hopkins, MN, January 1975.

3-7. Hanley, G. P., "Impact and Potential for Improvement for Stationary Engines," General Motors Statement at California Air Resources Board Workshop on Suggested Rules for the Control of NO_x Emissions in the South Coast Air Basin, Los Angeles, CA, May 1978.

3-8. Diesel and Gas Turbine Progress Worldwide, Vol. X, No. 8, October 1978, p. 10.

3-9. Diesel and Gas Turbine Progress Worldwide, Vol. IX, No. 8, October 1977, p. 22.

3-10. Letter from W. Bazen, Briggs & Stratton Corporation, Wauwatosa, WI, to S. Youngblood, Acurex Corporation, Mountain View, CA, 1974.

SECTION 4
CHARACTERIZATION OF INPUT MATERIALS, PRODUCTS, AND WASTE STREAMS

Stationary reciprocating internal combustion engines are sources of NO_x, HC, CO, and particulate emissions. Sulfur oxide emissions are also possible if the fuel burned contains sulfur. Most of these pollutants are formed during the combustion process and emitted through the engine exhaust. However, some emissions also escape from the engine crankcase, and others from fuel evaporation. This section presents the typical emission levels from stationary engines from these sources, then describes the effects of atmospheric conditions, fuel, and engine operation on these emissions.

4.1 BASELINE EMISSIONS

Because stationary engine sizes vary widely, emissions have always been presented on a mass-to-power basis. This allows comparisons of emissions between engines of various sizes. In addition, since some engines have already had emission controls applied, emissions are presented as controlled or uncontrolled.

Typical uncontrolled exhaust emission rates, based on the heavy duty engine 23-mode composite (see Federal Register, September 8, 1977, page 45132), for four-stroke naturally aspirated gasoline engines in g/brake kWh are: 3.2 to 17.6 for HC; 40 to 120 for CO; and 10.6 to 18.6 for NO_x (Reference 4-1). The 23-mode composite tests includes steady and transient operation. Similarly, brake specific emission rates from all types of spark ignited natural gas fired engines (two- and four-stroke, naturally aspirated and turbocharged engines) are: 0.5 to 9.4 for HC; 0.3 to 38.0 for CO; and 10 to 39 for NO_x (Reference 4-1).

For compression ignition engines, typical emission rates for all types of uncontrolled diesel engines (including four-stroke, either naturally aspirated or turbocharged and open- or divided-chamber, and two-stroke, either blower-scavenged or turbocharged) in g/brake kWh are: 0.1 to 3.9 for HC; 0.4 to 20 for CO; and 2.8 to 23 for NO_x (Reference 4-1).

Average emission factors have been developed for engines of various sizes and fuel types and are presented in Table 4-1 (References 4-1 and 4-2).

As previously mentioned, the engine exhaust is not the only source of emissions. Hydrocarbons escape from the crankcase because of blowby (gases which are vented from the oil pan after escaping from the cylinder past the piston rings) and from the fuel tank and carburetor due to evaporation. For diesel engines, crankcase blowby is minor (0.24 g/kWh HC + NO_x, 0.0084 g/kWh CO) because hydrocarbons are not present during compression of the charge (Reference 4-1). Evaporative losses are also insignificant due to the low volatility of diesel fuels. Evaporative losses are also negligible in engines using gaseous fuels because these engines usually receive their fuel continuously from a pipe rather than via a fuel storage tank and fuel pump. However, in gasoline fueled engines 20 to 25 percent of the total hydrocarbon emissions from uncontrolled engines come from crankcase blowby and another 10 to 15 percent from evaporation of the fuel in the storage tank and the carburetor (divided approximately equally between the two). Crankcase blowby emissions can be virtually eliminated through the use of positive crankcase ventilation as is demonstrated in the case of mobile engines.

Finally, oil and cooling water that are replaced during maintenance represents additional waste streams that are periodically discharged. These wastes need to be disposed of properly. However, combustion modifications should not have any major effects on these waste streams, and therefore disposal of these wastes is not discussed in this report.

4.2 EMISSIONS FORMATION

Oxides of nitrogen can be formed either from atmospheric nitrogen in the combustion air (thermal NO_x) or from nitrogen compounds in the fuel itself (fuel NO_x). The amount of thermal NO_x formed depends on the temperature, oxygen concentration, and residence time of the nitrogen at high temperatures and/or oxygen concentration. Fuel NO_x formation appears to be directly proportional to the amount of nitrogen compounds in the fuel. For burner (continuous) type combustion, it has been found that

TABLE 4-1. EMISSIONS FACTORS FOR IC ENGINES, g/kWh[a]
(References 4-1, 4-2)

Fuel		NO_x	CO	HC
Gasoline	>15 kW	11.9	137	11.2
	<15 kW	7.5	395	27.5
Diesel	>375 kW[b]	17.3	2.4	0.6
	<375 kW[c]	16.6	6.0	2.8
Natural gas		15.4	3.8	6.5
Dual Fuel		11.0	2.7	4.1

[a]Emission factors for gasoline and diesel engines are modal averages; those for natural gas and dual fuel are for rated conditions. The modal averages includes numbers from the constant output part of mobile transient tests.
[b]Based on an average of rated condition levels from engines considered.
[c]Weighted average of two- and four-stroke engines. Weighting factors = 2/3 for four-stroke and 1/3 for two-stroke.

as fuel nitrogen compounds increase, the NO_x also increases, although not as rapidly. The temperature, oxygen concentration, and residence time also influence how much fuel nitrogen is converted to NO_x. Because of the large amount of NO_x produced by an IC engine as compared to a distillate oil or natural gas burner, most of the NO_x emitted from an IC engine is probably thermal NO_x. For example a residual oil boiler burning oil with a nitrogen content of 0.5 percent would have a NO_x level of about 350 ppm at 3 percent O_2, while a diesel engine burning a diesel oil with a nitrogen content with less than 0.1 percent N, could have a NO_x level of 3000 ppm at 3 percent O_2.

Most of the other pollutants, HC, CO, and smoke are mainly the result of incomplete combustion. Hydrocarbon emissions are believed to be caused by three general mechanisms (Reference 4-3): wall quenching (fuel impingement on the walls causing the fuel to be cooled below the combustion temperature), variations in engine variables (mixing inside the cylinder, wrong air-to-fuel ratio, defective ignition, etc.), and, in two-cycle engines cooling down the exhaust gases by the scavenging air before combustion is completed. CO emissions can also form by the same general mechanisms.

Smoke formation is also related to incomplete combustion. The color of the smoke indicates the cause of the smoke (Reference 4-1). Bluish smoke occurs by incomplete combustion of crankcase oil forced past worn piston rings into the cylinder. White smoke usually occurs at low load or idle conditions and is mainly unburned liquid fuel or lubricating oil. Black smoke consists of carbon particles formed by incomplete combustion at high temperatures.

The other engine emissions such as SO_x, lead and other metals are directly related to the amount of these compounds in the fuel.

Atmospheric conditions, fuels used, and engine design and operation also affect emissions. These effects are described in the following sections.

4.3 ATMOSPHERIC EFFECTS

The effects of the atmospheric conditions on NO_x emissions have been evaluated by several sources, predominately by or for automotive engine manufacturers. Their test results indicate changes in NO_x of up to 25 percent caused by ambient temperature and humidity changes, and up to 40 percent changes in NO_x emissions caused by ambient pressure changes (Reference 4-1). Most of these effects are caused by changes in the air-to-fuel ratio as the density of the combustion air changes. However, humidity has an additional effect on NO_x in that high moisture conditions reduce the peak temperatures within the engine cylinder, decreasing NO_x emissions (References 4-3 through 4-8).

Because one of the variables of engine design is the air-to-fuel ratio, different engines respond differently to changes in atmospheric conditions. Thus it is quite difficult to quantify atmospheric effects on engine emissions. However, these general effects have been observed for engines operating close to stoichiometric conditions:

- Increases in humidity decrease NO_x emissions
- Increases in temperature increase HC and CO emissions
- Decreases in pressure increase HC and CO emissions.

It should be noted that most large engines are not operated close to stoichiometric conditions.

4.4 FUEL EFFECTS

Generally the difference in NO_x emissions from large bore engines operating with any of the primary fuels (diesel oil, gasoline, natural gas) is small. This is because the main source of NO_x is N_2 in the combustion air, rather than from the fuel.

However, there are significant differences in other pollutants. Hydrocarbons from natural gas engines can be five times greater than for diesel oil, although most of the emissions are methane. On the other hand, SO_x and particulate emissions are much greater with diesel oil. CO emissions are generally always high with gasoline. Characteristics of each primary fuel and its effects on emission are discussed in the following subsection.

4.4.1 Diesel Oil (No. 2)

Diesel oil is an important fuel for stationary reciprocating IC engines, since its use is increasing as users become concerned about natural gas supplies. Specifications for use in engines are well defined, as indicated in Table 4-2.

Diesel fuel composition is complex and hence the exhaust may contain between 9,000 and 12,000 different compounds (Reference 4-10). As indicated, diesel fuels contain impurities such as sulfur (up to 10 times

TABLE 4-2. SPECIFICATIONS FOR DIESEL FUELS (ASTM D975) (Reference 4-9)

Test	Limit
Flashpoint, K (°F), min	325(125)
Water and sediment, vol percent, max	0.10
Viscosity, kinematic, centistokes, 311K (100°F)	
min	2.0
max	5.8
Carbon residue, wt percent, max	0.35
Ash, wt percent, max	0.02
Sulfur, wt percent, max	1.0
Ignition quality, cetane number, min	40
Distillation, temp, deg K (°F) 90% evaporated	
min	555(540)
max	575(576)

more than gasoline -- Reference 4-10), ash, and carbon residue. Some fuel bound nitrogen may be included, and detergents and metal based additives may be added. All these fuel components affect emissions. The presence of sulfur and nitrogen in the fuel affects SO_x and NO_x emissions respectively although the maximum effect of fuel bound nitrogen is probably less than 15 percent (i.e., at least 85 percent of the NO_x is thermal -- Reference 4-1). Probably the most critical parameter indicating the fuel effects of diesel oils is the cetane number, or ignition quality. Increase in cetane number decreases NO_x, HC, and CO, but increases smoke through higher carbon formation (Reference 4-11). This is because the hydrogen of large fuel molecules burns away, leaving carbon which cannot be oxidized during the rapidly decreasing temperatures within the cylinder. Barium based additives can reduce smoke, but often increase other pollutants when impurities or deposits form.

4.4.2 Gasoline

Gasoline is used mostly for mobile engines, and emissions from mobile sources have been well documented. But gasoline has found use in stationary applications because many medium and small engines are either converted mobile engines or designed specifically for light stationary use. Thus gasoline, although not a significant fuel on an installed horsepower basis, is an important stationary fuel.

Like diesel oil, gasoline is a blend of many hydrocarbons. Because this fuel is more volatile than diesel oil, a characteristic of this fuel is an evaporative emission of 5 percent of the total HC emitted. Composition effects related to the H/C ratio of the fuel have been observed. The H/C ratio affects both NO_x and HC emissions, with NO_x being decreased as H/C increases because of changes in adiabatic flame temperature. HC emissions also decrease as H/C increases, probably due to changes in volatility and its effect on fuel combustion (References 4-3 and 4-12).

Fuel additive effects on emissions have also been observed. Tetraethyllead (TEL), an additive for antiknock purposes, is a major source of particulate, with smoke emission levels 10 to 20 times those for unleaded fuels. Lead additives also increase HC emissions (References 4-11, 4-12, 4-13).

CO emissions from gasoline are higher than from any other fuel. However, these are not necessarily caused by fuel composition, but are a function of stoichiometry, or combustion in fuel rich zones in the engine cylinder. Because gasoline engines generally operate close to a stoichiometric fuel ratio, CO emissions are therefore higher than emissions from other more lean burning fueled engines (References 4-3, 4-12).

4.4.3 Natural Gas

Natural gas is used more than any other fuel for stationary engines, although usage has been declining in recent years. There are no specific specifications for natural gas because properties vary by location. Natural gas is typically 85 percent methane with the rest of its composition consisting of other low boiling hydrocarbons, hydrogen, and nitrogen. NO_x emissions from natural gas fired engines are very similar to diesel engine emissions, primarily because natural gas engines and diesel engines often operate under similar conditions. However, total hydrocarbons are five times greater and CO is half as much in gas fired engines when compared to diesel engines. Most of the increase in HC emissions is due to blowby and fugitive sources since the fuel is gaseous. However, typically 80 percent of the HC emission is methane, a noncriteria pollutant. CO emissions are less because usually the gaseous fuel improves mixing within the engine cylinder, and oxidation is more complete (Reference 4-1).

4.4.4 Other Fuels

Other gaseous and liquid fuels may be used for stationary engines. Of course, emissions will vary depending on composition, additives, etc. Little data are available for other fuel types, thus it is difficult to predict general emission characteristics from other fuels.

4.5 EFFECTS OF ENGINE DESIGN AND OPERATION

Almost any variation in engine design or operation parameters will affect emissions. These parameters may be divided into four classes: (1) charging methods and air-to-fuel ratio; (2) engine combustion cycle; (3) combustion chamber design and valve and ignition timing; and (4) operating conditions of load and speed.

4.5.1 Charging Method and Air-to-Fuel Ratio

Charging method does not directly affect emissions, but it influences the air-to-fuel ratio(A/F). Air-to-fuel ratio, in turn, affects emissions significantly. These effects are best summarized in Figure 4-1. At air-to-fuel ratios below stoichiometric (rich), combustion occurs under conditions of insufficient oxygen and thus unburned hydrocarbon emission increase. CO increases because carbon is not sufficiently oxidized to CO_2. NO_x decreases both because of insufficient oxygen and lower temperatures.

At air-to-fuel ratios above stoichiometric (lean), combustion occurs under conditions of excess oxygen, thus essentially all carbon is oxidized to CO_2. NO_x first increases rapidly with A/F near stoichiometric, because of the excess oxygen and peak temperatures, then decreases rapidly with A/F as the excess air cools peak combustion temperatures. Hydrocarbons stay at a low level, then begin to increase as the air-to-fuel ratio is increased because the lower temperatures inhibit combustion.

Charging method is important because it often limits the range of air-to-fuel ratio. Naturally aspirated carbureted engines generally must operate with overall air-to-fuel equivalence ratios, defined as $(A/F)_{stochiometric}/(A/F)_{actual}$, greater than 0.7 because poor distribution among cylinders will allow some cylinders to go excessively lean. In contrast, turbocharged fuel injected engines, with precise control of air-to-fuel ratio to each cylinder, can operate at equivalence ratios of 0.5 to 0.3 without increasing hydrocarbon emissions significantly. Some blower scavenged engines operate at equivalence ratios below 0.25, although the actual ratio inside the cylinder is usually higher (Reference 4-3).

The choice of lean or rich operation often depends on engine use. Rich operating (meaning close to stoichiometry) engines give quicker response to changing conditions, and also produce maximum power. In addition, carbureted engines generally cost less than a comparable turbocharged model. Thus, these engines typically find use in construction and lightweight general industrial service.

Lean burning engines are much more economical to operate. Therefore, large industry applications where the engine will be operated for several thousand hours a year at constant load and speed typically have lean burning engines.

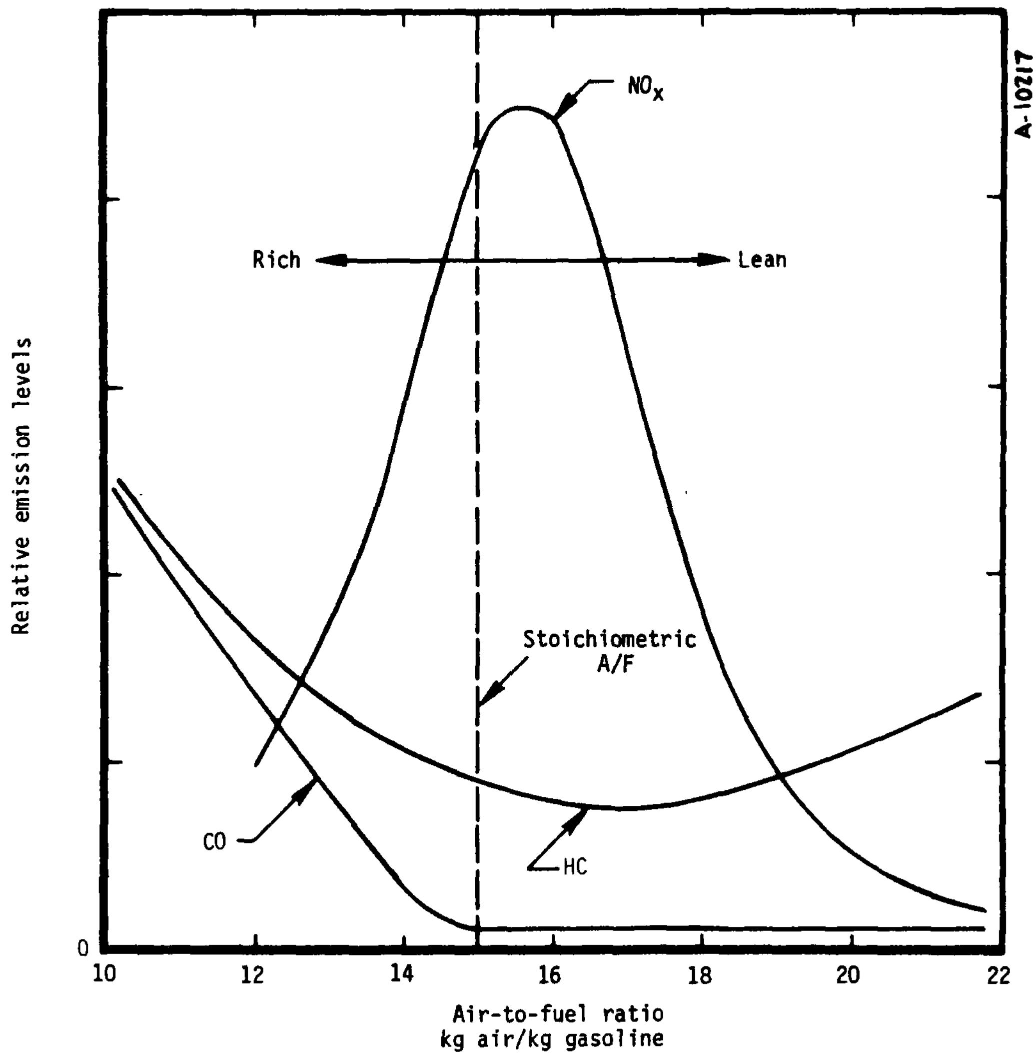


Figure 4-1. Effect of air-to-fuel ratio on emissions from a gasoline engine (Reference 4-1).

4.5.2 Combustion Cycle

As discussed in Chapter 3, reciprocating IC engines may be either two- or four-stroke cycle. During combustion, emissions from either type are essentially identical. However, during the charging of a two-cycle engine, several events can take place. On noninjected engines, the scavenging air, which purges the cylinder of exhaust gases and provides the combustion air, can also sweep out part of the fuel charge. Thus carbureted two-cycle engines often have higher HC emissions in the form of unburned fuel.

The two-stroke engine can also have lower NO_x emissions. If the cylinder is not completely purged of exhaust gases, the result is internal exhaust gas recirculation (EGR). The remaining inert exhaust gases absorb energy from combustion, lowering peak temperatures, and thereby lowering NO_x.

4.5.3 Combustion Chamber Design and Ignition Timing

Almost any variation in cylinder design, valving, or ignition timing will affect emissions. Unfortunately the effects cannot be quantified since each engine is different and some design variables will cancel any beneficial effects of others. However, some generalization can be made. Design variables which improve mixing within the cylinder tend to decrease emissions. Improvements in mixing may be accomplished through swirling the air or fuel and air mixture within the cylinder, improving the fuel atomization, and optimizing the fuel injection locations. Decreasing the cylinder compression ratio reduces the maximum temperatures achieved in the cylinder, lowering NO_x emissions.

Stratifying the charge into a fuel rich zone and fuel lean zone also reduces peak combustion temperature and NO_x. This usually requires a small antechamber apart from the cylinder itself. Fuel is injected or combustion initiated in the smaller (rich) chamber, then expands into the lean main chamber (cylinder). Combustion occurs slower with a lower pressure rise, thereby reducing peak temperatures.

Another technique of reducing temperature is to retard spark or injection timing. This is recognized as an effective NO_x control technique. By initiating combustion later in the cycle, pressures are reduced, thus lowering peak temperatures. However, there is a fuel penalty of 5 to 8 percent and the potential of excessive smoke from some engines (Reference 4-1).

Finally, engine size affects emissions, and is another variable of design. For instance, large bore, low- to medium-speed engines, independent of the fuel type (diesel, dual fuel, natural gas), are designed for low fuel consumption (i.e., high thermal efficiency). Their low rotational speed maintains the products of combustion near peak temperature for relatively long periods of time, so that fuel injection timing may be adjusted for optimum results. The larger engines also tend to have higher NO_x emissions. The large combustion volumes also allow more aerodynamic design freedom than is available in small- or medium-bore engines. Furthermore, these large engines are designed to operate under steady conditions and almost always at more than 50 percent of rated power; therefore, they do not have to contend with acceleration/deceleration or low power requirements .

4.5.4 Load and Speed Effects

Load and speed effects on emissions also vary from engine to engine. In general, diesel compression ignition engines exhibit decreasing brake specific emissions of NO_x with increasing load at constant speed. This is partly caused by changes in the air/fuel ratio. Some turbocharged engines show the opposite effect of increasing brake specific NO_x emissions as load increases. CO emissions first decrease with increasing load (equivalent to increasing temperature) and then increase as maximum load is approached. Brake specific HC emissions, decrease with increasing load as a result of increasing temperature, but smoke emissions reach their maximum at full load (References 4-1, 4-3, 4-11).

Natural gas engines follow the same trends as diesel engines for HC and CO, but generally have maximum NO_x at maximum power.

Gasoline engine results vary greatly, but generally show the same trends as above for HC and CO, with NO_x peaking at some intermediate load. Small (less than 11 hp) SI engines (gasoline) exhibit relatively high HC and CO emissions, particularly at low load operation. Speed effects generally will decrease HC and CO, and increase NO_x emissions. However, speed will also affect other design and operating variables which may reverse the positive effects (References 4-1, 4-4, 4-11).

4.6 PRODUCTS CHARACTERIZATION

The principal product of an IC engine is shaft power. The exhaust gas from an engine can also be considered a "product." The hot exhaust gas can be used to supply waste heat via a heat recovery device or be used in a supplementary fuel fired waste heat boiler. The product exhaust gas from an IC engine has been discussed earlier.

REFERENCES FOR SECTION 4

4-1. "Stationary Internal Combustion Engines: Background Information Proposed Standards," EPA 450/3-78-125a, April 1979.

4-2. Youngblood, S. B., and G. R. Offen, Acurex Internal Memorandum, "Emissions Inventory of Currently Installed Stationary Reciprocating Engines," Acurex Corp., Mountain View, CA, September 23, 1975.

4-3. Patterson, O. J., and N. A. Henein, "Emissions from Combustion Engines and Their Control," Ann Arbor Science Publishing, Inc., Ann Arbor, MI, 1972.

4-4. Grinberg, L., and L. Morgan, "Effects of Temperature on Exhaust Emissions," SAE 740527, June 1974.

4-5. Robson, J. A., "Humidity Effects on Engine Nitric Oxide Emissions at Steady State Conditions," SAE 700467, May 1970.

4-6. "Effect of Humidity of Air Intake on Nitric Oxide Formation in Diesel Exhaust," CRC Report No. 447, Coordinating Research Council, Inc., New York, NY, December 1971.

4-7. Krause, S. R., "Effect of Engine Intake-Air Moisture, Temperature, and Pressure on Exhaust Emissions," Ethyl Corp., Richmond, VA, December 1970.

4-8. Brown, W. J., et al., "Effects of Engine Intake-Air Moisture on Exhaust Emissions," SAE 700107, January 1970.

4-9. Baumeister, T., ed., Marks' Standard Handbook for Mechanical Engineers, Seventh Edition, McGraw-Hill, New York, NY 1976.

4-10. Ketcham, B., and S. Pinkwas, "Diesels and Man, Are We Creating a New Environmental Problem by Solving an Old One?" New Engineer, Vol. 7, No. 4, pg. 23, April 1978.

4-11. Roessler, W. U., et al., "Assessment of the Applicability of Automotive Emissions Control Technology to Stationary Engines," EPA-650/2-74-051, NTIS-PB 237 115, July 1974.

4-12. Harrington, J. A., and R. F. Shishu, "A Single-Cylinder Engine Study of the Effects of Fuel Type, Fuel Stoichiometry, and Hydrogen-to-Carbon Ratio on CO, NO, and HC Exhaust Emissions," SAE 730476, May 1973.

4-13. Salvesen, K. G., et al., "Emissions Characterization of Stationary NO_x Sources: Volume I. Results," EPA-600/7-78-120a, NTIS-PB 284 520, June 1978.

SECTION 5
PERFORMANCE AND COST OF CONTROL ALTERNATIVES

Emission controls for stationary reciprocating internal combustion engines can be divided into two classes; those which reduce NO_x emissions and those which reduce other pollutants. This is because the formation mechanisms which generally produce minimum HC, CO, and smoke emissions (high temperatures) often produce peak NO_x emissions, and vice versa. Thus tradeoffs between emission levels of NO_x and the other pollutants should be evaluated before any controls are applied.

The baseline emission levels reviewed in Section 4.1 indicate that NO_x emissions for diesel oil and natural gas are at least 2 to 3 times greater than other emissions. Gasoline engines also emit high NO_x levels in addition to high levels of CO. Thus NO_x is an essential emission control for all engines, and CO is important for stationary gasoline engines. HC emission control for very small engines may also be important.

From a nationwide emission standpoint, however, CO emissions become less important. Stationary engines account for approximately 3 percent of the total nationwide CO, versus nearly 7 percent for NO_x (Table 5-1) (Reference 5-1). Furthermore, large engines (greater than 75 kW/cyl) account for over 75 percent of all NO_x emitted from stationary engines. Also CO or HC emissions can be lowered by efficient operation, while NO_x reduction is harder to achieve. In addition, these large engines tend to be in a class by themselves (large bore, low rpm, essentially no transients), while the medium to small engines usually are modified mobile engines (smaller bore, greater rpm, and operate under more transient conditions). This division is not complete since some of the medium bore engines are as large as the large bore engines. Since much has been written about mobile engines and the larger size stationary engines are the main source of pollutants, this section will concentrate on these

TABLE 5-1. EMISSIONS FROM IC ENGINES (1975)[a]

Size	Emissions (10^6 metric tons/year)		
	NO_x	CO	HC
>5700 cc/cylinder	1.15	0.27	0.45
375 kW to 5700 cc/cyl	0.02	0.01	0.00
75 kW to 375 kW	0.16	0.35	0.05
11 kW to 75 kW	0.16	0.88	0.08
11 kW	0.04	1.95	0.14
Total	1.53	3.47	0.72
All Stationary Sources	14.5	32.6	10.4
All Sources	23.8	105.8	24.2

[a]Reference 5-1

larger engines. Also, since NO_x is the major pollutant from these engines, the following sections focus on emission reduction controls and their effectiveness on oxides of nitrogen. The performance and cost of these controls are estimated.

5.1 NO_x CONTROL TECHNIQUES

There are several demonstrated NO_x control techniques for large, stationary reciprocating internal combustion engines. Most of these reduce NO_x emissions by lowering the peak temperatures within the engine cylinder or altering the burning rate. These techniques include changes in air-to-fuel ratio or ignition timing, manifold air cooling, modifications to the combustion chamber, exhaust gas recirculation, and water injection. In addition, for many engines, derating the engine power is effective. Flue gas treatment controls such as catalytic reduction are under development.

Each control technique, its effectiveness, effects on other emissions, and operation and maintenance impacts are discussed in the following sections. All data are from Reference 5-1 except as noted.

5.1.1 Air-to-Fuel Ratio Changes

The air-to-fuel (A/F) ratio is defined as the mass flowrate of air divided by the mass flowrate of fuel. This ratio is termed stoichiometric if precisely enough oxygen is present in the mixture to just completely oxidize the fuel. Effects of air-to-fuel ratio changes were described in Section 4, and were summarized in Figure 4-1.

When the engine is operated rich, the lack of excess oxygen suppresses NO_x formation and so, despite the high cylinder temperatures, NO_x formation will drop sharply at increasingly rich mixtures. If the ratio is varied in the lean direction, the excess oxygen level will increase but so will the mass of the combustion mixture, enabling it to absorb more heat. This reduces the peak temperature, resulting in lower NO_x formation.

The most practical use of air-to-fuel ratio adjustment as a control technique is to change the setting toward leaner operation, since increasingly rich mixture operation increases both HC and CO emissions and fuel consumption. This technique is better suited to injection type engines. Carbureted engines will require better control of the air-to-fuel ratio between cylinders before they can operate at leaner A/F ratios. In fact, some current carbureted engines, when adjusted to leaner than normal air-to-fuel ratios (but still rich), tended to increase their NO_x emissions because they were moving towards the peak of the NO_x versus air-to-fuel ratio curve. They were not able to go beyond that point to lower NO_x levels in the lean region without misfiring (Reference 5-2).

Injection type engines achieve leaner air-to-fuel ratios by either reducing the fuel input (essentially derating) or by increasing the air input. More air flow can be accomplished by either installing a turbocharger or increasing the capacity of an existing turbocharger. For blower scavenged engines, the same is true. These changes are limited by increased parasitic horsepower which could increase fuel consumption.

Also the chance of misfiring becomes greater, which may require more sophisticated control of the air-to-fuel ratio such as O_2 sensors in the exhaust and a feedback control.

For carbureted engines, air flows can be increased by changing the venturi and fuel nozzles (Reference 5-3). Also, some modifications to the intake manifold may be required to allow a uniform fuel-air mixture distribution to each cylinder. The difficulty with distributing a uniform mixture to each cylinder via an intake manifold is why injector type engines allow leaner air-to-fuel ratios.

Figure 5-1 shows the effect of air-to-fuel ratio change on NO_x emissions and fuel consumption for diesel, gas, and dual fuel engines. Small changes in the air-to-fuel ratio, approximately 10 percent, generally reduce NO_x by about 30 percent with a fuel penalty of less than 5 percent. As discussed earlier, the limit to changes in the air-to-fuel ratio (i.e., the maximum ratio) is set by increased fuel consumption, onset of misfiring, and increases in organic emissions. Tests on installed gas pipeline engines have shown that increased air-to-fuel ratio is the most significant source of NO_x reductions from large bore gas engines (Reference 5-6).

5.1.2 Retarded Ignition Timing

Ignition in a normally adjusted engine is set to occur shortly before the piston reaches its uppermost position (top dead center or TDC). At TDC the air or air-fuel mixture is compressed to the maximum. The timing of the start of injection or of the spark is given in the number of degrees that the crankshaft must still rotate between this event and the arrival of the piston at TDC. The extent of retard is then expressed in degrees relative to normal ignition. Typical retard values achievable are 2^o to 6^o, depending on the engine. Beyond these levels, fuel consumption increases rapidly, power drops, misfiring (erratic ignition) occurs, and smoke from diesel engines becomes excessive (Reference 5-4). When ignition is retarded, the combustion process duration does not change significantly but extends longer to the power stroke. Thus combustion proceeds at a lower peak temperature, but raises the exhaust temperature. This lower peak combustion temperature decreases NO_x formation but does not affect CO and HC emissions, unless the temperature is lowered significantly. Smoke from diesel engines may

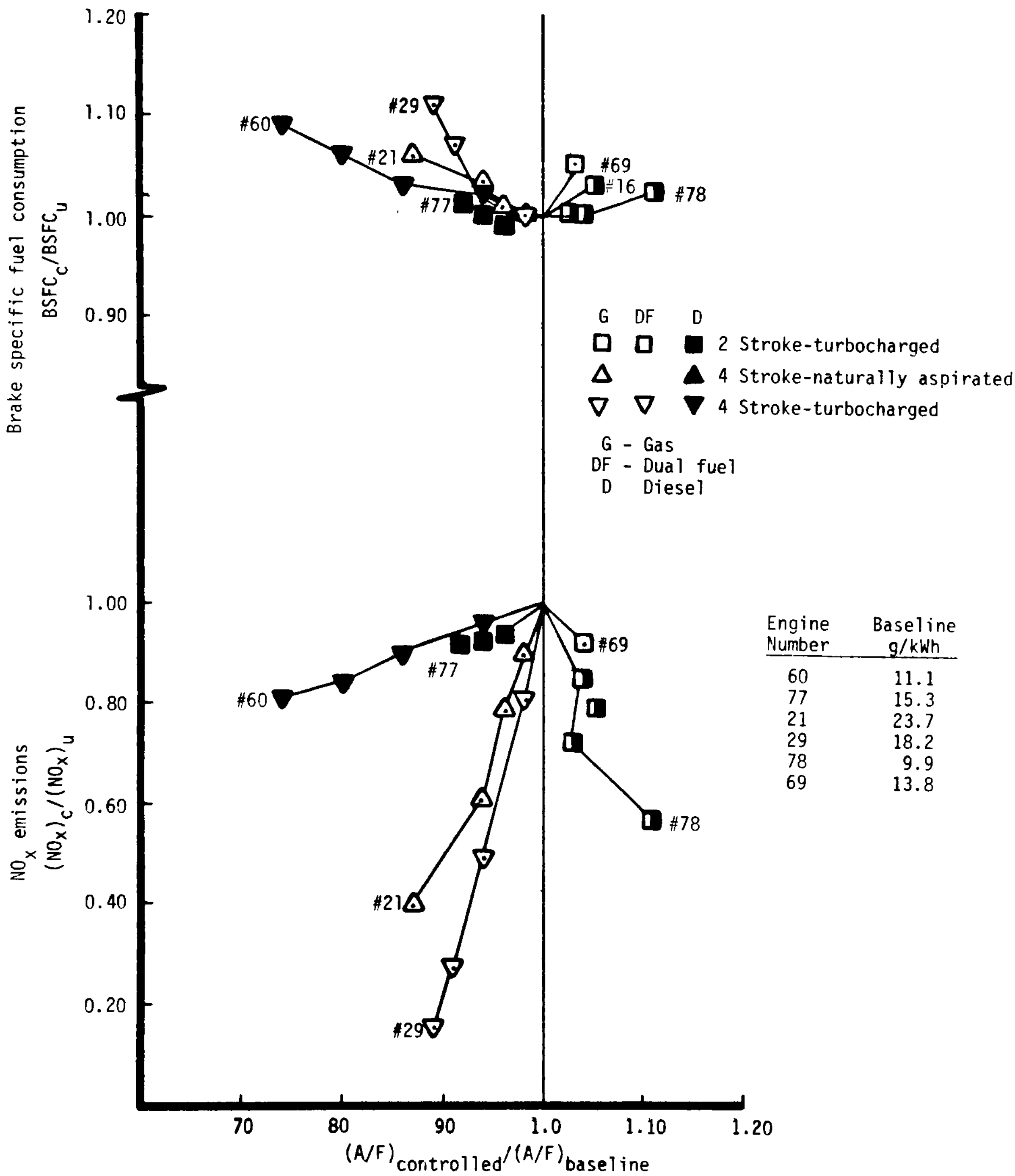


Figure 5-1. Effect of air-to-fuel (A/F) changes on NO_x emissions and fuel consumption (Reference 5-1).

increase due to this lower combustion temperature. The higher exhaust temperature could cause the exhaust valves and manifolds to require more maintenance, or possibly replacement with ones that withstand the higher temperatures. This higher exhaust temperature could also affect the turbocharger.

Figure 5-2 shows the effect of retard on NO_x emissions and fuel consumption for diesel engines; Figure 5-3 presents the same information for dual fuel and gas engines. On the average, diesel engines reduce NO_x by 25 percent for 4^o of retard and 40 percent for 8^o of retard. Fuel usage increases approximately 2 percent at 4^o of retard, while 8^o of retard raises fuel usage by about 6 percent. Gas and dual fuel engines show similar trends except the data are more scattered. Based on the limited data available, retard appears to be a more effective technique to reduce NO_x for dual fuel engines than gas engines.

5.1.3 Manifold Air Cooling and Turbocharging

Depending on how the engine is equipped, reducing manifold air temperature requires some or all of the following devices:

- Aftercooler or intercooler
- Coolant circulation device (fan for air or pump for water)
- Cooling tower or larger radiator if water cooled
- Temperature control mechanism.

Most turbocharged engines greater than 375 kW have an intercooler or an aftercooler which are heat exchangers between the turbocharger and intake manifold. Since compressing (turbocharging) air heats the air, cooling the air and thus increasing its density before it enters the cylinder allows a greater air mass flow rate. This in turn permits a higher fuel flowrate and thus more engine power output. Aftercooling of the air to the cylinders also reduces the peak combustion temperature. Aftercooling is required on a turbocharged, carbureted natural gas engine to prevent the hot gases from detonating the air fuel mixture before its entry into the cylinder (Reference 5-4).

Adding an aftercooler to a turbocharged engine or increasing the amount of aftercooling (lowering air temperature to the intake manifold), decreases the peak combustion temperature and lowers NO_x emissions.

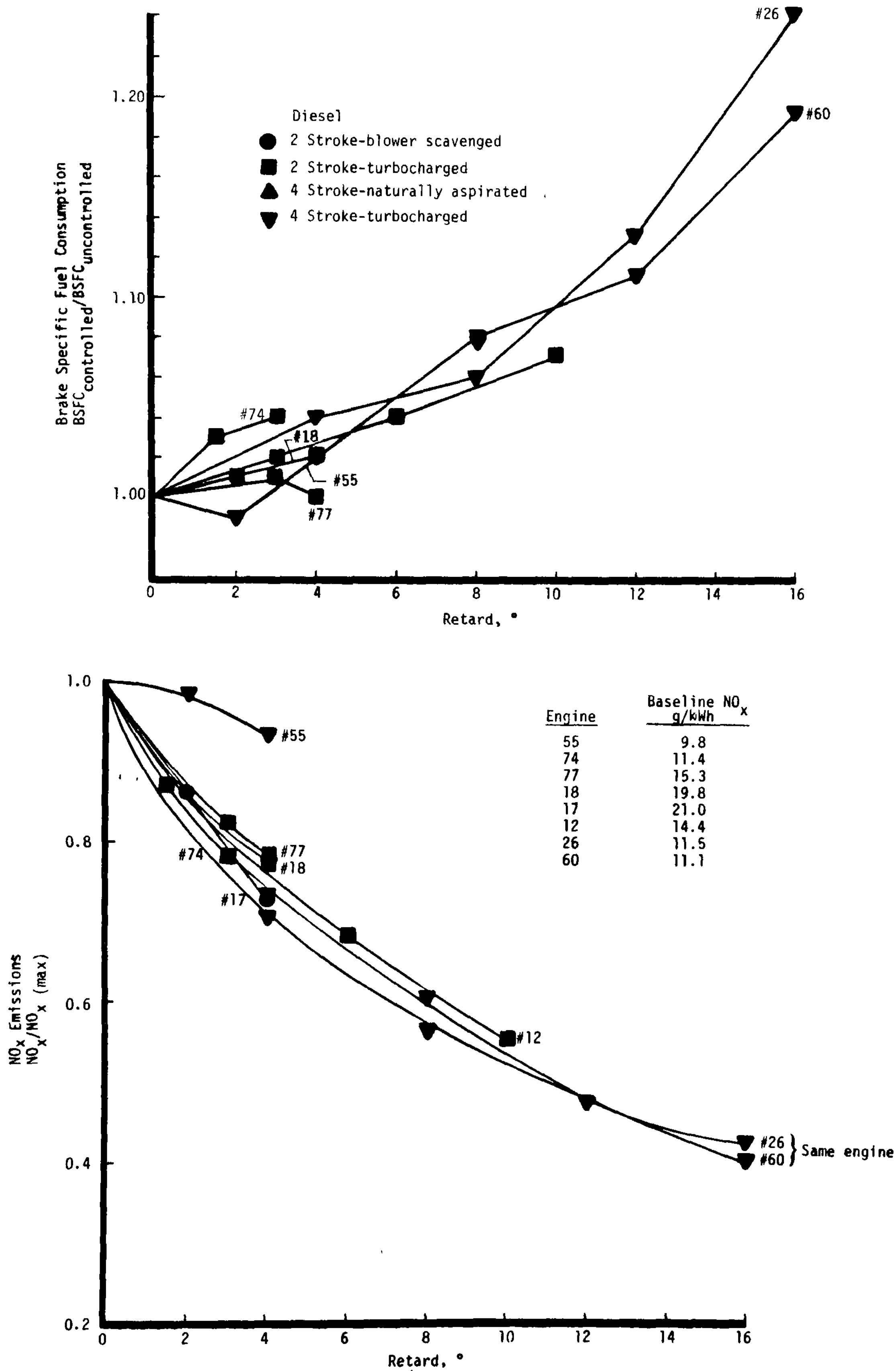


Figure 5-2. Effect of ignition retard on NO_x emissions and fuel consumption for diesel engines (Reference 5-1).

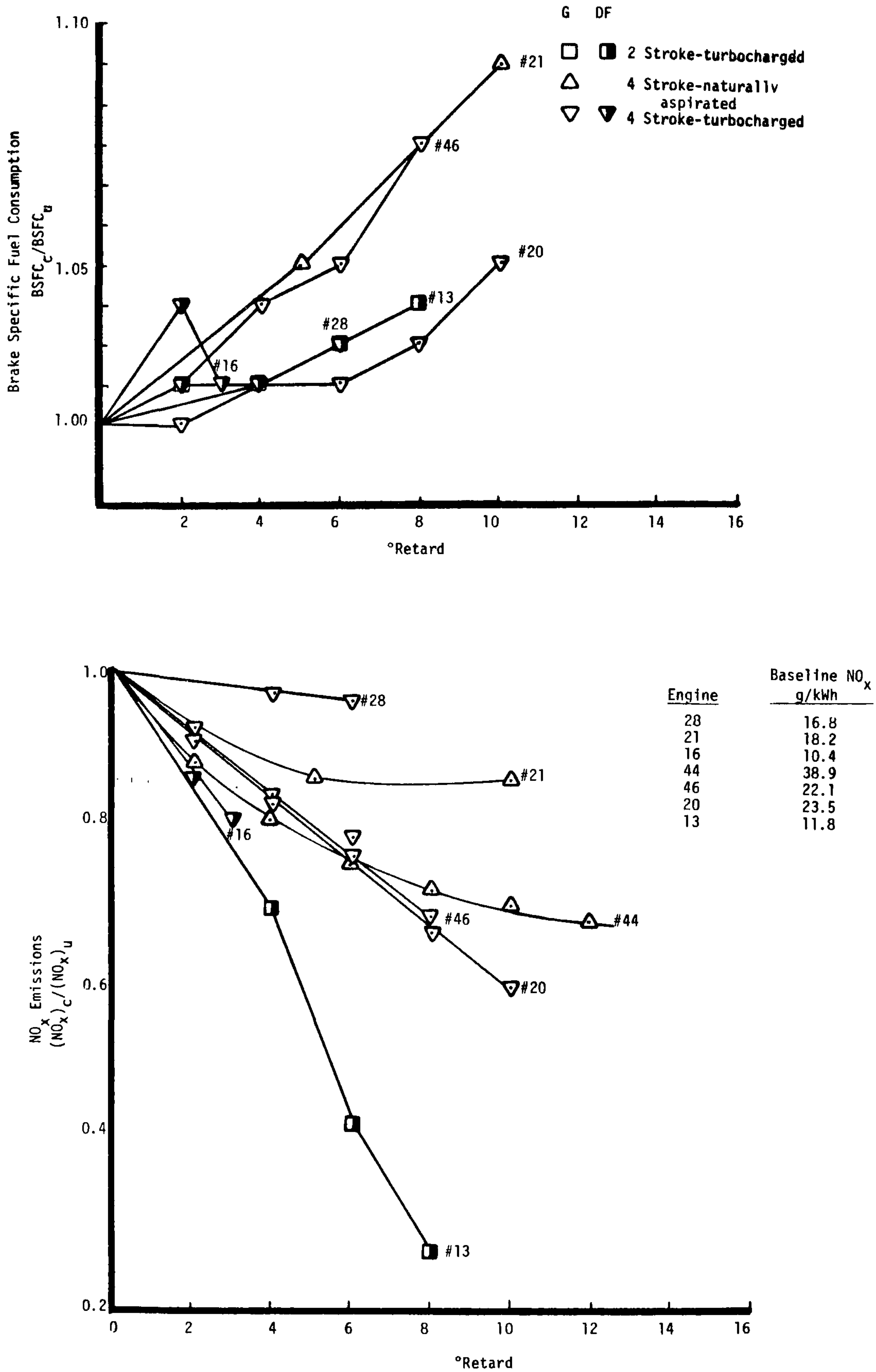


Figure 5-3. Effect of ignition retard on NO_x emissions and fuel consumption for gas and dual fuel engines (Reference 5-1).

However, this reduced peak combustion temperature can also increase CO, HC, and possible smoke emissions. Lowering manifold temperature in some of the diesels tested increased HC emissions but did not affect CO.

If the engine is equipped with an aftercooler, it may need to be enlarged and/or require a larger coolant circulation system. An enlarged coolant system could increase maintenance requirements. There is a limit to how much the manifold air temperature can be reduced, especially on hot days. Radiator cooling is limited by the ambient air temperature while cooling tower effectiveness is restricted by the ambient air dew point. Cooling tower systems use more water than radiators, which limits their use in dry areas. In all cases, a refrigeration system could be installed but might result in a large energy penalty.

Typical lower temperature limits for a hot, humid location with an ambient air temperature of 310 K ($100^{o}F$) are 320 K ($115^{o}F$) for an air cooled radiator, 310 K ($100^{o}F$) for a cooling tower, and lower if a refrigeration system is used. A typical gas exit design temperature from an aftercooler is 330 K ($130^{o}F$) (Reference 5-1).

The data presented in Figure 5-4 show that NO_x can be reduced 10 to 40 percent when manifold temperature is lowered from 330 K ($130^{o}F$) to 310 K ($100^{o}F$). This technique appears to be more effective on natural gas engines. Also note that the effect on fuel consumption is slight when compared to the other techniques.

Turbocharging an engine without air cooling sometimes reduces NO_x emissions. As the inlet air temperature rises, the peak cylinder temperature will be correspondingly higher; hence NO_x formation will generally increase. However, depending on the location of the air-to-fuel ratio of the nonturbocharged unit relative to the NO_x peak (as shown in Figure 5-1), the increase in power from turbocharging has the net result of a brake specific NO_x emissions reduction.

Adding a turbocharger requires installing the turbocompressor discussed earlier and, depending on the strength of the originally installed parts, may also require the replacement of a number of other engine components (piston rings, connecting rods, wrist pins and cylinder heads) with higher strength parts. This may be necessary because turbocharged engines operate with higher cylinder temperatures and pressures and thus experience higher thermal and structural loads.

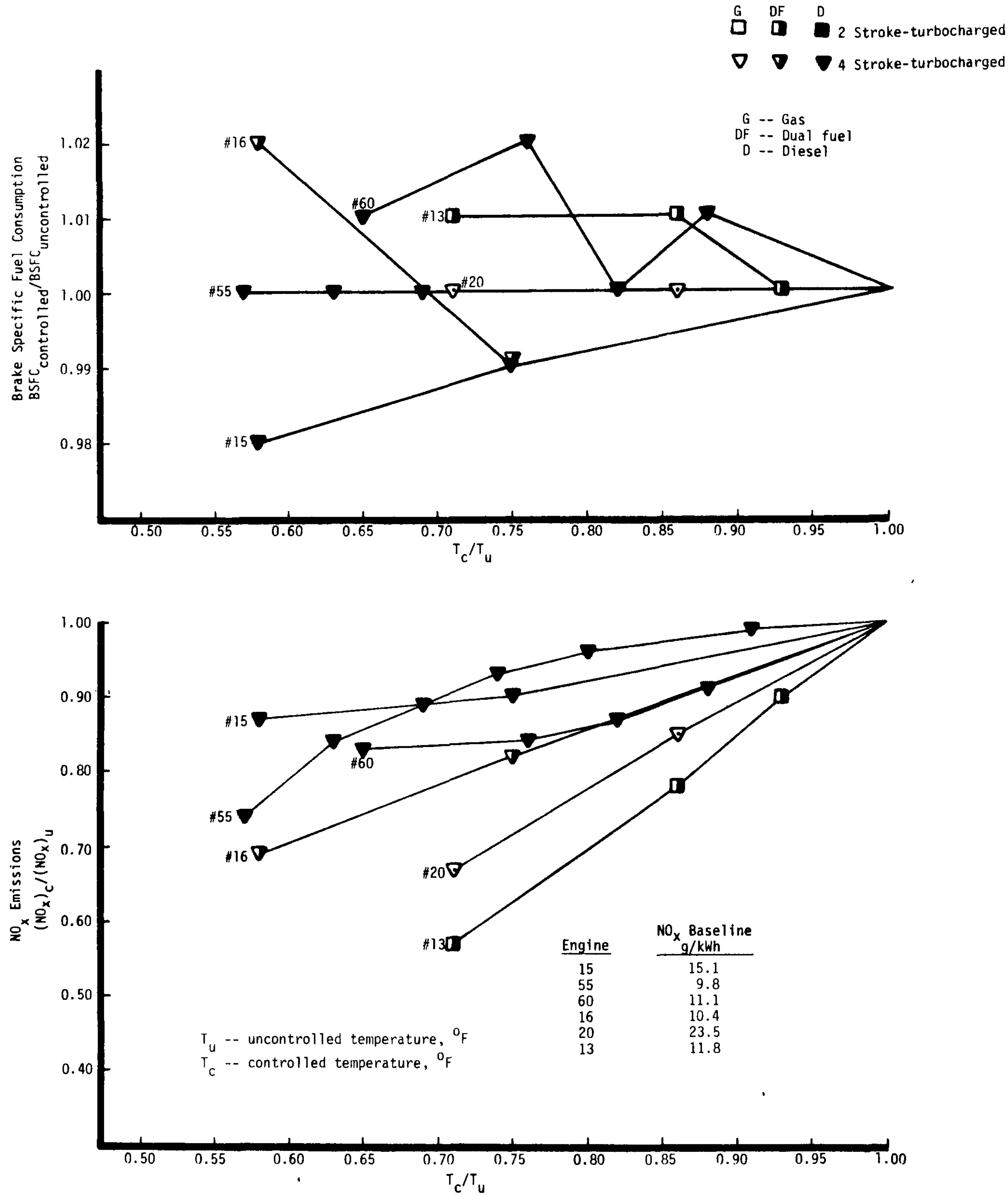


Figure 5-4. Effect of manifold air temperature reduction on NO_x emissions and fuel consumption (Reference 5-1).

An additional operational problem may occur if the turbocharged engine must respond to rapidly varying loads and speeds. The problem is increased smoke generation due to rich combustion during acceleration, since the fuel flowrate increases much more rapidly than the air flowrate. However, most large engines are not subject to rapidly varying loads and speeds so this problem generally does not arise and the problem can be designed away.

5.1.4 Exhaust Gas Recirculation

Exhaust gas recirculation (EGR) reduces peak combustion temperatures by increasing the mass of gas available to absorb heat. As compared to increasing the air-to-fuel ratio, this increased mass is achieved without raising the amount of excess oxygen. EGR can be accomplished by either restricting the exit of gases from the cylinder (internal EGR) or by reintroducing the exhaust gases into the intake manifold (external EGR). When external EGR is used the recirculated gases can be cooled to further reduce peak temperature.

Since CO and HC oxidation depend upon the availability of excess air and elevated temperatures, it might be expected that reducing both oxygen and temperature by EGR would increase emissions of these two pollutants. However, EGR traps or recirculates some of the unburned hydrocarbons in the exhaust gas and thus HC emissions frequently decrease when using EGR. Smoke levels increase with EGR, though, due to the reduction in excess air.

The primary durability consideration for external EGR systems, especially when applied to diesel engines, is the accumulation of solid exhaust products in the recirculating system. When EGR is applied to naturally aspirated engines, these deposits build up in the ducts, on any valves used to control the recirculation rate, and possibly on the intake valves. When external EGR is used in conjunction with a turbocharged and intercooled engine, further problems arise. Since the inlet charge, after the compressor, is at a higher pressure than the exhaust, a separate compressor and intercooler for the recirculating stream have to be provided or the recirculated gases have to be mixed with the incoming air before they pass through the turbocharger. Both approaches have similar problems, namely fouling of the compressor blades and the heat exchanger surface. If the compressor is designed to operate close to its optimum condition, its performance is very sensitive to the shape of the blades, which would be slightly changed by deposit build-up. Similarly, the effectiveness of heat exchangers is greatly reduced by any coating on exchange surfaces. In fact, moderate deposits can make the heat exchanger virtually useless. Such deposition problems do not necessarily preclude the use of EGR on these types of engines, but they would require significantly increased maintenance by the user.

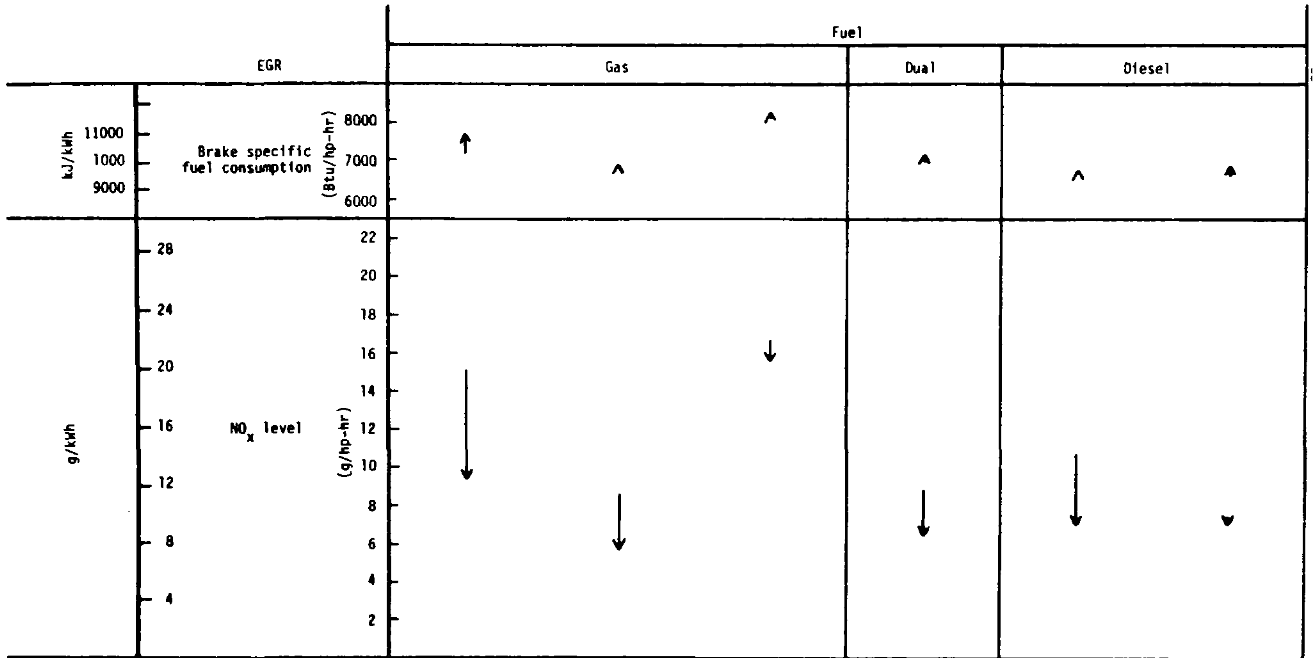
Figure 5-5 shows the effects of internal EGR on a naturally aspirated gas engine, a blower scavenged gas engine and a turbocharged diesel engine. For these three engines, internal EGR reduced NO_x emissions from 4 to 37 percent.

External EGR results from three tests on gas, dual fuel, and diesel turbocharged models are also shown on Figure 5-5 with reductions varying from 25 to 34 percent. These reductions were obtained with exhaust gas recirculation rates of 6.5 to 12 percent. The effect of varying EGR on NO_x emissions and fuel consumption is shown in Figure 5-6. At 6 percent EGR, NO_x reductions ranged from 10 to 22 percent. In general, fuel consumption remained unchanged for EGR rates less than 12 percent.

5.1.5 Water Injection

Water can be added to the fuel-air charge to lower the peak combustion temperature via both the heat used in vaporizing the water and the sensible heat absorbed by the water vapor. It is similar in principle to EGR except that water absorbs heat by vaporization, also. H_2O injection reduces NO_x emissions but increases HC emissions because of the lower peak temperature. In the engines tested, CO appeared to be unaffected.

The large bore engine maufacturers who tested water injection reported serious concerns about the adverse effects on engine durability. Their concern is based on observations of water in the crankcase (contaminated lubricating oil) and rapid buildup of mineral scale around the valves, water injection nozzles, and other components such as turbochargers (Reference 5-5). Because of these adverse effects, water injection will probably never be considered a viable technique. Demineralized water should decrease the mineral buildup problems but water in the crankcase would still be a problem.



	Gas			Dual	Diesel	
NO$_x$ reduction (%)	36.9	32.6	5.4	25.0	33.6	4.1
External/internal	I	E	I	E	E	I
EGR rate (%)	--	6.5	--	10.7	12.0	--
Uncontrolled fuel consumption (kJ/kWh)	10,091	9691	11,511	9967	9456	9439
Percent increase	7.7	0.03	1.1	0.8	0.5	2.2
Air charging	BS	TC	NA	TC	TC	TC
Stroke/cycle	2	2	4	2	2	4
Engine number	1	70	21	13	12	27

Figure 5-5. Effect of exhaust gas recirculation on NO$_X$ emissions and fuel consumption (Reference 5-1).

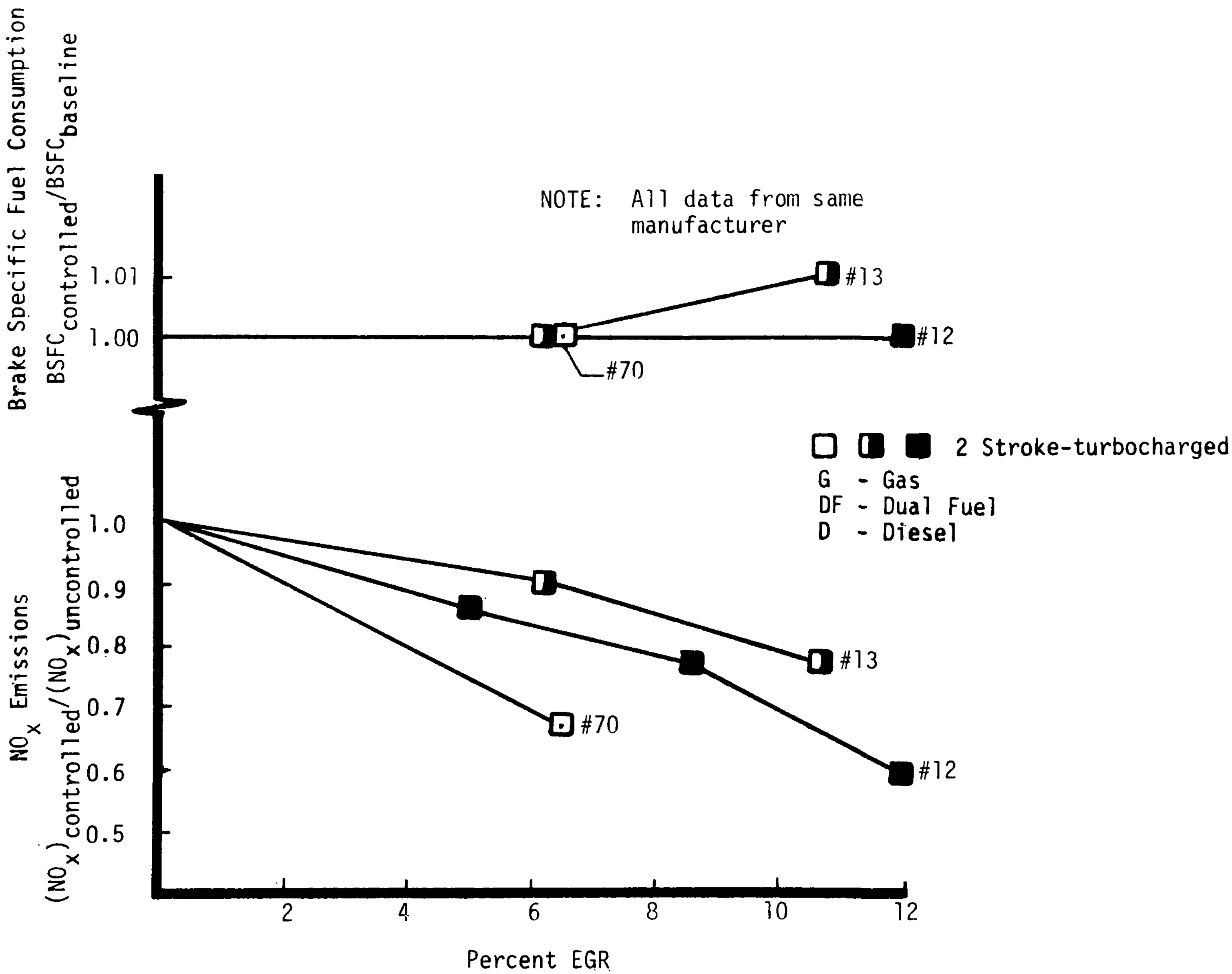


Figure 5-6. Effect of varying exhaust gas recirculation (EGR) on NO_x and fuel consumption for gas.

5.1.6 Derating

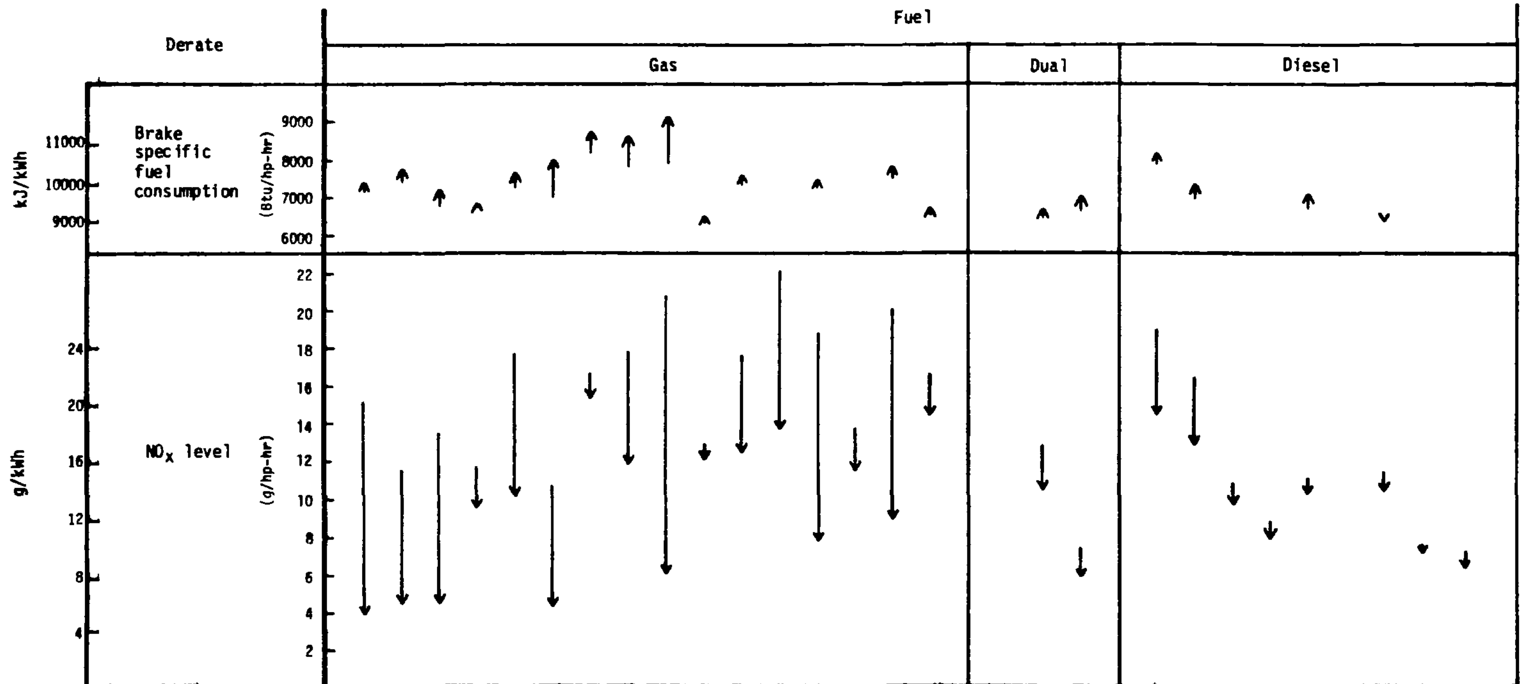
An engine can be derated by restricting its operation to lower levels of power production than normal for the given application. Derating reduces cylinder pressures and temperatures and thus lowers NO_x formation rates. Although NO_x exhaust concentrations (i.e., moles of NO_x per mole of exhaust) are reduced, it is quite possible for this reduction to be no greater than the power decrease. In such a case, brake specific emissions (i.e., grams of NO_x per kWh) are not reduced. This is especially true for four-stroke turbocharged engines. In addition, air-to-fuel ratios change less with derating for turbocharged engines than for naturally aspirated or blower scavenged units. Thus NO_x emissions are less responsive to derating for turbocharged engines. Derating also reduces the engine's operating temperature, which can result in higher CO and HC emissions.

Demonstrated NO_x emission reduction levels due to derating are shown in Figure 5-7 for a number of different engine types and fuels. These data show emission reductions ranging from 1.6 to 30.8 g/kWh for naturally aspirated or blower scavenged engines and from 0.3 to 14.0 g/kWh for turbocharged units. Since these results were obtained with varying amounts of derating, it is more informative to compare the effectiveness of this emission control technique on a normalized basis -- i.e., percent NO_x reduction per percent derate. On this basis, results for naturally aspirated or blower scavenged engines varied from 0.25 to 6.2, whereas those for turbocharged units varied from 0.01 to 2.6. No relationship was found between normalized effectiveness and uncontrolled emission level, number of strokes per cycle, or fuel. Fuel consumption increased by 0 to 15 percent. Some diesel engines showed increased NO_x emissions as load was decreased.

One big disadvantage of derating is that spare engine capacity may be needed which could require a large capital investment. For new engines, derate can be applied by designing the engine to operate under derated conditions. This could mean a larger, more expensive engine to do the same job.

5.1.7 Combining Control Techniques

Experiments on combining more than one emission control technique have been conducted by several engine manufacturers. Generally, the NO_x



	Gas																Dual		Diesel						
Percent derate	12	20	25	13	20	25	25	25	25	14	25	25	25	25	25	25	18	25	25	5	25	25	38	27	25
Normalized reduction [a]	6.2	3.0	2.6	1.4	2.1	2.3	0.3	1.3	0.4	0.3	1.1	1.5	2.3	0.6	2.2	0.5	0.9	0.8	0.9	4.2	0.3	0.5	0.2	0.3	0.2
	7127	7400	6775	6760	7265	6942	8130	7760	7840	6374	7300	7440	7220	7590	7500	6500	6400	6570	7844	6951	7175	6930	6677	6366	6667
Uncontrolled fuel consumption $\frac{Btu}{hp\text{-}hr}$	2.6	4.1	6.1	0.3	4.3	14.4	6.4	9.6	15.3	0.8	3.0	1.7	2.8	--	3.5	2.0	3.1	6.1	2.6	4.1	--	-5.5	4.8	-0.7	--
	BS	BS	TC	TC	TC	TC	NA	NA	NA	TC	TC	TC	TC	TC	TC	TC	TC	TC	BS	BS	TC	NA	TC	TC	TC
Percent increase	2	2	2	2	2	2	4	4	4	4	4	4	4	4	4	4	4	4	2	2	2	4	4	4	4
Air charging	1	71	2	64	65	69	21	22	44	4	20	23*	25	29	45	46	7	61	10	11	12	24*	5	15	27

[a]Percent NO_x reduction/percent derate
[b]Speed reduction, const. torque

Figure 5-7. Effect of derate on NO_x emissions and fuel consumption (References 5-1 and 5-20).

reductions are not additive, although diesel engines exhibit more additive behavior than other engine types. Results of a set of tests are shown in Figure 5-8.

For the large bore diesel engine shown in Figure 5-8, the maximum NO_x reduction for a single control (retard) is 2.3 g/kWh. When all the controls tested (retard, reduced inlet manifold air temperature, increased air-to-fuel ratio, and water injection) were applied simultaneously, NO_x was reduced 4.0 g/kWh. This is shown on the figure as an uninterrupted downward arrow. For comparison, to the left of this, is a multiple arrow line that represents the depiction in series of all the separate control effects, as they were individually measured. The difference between the length of these two lines is a measure of the relationship between the additive effects of the controls when applied simultaneously and the sum of their individual contributions. The figure shows the affects of the controls tested on this engine were essentially additive.

Figure 5-9 shows that the combination of retard and manifold air temperature reduction is nearly additive in dual fuel engines. However, additional use of increased air-to-fuel ratio does not decrease NO_x at much below the levels obtained by the first two controls as would be expected from the results with air-to-fuel ratio alone. Furthermore, adding water injection to these three controls has no effect whatsoever. If moderate NO_x reduction were required (e.g., 25 to 30 percent) for this engine, the air-to-fuel ratio would be changed; however, if a greater reduction were necessary, the combined controls of retard, manifold air temperature decrease, and air-to-fuel ratio modification would be used.

Gas engines also do not respond additively to the simultaneous application of several controls (Figure 5-10). Here, reducing manifold air temperature decreased emissions from the blower scavenged engine by 36 percent, whereas applying reduced temperature plus retard lowered emissions only 29 percent. Only the combination of the above two techniques with increased air-to-fuel ratio could reduce emissions below the level obtained by reduced manifold air temperature alone.

5.1.8 Combustion Chamber Modifications

Combustion chamber redesign is the control technique with the greatest potential for reducing NO_x emissions from large bore engines with little or no loss in efficiency. It is probably also the technique

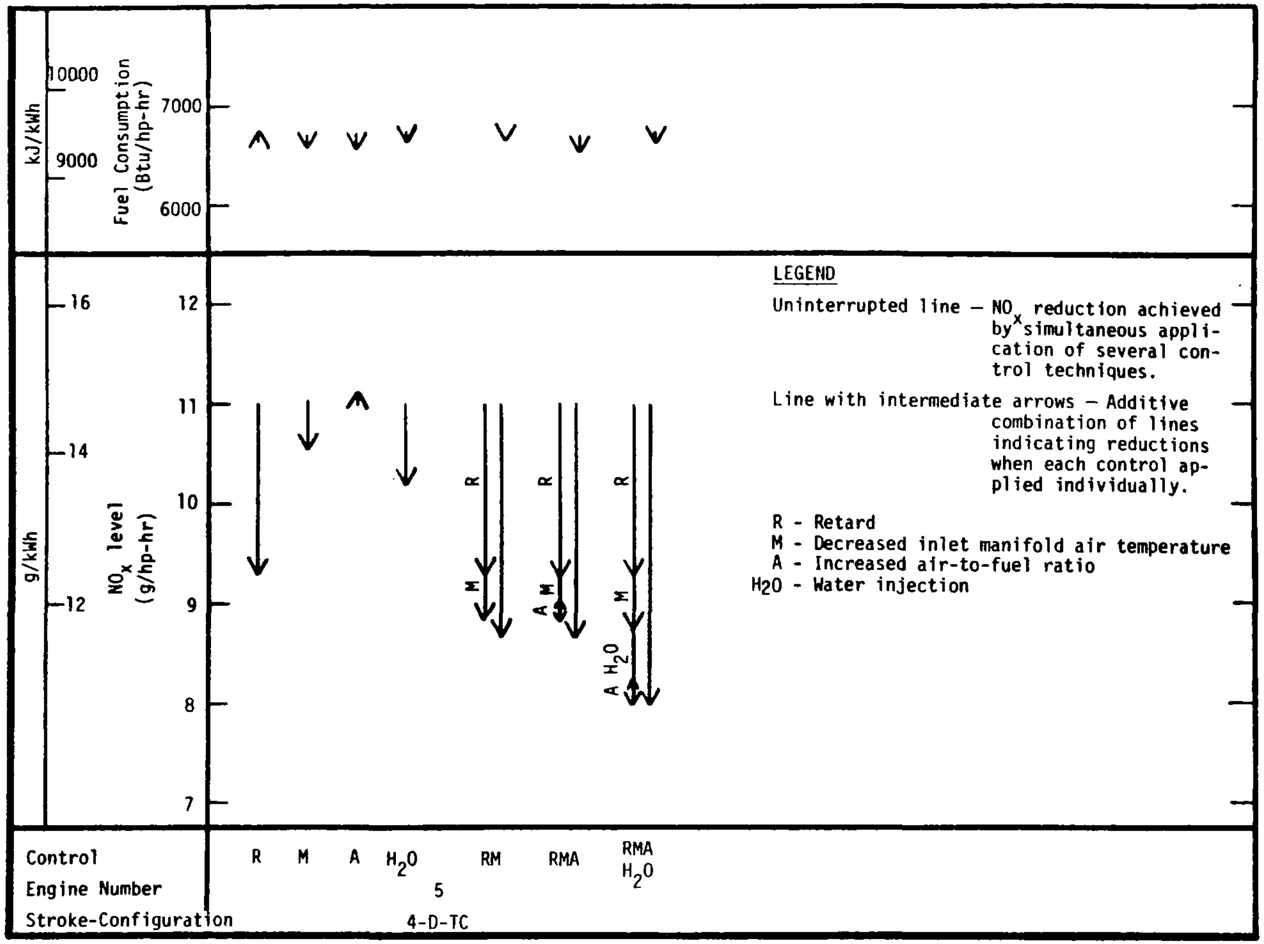


Figure 5-8. Additive effects of controls for a large bore diesel engine (Reference 5-1).

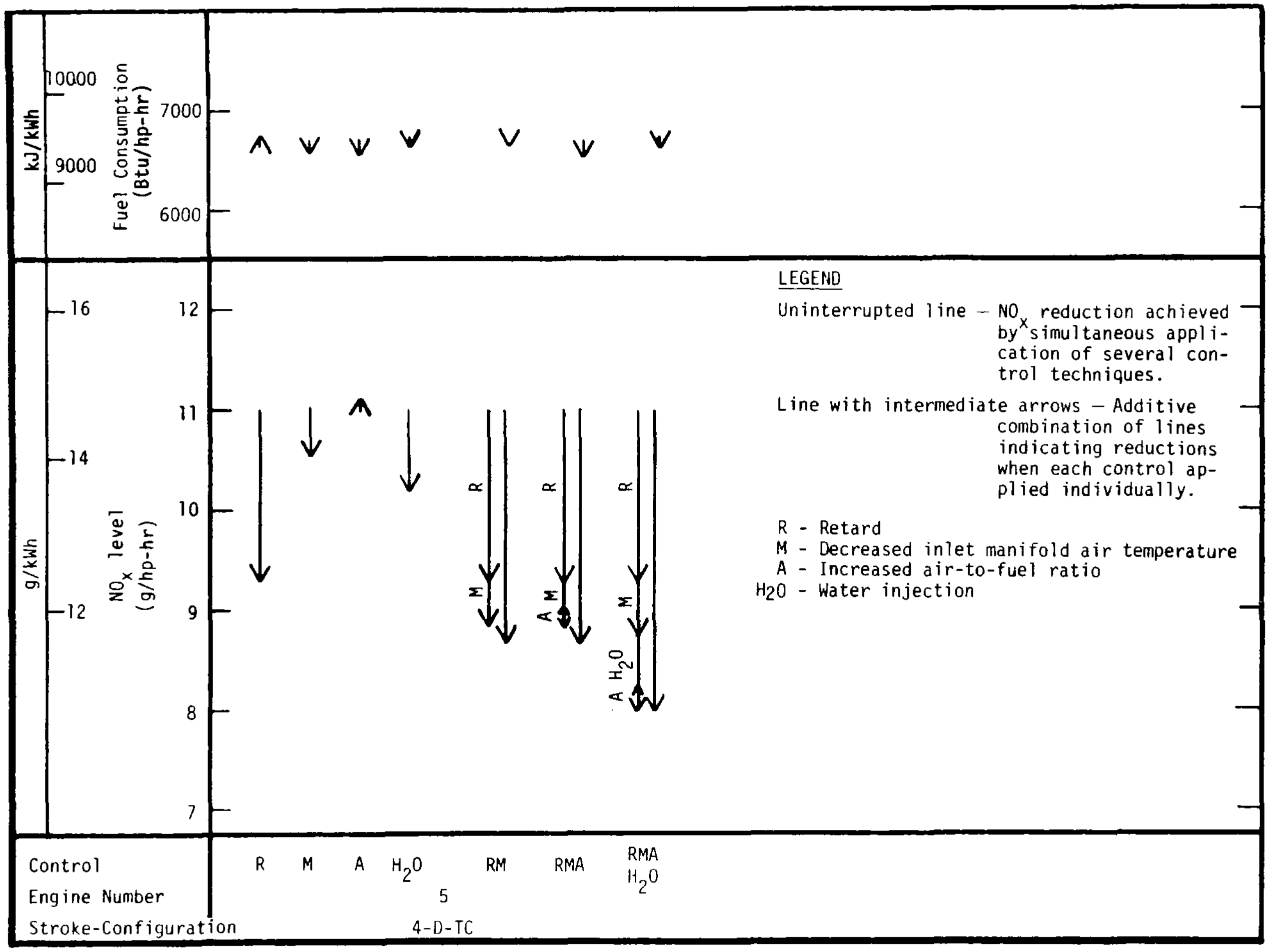


Figure 5-9. Additive effects of controls for a large bore dual fuel engine (Reference 5-1).

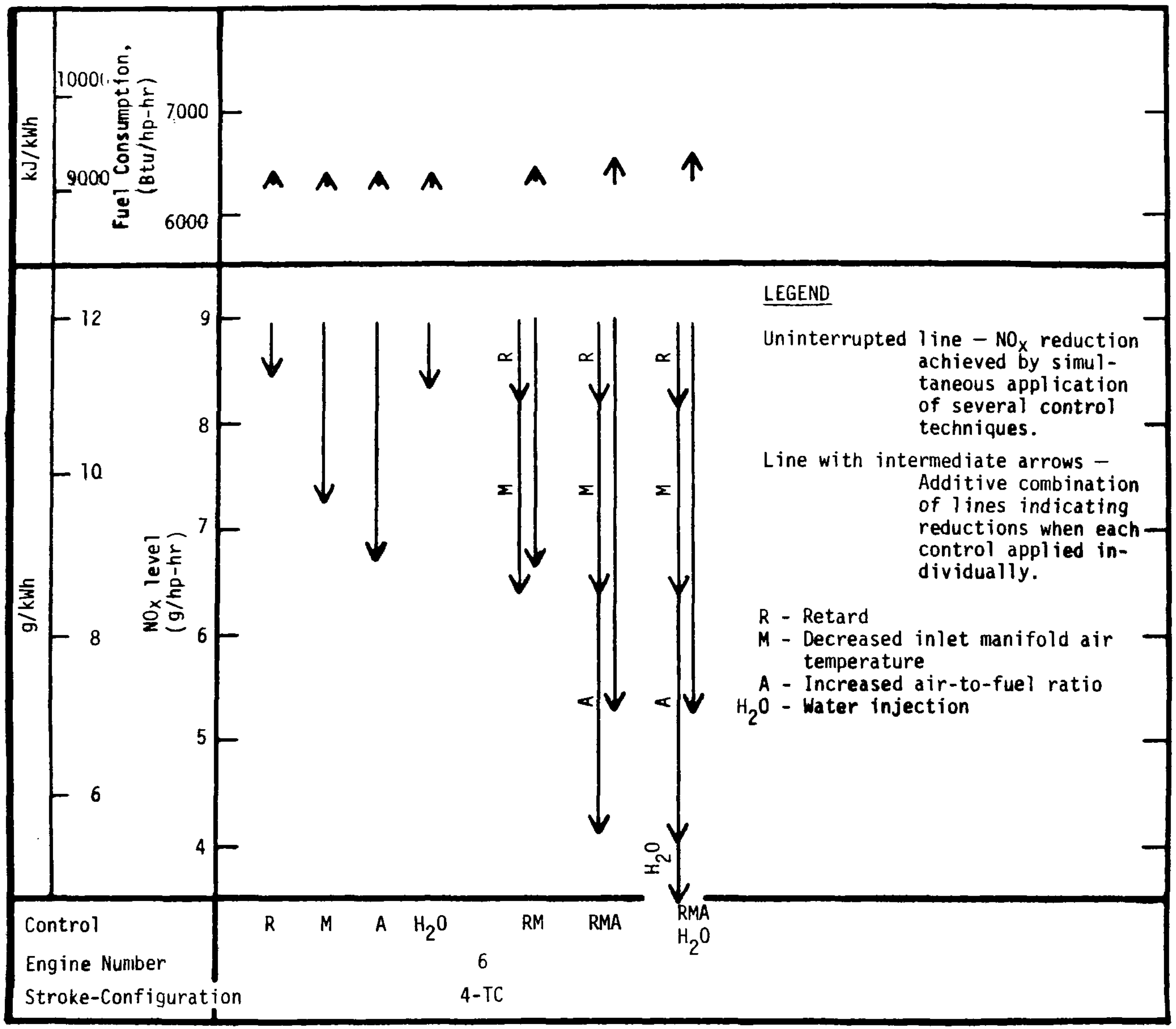


Figure 5-10. Additive effects of controls for large bore gas engines (Reference 5-1).

requiring the greatest amount of research and development, and could result in engine designs changing significantly. Thus, several years will be required to implement the changes and obtain reliable performance and durability data.

For some engines, the combustion process can be improved by redesigning chamber geometries to increase turbulence, which is conducive to good air-to-fuel mixing and, hence, efficient combustion. This increase turbulence may ensure that most of the combustion takes place under lean conditions rather than some combustion in regions where the air-to-fuel mixture is stoichiometric. Combustion in lean regions usually produces less NO_x.

Staged combustion, in which the fuel is first burned rich in a small chamber separate from the cylinder, then lean within the cylinder, is another way of reducing emissions through combustion chamber modifications. Rich combustion avoids excess oxygen at the time high temperatures are needed for ignition, then completes combustion at a temperature high enough for combustion but sufficiently low to limit NO_x formation.

Arthur D. Little, Inc. (Reference 5-7) is currently under EPA contract to evaluate combustion chamber modifications and other emission control concepts for stationary engines. They have identified potential chamber modifications which either improve mixing, enhance combustion, or represent some form of staged combustion. For diesel engines, mixing can be improved by circumferential injection, chamber shape, or a variable area prechamber. Improved combustion in gas engines can be achieved through torch ignition, multiple spark plugs, high energy spark, increased turbulence through swirl or "squish," or diesel fuel injection. Diesel fuel injection essentially converts the gas engine to a dual fuel engine. Existing dual fuel engines tend to give less NO_x than existing gas engines. The average dual fuel engine produces 30 percent less NO_x than the average natural gas engine. Swirl or "squish", besides increasing turbulence, is also a form of internal exhaust gas recirculation.

Staged combustion techniques include divided chambers, open chambers, or degraded mixing for gas engines, and a prechamber or pilot injection for diesel engines. Each technique is described below (from Reference 5-7), first for natural gas engines, then diesel engines.

5.1.8.1 Torch Ignition (Gas)

Torch ignition ignites a very lean mixture within the cylinder from a small (2 to 5 percent of the total volume) antechamber. Ignition takes placed throughout the torch boundary, rather than at a point source as with a spark (see Figure 5-11). Leaner mixtures can be used, the burn time is shortened, and turbulence is increased, all resulting in efficient combustion at reduced temperatures. This may reduce NO_x emissions 30 to 40 percent. Potential disadvantages are that hydrocarbon emissions may increase as a result of quenching, and fuel usage is projected to increase no more than 3 percent.

5.1.8.2 Multiple Spark Plugs (Gas)

This concept achieves shorter combustion times and allows leaner combustion by igniting the mixture at several points within the cylinder. Although less costly than torch ignition, a high energy spark should be used to allow reliable ignition. Emission reductions should be similar to torch ignition, with a slightly higher (4 percent) fuel penalty.

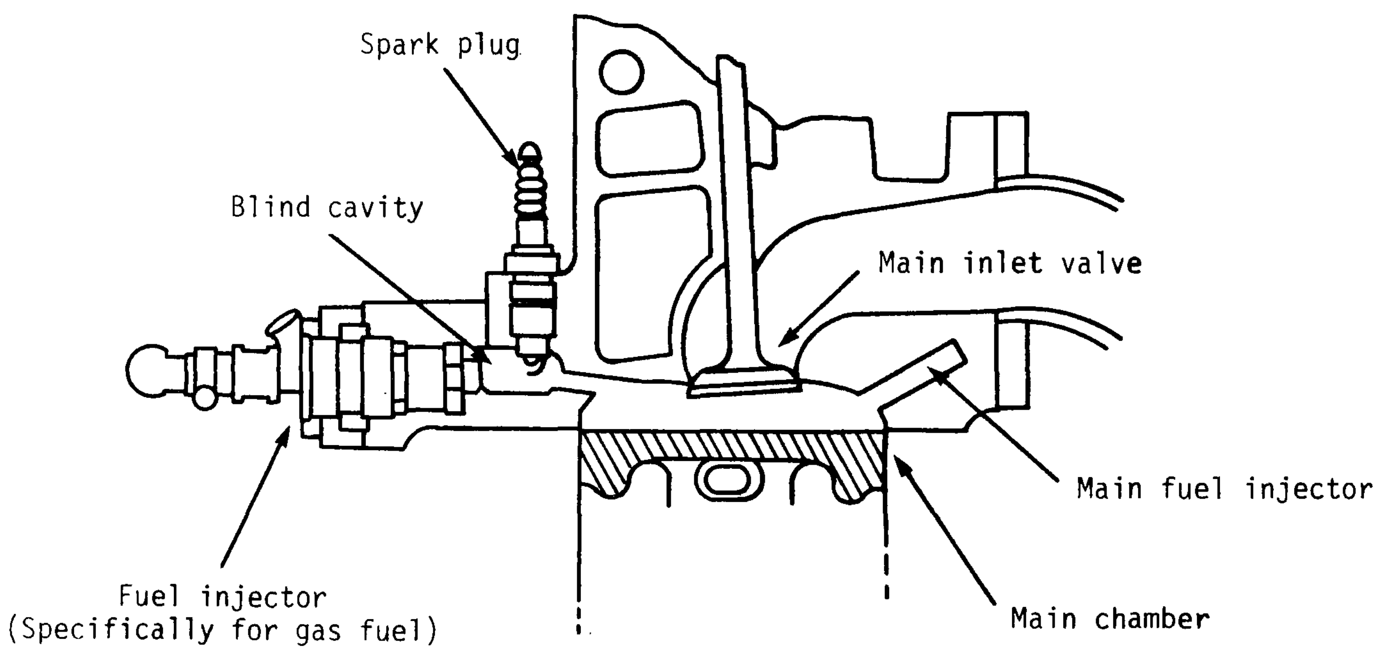


Figure 5-11. Fuel injection torch ignition concept (Reference 5-21).

5.1.8.3 High Energy Spark (Gas)

This concept is similar to torch ignition, except a plasma jet acts as the torch. The high energy spark is needed to ensure that a leaner mixture can be ignited. The concept appears to be the lowest cost method of achieving leaner combustion, but may have the greatest maintenance requirements. Emission reduction performance should be similar to torch ignition.

5.1.8.4 Increased Turbulence (Gas)

Increased turbulence also decreases the combustion duration, and allows combustion of leaner mixtures. If sufficiently high, no fuel penalty should be experienced. Turbulence can be increased by modifying the intake passages and by placing shrouds or fins around valves to swirl the mixture as it enters the cylinder. In addition, placing a cavity on top of the piston produces "squish" (a circumferential swirl). The disadvantages of increasing turbulence are the quench layer may be increased since more surface/volume will be added with a piston cavity; and increased heat transfer to the cylinder walls, requiring a larger heat rejection system. A 20 percent reduction in NO_x emissions is estimated.

5.1.8.5 Staged Combustion (Gas)

Staged combustion concepts all divide the mixture into a rich zone and lean zone, both of which avoid air-to-fuel ratios that produce high NO_x formation rates. The fuel rich zone is ignited first (it is much easier to ignite). NO_x formation is reduced both because of lower temperatures and lack of oxygen. Combustion continues as the rich zone ignites the lean zone, and NO_x formation is again reduced because of low flame temperatures.

There are two basic approaches to staged combustion: divided chamber, in which a smaller (20 percent to 30 percent of total volume), separate chamber is used for the rich mixture; and open chamber, which maintains a rich zone within the cylinders. Some type of fuel injection is required to ensure that there are rich zones. Emissions reductions for a large bore, divided chamber engine are estimated at 20 percent. For the open chamber engine, a 10 percent reduction in NO_x emissions is estimated. However, reductions up to 30 percent are expected for engines operating close to stoichiometric.

5.1.8.6 Prechamber (Diesel)

A prechamber allows staged combustion for diesel engines. Fuel is injected into a small chamber where it starts burning under rich conditions (5 percent to 40 percent of the total volume) and then expands into the main chamber where combustion is completed. (This differs from gas engines which have combustible mixtures in both chambers at the time of ignition.) NO_x emissions have been reduced up to 30 percent on small bore diesel engines. Large bore engines should achieve NO_x emission reductions of 20 percent with a 1 percent fuel savings based on computer models. However, potential problems include cavitation in the cylinder liner, shorter piston life, and some soot buildup.

5.1.8.7 Pilot Injection (Diesel)

Pilot injection places a small amount of the total fuel charge, 5 percent to 15 percent, into the cylinder before the main injection and ignition take place (see Figure 5-12). A premixed combustion mixture is formed from this pilot injection, and once ignition takes place ignition delay is shorter. This effectively allows ignition retardation and thereby lowers combustion temperatures, reducing the rate of NO_x formation. Timing of the pilot injection is critical; premature injection may produce HC emissions, while delayed injection may increase smoke emissions. Knocking is another potential problem since the pressure rise within the cylinder is greater.

NO_x emissions have been reduced 15 percent on one test, and if ignition is additionally retarded the reduction may reach 30 percent. Fuel economy should also slightly improve.

5.1.8.8 Combustion Chamber Shape (Diesel)

Improved mixing of the fuel and air increases turbulence and allows for greater ignition retard. Like gas engines, swirl, "squish", and manifold and cylinder shapes are used to increase turbulence. NO_x emission reductions of 20 percent are expected.

5.1.8.9 Circumferential Injection (Diesel)

This concept is similar to using multiple spark plugs for gas engines, in that ignition takes place simultaneously at several points within the cylinder. Fuel is injected tangentially which improves mixing, and the multiple point ignition shortens combustion time. Thus retard may again be used without a loss of fuel efficiency. The major disadvantage

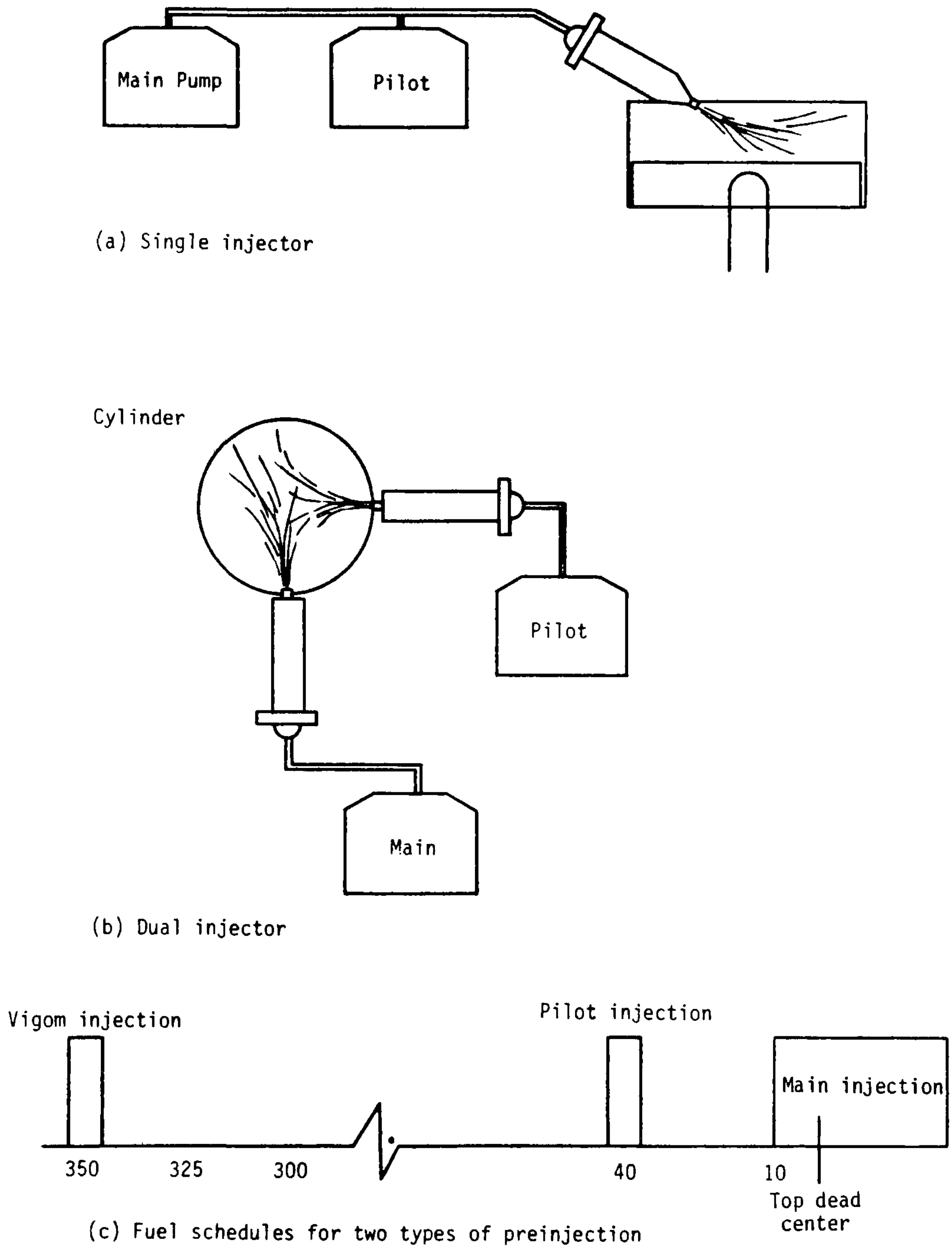


Figure 5-12. Alternative pilot injection schemes (Reference 5-7).

of this concept is the mechanical complexity of the ignition system. NO_x reductions are estimated to be 20 percent.

5.1.8.10 Summary of Combustion Chamber Modifications

Modifications which increase turbulence may reduce NO_x emissions 20 percent to 40 percent, with less than a 5 percent consumption penalty if all predictions are correct. Staged combustion techniques are predicted to reduce NO_x emissions 10 percent to 30 percent, with little or no fuel penalty.

5.1.9 Catalytic Reduction

Catalytic reduction is a flue gas treatment method of reducing NO_x emissions. The catalyst is enclosed in a chamber within the engine exhaust duct and the exhaust gases are passed over the catalyst. Under controlled conditions of air-to-fuel ratio and temperature, reduction of NO_x to N_2 and O_2 takes place. The concept has been tested by the automobile industry for some time; one example is a rhodium catalyst selectively reducing NO_x. However, this concept requires that essentially no oxygen exist in the exhaust stream. Most large stationary engines, except for carbureted engines which operate at air-to-fuel ratios on the rich side of the NO_x peak, operate at lean air-to-fuel ratios with large amounts of excess oxygen. Therefore, such catalysts cannot be straightforwardly applied to these large engines.

One approach to overcome this limitation is to inject ammonia, hydrogen, CO, or natural gas into the exhaust to create the required reducing atmosphere. Ammonia appears to work best because it reacts preferentially with the NO rather than the excess oxygen. Thus only slightly more ammonia would be required for stoichiometric reaction with the NO, whereas the other three substances need additional quantities to deplete the excess oxygen in the exhaust.

The overall ammonia reaction is as follows:

$$2\ NO + 2\ NH_3 + 1/2\ O_2 \longrightarrow 2\ N_2 + 3\ H_2O$$

This reaction proceeds with a sufficiently high rate at temperatures above approximately 1090 K (1500^{o}F), but since exhaust gas temperatures are below this value, a catalyst is required. With the catalyst the reaction can be promoted in the temperature range 590 to 700 K (600^o to 800^oF). The actual temperature window is catalyst dependent.

Published emissions data from such catalyst systems are not yet available. However Engelhard, the only current manufacturer of a catalyst for this system, has reported NO_x emission reductions of 70 percent to 80 percent at an industrial boiler installation in Japan (Reference 5-8). This unit had flue gas oxygen levels similar to exhaust gas levels from stationary engines, but operated at higher pressures (500 to 600 kPa). The reaction is usually complete with the reduction in NO_x directly dependent on the amount of ammonia injected.

Several potential problems must still be solved before such systems are ready for commercialization. First, the catalyst is predicted to have a limited lifetime (1 to 3 years), leading to high operation and maintenance costs. A reliable ammonia storage and injection system must still be developed. Materials and design of the exhaust system must also be improved, because the reaction of NH_3 and O_2 is exothermic, and heat may need to be removed from the system. Finally, toxic products such as hydrogen cyanide and raw ammonia can result, and the allowable range of parameters such as temperature, NH_3/NO_x ratio, and catalyst surface area must be fully understood (Reference 5-6). Several engine users are starting to test these catalytic systems on engines out in the field (Reference 5-22). Both lean burning and rich burning engines are to be tested, i.e., reduction catalyst with and without ammonia injection.

5.2 COST OF POLLUTION CONTROLS

Since this report concentrates on the large size stationary engines, this cost section focuses on those size categories. Medium and small size engines (which are essentially modified mobile engines) would use mobile control techniques, and these costs may be found in reports on engines of that size (e.g., References 5-18 and 5-19). Although it might be assumed that mobile engine control costs would also apply to stationary engines, this might not always be the case.

One special problem that might arise when applying mobile engine control costs to stationary engines concerns certification requirements. In many cases, stationary engine manufacturers satisfy user engine size needs required by varying the number of cylinders. If each combination of cylinders is considered a separate type of engine that must be certified, large certification costs may be incurred. This could greatly impact the pollution control device cost (Reference 5-10).

5.2.1 Cost Evaluation Procedures

The two major cost components are capital costs and the cost of "operating" the control. Capital cost is the cost to purchase and install the pollution control device. In addition, capital cost could include certification charges and factors for recovering research and development cost for the control technique. This report does not include R&D, recovery, and certification costs in the estimated capital costs. However, the installation portion of capital costs include indirect charges such as engineering and overhead in addition to charges for the labor to actually install the control device.

The annualized incremental cost for pollution control devices consists of annualized initial costs, charges resulting from maintenance and operating cost changes, and alterations in fuel and lubrication use. Annualized costs are calculated by the following formula:

$$AC = AIC + M + F$$

where

AC = annualized costs due to control device

AIC = annualized initial costs (capital cost)

M = maintenance and operating costs

F = incremental fuel and lubrication charges

In units of $/kWh, the terms are calculated by the following:

$$AIC = \frac{\text{(Initial cost)(capital cost factor)}}{\text{(Average power output)(hours/year operation)}}$$

M - will be given in terms of $/kWh (operating refers to operator time)

F = (percent change in fuel consumption)(fuel consumption, kJ/kWh) · (fuel cost, $/kJ) + Changes in lubrication use costs.

A capital cost factor of 20 percent will be used as is common in industrial practice. Fuel costs used are $2.00/GJ ($2.11 \times 10^6$ Btu) for natural gas and $3.10/GJ ($3.27 \times 10^6$ Btu) for diesel fuel (Reference 5-11). These costs are based on the lower heating value of the fuel. To show how controls affect engine costs, the capital and annualized costs are given as a percent of total engine costs.

Baseline costs are presented for four typical IC engines in Table 5-2. The annualized costs in the table do not include any charges for operator attendance. Since these engines are designed to essentially operate without any operator attention, this is not a large factor. Also, costs are for the engines alone and do not include charges for devices such as generators. The four engines described in Table 5-2 are: a large bore diesel engine used to generate electricity; a large bore dual fuel engine used to generate electricity; a large bore natural gas engine used as a pipeline compressor; and a natural gas engine used at an oil and gas production site. The average size of an engine used at the oil and gas production site is about 750 kW, while the other engines are of average size about 3000 kW. All the engines are assumed to operate for 8000 hours/year which is typical of baseload operation. The maintenance costs, initial costs, and lubrication costs are based on estimates from engine manufacturers as cited in Reference 5-1. All cost data are in 1978 dollars. Cost data that were not in 1978 dollars were updated using Nelson cost indices (Reference 5-12).

As already discussed, the three main types of combustion modification control techniques are: operational adjustments, exhaust cleaning, and combustion chamber modifications. Since operational adjustments have been studied in the greatest detail, the most cost information exists for these control techniques. Cost data also exist for the other control techniques, but since most of these techniques are in the developmental stage, only rough cost estimates can be given.

5.2.2 Estimated Costs of Operational Adjustments

Operational adjustment techniques for NO_x control include retard, derate, air-to-fuel ratio changes, manifold air temperature reduction and external exhaust gas recirculation. The capital and maintenance cost of these techniques were compiled from confidential communications with engine manufacturers (Reference 5-1). The changes in efficiency or fuel use were discussed in the preceding sections.

5.2.2.1 Retard

Retard involves only changing the ignition or injection timing, thus, in principle, no capital equipment changes are necessary. However,

TABLE 5-2. TYPICAL COSTS FOR UNCONTROLLED ENGINES (1978)

Parameters	Typical Engine			
	3000 kW Diesel (Electrical Generation)	3000 kW Dual Fuel (Electrical Generation)	Natural Gas	
			3000 kW (Gas Transport)	750 kW (Gas Production)
Initial Cost, $/kW[a]	240	240	240	80
Capital Cost Factor	0.2	0.2	0.2	0.2
Operating Hours hr/yr	8,000	8,000	8,000	8,000
Maintenance, $/kW[a]	0.005	0.005	0.005	0.005
Fuel cost, $/GJ[b]	3.10	2.00	2.00	2.00
Fuel Consumption, kJ/kWh	9,900	9,200	9,900	11,300
Lubrication, % Fuel Cost[a]	5	10	10	10
Annualized Costs ($/kWh)				
Capital	0.006	0.006	0.006	0.002
Maintenance	0.005	0.005	0.005	0.005
Fuel & Lubrication	0.032	0.020	0.022	0.025
Total	0.043	0.031	0.033	0.032

[a]Reference 5-1
[b]Reference 5-11

because retard can raise exhaust temperatures, engine manufacturers may want to replace existing exhaust valves or modify the exhaust system. If this is done, there would be a capital cost. The cost estimates assume that the valves are not replaced but maintenance increases by 33 percent. Two-cycle diesel engines do not always have valves, but large four-cycle engines do; thus maintenance costs for diesel engines are assumed to increase 33 percent, also. For all engine types ignition retard reduces engine efficiency zero to 4 percent for a 20 to 30 percent reduction in NO_x, thus a 4 percent efficiency reduction will be assumed as a worst case. Table 5-3 shows that the estimated incremental annualized cost would increase by about 6 to 7 percent. Recall that this is based on the cost of the basic engine and thus the percent increase in operating a generator or whatever would be less.

5.2.2.2 Air-to-Fuel Ratio Change

The incremental cost of air-to-fuel ratio changes depends highly on whether a turbocharger is added. Since most large engines are equipped with turbochargers, the initial cost of the engines would not change. Adding a turbocharger to a medium size diesel engine would be ~5 percent of the initial engine cost (assuming that there is space to add the turbocharger), but this would be somewhat offset by the increased efficiency (Reference 5-10). Since the turbocharger handles more air as the air-to-fuel ratio increases, the maintenance charge for cleaning the turbocharger could rise by ~0.0002 $/kWh. For a large bore diesel engine, an air-to-fuel change to reduce NO_x by ~20 percent would increase fuel consumption by about 10 percent. A dual fuel or natural gas engine air-to-fuel change to reduce NO_x 40 percent should only increase fuel consumption by ~2 percent.

Table 5-4 shows that the annualized cost of diesel engines would increase by about 7 percent, with 3 percent increases for dual fuel or natural gas engines. Remember that these costs are only for the basic engine; the cost percentage increase for the total engine/generator installation would be less.

5.2.2.3 Derate

Derate is a viable control technique for new units only if extra power is available. However, derate can be used for new installations by buying an engine larger than normally required. The initial cost of this control technique is the cost of the additional capacity purchased. Maintenance costs are almost directly proportional to the number of cylinders to maintain; thus if the derated engine has more cylinders than an engine that would normally be purchased, maintenance costs would increase (Reference 5-13). Also, since derate usually involves operating the engine at a lower rating than would provide maximum efficiency, brake specific fuel consumption would increase and raise fuel operating cost.

TABLE 5-3. ESTIMATED INCREMENTAL COST DUE TO RETARD (1978)

	Engine/Fuel Type			
			Natural Gas	
	Diesel (Electrical Generation)	Dual Fuel (Electrical Generation)	(Oil and Gas Pipeline)	(Oil and Gas Production)
Capital ($/kW)	0	0	0	0
Maintenance ($/kWh)	0.0016	0.0016	0.0016	0.0016
Fuel ($/kWh)	0.0012	0.0007	0.0008	0.0009
Total Annualized ($/kWh)[a]	0.003	0.002	0.002	0.002
Percent increase in initial cost	0	0	0	0
Percent increase in total annual past cost	7.0	6.4	6.1	6.2
Percent reduction in NO_x	20-30	20-30	20-30	20-30

[a]Assumes 8000 hours operating year.

TABLE 5-4. ESTIMATED INCREMENTAL COST OF AIR-TO-FUEL INCREASE (1978)

	Engine/Fuel Type			
			Natural Gas	
	Diesel (Electrical Generation)	Dual Fuel (Electrical Generation)	(Oil and Gas Pipeline)	(Oil and Gas Production)
Capital ($/kW)	0	0	0	0
Maintenance ($/kWh)	0.0002	0.0002	0.0002	0.0002
Fuel ($/kWh)	0.0031	0.0004	0.0004	0.0004
Total Annualized ($/kWh)[a]	0.003	0.001	0.001	0.001
Percent increase in initial cost	0	0	0	0
Percent increase in annual cost	7.0	3.0	3.0	3.0
Percent reduction in NO_x	20	40	40	40

[a]Assumes 8000 hours operating year.

Specific figures are not presented here because of the difficulty in specifying costs that are very site dependent.

5.2.2.4 Manifold Air Temperature Reduction

If the engine has an aftercooler or an intercooler, reducing manifold air temperature could require a larger heat exchanger, more coolant circulation, or perhaps a temperature control system. These modifications would cost about 1.5 percent of the initial cost of the engine (Reference 5-1). Maintenance costs would also rise because of the additional cooling water which requires chemical treatment to prevent sludge and scale buildup, or the increased service of radiators to maintain the lower temperatures. However, these increased maintenance costs are expected to be small: only on the order of $0.0008/kWh for a diesel or dual fuel engine and $0.0002/kWh for a gas engine. There would also be a small charge for pumping the additional cooling air or water. As already discussed, manifold air cooling is expected to negligibly affect engine efficiency. To reduce NO_x by 20 percent, the largest increase in brake specific fuel consumption for the tests made was only 1 percent. In some cases, brake specific fuel consumption even decreased since manifold air cooling allows more air to be charged, thus increasing efficiency. The largest increase in cost of power from these engines due to increased manifold air cooling is expected to be about 1 percent.

5.2.2.5 External Exhaust Gas Recirculation

Installing an external exhaust gas recirculation system could increase the initial engine cost 5 percent. That price includes adding the required piping to recirculate the exhaust gases, a heat exchanger for cooling the recirculated exhaust gases, and a control system. Maintenance costs are expected to rise about 40 percent for a diesel engine and 80 percent for dual fuel or natural gas engines. This cost escalation includes the additional system to maintain recirculating exhaust gases which can foul various parts of the system (References 5-6, 5-14, 5-15). The small number of tests has shown almost no change in fuel consumption due to EGR (see preceding sections). Table 5-5 shows that EGR could increase the cost of operating a diesel engine by over 5 percent and the other types by over 13 percent.

5.2.2.6 Combination of Operational Adjustment Controls

By combining control techniques it may be possible to achieve the same NO_x reductions with a smaller fuel penalty, or reduce NO_x levels more than could be achieved by each technique alone. Except for brake specific fuel consumption (BSFC), the initial and operating costs can be assumed to be additive. The cost advantage of combined controls over single techniques will depend heavily upon the actual change in BSFC. There are other reasons, such as operational impacts and control of other emissions, that may favor combined control even if there are no cost advantages. Table 5-6 compares three different methods for reducing NO_x by 40 percent from a large bore diesel engine. Air-to-fuel changes combined with manifold air cooling will only give 40 percent reduction in special cases. For this case, there is a definite advantage to using combined controls since the two combined techniques, air-to-fuel ratio change and manifold air cooling or air-to-fuel ratio change and retard , had a lower BSFC than retard alone in the tests.

5.2.3 Exhaust Gas Treatment Techniques

Catalysts for oxidizing CO and HC are available, but when used with diesel engines must be regenerated periodically because of particle matter in the exhaust. The initial cost of the catalyst is about $10/kW (Reference 5-1) and because of the regeneration problem, maintenance costs can increase.

Catalysts for reducing NO_x are available for rich burning engines and have been used for purifying gases for injection into wells (Reference 5-16). Engines are sometimes used as a source of inert gases for injecting into gas wells because the engine can also be used to operate the compressor. Because NO_x can cause corrosion problems in the compressor (due to water formed when gas compresses), catalysts have been used to reduce NO_x. The engines are usually specially tuned so that very little oxygen is present in the exhaust. Thus these catalysts cannot be applied to a lean burning engine such as most large stationary engines.

Some manufacturers are developing catalysts that use NH_3 as a reducing agent but these catalysts have not yet been tested in actual engines (Reference 5-17). Since these catalysts are only in the developmental stage, costs can only be roughly estimated. The catalyst and container is estimated to be about 4 percent of the initial cost of

TABLE 5-5. ESTIMATED INCREMENTAL COST OF EXTERNAL EXHAUST GAS RECIRCULATION (1978)

	Engine/Fuel Type			
	Diesel (Electrical Generation)	Dual Fuel (Electrical Generation)	Natural Gas (Oil and Gas Pipeline)	Natural Gas (Oil and Gas Production)
Capital ($/kW)	12	12	12	4
Maintenance ($/kWh)	0.002	0.004	0.004	0.004
Fuel ($/kWh)	0	0	0	0
Total Annualized ($/kWh)[a]	0.0023	0.0043	0.0043	0.0041
Percent increase in initial cost	5	5	5	5
Percent increase in annual cost	5.3	14	14	13
Percent reduction in NO_x	20	20	20	20

[a]Assumes 8000 hours operating year.

TABLE 5-6. ESTIMATED INCREMENTAL COST OF COMBINED CONTROLS FOR LARGE BORE DIESEL ENGINES AT 40 PERCENT NO_x REDUCTION (1978)

	Control Technique		
	Retard	Air-to-Fuel Changes and Manifold Air Cooling	Air-to-Fuel and Retard
Capital ($/kW)	0	3.6	0
Maintenance ($/kWh)	0.0016	0.001	0.0018
Fuel ($/kWH)	0.0024	0.0015	0.0015
Total Annualized ($/kWh)[a]	0.0040	0.0026	0.0033
Percent increase in capital cost	0	1.5	0
Percent increase in annual cost	9.3	6.0	7.7
Percent reduction in NO_x	40	30-40	40

[a]Assumes 8000 in operating year.

the engine (Reference 5-1), in addition to a charge for the ammonia handling and injection system. Besides operating charges for maintaining the NH_3/catalyst system, there will be a charge for the ammonia and a small cost for operating the injection system.

5.2.4 Combustion Chamber Redesign

Combustion chamber redesign techniques are only now being developed and thus only very rough price estimates can be given. A. D. Little, under contract to the EPA, is presently planning to test some combustion chamber modifications and they have presented some cost estimates of these techniques (Reference 5-7).

The cost estimates of the required hardware range from about \$1/kW to \$47/kW with \$8/kW being a typical estimate. Of course the price will depend on exactly which technique is used. The operating cost will be very dependent on the BSFC change. The goal is to keep the change in BSFC to less than 4 percent. If this goal is met and the initial cost is about what is estimated above, combustion chamber redesign could be the most cost effective method of controlling NO_x for achieving 30 to 40 percent reduction.

5.3 SUMMARY OF NO_x EMISSION REDUCTIONS AND OPERATION AND MAINTENANCE IMPACTS

5.3.1 NO_x Reduction Techniques

Table 5-7 shows the NO_x reductions and fuel penalties associated with each control for each engine type. Combinations of controls which may be readily applied are also shown. From this table it is apparent that, with the exception of catalytic reduction, which is a flue gas treatment concept, fuel injection retard is the most effective engine control for diesel engines and EGR looks good if the additional maintenance is not a problem. Air-to-fuel ratio changes are most effective for gas engines and EGR and retard are also effective controls. With combinations of controls, NO_x emissions reductions of 40 percent should be achievable with further reductions possible with a catalyst-NH_3 system. Some engines may even achieve 60 percent NO_x reduction.

On July 23, 1979 (Federal Register Vol. 44, No. 142, p. 43153), it was proposed that diesel and dual fuel engines larger than 560 cubic inches (0.0092 m^3) per cylinder be controlled to 600 ppm at 15 percent

O_2, and gas engines larger than 350 cubic inches (0.0057 m^3) per cylinder be controlled to 700 ppm. Recall that the average emissions from these engines is about 17 g/kWh (approximately 930 ppm at 15 percent O_2) for diesel and slightly less for the others. Thus a reduction of approximately 35 percent is required. As has just been discussed, there are several techniques which should be able to achieve this level of reduction.

5.3.2 Operational and Maintenance Impacts

Since engines are currently optimized for a minimum of maintenance and fuel usage, any control technique which varies engine parameters from standard conditions will impact operation and maintenance. Some of these impacts have been well characterized, especially those from control techniques which involve engine operational changes. Other control techniques will require various degrees of evaluation before impacts are clearly understood.

Derate, air-to-fuel ratio changes, manifold air cooling and ignition retard present the least problems in operation and maintenance. Derate has no impact mechanically and can improve durability because of lower temperatures and pressures. However, additional engines may be required to replace the lost power. Fuel penalties are usually low.

Air-to-fuel ratio changes in the lean direction cause a power loss if larger blowers or turbochargers must be used. If the engine must be operated richer to reduce NO_x emissions, other emissions such as smoke and deposits can increase. This could cause an increase in engine maintenance. Changes in air-to-fuel ratio in either direction will increase fuel consumption. Finally, operation at air-to-fuel ratios where misfiring or detonation occur can cause severe engine damage.

Manifold air cooling has little operational or maintenance impact for a unit that is already intercooled, but will increase the size of the heat exchanger, water or air pump, control system, and other system components. Of course, installing intercooling on an engine originally without the system will add maintenance to achieve additional temperature reductions. Changes in fuel consumption are small.

When properly applied, ignition retard has no serious mechanical drawbacks. Some increase in operational and maintenance time would ensure the degree of retard is always within safe limits. Increases in fuel

TABLE 5-7. NO_x REDUCTION AND FUEL CONSUMPTION PENALTIES FOR DIESEL, DUAL FUEL, AND GAS ENGINES

Control Approach		Diesel % NO_x Reduction	Diesel ΔBSFC, %[a]	Dual Fuel % NO_x Reduction	Dual Fuel ΔBSFC, %[a]	Natural Gas % NO_x Reduction	Natural Gas ΔBSFC, %[a]
Derate (D)	3%	--	--	--	--	<20	2
	6%	--	--	--	--	<40	3
	10%	--	--	<20	4	--	--
	20%	<20	4	--	--	--	--
	25%	5-23	1-5	1-33	1-7	5-90	2-12
Retard (R)	2°	<20	4	<20	3	--	--
	4°	<40	4	<40	1	<20	3
	8°	28-45	2-8	50-73	3-5	8-40	2-7
Air-to-Fuel (A)	2%	--	--	--	--	<20	2
	3%	--	--	<20	0	--	--
	5%	--	--	--	--	<40	7
	±10%	7-8	3	25-40	1-3	20-80	5-12
Manifold Air Temperature (M')	311K(100°F)	7-15	0-2	18-37	0-1	28	0
	315K(107°F)	--	--	--	--	<20	0
	318K(113°F)	--	--	<20	1	--	--
Internal EGR		5	2	--	--	<20	5
		--	--	--	--	5-35	0-8
External EGR	10%	<20	1	20	1	<20	0
		33	1	--	--	33	0
Retard and Manifold Air Temperature		<20	1	<20	1	<20	3
		10-24	0-1	25	2	30-40	5-6
Retard & Air-to-Fuel		<20	8	<20	1	<20	4
		<40	16	<40	2	<40	8
		35-65	5-26	56	2	17-52	4-11
Retard and Manifold Air Temperature and Air-to-Fuel		20	0	<20	2	<20	2
		--	--	40	3	<40	4
		--	--	--	--	40-65	6-7
Air-to-Fuel and Manifold Air Temperature		<20	2	--	--	--	--
		20-30	3	--	--	--	--
Water Injection H_2O/fuel ratio	50%	25-35	2-4	--	--	25-35	1-2
	100%			--	--	60-75	2-5
Catalytic Reduction (Projected)		50-80	0	50-80	0	50-80	0
Combustion Chamber Modifications (Projected)	Increased Mixing	10-30	<5	20-40	<5	20-40	<5
	Staged Combustion	10-30	0	10-30	0-7	10-30	0-2

[a]ΔBSFC is increase in brake specific fuel consumption.

consumption are moderate. Excessive amounts of retard, however, can create severe engine problems. Fuel consumption will increase rapidly, power drops, misfiring can occur, and smoke levels increase. In addition, mechanical maintenance will increase if the exhaust temperature exceeds the safe limits for valves or the turbocharger, usually 920 K ($1200^{\circ}F$). More frequent engine teardown will be required, and higher initial costs will result for higher temperature materials.

Exhaust gas recirculation and water injection are two effective control techniques that will require some development before operation and maintenance problems make the techniques useful. Exhaust gas recirculation requires new hardware components which mandate new maintenance techniques. Problems of fouling the flow passages of the cooling heat exchanger, the engine turbocharger and aftercooler with particulate must be solved, or frequent engine teardown will be required. Under varying load conditions a sophisticated control system is required or the engine may stall or emit unacceptable smoke levels. Fuel consumption penalties with EGR are small.

Water injection can cause severe maintenance problems. Deposits from untreated water will build up on internal engine surfaces, and also foul the lubricating oil. These problems lead to major engine maintenance. Water injection also adds another system to the engine which must be maintained and controlled.

Although not demonstrated, combustion chamber modifications will present the least impact to operation and maintenance. Maintenance can be expected to increase slightly if additional injectors, spark plugs and valves are added to the chamber. However, because this control technique involves new design, many of the additional maintenance requirements can be designed out. Fuel penalties are expected to be small.

Catalytic reduction will require no additional engine maintenance, since it is a flue gas treatment technique, rather than an engine modification. However, operating the catalyst may be expensive. Fouling the catalytic surfaces with particulate may require frequent regeneration. The catalyst may also have a relatively short life and need to be replaced. Another system, ammonia injection, must be added and maintained for most applications, and the cost of ammonia must be included in engine operation. Finally, harmful products of the reaction may be produced if the catalyst temperature varies from the proper level, or if excessive ammonia is injected. This catalyst must be installed and operated on an engine before all these effects can be quantified.

REFERENCES FOR SECTION 5

5-1. "Stationary Internal Combustion Engines. Background Information: Proposed Standards," EPA-450/3-78-125a, April 1979.

5-2. Letter from W. E. Snyder, Waukesha, WI to D. Bell, EPA, RTP, N.C., May 18, 1978.

5-3. Patterson, D. J., and N. A. Henein, Emissions from Combustion Engines and their Control, Ann Arbor Science Publishers, Inc., Ann Arbor, MI, 1974.

5-4. Roessler, W. U., et al., "Assessment of the Applicability of Automotive Emission Control Technology to Stationary Engines," EPA-650/2-74-051, NTIS-PB 237 115, July 1974.

5-5. Shaw, J. C., "Emission Reduction Study on a Carbureted Natural Gas Fueled Industrial Engine," Draft ASME Paper, White Superior Division, White Motor Corporation, Eastlake, OH, November 1974.

5-6. Urban, C. M., and K. J. Springer, "Study of Exhaust Emissions from Natural Gas Pipeline Compressor Engines," Southwest Research Institute, PR-15-61, San Antonio, TX, February 1975.

5-7. "Potential Emission-Control Concepts for Large-Bore Stationary Engines," Draft Report, EPA Contract 68-02-2664, Arthur D. Little, Inc., Cambridge, MA, November 1978.

5-8. Acurex meeting notes, California Air Resources Board, "Public Workshop for the Discussion of a Tentative Proposal for Control of NO_x Emissions from Stationary Internal Combustion Engines," Sacramento, CA, October 1978.

5-9. Personal communication with D. Brown, Briggs and Stratton, Wauwatosa, WI, October 1978.

5-10. Personal communication with G. P. Hanley, General Motors, Warren, MI, December 19, 1978.

5-11. "Monthly Energy Review," DOE/EIA-0035/7, NTIS-PB 127 007, July 1978.

5-12. "Nelson Cost Indices," The Oil and Gas Journal, Vol. 76, No. 40, pg. 125, October 1978.

5-13. "Diesel Power -- MAN Supplement," Diesel and Gas Turbine Progress Worldwide, Vol. 10, No. 9, pg. 1-44, November-December 1978.

5-14. "Characterization and Control of Emissions from Heavy Duty Diesel and Gasoline Fueled Engines," Report under EPA-IAG-0219(D), Fuels Combustion Research Group, Bartlesville Energy Research Center, Bartlesville, OK, December 1972.

5-15. Bosecker, R. E., and D. F. Webster, "Precombustion Chamber Diesel Emissions -- A Progress Report," SAE 710672, August 1971.

5-16. Barstow, W. F., and G. W. Watt, "Fifteen Years of Progress in Catalytic Treating of Exhaust Gas," Society of Petroleum Engineers Paper 5347, 1975.

5-17. Personal communication with R. Leohelt, Engelhard Industries, Newark, NJ, November 17, 1978.

5-18. "Environmental Impact Statement and Economic Impact Analysis, Revised Heavy-Duty Engine Regulations for 1979 and Later Model Years," Prepared by Office of Mobile Source Air Pollution Control, EPA, Washington, DC, August 1977.

5-19. Lindgren, L., "Cost Estimations for Emission Control Related Components/Systems and Cost Methodology Description," EPA-460/3-78-002, NTIS-PB 279 195, March 1978.

5-20. Hunter, S. C., "Application of Combustion Modifications to Industrial Combustion Equipment," EPA-600/7-79-015a, January 1979.

5-21. Wyczalek, F. A., et al., "EFI Prechamber Torch Ignition of Lean Mixtures," SAE Paper 750351, February 1975.

5-22. Personal communication with Pat Bradley, Southern California Gas, Los Angeles, California, February 22, 1980.

SECTION 6
ENVIRONMENTAL IMPACTS

One very important criterion when evaluating the effectiveness of a pollutant control technique is its effect on other pollutant emissions. A NO_x control technique should limit NO_x emissions without adversely affecting the emissions of other pollutants such as smoke. This section discusses pollutant formation mechanisms, and reviews the available data on the effect of NO_x controls on incremental emissions. To help quantify how NO_x controls affect the environmental impact of a combustion source, a Source Analysis Model, SAM IA, was applied to typical effluent stream emissions from a large bore diesel engine since a diesel exhaust probably has the most pollutants.

6.1 POLLUTANT FORMATION MECHANISM

This section discusses how various pollutants may form during the combustion process. The effect of postcombustion techniques such as catalytic treatment will not be covered here since any effects are very catalyst dependent.

6.1.1 NO_x Formation

As mentioned previously, the two main sources of NO_x are: NO_x formed from the oxidation of N_2 introduced in the combustion air, or thermal NO_x; and NO_x formed from nitrogen containing compounds in the fuel or fuel NO_x. Because of the low nitrogen content of the fuels normally used (see Section 4), most NO_x emissions are due to thermal NO_x. The residence time of the N_2 at high temperatures in contact with oxygen strongly influences the amount of NO_x formed. Internal cylinder conditions such as local air-to-fuel mixture, fuel injection, and heat loss could all affect NO_x formation.

6.1.2 HC Formation

The main sources of hydrocarbon emissions are unburned or partially burned fuel components. Other sources such as blowby and evaporative losses should not be important in large bore engines. Diesel fuel has a relatively low vapor pressure, and since natural gas is mainly methane (which is not considered a reactive hydrocarbon) evaporative losses are not expected to be significant. Blowby emissions should not be important because diesel fuel is injected after the compression stroke. Therefore, any blowby during the compression stroke would contain little fuel. For natural gas engines blowby would be mainly methane. Also, for these large bore engines the amount of surface area between the rings and cylinder walls is relatively small.

Several mechanisms exist which could cause hydrocarbon emissions. Fuel droplets could be transported or injected into the air layer near the cylinder walls, which could be at too low a temperature to sustain combustion. There could also be incomplete mixing or an improper air-to-fuel ratio. For diesel engines, the fuel droplets could be too large and thus not have enough time for complete volatilization and combustion. Any mechanism that will keep the fuel from being in contact with oxygen at a high enough temperature for a sufficient time can cause hydrocarbon emissions.

6.1.3 CO Formation

CO emissions are caused by lack of oxygen or the CO cooling off before it can be converted to CO_2. Mechanisms that cause hydrocarbon emissions can also cause CO emissions. Since the CO comes from partial oxidation of the fuel while HC emissions may be caused by completely unoxidized fuel, there will not always be a 1 to 1 correspondence between CO and HC emissions. But as a first approximation, mechanisms that usually cause hydrocarbon emissions will also cause CO emissions.

6.1.4 Smoke Formation

As is the case for HC and CO emissions, smoke can be caused by the lack of oxygen or low temperature. Smoke can be a mixture of soot particles, which are mainly carbon particles, and fuel droplets. Smoke can be caused by unburned or partially burned fuel from the center of large fuel droplets and the core of the injector spray. The quenching of the burning fuel by the low temperature region near the walls can also cause smoke. Diesels may form polycyclic organic matter (POM) as mentioned in Section 2, but the formation mechanisms are not understood.

6.1.5 SO_x and Trace Elements

SO_x emissions are expected to be directly proportional to the amount of sulfur in the fuel. Trace elements emissions can come from the fuel, from particles picked up by the combustion air, or from leaching of engine surfaces. Trace element emissions are expected to be very small.

6.1.6 Baseline Emissions

Pollutant emission levels are given here in terms of g/kWh power output instead of ng/J heat input since some NO_x control techniques can cause large changes in fuel consumption (even up to 10 percent as discussed in Section 5). Baseline emissions are listed for comparison in Table 6-1 (Reference 6-1).

TABLE 6-1. AVERAGE BASELINE EMISSIONS FROM STATIONARY INTERNAL COMBUSTION ENGINES[a], g/kWh

Engine Fuel	NO_x	CO	HC
Large bore engines			
Diesel	17.3	2.4	0.6
Natural gas	15.4	3.8	6.5
Dual fuel	11.0	2.7	4.2
Medium and Small Bore Engines			
Gasoline	11.9	13.7	11.3
Diesel	16.6	6.0	2.8

[a]Reference 6-1

Recall that most of the hydrocarbon emissions from natural gas and dual fuel engines are methane. Also, these emissions are only for an average engine and emissions from a particular engine can be very different since emissions depend on such factors as swirl, types of valves and ports, type of injector or spark plug and their location, type of piston head, combustion chamber and shape, operation conditions, etc. Also notice that only NO_x, CO, and total hydrocarbon emissions are listed since the other emissions and the breakdown of hydrocarbon emissions are not very well quantified. As in the other sections, only the potential environmental impacts of combustion modifications on large bore engines are discussed in the following.

6.2 INCREMENTAL ENVIRONMENTAL IMPACTS ON AIR

Figures 6-1 through 6-6 summarize the changes in CO and total HC emissions as a function of controlled NO_x level for the large bore engines measured. The graphs list the values for the emissions before and after the controls were applied. The numbers by the dots refer to the engine numbers as used in Section 5. Also recall that except for diesel engines, most hydrocarbon emissions are methane as discussed in Reference 6-2. These data are from publications and reports from engine manufacturers (Reference 6-3).

6.2.1 Derate

Derate can lower the maximum engine temperature which could increase the amounts of HC, CO, and smoke produced. For most of the large bore diesel engines measured, CO only increased slightly, while two of the engines actually showed large decreases in CO emissions. The natural gas and dual fuel engines tested measured small to large increases in CO emissions. The same effects were noticed for total hydrocarbon emissions. For HC emissions, some engines showed large increases while some showed large decreases as the engine was derated, with the unexpected results that the engine showing large changes in HC emissions was not necessarily the one showing large CO changes. For dual fuel and natural gas engines tested, the total hydrocarbon emissions showed small to large changes (some showed increases, some showed decreases) as the engine was derated. The changes in CO and HC emissions from these engines were more closely correlated than for diesel, though there was not a direct correspondence for each engine. Again, for all types of engines the preceding comments are based on only a limited number of tests. For all tests conducted, diesel engines showed lower smoke levels as they were derated, but results were based on three tests only. The fact that some engines showed increased HC indicates that smoke might also increase.

6.2.2 Retard

Retard shortens the residence time, which would tend to increase HC and CO but also increases the exhaust temperature, which would tend to lower HC and CO emissions. As long as the engine is not retarded to the point where misfiring occurs, tests have usually shown only small

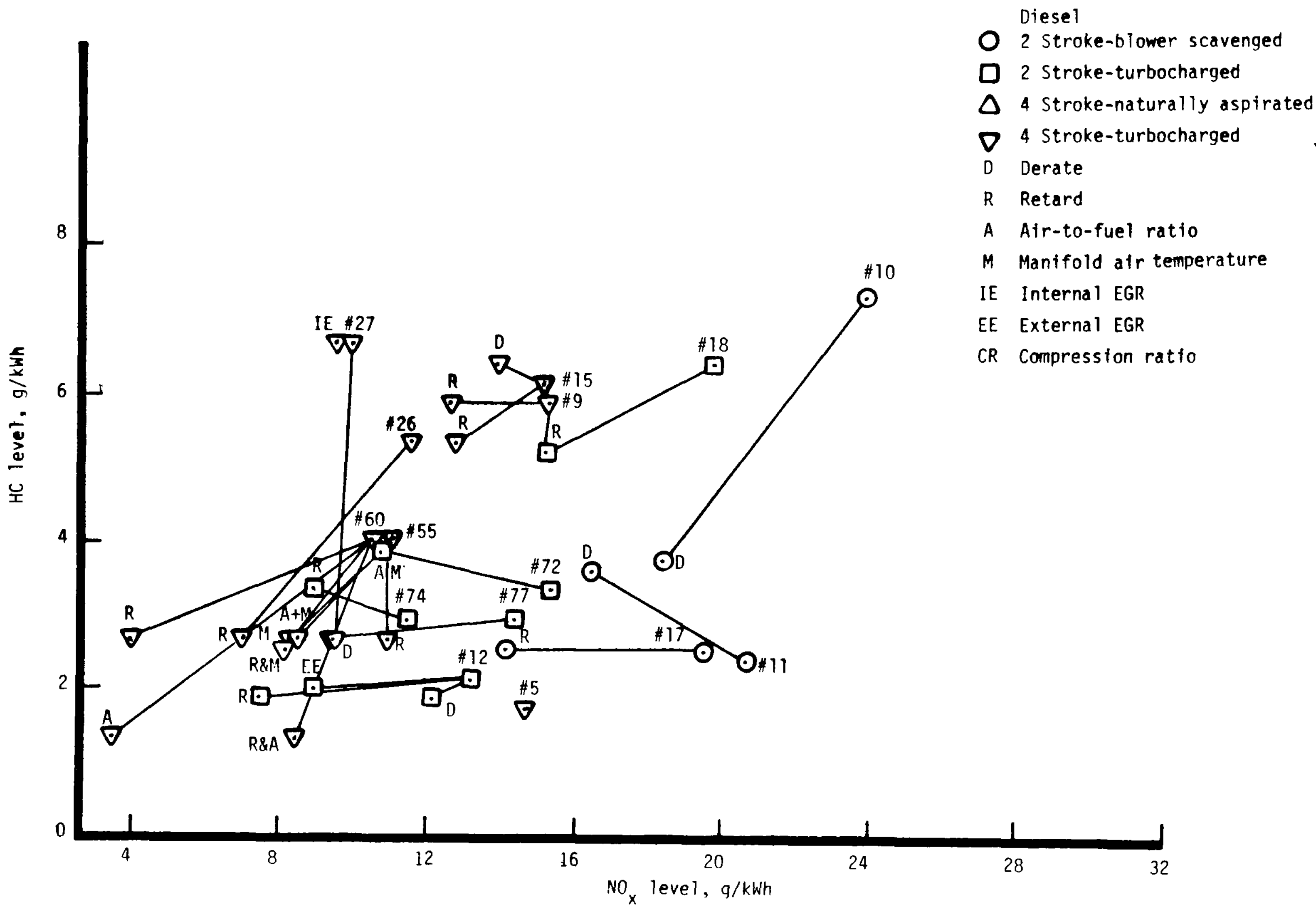


Figure 6-1. HC levels versus controlled NO_x levels for diesel engines (Reference 6-3).

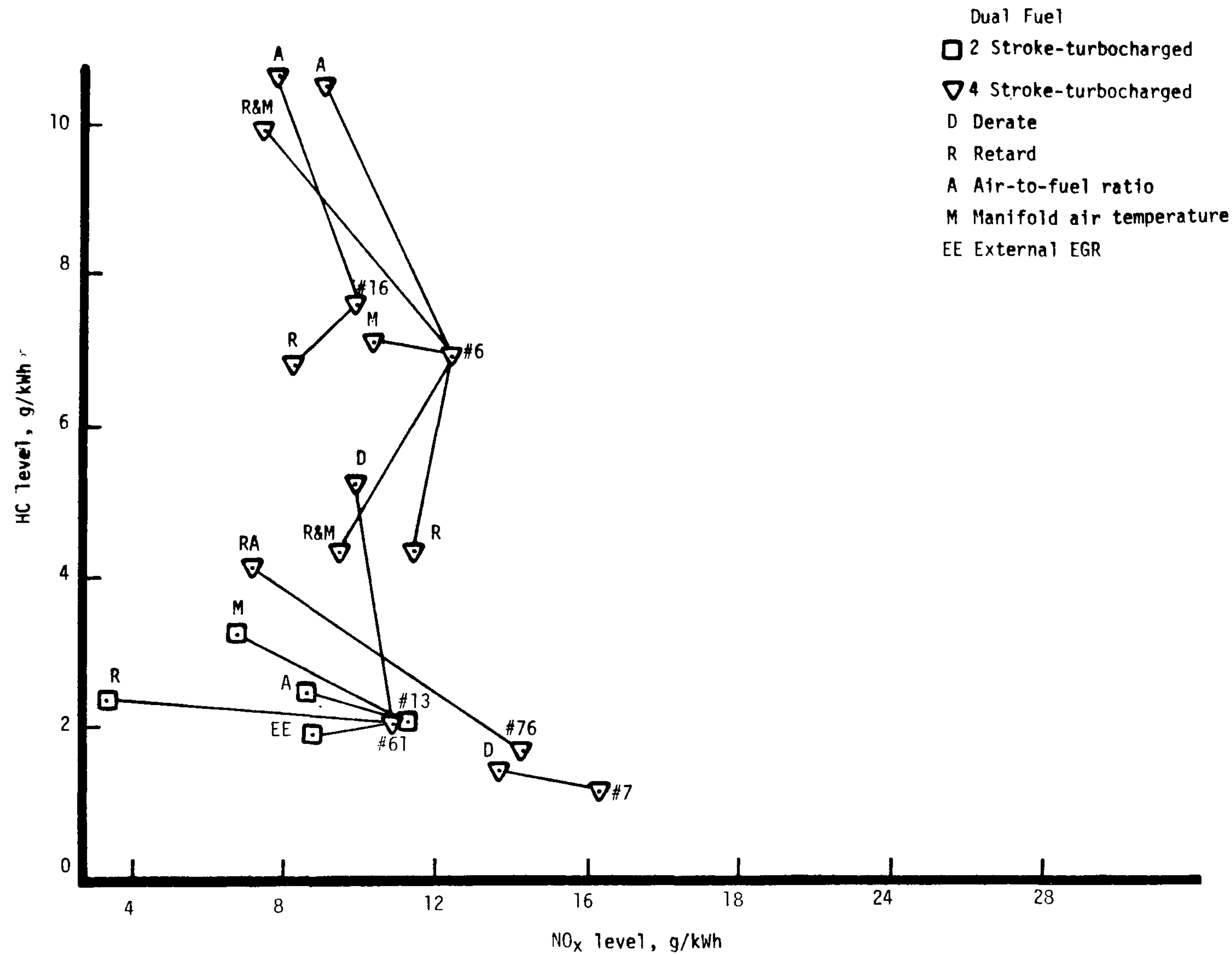


Figure 6-2. HC levels versus controlled NO_x levels for dual-fuel engines (Reference 6-3).

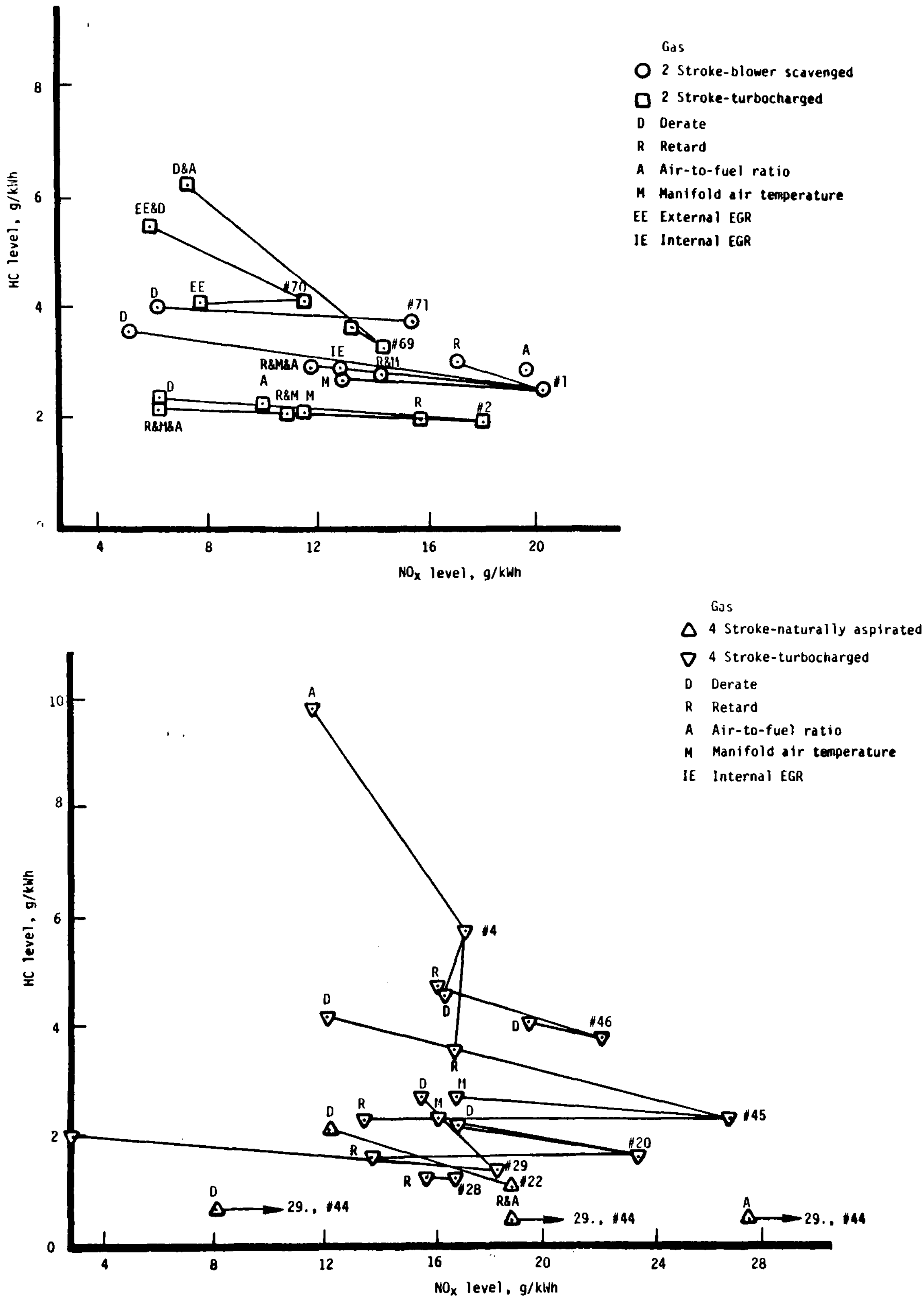


Figure 6-3. HC levels versus controlled NO_x levels for gas engines (Reference 6-3).

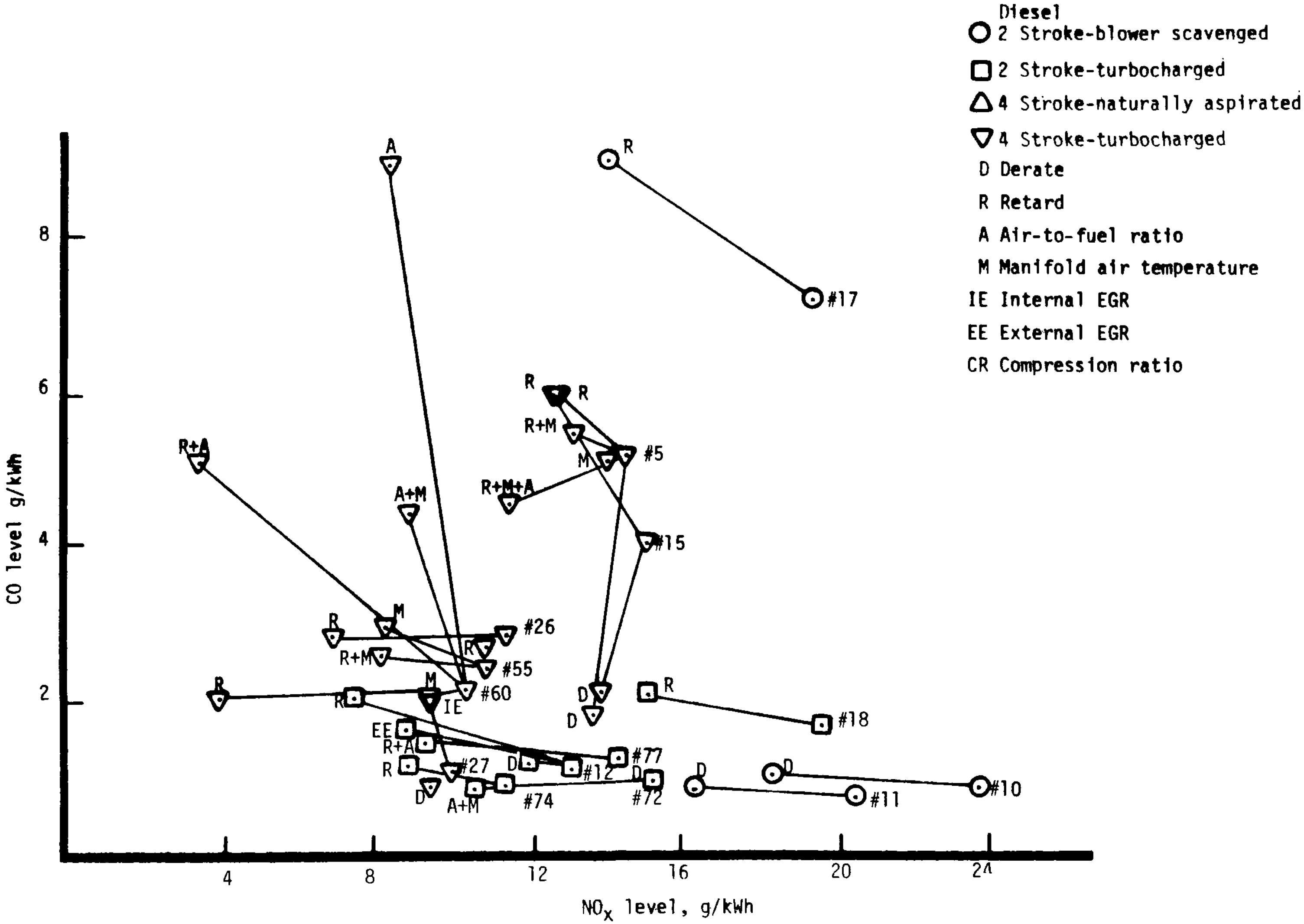


Figure 6-4. CO emissions versus controlled NO_x levels for diesel engines.

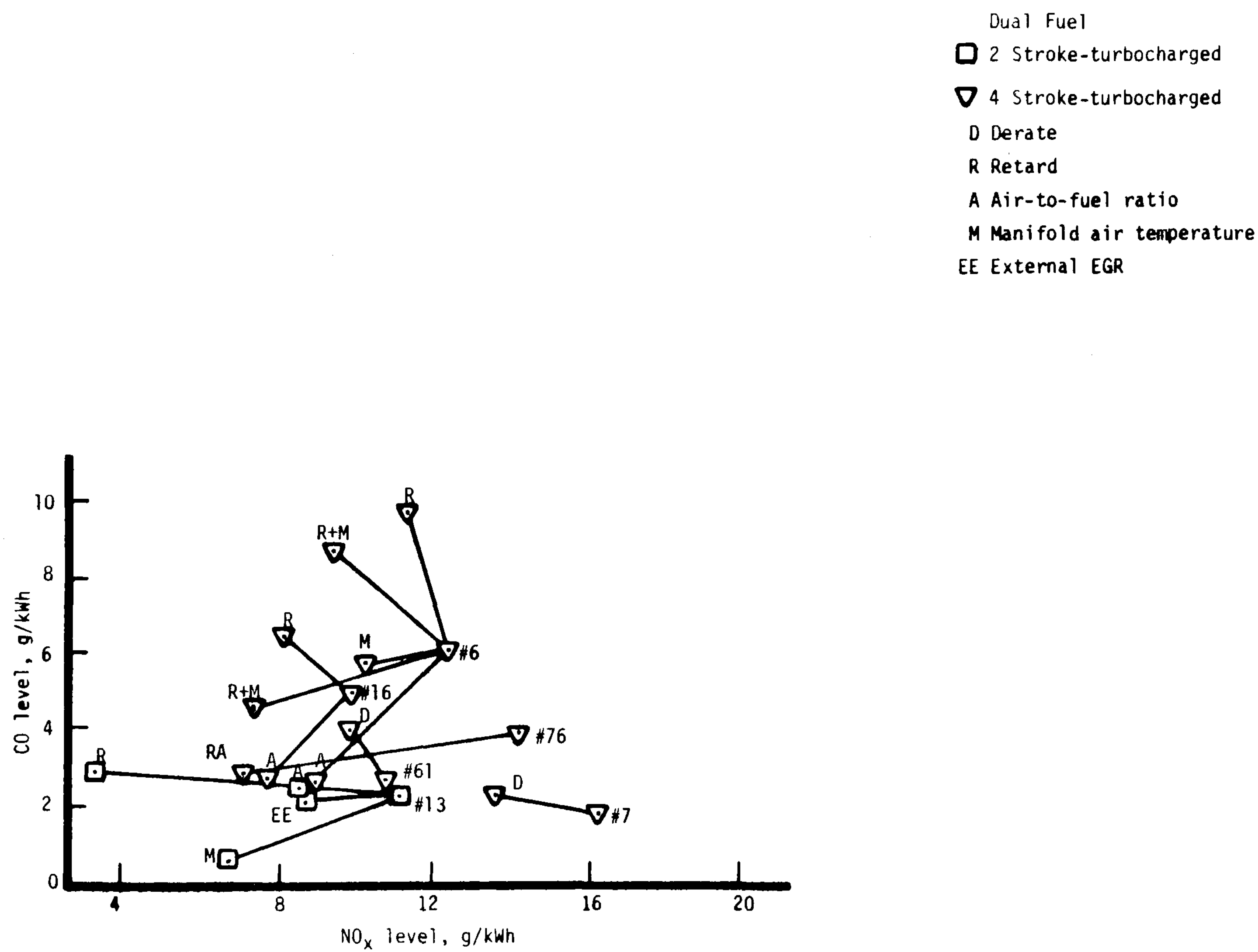


Figure 6-5. CO emissions versus controlled NO_x levels for dual fuel engines (Reference 6-3).

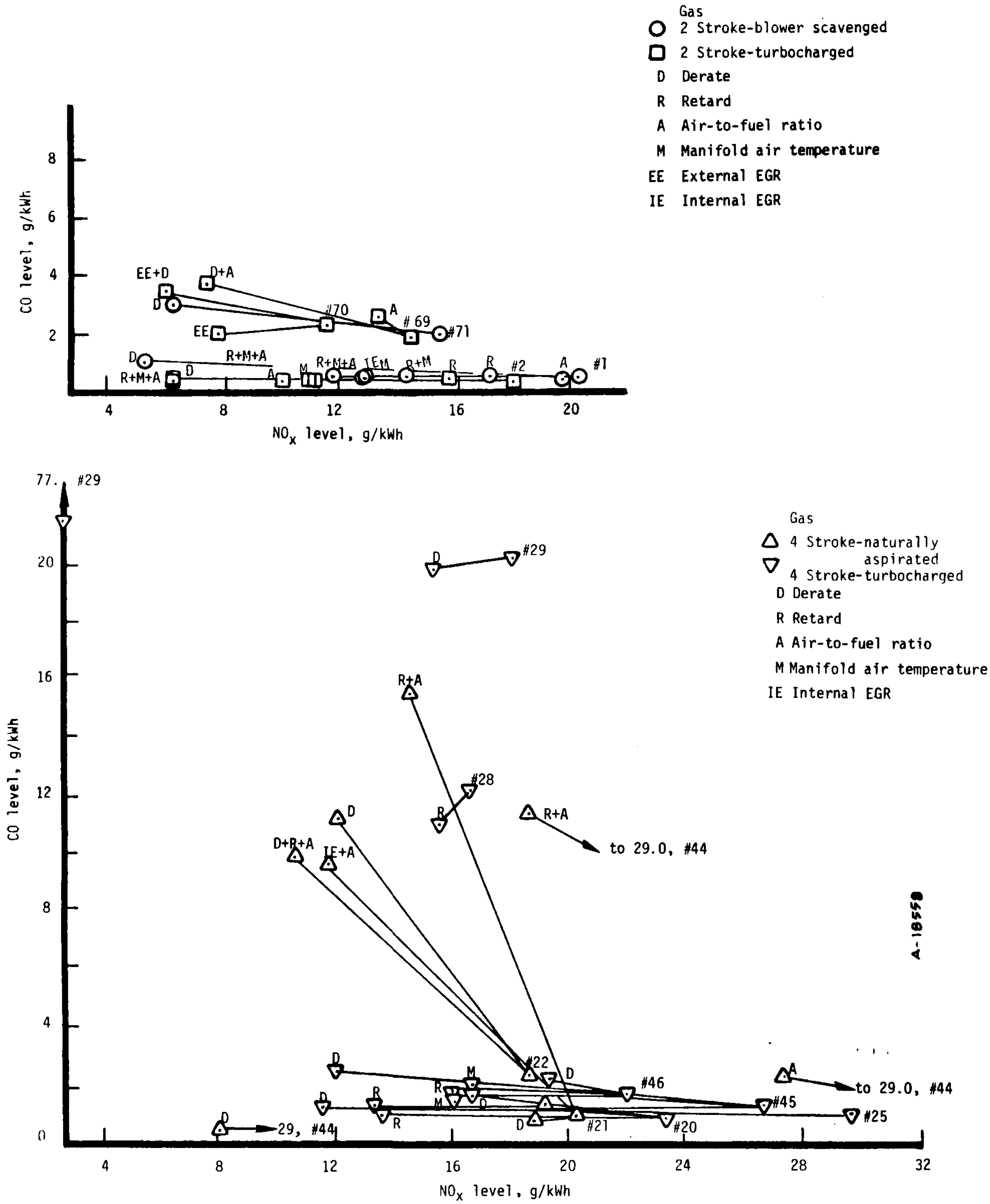


Figure 6-6. CO emissions versus controlled NO_X levels for gas engines (Reference 6-3).

increases in CO and HC emissions. In some tests, retard even decreased HC emissions. Up to a certain point (2^0 to 6^0 retard) only a small increase was noticed in smoke levels but if the diesel engine was retarded further, smoke levels greatly increased (Reference 6-4).

6.2.3 Air-to-Fuel Changes

Most large bore engines operate on the lean side, i.e., greater than stoichiometric air, and air-to-fuel changes for NO_x control mean operating under even leaner conditions. As the mixture becomes leaner, the peak temperature decreases while the amount of excess oxygen increases. Thus the HC emissions are expected to increase because of this decreased temperature. However, CO emissions are not expected to increase because the increased amounts of oxygen should counteract the effect of lower temperature. Except for one diesel engine which showed a large CO increase, the engines tended to follow the above, i.e., small HC increases and very small CO increases. The test which showed large CO increases was where the air-to-fuel ratio was decreased instead of increased as in the other cases. NO_x was still reduced.

6.2.4 Reduced Manifold Air Temperature

Reduced manifold air temperature tends to lower the maximum combustion temperature, which may raise CO and HC emissions. Only small increases in HC and CO were measured from the engines tested, and in a small number of cases, there was even a decrease. There are very few smoke data but no significant change was noticed in the available data, though smoke probably would increase along with increased HC emissions. One engine manufacturer claims smoke increased as a result of reduced manifold air temperature (Reference 6-4).

6.2.5 Exhaust Gas Recirculation

EGR will lower the combustion temperature and the amount of oxygen present, which could lead to increased CO, HC, and smoke emissions. At the same time, recirculating the exhaust gas could lower HC since some of the HC in the exhaust would be put back into the cylinder and get a second chance to burn. Tests on diesel engines used as locomotive engines showed increased CO and smoke while HC was unchanged (Reference 6-5). Tests on truck engines have shown the same trends (Reference 6-6).

6.2.6 Flue Gas Treatment

The effect on CO and HC caused by a NO_x reduction catalyst will be very catalyst dependent. Since most of these engines have large amounts of excess oxygen, it is not expected that CO or HC would be effected. If NH_3 is used as the reducing agent, there is the danger of HCN or NH_3 emissions. In contrast, in one test on an oxidation catalyst for reducing CO and HC emissions, NO_x emissions increased (Reference 6-7).

6.2.7 Source Assessment Model Results

To help quantify the change in potential environmental impact of an IC engine which switches from baseline to low NO_x operation, a source analysis model, SAM IA (References 6-8 and 6-9), was applied to typical air emissions data from a large bore diesel engine. EPA has been developing a series of source analysis models to define methods of comparing emission data to environmental objectives, termed multimedia environmental goals (MEG's) (Reference 6-10). The model selected for the level of data detail obtainable was SAM IA, designed for rapid screening purposes. As such, it includes no treatment of pollutant transport or transformation. Goal comparisons employ threshold effluent stream concentration goals, termed discharge multimedia environmental goals (DMEG's).

For the purposes of screening pollutant emissions data to identify species requiring further study, a discharge severity (DS) is defined as follows:

$$DS_i = \frac{\text{Concentration of Pollutant i in Effluent Stream}}{\text{DMEG of Pollutant i}}$$

The DMEG value, the threshold effluent concentration, is the maximum pollutant concentration considered safe for occupational exposure. When DS exceeds unity, more refined chemical analysis may be required to quantify specific compounds present.

To compare waste stream potential hazards, a weighted discharge severity (WDS) is defined as follows:

$$WDS = (\sum_i DS_i) \times \text{Stream Mass Flow Rate},$$

where the DS's are summed over all species analyzed. The WDS is an indicator of output of hazardous pollutants and can be used to rank the needs for controls for waste streams. It can also be used as a preliminary measure of how well a pollutant control, say a combustion modification NO_x control, affects the overall environmental hazard of the source. An extensive exposition of SAM IA and list of DMEG's are presented in References 6-8 through 6-11 and will not be repeated here.

Using SAM IA, the effect of combustion modification controls upon the total emissions from a large bore diesel engine was estimated. The only pollutants for which there are quantified data showing these effects are CO, NO_x, and total hydrocarbons. A partial hydrocarbon breakdown has been measured for the baseline (uncontrolled) case but not for the engine operating under NO_x control conditions (Reference 6-12). Because of this, SAM IA was applied assuming that the same kinds and distribution of organics are emitted under NO_x control as those under baseline conditions. Only the total amount of HC emitted is assumed to change. The results are summarized in Table 6-2. Appendix A presents the assumed waste stream characterizations and SAM IA worksheet calculations. An average NO_x control technique (such as retard, increased manifold air cooling, etc.) is assumed to be applied to the engine.

If the above assumption about organic emissions is correct, these NO_x controls decrease the total toxic level of emissions from the engine. The decrease in NO_x emissions is not offset by increases in CO and HC. Some authors have measured mutagenic compounds in diesel exhaust gases from mobile sources (Reference 6-9). Tests on the exhaust from a 150 kW stationary diesel have shown the exhaust to have a small mutagenic effect but less than that found for mobile engines (Reference 6-14). The tests on this engine operating under NO_x control conditions has not been completed yet. Thus more tests are needed before quantifying the results to give the true health effects of lowering NO_x emissions.

6.2.8 Concluding Remarks on Health Effects

An additional comment should be made about the toxicity of diesel exhaust. As just mentioned, some authors have found mutagenic compounds in exhaust from mobile diesel engines. A recent conference discussed this potential problem (Reference 6-15). Tests where diesel exhaust was assayed using bacterial mutagenecis tests elicited mutagenic responses. But studies on workers in a London bus garage and on workers in mines where diesels are used have showed no increases in the cancer rate as compared to the general public. Still, benzo(a)pyrene, a potent carcinogen, is one of the compounds found in the exhaust from diesel

TABLE 6-2. SAM IA RESULTS FOR DIESEL EXHAUST

Pollutant	Discharge Severity		Weighted Discharge Severity	
	Uncontrolled	Controlled	Uncontrolled	Controlled
CO	3.5	7	13650	27300
NO_x	86	54	333450	212160
HC[a]	22	35	680000	125000
Total	112	96	4.3×10^6	3.7×10^6

[a]Composition of organic emissions is approximated from limited emission data and is assumed not to change under controlled conditions. the total amount of organic emissions increases under controlled operation.

vehicles. Thus data indicate that diesel exhaust has the potential for being a serious health hazard, but epidemiological data have not confirmed this to date.

6.3 WATER AND SOLID POLLUTION IMPACTS

Combustion modification control techniques are not expected to cause any additional water solid pollution impact. Since some, if not all, of the techniques decrease fuel efficiency, the additional heat not converted to mechanical energy must be disposed of. This could have a small impact on the cooling water. Also, if reduced manifold air temperature is achieved by the use of cooling water, there could be a small impact on water discharges. These impacts are expected to be minor except in very unusual cases.

6.4 SUMMARY OF ENVIRONMENTAL IMPACTS

Table 6-3 and 6-4 summarize the qualitative effects of NO_x control techniques on CO and HC emissions, respectively. In general, CO and HC emissions are expected to slightly increase as NO_x is reduced. This will, of course, not always be the case and the actual changes will be very engine dependent. The increases in CO and HC emissions are expected to be less than the decrease in NO_x emissions. The distribution of hydrocarbon emissions is not very well quantified; thus it cannot definitely be said that the health effect improvement caused by the decrease in NO_x is not offset by the HC increase, although at present this does appear to be the case.

TABLE 6-3. EFFECTS OF CONTROLS ON ENGINES LARGER THAN $5.7 \times 10^{-3} m^3/cyl$: CO EMISSIONS

Fuel	Diesel				Dual Fuel				Natural Gas			
Strokes/Cycle	Two		Four		Two		Four		Two		Four	
Control \ Air Charging	BS	TC	NA	TC	BS	TC	NA	TC	BS	TC	NA	TC
Derate	↑			↓				↑↓	↑	↑	↑	↑
Retard	↑	↑		↑		↑		↑	↑	↑		↓
A/F				↑↓		↑		↓		↑	—	
TC	↑								↓		↓	
MAT	↑	↓		↑↓		↑		↑↓	↑	↑		↑
INJ	↓	↑↓										
EGR(I)				↓				↓	↑		—	
EGR(E)		↑				↑						
H_2O	↓			↑↓				↓			↑	↑
CR												

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

TABLE 6-4. EFFECTS OF CONTROLS ON ENGINES LARGER THAN 5.7 × $10^{-3}m^3$/cyl: HC EMISSIONS

Fuel	Diesel				Dual Fuel				Natural Gas			
Strokes/Cycle	Two		Four		Two		Four		Two		Four	
Control \ Air Charging	BS	TC	NA	TC	BS	TC	NA	TC	BS	TC	NA	TC
Derate				↑				↑	↑	↑	↑	↑
Retard	—	↑↓		↑↓		↑		↓	↑	↑	↑	↑
A/F				↑↓		↑		↑		↑		↑
TC	↓								↓			
MAT	↑	↑↓		↑↓		↑		↑↓	↑	↑		↑
INJ	↓	↓	↓									
EGR(I)				↓				↑	↑			
EGR(E)		—				↓						
H_2O	↑			↑↓				↑			↑	↑
CR												

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

REFERENCES FOR SECTION 6

6-1. Youngblood, S. B., and G. R. Offen, Acurex Interoffice Memorandum, "Emissions Inventory of Currently Installed Stationary Reciprocating IC Engines," Acurex Corporation, Mountain View, CA, September 23, 1975.

6-2. Urban, C. M., and K. J. Springer, "Study of Exhaust Emissions from Natural Gas Pipeline Compressor Engines," Southwest Research Institute Report PR-15-61, San Antonio, TX, February 1975.

6-3. "Stationary Internal Combustion Engines. Background Information: Proposed Standards," EPA-450/3-78-125a, April 1979.

6-4. Letter from W. C. Passie, Caterpillar Tractor Co., Peoria, IL, to H. Mason, Acurex Corporation, Mountain View, CA, March 15, 1979.

6-5. Storment, J. O., and K. J. Springer, "Assessment of Control Techniques for Reducing Emissions from Locomotive Engines," Southwest Research Institute Report AR-884, San Antonio, TX, April 1973.

6-6. Roessler, W. U., et al., "Assessment of the Applicability of Automotive Emissions Control Technology to Stationary Engines," EPA-650/2-74-051, NTIS-PB 237 115, July 1974.

6-7. Wasser, J. H., and R. M. Statnick, "Emission Characteristics of Small Stationary Diesel Engines," Proceedings of the Second Stationary Source Combustion Symposium, Volume V, EPA-600/7-77-073e, NTIS-PB 274 897, July 1977.

6-8. Schalit, L. M., and K. J. Wolfe, "SAM/IA: A Rapid Screening Method for Environmental Assessment of Fossil Energy Process Effluents," EPA-600/7-78-051, NTIS-PB 277 088/AS, February 1978.

6-9. Hangebrauck, R. P., et al., "Nomenclature for Environmental Assessment Projects: Part 1 -- Terminology for Environmental Impact Analyses," U.S. Environmental Protection Agency, Industrial Environmental Research Laboratory, Research Triangel Park, NC, August 1979.

6-10. Waterland, L.R., and L.B. Anderson, "Source Analysis Modeling for Environmental Assessment," Presented to Fourth Symposium on Environmental Aspects of Fuel Conversion Technoloy, Hollywood, FL, April 17, 1979.

6-11. Cleland, J. G. and G. L. Kingsburg, "Multimedia Environmental Goals for Environmental Assessment," Volumes I and II, EPA 600/7-77-136a and b, NTIS-PB 276 919 and NTIS-PB 276 920, November 1977.

6-12. Shih, C. C., et al., "Emissions Assessment of Conventional Stationary Combustion Systems; Volume II Internal Combustion Sources," EPA-600/7-79-029c, NTIS-PB 296 390, February 1979.

6-13. Barth, D. S., and S. M. Blacker, "The EPA Program to Assess the Public Health Significance of Diesel Emissions," JAPCA, Vol. 28, No. 8, pg. 769, August 1978.

6-14. Personal communication with J. Wasser, U.S. Environmental Protection Agency, Research Triangle Park, NC, April 30, 1980.

6-15. Edited Transcript of Proceedings, "Workshop on Unregulated Diesel Emissions and Their Potential Health Effects," DOT HS-803 27, April 1978.

SECTION 7
SUMMARY OF NEEDS FOR ADDITIONAL DATA

There are two major weaknesses in the available data. The information on operational effects of these control techniques is not very complete, especially information about combustion chamber redesign and catalytic exhaust gas treatment. Also, the data available on emissions other than NO_x, CO, and total hydrocarbons and specifications of HC emissions are very limited.

Research is needed on designing a low NO_x emitting engine (a program sponsored by the EPA to address this need is discussed below). Even with the best available controls applied, the large bore stationary reciprocating internal combustion engine is the worst NO_x emitter on a heat input basis of all the major combustion sources. From an energy viewpoint, these engines tend to be very efficient when compared to other sources, especially if a heat recovery device is used on the exhaust from the engine. Even with this high relative efficiency, the NO_x emitted per power output is still very high. Other emissions from these engines may also be important but more data are needed for quantification. In spite of their high emissions, their engines are still preferred in many operations because of their ease of operation (as compared to a boiler for example).

7.1 DATA NEEDS

Operational effects and long term effects of combustion modification control techniques need to be further evaluated. The effects of retard, derate, air-to-fuel ratio changes, manifold air cooling and EGR on engine operation are understood but some long term tests should be conducted to verify that there are no unforeseen problems. Of perhaps of more importance are tests on catalytic NO_x reduction and combustion chamber redesign techniques. Since these last two techniques have not been tested extensively even on a short term basis, much work is required to identify operation and maintenance problems, as well as to evaluate emission control effectiveness. Combustion chamber redesign techniques share the promise of giving the same emission reductions as the operational adjustment methods but without the fuel penalties. Exhaust gas catalytic NO_x reduction techniques have the potential of giving very large emission reductions.

A. D. Little under EPA contract has identified the most promising combustion chamber modification techniques and a report on this work has been published (Reference 7-1). These most promising techniques are at present being tested on large bore one cylinder engines. The tests should be finished by the end of 1981. Modification and testing of a full size engine will follow (Reference 7-2).

Other than CO, NO_x, and total hydrocarbons, there is only a small amount of information on emissions from these large bore engines. The type of organics emitted both as vapor phase and absorbed on particulate matter is not quantified under NO_x control conditions. Mobile diesels have been found to emit mutagenic compounds. Large bore stationary diesels may also emit these compounds. The amount and size distribution of the particulate matter need to be quantified.

REFERENCE FOR SECTION 7

7-1. Wilson, Jr., R. P., "Single-Cylinder Tests of Emission Control Methods for Large-Bore Stationary Engines," in Proceedings of the Joint Symposium on Stationary Combustion NO_x Control, EPA IERL-RTP-1087, October 1980.

7.2 Personal communication with R. P. Wilson, Jr., Arthur D. Little Co., Cambridge, MA, February 2, 1981.

APPENDIX A

SAM/IA WORKSHEETS

TABLE A-2. SAM IA WORKSHEET FOR CONTROLLED DIESEL ENGINE EXHAUST

1. SOURCE/CONTROL OPTION:

Diesel Engine/Controlled

LEVEL 1 X
LEVEL 2 ____

Page 1 of 1

2. EFFLUENT STREAM

101 — CODE NO.

Exhaust — NAME

3. TOTAL MASS RATE OF DISCHARGE (g/s)

Q = 3900 g/s

4. COMPLETE THE FOLLOWING TABLE FOR THE EFFLUENT STREAM OF LINE 2.

A	B	C	D	E	F	G	H	I	J	K	L	M
											WEIGHTED DISCHARGE SEVERITY	
SPECIES OR MEG CATEGORY NAME	MEG NUMBER OR CATEGORY	SPECIES OR CATEGORY CONCENTRATION	HEALTH DMEG CONCENTRATION	COMPOUND MEG NO. ASSUMED FOR LEVEL 1 — HEALTH	ECOLOGICAL DMEG CONCENTRATION	COMPOUND MEG NO. ASSUMED FOR LEVEL 1 — ECOLOGICAL	DISCHARGE SEVERITY — HEALTH (C/D)	DISCHARGE SEVERITY — ECOLOGICAL (C/F)	✓ IF HEALTH DMEG EXCEEDED	✓ IF ECOL. DMEG EXCEEDED	(HEALTH BASED) (H x LINE 3)	(ECOLOGICAL BASED) (I x LINE 3)
UNITS	—	μg/m³	μg/m³	—		—	—	—	—	—	g/s	g/s
CO	42B100	2.8×10^5	4×10^4				7.0				27300	
NO_x	47B150	4.9×10^5	9×10^3				54.4				212160	
Organics												
Aliphatic Hydrocarbons	0.1	6×10^4	2×10^5	B120			0.3				1170	
Substituted Benzene	1.5	3.7×10^4	1×10^5	A140			0.4				1560	
Aldehydes Ketones	0.7	0.8×10^4	2.5×10^2	A060			32				124800	
POM	21	0.3×10^4	1.6×10^3	A180			1.9				7410	

(IF MORE SPACE IS NEEDED USE A CONTINUATION SHEET)

5. TOTAL DISCHARGE SEVERITY

HEALTH BASED (Σ col. H) 5a 96.0

ECOLOGICAL BASED (Σ col. I) 5b ____

(ENTER HERE AND AT LINE 8 OF THE SUMMARY SHEET)

6. TOTAL WEIGHTED DISCHARGE SEVERITY

HEALTH BASED (Σ col. L) 6a 37×10^5

ECOLOGICAL BASED (Σ col. M) 6b ____

(ENTER HERE AND AT LINE 8 OF THE SUMMARY SHEET)

TABLE A-1. SAM IA WORKSHEET FOR UNCONTROLLED DIESEL ENGINE EXHAUST

1. SOURCE/CONTROL OPTION: Diesel Engine/Uncontrolled

LEVEL 1 X
LEVEL 2 ____

2. EFFLUENT STREAM
CODE NO. 101 NAME Exhaust

3. TOTAL MASS RATE OF DISCHARGE (g/s)
Q = 3900 g/s

4. COMPLETE THE FOLLOWING TABLE FOR THE EFFLUENT STREAM OF LINE 2.

A	B	C	D	E	F	G	H	I	J	K	L	M
											WEIGHTED DISCHARGE SEVERITY	
SPECIES OR MEG CATEGORY NAME	MEG NUMBER OR CATEGORY	SPECIES OR CATEGORY CONCENTRATION	HEALTH DMEG CONCENTRATION	COMPOUND MEG NO ASSUMED FOR LEVEL 1 – HEALTH	ECOLOGICAL DMEG CONCENTRATION	COMPOUND MEG NO. ASSUMED FOR LEVEL 1 – ECOLOGICAL	DISCHARGE SEVERITY – HEALTH (C/D)	DISCHARGE SEVERITY – ECOLOGICAL (C/F)	✓ IF HEALTH DMEG EXCEEDED	✓ IF ECOL. DMEG EXCEEDED	(HEALTH BASED) (H x LINE 3)	(ECOLOGICAL BASED) (I x LINE 3)
UNITS	—	$\mu g/m^3$	$\mu g/m^3$	—		—	—	—	—	—	g/s	g/s
CO	42B100	1.4×10^5	4×10^4				3.5				13650	
NO_x	47B150	7.7×10^5	9×10^3				85.5				333450	
Organics												
Aliphatic Hydrocarbons	01	4×10^4	2×10^5	B120			0.2				780	
Substituted Benzene	15	2.5×10^4	1×10^5	A140			0.2				780	
Aldehydes Ketones	07	0.5×10^4	2.5×10^2	A060			20				78000	
POM	21	0.2×10^4	1.6×10^3	A180			1.2				4680	

(IF MORE SPACE IS NEEDED USE A CONTINUATION SHEET)

5. TOTAL DISCHARGE SEVERITY
HEALTH BASED (Σ col. H) 5a 110.6
ECOLOGICAL BASED (Σ col. I) 5b ____
(ENTER HERE AND AT LINE 8 OF THE SUMMARY SHEET)

6. TOTAL WEIGHTED DISCHARGE SEVERITY
HEALTH BASED (Σ col. L) 6a 43×10^5
ECOLOGICAL BASED (Σ col. M) 6b ____
(ENTER HERE AND AT LINE 8 OF THE SUMMARY SHEET)

TECHNICAL REPORT DATA
(Please read Instructions on the reverse before completing)

1. REPORT NO. Epa-600/7-81-127	2.	3. RECIPIENT'S ACCESSION NO.
4. TITLE AND SUBTITLE Environmental Assessment of Combustion Modification Controls for Stationary Internal Combustion Engines		5. REPORT DATE July 1981
		6. PERFORMING ORGANIZATION CODE
7. AUTHOR(S) H. I. Lips, J. A. Gotterba, K. J. Lim, and L. R. Waterland		8. PERFORMING ORGANIZATION REPORT NO.
9. PERFORMING ORGANIZATION NAME AND ADDRESS Acurex/Energy and Environmental Division 485 Clyde Avenue Mountain View, California 94042		10. PROGRAM ELEMENT NO. EHE624A
		11. CONTRACT/GRANT NO. 68-02-2160
12. SPONSORING AGENCY NAME AND ADDRESS EPA, Office of Research and Development Industrial Environmental Research Laboratory Research Triangle Park, NC 27711		13. TYPE OF REPORT AND PERIOD COVERED Task Final; 9/78-7/79
		14. SPONSORING AGENCY CODE EPA/600/13

15. SUPPLEMENTARY NOTES IERL-RTP project officer is Joshua S. Bowen, Mail Drop 65, 919/541-2470.

16. ABSTRACT The report gives results of an environmental assessment of combustion modification techniques for stationary internal combustion engines, with respect to NOx control reduction effectiveness, operational impact, thermal efficiency impact, capital and annualized operating costs, and effects on emissions of pollutants other than NOx. Currently available operational adjustments for NOx control can reduce emissions by about 40%, but significantly increase operating costs. The total annualized cost to control can increase the cost of power by 3-14%, due to additional fuel and maintenance requirements. Combustion modifications can reduce NOx emissions without significantly increasing CO and hydrocarbon emissions; however, the kinds and distribution of organic compounds emitted from stationary diesel engines are not well characterized, and therefore are of concern.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Pollution	Pollution Control	13B
Internal Combustion Engines	Stationary Sources	
	Combustion Modification	21K
Combustion Control	Environmental Assessment	21B
Assessments		14B
Nitrogen Oxides		07B

18. DISTRIBUTION STATEMENT Release to Public	19. SECURITY CLASS (This Report) Unclassified	21. NO. OF PAGES 141
	20. SECURITY CLASS (This page) Unclassified	22. PRICE

EPA Form 2220-1 (9-73)

United States
Environmental Protection
Agency

Industrial Environmental Research
Laboratory
Research Triangle Park NC 27711

Research and Development

EPA-600/S7-81-127 July 1982

EPA

Project Summary

Environmental Assessment of Combustion Modification Controls for Stationary Internal Combustion Engines

H. I. Lips, J. A. Gotterba, K. J. Lim, and L. R. Waterland

This report gives results of an evaluation of combustion modification techniques for stationary internal combustion (IC) engines, with respect to NO_x control reduction effectiveness, operational impact, thermal efficiency impact, capital and annualized operating costs, and effects on emissions of pollutants other than NO_x. Currently available operational adjustments for NO_x control can reduce emissions by about 40 percent, but significantly increase operating costs. The total annualized cost to control can increase the cost of power by 3 to 14 percent, due to additional fuel and maintenance requirements. Combustion modifications can reduce NO_x emissions without significantly increasing CO and hydrocarbon emissions for most engines. However, the kinds and distribution of organic compounds emitted from stationary diesel engines are not well characterized, and therefore are of concern.

This Project Summary was developed by EPA's Industrial Environmental Research Laboratory, Research Triangle Park, NC, to announce key findings of the research project that is fully documented in a separate report of the same title (see Project Report ordering information at back).

Introduction

With the increasing extent of NO_x control applications in the field, and expanded NO_x control development anticipated for the future, there is currently a need to ensure that: (1) the current and emerging control techniques are technically and environmentally sound and compatible with efficient and economical operation of systems to which they are applied, and (2) the scope and timing of new control development programs are adequate to allow stationary sources of NO_x to comply with potential air quality standards. With these needs as background, EPA's Industrial Environmental Research Laboratory, Research Triangle Park (IERL-RTP) initiated the "Environmental Assessment of Stationary Source NO_x Combustion Modification Technologies Program" (NO^x EA) in 1976. This program has two main objectives: (1) to identify the multimedia environmental impact of stationary combustion sources and NO_x combustion modification controls applied to these sources, and (2) to identify the most cost-effective, environmentally sound NO_x combustion modification controls for attaining and maintaining current and projected NO_2 air quality standards to the year 2000.

The NO_x EA's assessment activities have placed primary emphasis on major stationary fuel combustion NO_x sources (utility and industrial boilers, gas turbines, IC engines, and commercial and residential warm air furnaces); conventional gaseous, liquid, and solid fuels burned in these sources; and combustion modification controls applicable to these sources with potential for implementation to the year 2000.

This summary outlines the environmental, economic, and operational impacts of applying combustion modification controls to this source category.

Conclusions

Source Characterization

Stationary reciprocating IC engines are the second largest contributor of NO_x emissions from stationary sources in the U.S. Figure 1 shows that this source constituted about 19 percent in 1977. Because of this high level of NO_x emissions and their potential for control, stationary IC engines represent a priority source category for control evaluation in the NO_x EA.

Stationary IC engines can be classified into three characteristic size ranges: large bore, high power, low to medium speed; medium bore, high speed; and small.

Large bore engines (>75 kW/cyl) operate at lower speeds (usually less than 1000rpm) and burn three major types of fuel: diesel, natural gas, and dual fuel (mixture of diesel and gas). Natural gas engines are spark ignited, and diesel and dual fuel engines are compression ignited. Both two- and four-stroke models are in this size range, and the engine may be turbocharged, which usually increases efficiency. Typical heat rates are 9 to 11 MJ/kWh (8500 to 10,500 Btu/kWh). Typical industries using these large bore engines are municipal electric power generation, oil and gas pipeline transmission, and oil and gas production. In these industries, the engine is run continuously. Based on 1976 data, only about 1000 to 2000 of these engines are sold per year, with a total production value of $80 to $150 million (1976 dollars). Sales have generally been declining, although sales of diesel engines for electric power generation are up.

Medium power engines (7.5 to 75 kW/cyl) exhibit the greatest variety; some large units equal the power of large bore engines. However, where large bore engines produce high power output at low speeds due to their large displacement and consequently high power per cylinder, medium bore engines have lower power per cylinder and, therefore, more cylinders for the same engine horsepower. Fuels burned in medium power engines are typical mobile fuels, either diesel oil or gasoline, although there are a few (usually modified) natural gas engines of this size. These engines are used in miscellaneous industrial, commercial, nonpropulsive marine, and agricultural applications where shaft power is needed and electric motors cannot be used.

Small engines are mostly one- and two-cylinder engines of less than 40 hp. These engines are mostly diesel and gasoline, one- and two-cylinder models, with some four-cylinder models. Almost all have four stroke cycles and are usually air cooled. Small engines are used typically in generator sets, small pumps and blowers, off-the-road vehicles, and refrigeration compressors for trucks and railroad cars.

This report focuses on large bore engines since these represent the largest NO_x emitters in the category, and they are most amenable to combustion modification control.

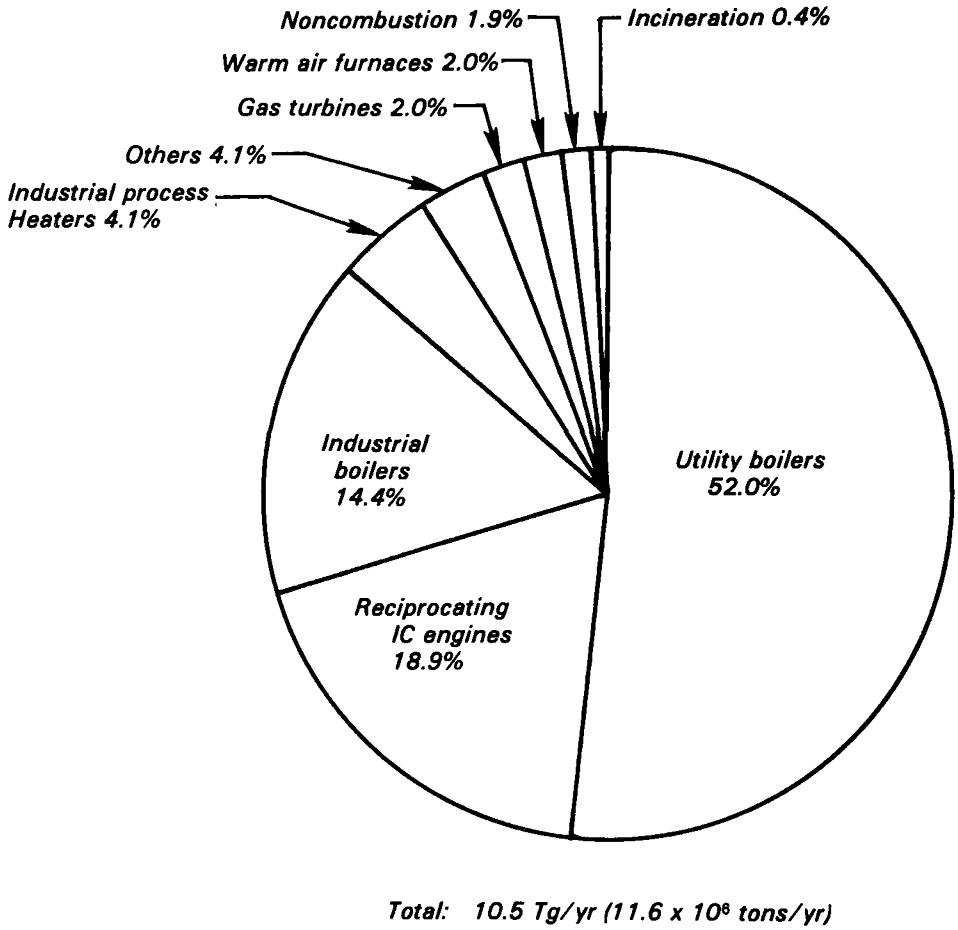


Figure 1. ***Distribution of stationary anthropogenic NO_x emissions for the year 1977 (controlled NO_x levels).***

Source Emissions

Air emissions in the form of exhaust gases are essentially the only effluent stream from stationary IC engines. Hydrocarbons (HC) can be emitted from the fuel before combustion, especially from natural-gas-fired engines, but these emissions are considered minor. There may also be some emissions from the crankcase caused by blowby, but this is also a minor source. The cooling system may release minor water pollutant emissions, and liquid wastes in the form of used crankcase oil may be another pollutant. Neither of these is a major release.

NO_x, CO, and HC are the major pollutants of concern in the exhaust gases from stationary IC engines. SO_x emissions are possible if the fuel burned has appreciable sulfur content, but this is rarely the case with the clean fuels burned in these engines. Particulate emissions are low from stationary engines. Diesel engines may also emit polycyclic organic matter (POM) at low levels, but even low level emissions of these compounds would be of concern because of their mutagenicity and potential carcinogenicity.

NO_x in IC engines, as in all combustion sources, is formed primarily by two mechanisms — thermal fixation and fuel NO_x formation. Thermal NO_x results from the termal fixation of molecular nitrogen and oxygen in the combustion air, and the rate of formation increases exponentially with local flame temperature. Fuel NO_x results from the oxidation of organically bound nitrogen found in certain fuels and primarily depends on the nitrogen content of the fuel. Since IC engines generally burn clean fuels, with correspondingly low nitrogen contents, thermal NO_x predominates.

Of the other pollutants, HC and CO are mainly the result of incomplete combustion. HC emissions are believed to be caused by three general mechanisms: wall quenching (fuel impingement on the walls causing the fuel to be cooled below the combustion temperature), variations in engine operation (mixing inside the cylinder, wrong air-to-fuel ratio, defective ignition, etc.), and, in two-cycle engines, cooling the exhaust gases by scavenging air before combustion is completed. CO emissions are also formed by the same general mechanisms.

Typical uncontrolled emission factors for IC engines are listed in Table 1. The HC emissions listed are total HCs; for natural gas engines, these are mainly methane. Although Table 1 lists factors for all engine sizes, this report focuses on the larger engines. Note that NO_x is the major pollutant for large engines.

Table 1. *Emissions Factors for IC Engines, g/kWh*[a]

Fuel		*NO_x*	*CO*	*HC*
Gasoline	*> 15 kW*	*11.9*	*137*	*11.2*
	< 15 kW	*7.5*	*395*	*27.5*
Diesel	*>375 kW*[b]	*17.3*	*2.4*	*0.6*
	<375 kW[c]	*16.6*	*6.0*	*2.8*
Natural gas		*15.4*	*3.8*	*6.5*
Dual Fuel		*11 0*	*2.7*	*4.1*

[a] *Emission factors for gasoline and diesel engines are modal averages; those for natural gas and dual fuel are for rated conditions. Modal averages mean that some of the NO_x numbers are taken from the constant power out portion of mobile tests.*

[b] *Based on an average of rated condition levels from engines considered.*

[c] *Weighted average of two- and four-stroke engines. Weighting factors = 2/3 for four-stroke and 1/3 for two-stroke.*

Control Alternatives

Since NO_x is the major pollutant emitted by stationary large bore IC engines, control development has focused on limiting NO_x emissions. There are three major approaches to controlling NO_x from IC engines: operational adjustment, combustion chamber redesign, and catalytic exhaust gas treatment. Operational adjustment techniques can be considered demonstrated and are finding current application. Combustion system redesigns are currently being developed and have seen, at best, laboratory scale testing. The use of catalysts to reduce NO_x emissions from lean-running engines (selective catalytic reduction) has seen only laboratory scale testing. Similarly, early limited testing of NO_x reduction catalysts for rich-running engines (non-selective catalytic reduction) has been performed.

The operational adjustment techniques are derate, ignition retard, air-to-fuel ratio change, reduced manifold air temperature, exhaust gas recirculation (EGR) (both internal-restricting the exit of exhaust gases from the cylinder, and external-reintroducing exhaust gases into the intake manifold), and water injection. All these techniques essentially act to lower the peak combustion temperatures, thereby limiting thermal NO_x formation. These techniques can be seen used in combustion, although NO_x reductions are not always additive.

Combustion system redesigns have been aimed at improving cylinder mixing, enhancing combustion, or establishing some form of staged combustion. The first two allow efficient combustion to occur under leaner lower-temperature conditions. The third, in addition to lowering peak temperature, lowers oxygen availability at peak temperature.

For diesel engines, mixing can be improved by circumferential injection, chamber shape, or a variable area prechamber. Combustion in gas engines can be improved by torch ignition, multiple spark plugs, high energy spark, increased turbulence through swirl or "squish," or diesel fuel injection. Staged combustion techniques include divided chambers, open chambers, or degraded mixing for gas engines, and a prechamber or pilot injection for diesel engines.

Catalytic reduction is a flue gas treatment technique in which exhaust gas is passed over a reduction catalyst which reduces NO_x to NO_2. Nonselective reduction catalysts can be used with rich-running engines since very little oxygen exists in their exhaust. However, lean-running engines require selective reduction catalysts which further require injecting a reducing agent, ammonia, into the exhaust stream.

Table 2 lists the various combustion modifications that have been investigated for IC engines and shows the NO_x reduction and fuel penalties associated with these controls as a function of engine type.

Currently, the best demonstrated controls, the only ones sufficiently demonstrated to allow meeting the proposed IC new source performance standards (NSPS), are: (1) retarded ignition or retarded fuel injection, (2) air-to-fuel ratio changes, (3) increased manifold air, or (4) in combinations with the others. The best combination will be very engine dependent. But in general, retard is best for diesel-fueled engines, air-to-fuel ratio changes for natural gas, and either control for dual fuel. A 40 percent reduction in NO_x can usually be achieved without causing any major operational problems, but there are fuel consumption penalties.

For the future, combustion system redesigns have the potential for obtaining the same level of NO_x reduction (40 percent) but with lower costs and fuel penalty. For very low NO_x emissions, only catalytic reduction techniques show promise.

Table 3 compares the estimated annualized incremental costs of retard, air-to-fuel increase, and exhaust gas recirculation applied to various engines to those of the corresponding uncontrolled engines. Costs in Table 3 represent annualized costs in mills/kWh (assuming 8000 hours of operation per year) and are in 1978 dollars. Table 3 shows that ignition retard increases the total cost of power 6 to 7 percent, air-to-fuel increase increases power costs 3 to 7 percent, and EGR increases power costs 5 to 14 percent. Though not shown in Table 3, manifold air temperature reduction should only have a small cost impact, about a 1.5 percent increase in initial engine cost and an increase in the cost of power of about 1 percent. Derate is a viable technique only if spare power is available elsewhere. Though derating

Table 2. *NO_x Reduction and Fuel Comsumption Penalties for Diesel, Dual-Fuel, and Gas Engines*

Control Approach		Diesel		Dual Fuel		Natural Gas	
		% NO_x Reduction	ΔBSFC, %[a]	% NO_x Reduction	ΔBSFC, %[a]	% NO_x Reduction	ΔBSFC, %[a]
Derate	3%	—	—	—	—	<20	2
	6%	—	—	—	—	<40	3
	10%	—	—	<20	4	—	—
	20%	<20	4	—	—	—	—
	25%	5-23	1-5	1-33	1-7	5-90	2-12
Retard	2°	<20	4	<20	3	—	—
	4°	<40	4	<40	1	<20	3
	8°	28-45	2-8	50-73	3-5	8-40	2-7
Air-to-Fuel	2%	—	—	—	—	<20	2
	3%	—	—	<20	0	—	—
	5%	—	—	—	—	<40	7
	±10%	7-8	3	25-40	1-3	20-80	5-12
Manifold	311k(100°F)	7-15	0-2	18-37	0-1	28	0
Air	315k(107°F)	—	—	—	—	<20	0
Temperature	318k(113°F)	—	—	<20	1	—	—
Internal EGR		5	2	—	—	<20	5
		—	—	—	—	5-35	0-8
External EGR	10%	<20	1	20	1	<20	0
		33	1	—	—	33	0
Retard and Manifold		<20	1	<20	1	<20	3
Air Temperature		10-24	0-1	25	2	30-40	5-6
Retard & Air-to-Fuel		<20	8	<20	1	<20	4
		<40	16	<40	2	<40	8
		35-65	5-26	56	2	17-52	4-11
Retard and Manifold		20	0	<20	2	<20	2
Air Temperature and		—	—	40	3	<40	4
Air-to-Fuel		—	—	—	—	40-65	6-7
Air-to-Fuel and		<20	2	—	—	—	—
Manifold Air Temperature		20-30	3	—	—	—	—
Water Injection	50%	25-35	2-4	—	—	25-35	1-2
(H_2O/fuel ratio)	100%			—	—	60-75	2-5
Catalytic Reduction (Projected		50-80	0	50-80	0	50-80	0
Combustion Chamber Modifications	Increased Mixing	10-30	<5	20-40	<5	20-40	<5
(Projected)	Staged Combustion	10-30	0	10-30	0-7	10-30	0-2

[a] *Brake specific fuel consumption penalty.*

would increase fuel consumption and raise operating costs, specific figures are not given because of the difficulty in specifying highly site dependent costs.

In general, as shown in Table 3, the incremental initial capital costs of the available controls range from 0 to 5 percent of an uncontrolled engine's cost. However, the total annualized cost to control can increase the cost of power from an engine by 3 to 14 percent, the significant impact due to additional fuel and maintenance requirements.

By combining control techniques, it may be possible to achieve the same NO_x reductions with a smaller fuel penalty, or reduce NO_x levels more than could be achieved by each technique alone. Table 4 compares three different methods for reducing NO_x by 40 percent from a large bore diesel engine. For this case, there is a definite advantage to using combined controls since the two combined techniques, air-to-fuel ratio change and manifold air cooling (or air-to-fuel ratio change and retard), had a lower brake specific fuel consumption penalty (BSFC) than retard alone.

Cost estimates for combustion system redesign controls vary significantly due to the developmental state of these techniques. Estimates indicate that these redesigns will fall between 0.5 and 20 percent of the capital cost of a large engine, with 3 percent being typical. Operational and maintenance costs should increase very little because the goal of development is to keep BSFC changes negligible.

Operational Impacts of Controls

Since engines are currently optimized for minimum maintenance requirements and fuel use, any control technique which varies engine parameters from standard conditions will impact operation and maintenance. Some of these impacts have been well characterized, especially from control techniques which involve engine operational changes.

Table 3. *Annualized Control Costs for IC Engines*[a]

Typical Engine	Uncontrolled Engine Cost, Mills/kWh		Retard: Percent NO_x Reduction	Retard: Incremental Cost, mills/kWh	Air-to-Fuel Ratio Change: Percent NO_x Reduction	Air-to-Fuel Ratio Change: Incremental Cost, mills/kWh	External EGR: Percent NO_x Reduction	External EGR: Incremental Cost, mills/kWh
3000kW Diesel	Capital	6		0		0		0.3
(Electrical	Maintenance	5	20-30	1.6	20	0.2	20	2
Generation)	Fuel	32		1.2		3.1		0
	Total	43		2.8		3.3		2.3
3000 kW Dual Fuel	Capital	6		0		0		0.3
(Electrical	Maintenance	5	20-30	1.6	40	0.2	20	4
Generation)	Fuel	20		0.7		0.4		0
	Total	31		2.3		0.6		4.3
3000 kW Natural Gas	Capital	6		0		0		0.3
(Gas Transport)	Maintenance	5	20-30	1.6	40	0.2	20	4
	Fuel	22		0.8		0.4		0
	Total	33		2.4		0.6		4.3
750 kW Natural Gas	Capital	2		0		0		0.1
(Gas Production)	Maintenance	5	20-30	1.6	40	0.2	20	4
	Fuel	25		0.9		0.4		0
	Total	32		2.5		0.6		4.1

[a]Assumes 8000 hours of operation per year, 1978 dollars.

Table 4. *Estimated Incremental Cost of Combined Controls for a Large Bore Diesel Engine at 40 percent NO_x Reduction*

Incremental Annualized Control Cost,[a] mills/kWh	Retard	Air-to-Fuel Changes and Manifold Air Cooling	Air-to-Fuel and Retard
Capital	0	0.1	0
Maintenance	1.6	1.0	1.8
Fuel	2.4	1.5	1.5
Total	4.0	2.6	3.3

[a]Assumes 8000 hours in operating year, 1978 dollars.

Other control techniques will require various degrees of evaluation before impacts are clearly understood.

Derate, air-to-fuel ratio changes, manifold air cooling, and ignition retard present the fewest problems in operation and maintenance. Derate has no impact mechanically and can improve durability because of lower operating temperatures and pressures. However, additional engines may be required to replace the lost power. Fuel penalties are usually low.

Air-to-fuel ratio changes in the lean direction cause a power loss if larger blowers or turbochargers must be used. If the engine must be operated richer to reduce NO_x emissions, other emissions (e.g., smoke, CO, and HC) can increase. This could cause an attendant increase in engine maintenance. Changes in air-to-fuel ratio in either direction will also generally increase fuel consumption. Finally, operation at air-to-fuel ratios where misfiring or detonation occur can cause severe engine damage.

Increased manifold air cooling has little operational or maintenance impact for a unit that is already intercooled, but will increase the size of the heat exchanger, water or air pump, control system, and other system components. Of course, backfitting intercooling on an engine will add maintenance attendant to additional temperature reductions. Changes in fuel consumption are small.

When properly applied, ignition retard has no serious mechanical drawbacks. Some increase in operational and maintenance time would be needed to ensure that the degree of retard is always within safe limits. Increases in fuel consumption are moderate. Excessive amounts of retard, however, can create severe engine problems. Fuel consumption increases rapidly, power drops, misfiring can occur, and smoke levels increase. In addition, mechanical maintenance will increase if the exhaust temperature exceeds the safe limits for valves or the turobcharger (usually 920 K-1200°F). More frequent engine teardown will be required, and higher initial costs will result for higher temperature materials.

Exhaust gas recirculation requires new hardware components which may require added maintenance, Problems of fouling the flow passages of the cooling heat exchanger, the engine turbocharger, and the aftercooler with particulate must be solved, or frequent engine teardown will be required. Under varying load conditions a sophisticated control system is required or the engine may stall or emit unacceptable smoke levels. Fuel consumption penalties with EGR are small.

Water injection can cause severe maintenance problems. Deposits from untreated water can build up on internal engine surfaces, and also foul the lubricating oil. The problem can lead to major engine maintenance. Water injection also adds another system to the engine which must be maintained and controlled.

Although not demonstrated, combustion system modifications are expected to present the least impact to operation and maintenance. Maintenance requirements can be expected to increase slightly if additional injectors, spark plugs, and valves are added to the chamber. However, because this control technique involves new design, many of the additional maintenance requirements can be designed out. Fuel penalties are expected to be small.

Catalytic reduction will require no additional engine maintenance, since it is a flue gas treatment technique, rather than an engine modification. However, operating the catalyst system may be expensive. Fouling the catalytic surfaces with particulate may require frequent regeneration. The catalyst may also have a relatively short life and need to be replaced. Another system, ammonia injection, must be included in engine operation. Finally, harmful products of the reaction may be produced if the catalyst temperature varies from the proper level, or if excessive ammonia is injected. The catalyst must be installed and operated on an engine before all these effects can be quantified.

Currently available operational adjustment NO_x controls can only reduce emissions by approximatley 40 percent, while significantly increasing operating cost and maintenance. Advanced combustion chamber redesigns have the potential of achieving similar NO_x reductions but at lower cost and smaller fuel penalty. If very low NO_x emissions are required, catalytic exhaust gas treatment is the only developing technique with that potential.

Table 5 lists achievable control levels and associated control techniques and costs for typical diesel, natural gas, and dual fuel engines, all assumed to be turbocharged. In the case of natural gas and dual fuel engines, the obvious preferred approach from a cost-effectiveness view would be to go directly to the more stringent control level with air-to-fuel adjustment. Note that all values discussed are typical, and may vary from engine to engine.

Combustion modification controls can reduce NO_x emissions without significantly increasing CO and HC emissions from most engines. However, the kinds and distribution of organic compounds emitted from diesel engines are not well characterized and, therefore, are of potential concern.

Table 5. Projected Control Requirements and Costs for Alternate NO_x Emission Levels

Type	NO_x Emission, g NO_x/kWh output	Control Techniques	Control Cost, mills/kWh output
Diesel	17	Baseline	—
	14	Exhaust gas recirculation	2.3
	12	Retard	2.8
	10	A/F increase + retard	3.3
Natural Gas	15	Baseline	—
	12	Exhaust gas recirculation	4.3
	11	Retard	2.4
	9	A/F increase	1.0
Dual Fuel	11	Baseline	—
	9	Exhaust gas recirculation	4.3
	8	Retard	2.3
	7	A/F increase	1.0

Recommendations

There are two major weaknesses in the data base for combustion modification controls on IC engines. The information on operational effects and long-term durability of these control techniques is incomplete, especially concerning combustion system redesign and catalytic exhaust gas treatment. Information on combining these controls to achieve an optimum of low emissions other than NO_x, CO, and total HC is very limited. The amounts and types of organics emitted from these large bore engines are not very well characterized. The potential mutagenicity of organic emissions in diesel exhaust is of major concern.

Research is needed on designing a high efficiency low-NO_x emitting engine. Even with the best available control applied, the large bore stationary reciprocating IC engine is the highest NO_x emitter on a heat input basis of all major combustion sources.

EPA is currently sponsoring several programs in the health effects area as well as new engine designs for low-NO_x and high efficiency. These programs should help resolve many of the major data gaps in the operational and environmental impacts of NO_x controls.

H. I. Lips, J. A. Gotterba, K. J. Lim, and L. R. Waterland are with Acurex Corporation, Energy and Environmental Division, Mountain View, CA 94042.
***J. S. Bowen** is the EPA Project Officer (see below).*
The complete report, entitled "Environmental Assessment of Combustion Modification Controls for Stationary Internal Combustion Engines," (Order No. PB 82-224 973; Cost: $13.50, subject to change) will be available only from:
National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161
Telephone: 703-487-4650
The EPA Project Officer can be contacted at:
Industrial Environmental Research Laboratory
U.S. Environmental Protection Agency
Research Triangle Park, NC 27711

*US.GOVERNMENT PRINTING OFFICE:1982-559-092-423