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Air Pollution Emissions and Control Technology: Thermal Power Generation Industry. Vol. 1 Internal Combustion Engines

Economic and Technical Review
Report EPS 3-AP-76-5

Air Pollution
Control Directorate
November 1977

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AIR POLLUTION EMISSIONS AND CONTROL TECHNOLOGY: THERMAL POWER GENERATION INDUSTRY. VOL. I – INTERNAL COMBUSTION ENGINES

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Report EPS 3-AP-76-5
November 1977

Cat. No. En 42-3/76-5
ISBN 0-662-01587-8

ERRATA

Report EPS 3-AP-76-5

Air Pollution Emissions and Control Technology: Thermal Power Generation Industry.

Vol. 1 - Internal Combustion Engines

Page 3, line 10: *for* = *read* <

Page 54, line 33: *for* dual-fired *read* dual-fuel

Page 62, line 8: *for* Table 6 *read* Table 4

Page 79, line 6: *for* (lb/Bhp-h) *read* (Btu/Bhp-h)

Page 80, line 4: *for* (lb/Bhp-h) *read* (Btu/Bhp-h)

ABSTRACT

The fundamentals of combustion and emission formation in both reciprocating stationary engines and stationary gas turbine systems are delineated. These fundamentals are used to examine the emission control techniques which have been applied to control oxides of nitrogen, hydrocarbons, aldehydes, carbon monoxide and smoke emissions. In many cases the reader is referred to cited literature for further detail.

Because experience shows that emission control tends to be unique to individual engine types, ready-made solutions are not given. Instead, the underlying principles of emission formation are presented, together with various control methods so that the reader may bring them to bear upon individual emission problems.

RÉSUMÉ

Le présent rapport esquisse les principes de base intervenant dans la combustion et la formation d'émissions entretenues par les moteurs alternatifs fixes et les machines fixes mues par des turbines à gaz. Ces principes font l'objet de rappels constants tout au long de l'exposé des techniques de lutte contre les émanations d'oxyde azoteux, d'hydrocarbures, d'aldéhydes, de monoxyde de carbone et de fumées. Dans de nombreux cas, l'ouvrage renvoie le lecteur aux travaux cités en annexe.

L'expérience semble démontrer qu'il y a une technique antipolltuion unique convenant à chaque type de moteur; le rapport ne préconise aucune solution toute faite. Il tente plutôt d'initier le lecteur aux principes et aux méthodes de lutte pour qu'il soit en mesure de solutionner lui-même ses problèmes de pollution.

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1.1 Purpose

The purpose of this report is to provide the technical information necessary for the development of effective national emission guidelines relating to air pollution from operations associated with the generation of thermal electric energy in Canada.

This entails the identification and classification of types of stationary internal combustion engines associated with the generation of thermal electric energy, and the acquisition of detailed information on two such engine types, namely, gas turbines and reciprocating engines.

1.2 Scope

The scope of the study was specifically defined as follows:

a) To develop a comprehensive list of stationary internal combustion equipment;
b) To analyse gas turbines and reciprocating engines in detail and classify the various types and subtypes according to such factors as combustion processes, fuels burned, and size, and to include in the analysis a discussion of the relative importance of the processes as emission sources;
c) To determine typical emission factors for gas turbines, diesel engines, dual-fuel engines and gas engines;
d) To determine potential emission control techniques;
e) To determine existing emission regulations affecting stationary internal combustion engines;
f) To determine existing exhaust emission measurement procedures for stationary internal combustion engines;
g) To determine installed capacity, power generation, and emissions from Canadian power generation facilities.

2 ENGINE TYPES AND COMBUSTION PROCESSES

The electric utility industry uses gas turbines and reciprocating engines for both continuous and peaking power service. The emphasis is on peaking power in large utility companies and on continuous power in smaller municipal utilities.

2.1 Reciprocating Engines

Stationary reciprocating engines can be classified into several categories depending upon the method of ignition of the air-fuel mixture, number of strokes per cycle, and methods of air and fuel charging. Table 1 summarizes the various alternatives.

TABLE 1 TYPES OF STATIONARY RECIPROCATING ENGINES

	Engine type		
	Gaseous fuel (i.e. gas engine)	Diesel	Dual fuel
Ignition type	Spark	← Compression →	
Fuel	Natural gas	Diesel oil	Natural gas 95% + Diesel oil 5%
Strokes/cycle	← 2 or 4 →		
Air charging:			
2-cycle	← Scavenged or Supercharged →		
4-cycle	← Naturally Aspirated or Supercharged →		
Fuel charging	Carburation	Direct injection or precombustion chamber	Direct injection and carburation
Engine speed	← High or Low →		

The air-fuel mixture is ignited by either an electric spark discharge (spark-ignition engines) or compression heating (diesel engines). Either two or four strokes per cycle are used. Air is introduced by natural aspiration, an air blower, supercharging, or turbocharging. Fuel is introduced by carburation or direct injection into the cylinder. Fuels include natural gas, distillate fuel oil, residual fuel oil, and to some extent crude oil.

2.1.1 Diesel Engines. Regardless of design, all diesel engines operate on the compression ignition principle in which air is compressed and liquid fuel is injected under high pressure. The high-temperature mixture ignites spontaneously, resulting in power output from the engine. Both two-stroke (one power stroke per cylinder for each engine revolution) and four-stroke (one power stroke per cylinder for every two revolutions) engines are used. Diesel engines are either naturally aspirated or supercharged. In naturally aspirated engines, air is sucked in from the atmosphere by pistons without external assistance and the amount of air taken in depends on engine speed and volumetric efficiency. In supercharged engines, a blower or air compressor is used to increase the amount of air induced per engine stroke.

The amount of fuel injected determines the power output. During idle, very little fuel is needed, but at high load and rated speed more fuel is required. It is convenient to think of the diesel in terms of the air/fuel ratio. The air/fuel ratio can be expressed in terms of either weight or volume (gaseous fuels) and expresses the relative fractions of air and fuel present in the mixture burned in the cylinders of reciprocating engines and in the combustors of gas turbines. A more meaningful parameter is the equivalence ratio, λ, which is the ratio of the actual and stoichiometric air/fuel ratios:

$$\lambda = \frac{(A/F)\ \text{actual}}{(A/F)\ \text{stoich.}} \qquad \text{(Equation 1)}$$

For $\lambda = 1$, the air–fuel mixture is lean.

An air/fuel ratio of about 100:1 is present during idle in diesels, but at high power output the ratio is closer to 20:1. When the air/fuel ratio is about 15:1, the chemically correct amounts of air and fuel are present for complete combustion. The ratios in diesel engines are greater than this, and the diesel operates air rich.

Classification of diesel engines according to combustion chamber design is based upon the method used to control the combustion process.

In *Open Combustion Chambers,* (direct injection) fuel is usually injected through a multiple-orifice nozzle directly into the clearance space between the piston and the cylinder head. The piston head is usually conformed to fit the fuel spray.

Precombustion chamber engines are so called because they incorporate a volume, small relative to the main combustion chamber, the two being connected by a duct. Fuel is injected into the prechamber rather than into the combustion chamber as in direct-injection engines. After initial ignition in the prechamber, combustion continues into the main chamber. Many modifications of divided combustion chambers are available. This system is analogous to the stratified-charge-combustion system being considered for control of automotive NO_x (oxides of nitrogen) emissions. Indeed, one advantage of the precombustion chamber system is that emissions of smoke, NO_x, carbon monoxide (CO) and exhaust odour are lower.

2.1.2 ***Dual-Fuel Engines.*** Dual-fuel engines can operate on either 100% fuel oil or a mixture of natural gas and fuel oil (usually up to 95% natural gas on the basis of heating value). The fuel oil serves as a pilot for ignition of the gas, which is difficult to ignite by compression heating alone. Dual-fuel engines have at least two advantages over spark-ignition gas engines and full diesel engines - greater fuel flexibility and lower fuel consumption.

2.1.3 ***Gaseous Fuel Engines (Gas Engines).*** In four-cycle engines using gaseous fuel, gas is mixed with air in a carburetor and passed into the cylinder through an intake valve.

2.1.4 Four-Stroke Engines. In four-stroke engines, the fuel is either mixed with air in a carburetor and passed into the cylinder through an intake valve or it is injected directly into the cylinder. Four-stroke engines can be naturally aspirated, i.e., the air-fuel mixture or air alone is driven into the engine by the natural pumping action of the cylinders. Supercharging and turbocharging are used to supply air to the engine above atmospheric pressure and increase the power output. The turbocharger is powered by an exhaust-driven turbine, while the supercharger is driven by the engine crankshaft.

2.1.5 Two-Stroke Engines. In two-stroke engines, the fuel is either injected into the cylinder, as in four-stroke engines, or directed through intake ports in the cylinder wall which are uncovered as the piston nears the bottom of its stroke (carburetion). The charge of air is compressed in a separate crankcase compartment for each cylinder, or is compressed by a compressor or blower, to a few pounds above atmospheric pressure. Intake ports uncovered by the piston are open soon after the opening of the exhaust, and the compressed charge flows into the cylinder, expelling most of the exhaust products, some charge escaping with the exhaust.

Two-stroke engines can be either 'uniflow' or 'loop' scavenged. In the former, incoming air dilutes the exhaust gases and the mixture exits through an open exhaust valve in the cylinder head. In loop-scavenged engines, the scavenging air-exhaust mixture leaves through exhaust ports in the cylinder wall. A ridge on top of the piston causes air to loop through the cylinder and sweep out the exhaust gases. As a result of exhaust scavenging, exhaust pollutants are diluted 50% - 70% of their original concentrations. Thus, in the case of two-stroke engines, the air/fuel ratio cannot be estimated directly from exhaust composition.

In some cases an exhaust turbine is used to supercharge the engine.

2.2 Gas Turbines

The gas turbine operates by burning a mixture of fuel and compressed air in a manner that forces the hot combustion gases to expand through a series of turbine fans which drive both the compressor and the power output shaft. The combustion gases, at about 480°C or hotter, then exhaust to the atmosphere.

A typical stationary gas turbine, Figure 1, consists of three separate sections: the compressor, the combustion chamber and the turbine (powered by combustion gases from the combustion chamber).

A separate flame tube is contained within an external casing of the combustion chamber, Figure 2. Air is led into the primary zone through a system of holes and slots such that a recirculating flow is set up. This is necessary to preheat air and fuel to the required reaction temperature by means of the hot previously burned products. Fuel is usually injected by a nozzle designed to produce a fine spray of fuel droplets. Combustion in the primary zone occurs at about 1900°C. Some of the air is injected at intervals along the flame-tube walls for cooling. The remaining air is admitted to dilute the combustion products down to a temperature which the turbine blades can tolerate.

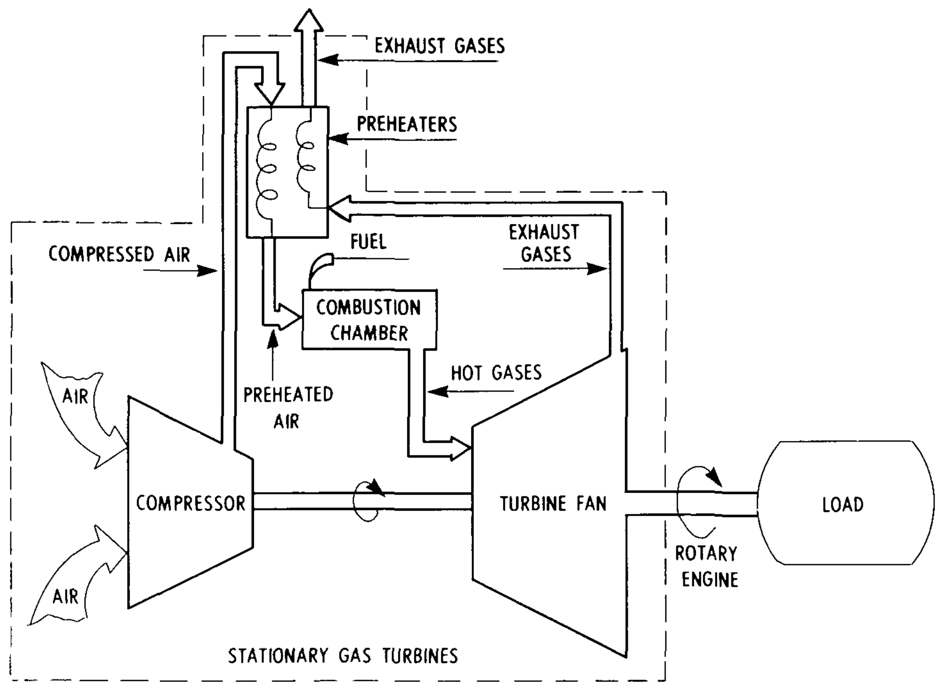


FIGURE 1 REGENERATIVE CYCLE GAS TURBINE SYSTEM

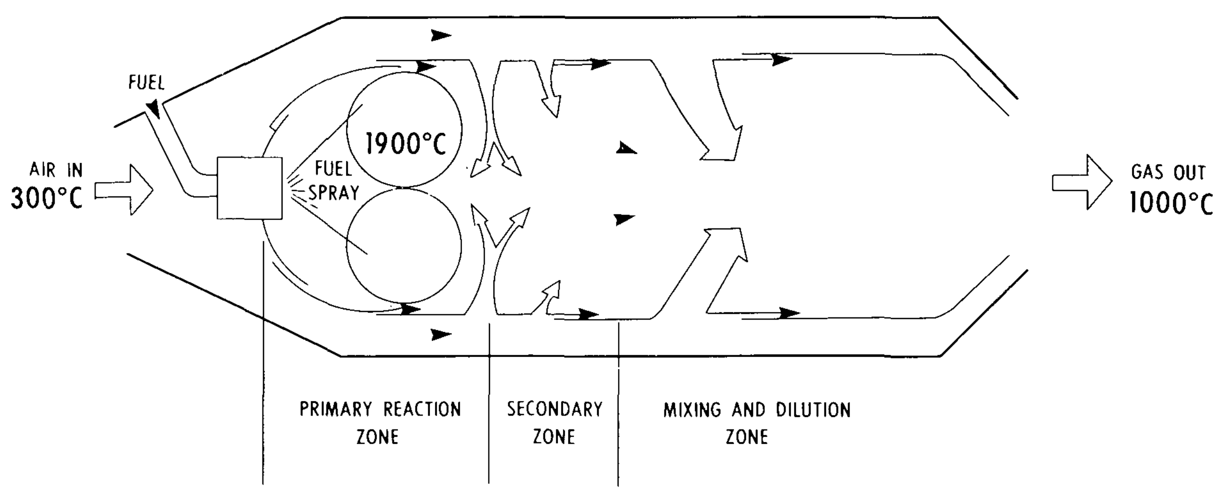


FIGURE 2 GAS TURBINE COMBUSTION CHAMBER

For most turbines currently in use, this is in the range of 800° - 1000°C. To increase thermal efficiencies, continuing efforts are being made by turbine manufacturers to increase this temperature. The short-term objective is 1100° - 1200°C and the long-term, 1600°C. Hence, overall air/fuel ratios are high, ranging from 50:1 - 100:1; too high in fact for combustion to take place, so that in the combustion chamber, the airflow from the compressor is split into two major parts. Approximately one third is led into the primary zone where combustion takes place at near stoichiometric conditions (15:1).

Gas turbine compressors may be high-efficiency models. Axial-flow compressors are used on all large gas turbine units because of their high efficiency and capacity. They have been constructed to handle airflows up to 8000 m^3/min. The energy absorbed by the compressor is two to four times the net output of the plant.

The various gas turbine cycles and types can be constructed for single- or multiple-shaft arrangements. Single-shaft machines are the simplest, with all rotating elements operating as a single assembly. Multiple-shaft machines may have a separate output turbine in series or parallel, with one or more turbines driving the compressor. The latter may operate at any speed, independent of the output turbine. The range of thermal efficiency of gas turbines varies between 20% - 37% (1).

A relatively flat efficiency curve for the gas turbine can be obtained at a cost of added complexity, using regeneration and regulation of airflow by variable area turbine nozzles or variable supercharging.

Over half of the large industrial gas turbines are in electric-generating use. Electric utility companies in Canada use gas turbines primarily for peak-load duty. They are used for base-load electric generation where additional capacity is needed quickly; where refined fuel, such as natural gas, is available at low cost; or where turbine exhaust energy can be utilized. Combining the gas turbine with a steam power plant allows improved overall efficiency.

The major types of gas turbines can be described as follows:

Open-cycle turbine - simple. No exhaust heat recovery equipment is employed. This system includes a rotating compressor for pressurizing atmospheric air, a combustor where the compressed air is mixed with fuel and burned, and a turbine where the hot gases are expanded against the turbine blades to generate rotation. The expended hot gases are then exhausted directly to the atmosphere.

Open-cycle turbine - regenerative. This system employs a regenerator or heat exchanger, between the compressor and the combustor, through which the exhaust gases are circulated to heat air coming into the combustor (Figure 1). This, of course, uses some of the heat which would normally be exhausted, thereby reducing fuel consumption. Thermal efficiency is, consequently, increased 5% to 10% above that of the simple system.

Combined cycle. In this system the exhaust gases are routed directly from the turbine to a boiler where steam is generated for further production of power or for process uses. This increases thermal efficiency approximately 10% to 15% above that of the simple system.

The simple gas turbine is by far the most widely used. It requires the least investment and is the most flexible in application. The other types are less generally used, and are primarily employed where the usage requirements are very intermittent, e.g., for standby power. The combined-cycle turbine appears to be gaining acceptance among power utilities for base power utilities and extended peaking requirements, and it will probably become more widely used in the future (1).

Gas turbines range in size from below 800 kW to 80 000 kW. Capacities of the largest units manufactured have been increased, using the combined cycle, to above 80 000 kW. In the largest size category, however, the average size of units sold from 1968 to 1970 was between 25 000 and 30 000 kW (1).

The primary fuels used in gas turbines are natural gas and light and heavy distillate oil. Residual oils can be used; however, this depends upon the removal of certain contaminants in the fuel (primarily salts of light metals) and the inhibition of certain corrosive components of the fuel (primarily V_2O_5) There is also some use of waste gases from chemical processing as fuel.

It should be noted that certain industries are generally limited to certain types of fuel, e.g., the natural gas transmission industry uses primarily natural gas because of its availability and low cost. In many applications, gas turbines are equipped to use either natural gas or oil.

Because of the relatively low capital cost of gas turbines, general industrial use is continuing to expand, primarily for captive power generation (1). Geographic location of gas turbines used by power companies would follow the location of power plants in general, i.e., close to large metropolitan centres. Location of those used in the natural gas transmission industry would, correspondingly, be along the transmission lines of that industry, primarily in rural areas.

3 POLLUTANTS FROM DIESEL ENGINES AND GAS TURBINES

Exhaust emissions from diesel engines and gas turbines consist of the same type of pollutants emitted by other combustion systems operating on petroleum-based fuels. The level of exhaust emissions, however, differs from engine to engine, depending on engine design and operating parameters. Fuel preparation, distribution and composition are also factors. In many cases it is impossible to completely isolate the effect of a single design variable or operating parameter.

3.1 Formation of Combustion Emissions

Major emissions from diesel engines and gas turbines are those common to many combustion sources: oxides of nitrogen (NO_x), carbon monoxide (CO), sulphur dioxide (SO_2), hydrocarbons (HC), particulates, aldehydes, and odour.

The concentration of the different emission species in the exhaust is the result of their formation, and their further elimination or further formation in the combustion chamber and exhaust system. In general, further formation applies to NO_x and further elimination applies to incomplete oxidation products. The further formation and elimination reaction rates are functions of the oxygen (or oxidants) concentration, the local mixture temperature, mixing, and residence time. This process can be expressed as follows:

$$\begin{matrix}\text{Exhaust} \\ \text{Emissions}\end{matrix} = \begin{matrix}\text{Emission} \\ \text{Formation}\end{matrix} + \begin{matrix}\text{Further} \\ \text{Formation}\end{matrix} - \begin{matrix}\text{Further} \\ \text{Elimination}\end{matrix}$$

3.2 Oxides of Nitrogen (NO_x)

The precise mechanism of NO_x formation in a combustion chamber is not properly understood, and is the basis of a great deal of research. It is clear, however, that formation is related to the temperature in the combustion chamber. As a result, emissions are relatively low at idle, rising at maximum power.

The production of NO_x increases as the air/fuel ratio is increased, reaching a peak on the weak side of stoichiometric; the result of increasing temperature and increased availability of oxygen. This is unfortunate because this would be a desirable method for reducing the other pollutants.

The principal oxide of nitrogen formed during combustion is nitric oxide (NO). It is formed by the oxidation of nitrogen at high temperatures. Most oxides of nitrogen react in the atmosphere to form nitrogen dioxide (NO_2). In this study, all emission rates of NO_x are reported as though all the nitrogen present had combined to form NO_2.

3.3 Carbon Monoxide (CO)

Carbon monoxide is a colourless, odourless, highly toxic gas, frequently formed during oxygen-deficient combustion. The formation of CO is also dictated by reaction kinetics as in the case of diesel and gas turbine combustion processes. Fortunately, emissions from diesel engines and especially those from gas turbines contain only very small quantities of CO, due to the large quantities of excess air available. It is in regard to this pollutant, that the gas turbine has the most spectacular advantage over the piston engine.

3.4 Sulphur Dioxide (SO_2)

Sulphur dioxide is a stable compound which is not inflammable nor supportive of combustion. It is a colourless gas or liquid with a suffocating odour and combines with water to form sulphurous acid. In the presence of certain catalysts and oxygen, it is oxidized to sulphur trioxide (SO_3) and subsequently hydrated to sulphuric acid.

All hydrocarbon fuels found in nature contain some sulphur and when they are burned, SO_2 and SO_3 are formed. Because virtually all sulphur is converted to the oxide, emissions of sulphur oxides are a direct function of the sulphur content of the fuel. Diesel engines and turbines fired with low-sulphur fuels have negligible SO_2 emissions. The trend, however, is toward the utilization of poorer, less refined fuels which have increasingly high sulphur values.

Sulphur oxides cause plant damage and corrosion, as well as bronchial or asthmatic troubles. Sulphuric acid in the air causes corrosion of exposed metal structures.

3.5 Hydrocarbons (HC)

Unburned HC in diesel and gas turbine exhaust consist of either original or decomposed fuel molecules, or of recombined intermediate compounds. A small portion of these HC emissions originate from lubricating oil. The mechanisms of formation and oxidation of the HC molecules depend upon most of the engine operating variables, such as air/fuel ratio, turbocharging, injection timing, swirl, injection system design and load. As with CO, these products result from insufficient residence time in the combustion chamber and subsequent quenching of the reaction products either by the cooler mixing air, scavenging air or against the relatively cool walls of the combustion chamber. In brief, the foregoing HC emissions are all the result of incomplete combustion. Hydrocarbon emission from gas turbines is relatively low when compared with that from the diesel engine.

3.6 Aldehydes and Odour

Recent studies (2) show that aldehydes probably do not contribute significantly to the characteristic diesel odour. The total intensity of the odour is described by two dominant odour-character groups, 'smokey-burnt' and 'oily-kerosene', each contributing equally to the odour. 'Oily-kerosene' is comprised principally of alkyl benzenes, a group of indans and tetralins, and the major naphthalene components. The 'smokey-burnt' odour character is most consistently associated with hydroxy and methoxy indanones with some contributions from methyl and methoxy phenol.

It is estimated that about 200 different distinct chemical species may be responsible for the oily-kerosene odour, while approximately 2000 species contribute to the smokey-burnt odour. The major source of the most significant odour contributors appears to be the aromatic portion of the fuel, although some contribution is also made by the paraffin portion. Odorous substances, even in low concentrations, are very irritating to the eyes and to mucous membranes.

A very limited amount of work has been done to examine the effects of fuel and engine variables on the odour of diesel exhaust. Odorous emissions from gas turbines appear to be comparable to those from piston engines. At this time, little effort is being made to produce effective means of eliminating odours from diesels or gas turbines.

3.7 Particulates (Smoke)

Smoke from diesel engines and gas turbines was the first emission to come to public notice and has received the most attention to date.

Different types of particulates causing smoke are emitted from the combustion chamber under different operating conditions. These particulates can be divided as follows:

(a) Liquid particulates appear as white clouds of vapour emitted under cold starting, idling, and low-load conditions. These consist mainly of fuel and a small portion of lubricating oil, emitted without combustion; they may be accompanied by partial oxidation products.

(b) Soot or black smoke is emitted as a product of incomplete combustion, particularly at maximum loads.

(c) Other particulates include unburned lubricating oil and fuel additives.

Black smoke emissions consist mainly of irregularly shaped, agglomerated, fine carbon particles. Consideration was limited to these dry particulates, because the large majority of engine emission data is confined to this topic. According to current theories, these carbon particulates can be formed from hydrocarbon fuels in the presence or absence of oxygen depending on the temperature and the ratio of CO and CO_2 concentrations.

The smoke intensity is affected by many parameters including fuel composition, injection timing (in diesels), rate of injection, injection nozzle design, inlet air temperature and pressure, after-injection (in diesels), and insufficient mixing of air and fuel.

Once carbon is formed, it does not burn readily and is ejected from the engine as smoke. Gas turbine smoke is pure carbon (3), unlike diesel smoke which contains known carcinogens (e.g. 3.4 benzopyrene). Although very visible, the actual quantity of particulate matter is very small. Generally, smoke is a sign of poor fuel economy, or engine malfunction.

3.8 Ice Fog

In extremely cold regions, local 'ice fog' can form in areas of saturated air. Ice fog, tiny ice crystals nucleated by airborn particulates, severely restricts visibility and plagues automobiles and air traffic. It occurs naturally over lakes and rivers and in the exhaust of automobiles, power plants and other sources. The occurrence of ice fog is difficult to predict.

4 EMISSION FACTORS FOR GAS TURBINES AND DIESEL, DUAL-FUEL, AND GAS ENGINES

In this section the emission factors for estimation of total NO_x, CO, HC, SO_2 and particulate emissions from stationary engines used by Canadian electric power generating stations are described. Before presenting the data, several important points which greatly affect the magnitude of pollutant emissions from engines are discussed. These include engine operating conditions at the time of emission tests, and measurement methods.

4.1 Effect of Engine Operating Conditions

Exhaust emissions can vary over a considerable range depending upon the condition of the engine, operating conditions, and various design factors. The most important operating conditions are the air/fuel ratio of the trapped charge mixture and the load of the engine. Other factors of less importance include ignition timing (reciprocating engines) and air temperature, pressure and humidity.

4.1.1 Air/Fuel Ratio. Data in Figure 3 demonstrate how the air/fuel ratio affects exhaust concentrations of NO_x, CO, and HC for four-stroke, gasoline-fueled laboratory engines. The NO_x concentration reaches a maximum for an air/fuel ratio slightly on the lean side of stoichiometric (S in Figure 3). Richer mixtures result in lower available oxygen concentrations, while leaner mixtures result in lower flame temperatures. Both factors result in less favorable conditions for the formation of NO_x. Carbon monoxide emissions are essentially functions of oxygen availability and thus are significant only for rich mixtures. Hydrocarbon emissions result principally from quenching of the combustion reaction at the cylinder wall and tend to be higher for both rich and very lean mixtures. The former effect results from lack of available oxygen for combustion and the latter from lean misfires under oxygen-rich conditions.

The effect of air/fuel ratio or equivalence ratio, (Section 2.1.1) on NO_x emission level in the gas turbine is shown in Figure 4, and is similar in reciprocating engines.

4.1.2 Engine Load. Varying load is the predominant factor causing emissions from reciprocating engines and gas turbines to vary with time. Laboratory studies suggest, however, that the effect is primarily one of simultaneous changes in the air/fuel ratio (4). Nevertheless, load has a primary effect on emission via its effect on combustion pressure and temperature which in turn affect the rate of formation of NO_x and the combustion of CO and HC.

Figure 5 shows the effect of load on the brake-specific mass emissions of NO_x for three Cooper Bessemer engines (4). The two spark-ignition gas engines (GMVA-8 and GMVH-8) show great sensitivity of specific NO_x emissions to load, while the diesel engine (KSV-12) is less sensitive to load in both the full-diesel and dual-fuel mode. Data from other sources show the same relative effects for gas and diesel engines (5). As load increases, the air/fuel ratios decrease and combustion temperatures increase in both gas and diesel engines. In the case of natural gas combustion, both effects tend to increase NO_x emissions. However, the two effects tend to cancel in liquid fuel combustion in diesel engines (5). The air/fuel ratio is already on the rich side of the peak NO_x settings, and further enrichment leads to lower NO_x formation. Thus, gas engines exhibit greater NO_x sensitivity to load than diesel engines.

Gas turbines are also known to exhibit NO_x sensitivity to load (4). Combustion intensity and temperature increase with load, leading to higher NO_x emission. However, airflow does increase with load. Thus, specific mass emissions of NO_x from gas turbines are less sensitive to load than those from gas engines.

At a given speed, power output is proportional to load. Hence, derating an engine, i.e., operating below rated load, would be expected to be an effective NO_x control method only in the case of gas engines. As load is reduced, the magnitude of diesel and dual-fuel NO_x emissions per unit power output does not change significantly. It is also significant that as load is reduced, HC and CO emissions both increase slightly in all reciprocating engines (4).

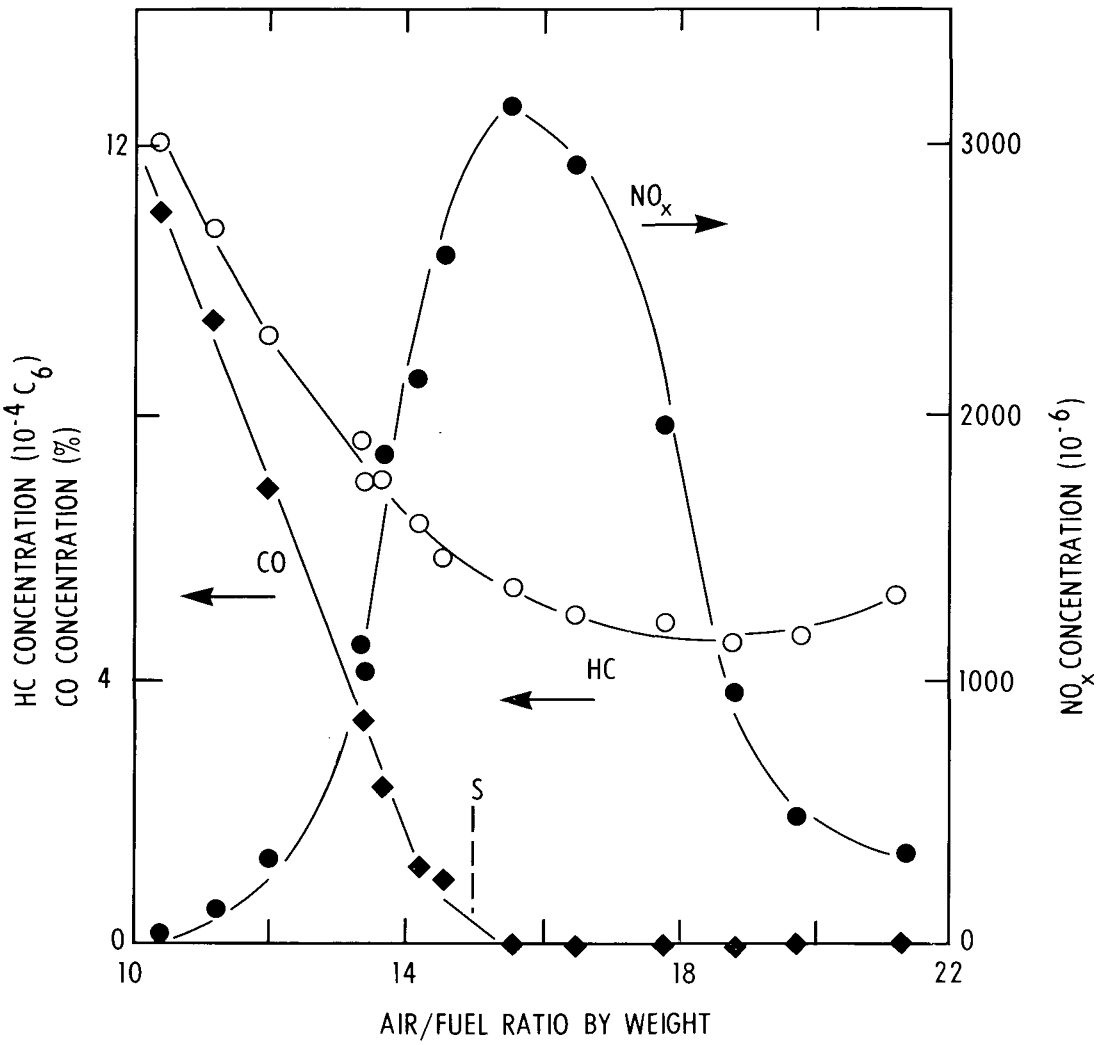


FIGURE 3 THE EFFECTS OF AIR/FUEL RATIO ON HC,CO, AND NO_x EXHAUST EMISSIONS

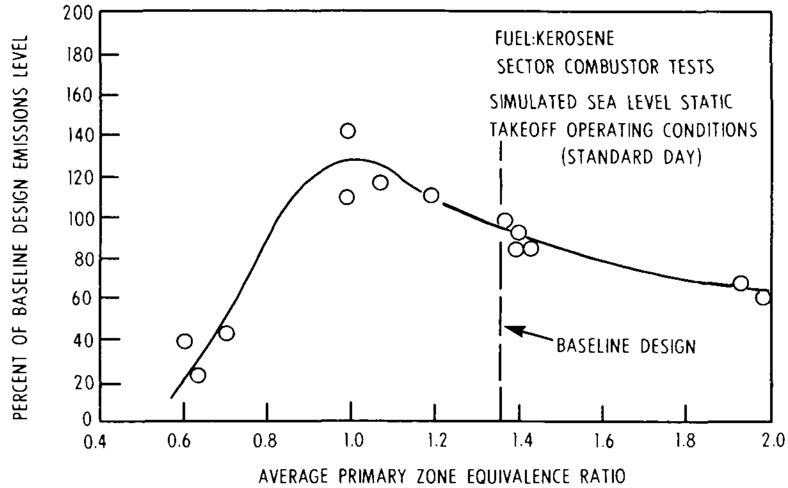


FIGURE 4 EFFECT OF PRIMARY COMBUSTION ZONE EQUIVALENCE RATIO ON NOx EMISSION LEVEL. EQUIVALENCE RATIO SPECIFIED AS $\frac{\text{A/F ACTUAL}}{\text{A/F STOICHIOMETRIC}}$

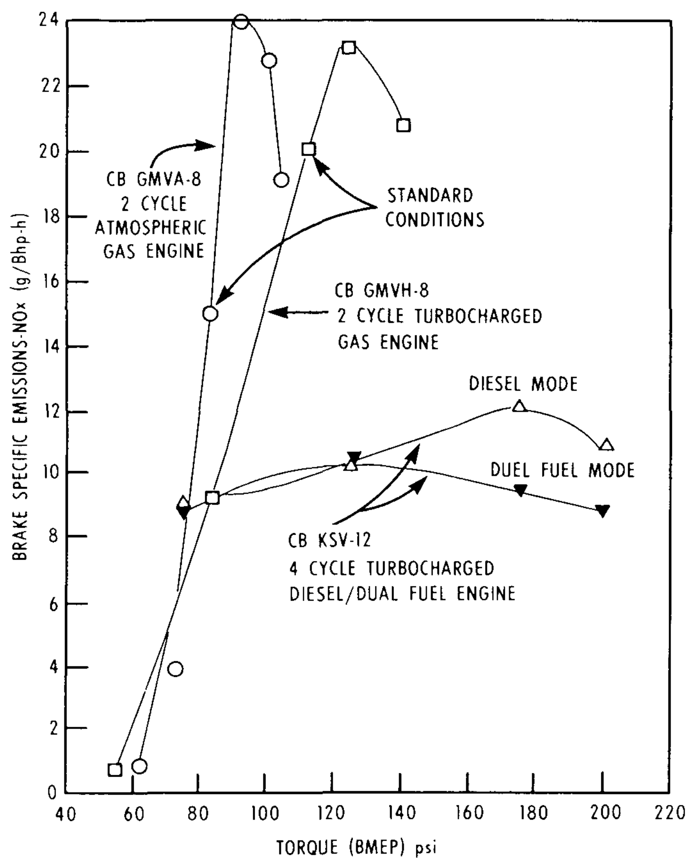


FIGURE 5 EFFECT OF LOAD ON SPECIFIC NOx EMISSIONS - COOPER BESSEMER GAS AND DIESEL ENGINES

4.2 Test Procedures

Considerable variations in exhaust emissions can be the result of seemingly minor differences in test procedures. In addition to the exhaust concentrations of pollutants, it is necessary to determine engine power output and exhaust flow rate in order to calculate specific mass emission rates (grams/brake horse power-hour). The latter requires accurate measurement of fuel flow rate and air flow rate or exhaust flow rate. Different analytical instruments may lead to different flow rates and emission concentration values. When the engine is driving additional equipment, it may be difficult to determine power output, which will introduce some uncertainty into the results. Likewise, different exhaust sample treatment and probe location will cause variations. There is, consequently, a great deal of uncertainty in the emission factors presented and they should only be considered order-of-magnitude estimates.

There has been some effort within the United States' engine manufacturing industry to standardize emission testing procedures for stationary engines. The Diesel Engine Manufacturers Association (DEMA) has developed an emission test code for stationary diesel engines (6). It is hoped that the DEMA Emissions Test Code will eliminate much of the variation found in emission test data.

4.3 Emission Factors

It is important to note that diesel and gas turbine emissions are most frequently measured over some kind of test cycle. Few measurements are based on constant speed or load. Unfortunately, the latter should be used in those cases where engines are run for long periods at constant speed, e.g., electric power generation.

The accuracy of exhaust emission factors is directly related to the number of field studies performed on the particular sources, and, in general, the most intensive field efforts have been concentrated on the major sources of pollution. As the emission factor is based on a statistical average of several studies conducted on a particular combustion engine, gross inaccuracies could arise when applying the factor to a specific engine. Field test results have shown that the apparent particulate emissions from two similar combustion processes, using the same fuel to produce the same power output, in the same engine, can vary by a factor as high as 5, depending on the technique used (7).

The emission factors in this study are composites of data collected from literature sources (see Appendix). Some emission results were published in standard mass form, i.e., g/Bhp-h; some were calculated from concentration form, i.e., ppm, using exhaust flow rate reported or estimated according to Section 7.1.

4.3.1 Method of Calculation. Average specific mass emission factors for each type and for each group of engines were calculated from full-load specific mass emission (SME) weighted by the number of tests included in the samples:

$$EF_1 = \frac{\Sigma \text{ SME}}{\text{No. of tests}} \quad \text{(g/kWh)} \ldots\ldots\ldots\ldots \text{(Equation 2)}$$

This approach is justified since the brake-specific results are based on mass flow rates that can be combined and averaged.

Emissions of sulphur oxides are a direct function of the sulphur content of the fuel.

Average specific emission factors related to fuel consumed are derived by dividing the mass emission factors by the fuel consumption and converting grams to pounds:

$$EF_2 = \frac{SME \times 10^6}{SFC \times 453.59} \quad (lb/10^6\ Btu) \text{..........................(Equation 3)}$$

$$EF_2 = \frac{SME \times 10^3}{SFC \times 453.59} \quad (lb/10^3\ lb\ fuel) \text{....................(Equation 4)}$$

Because particulate emission data are very limited, the soot emission factor in the study was calculated as follows:

One reference (8) states that 0.5% of full-load fuel input appearing as carbon gives a smoke opacity of 46%. Taking into account a correlation between opacity and soot density (9), and assuming an average opacity of 10% (from Figure 12 average opacity is 7.2%), an average fuel consumption of 0.40 lb/Bhp-h, and an LHV of diesel fuel of 18 400 Btu/lb, the resulting particulate emission factor will be 0.05 lb/10^6 Btu or 0.22 g/kWh.

Figures 6 to 12 and Tables 2,3 and 4 summarize the emission factors resulting from emission data in the Appendix.

Because there are currently no emission standards or emission units for stationary diesels and gas turbines, the most reasonable unit for estimation of emission rates from existing engines would be pounds of pollutants emitted according to some amount of fuel consumed, e.g., pounds of pollutants per 10^6 Btu heat input. It should be noted, however, that it is not suggested that this basis be used for emission standards, because it does not encourage development of more efficient engines (if variation in plant efficiency is not compensated by an efficiency factor).

Except for gas turbines, and precombustion chamber diesel engines, NO_x factors exhibit only minor differences among the different engine types, and NO_x factors are in the range 3.0 - 3.5 lb/10^6 Btu fuel burned. Gas turbines are one order of magnitude lower, primarily as a result of the lower peak temperatures in the combustion chamber. Emissions of NO_x are about 30% higher from oil-fired gas turbines than from gas-fired, (Table 4). Precombustion chamber-type diesels emit about half as much NO_x as direct-injection diesels, see Figures 6, 7 and 13. This is probably due to a stratified-charge combustion effect whereby combustion is initiated in a fuel-rich precombustion chamber.

Carbon monoxide emissions data also show only minor differences between direct-injection diesel engines and gas engines. Precombustion chamber engines generally show lower CO emissions,

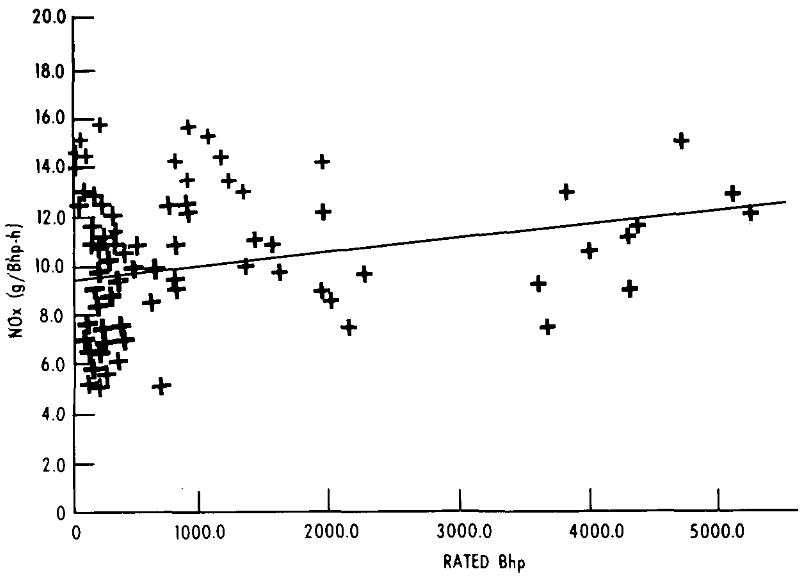


FIGURE 6 NOx EMISSIONS FROM DIRECT-INJECTION DIESELS AND GASEOUS FUEL ENGINES (146 ENGINES TESTED)

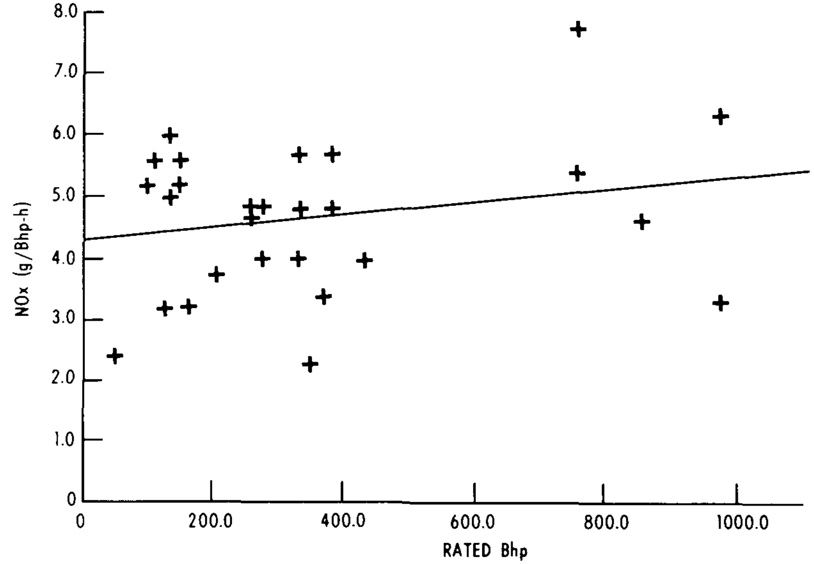


FIGURE 7 NOx EMISSIONS FROM PRECHAMBER DIESEL ENGINES (35 ENGINES TESTED)

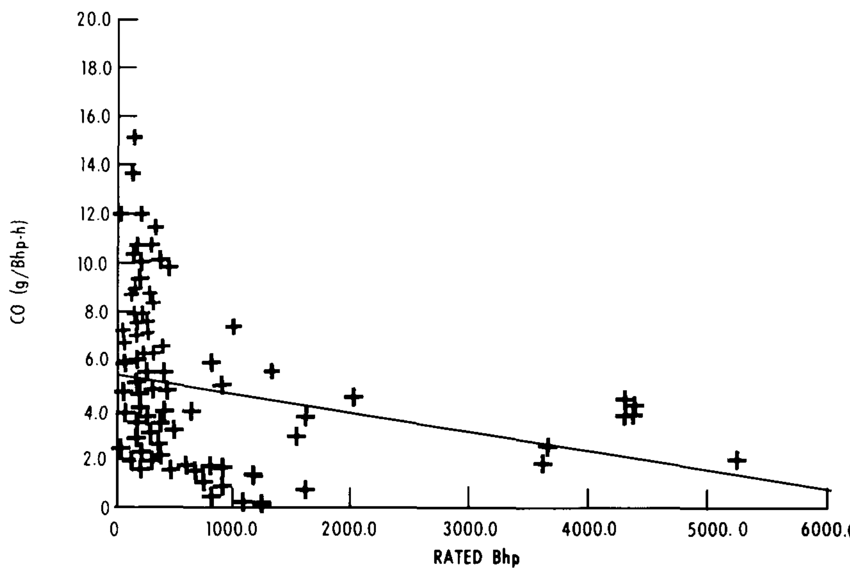


FIGURE 8 CO EMISSIONS FROM DIRECT-INJECTION DIESELS AND GASEOUS FUEL ENGINES (123 ENGINES TESTED)

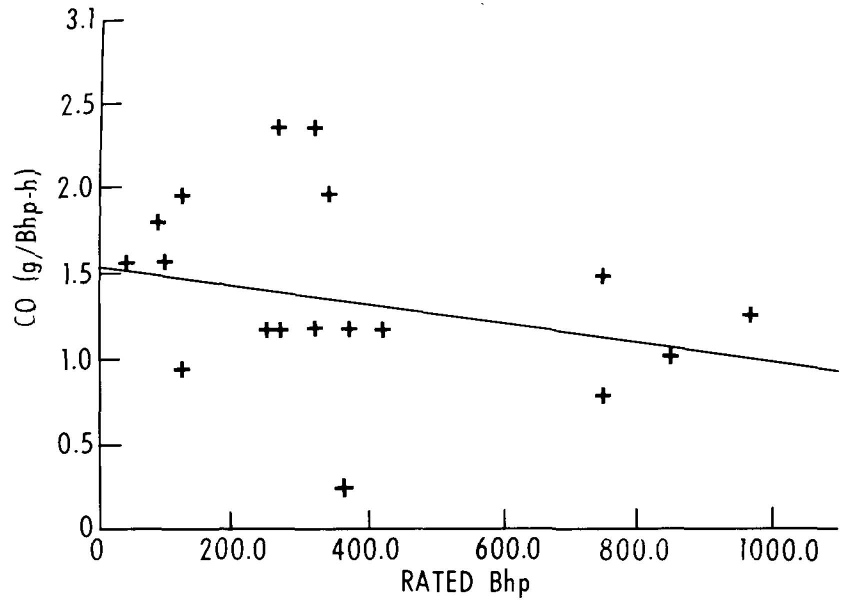


FIGURE 9 CO EMISSIONS FROM PRECHAMBER DIESEL ENGINES (26 ENGINES TESTED)

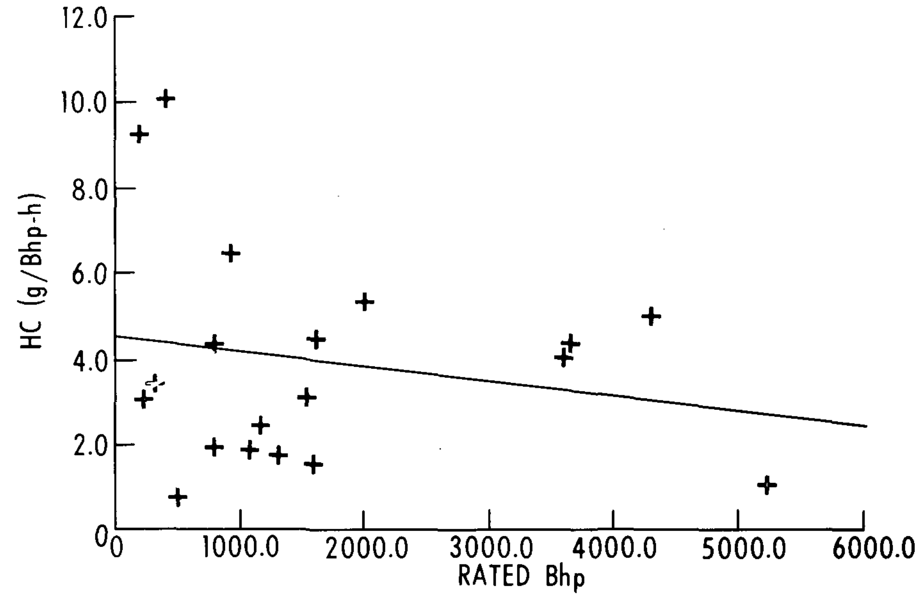


FIGURE 11 HC EMISSIONS FROM GASEOUS FUEL ENGINES (19 ENGINES TESTED)

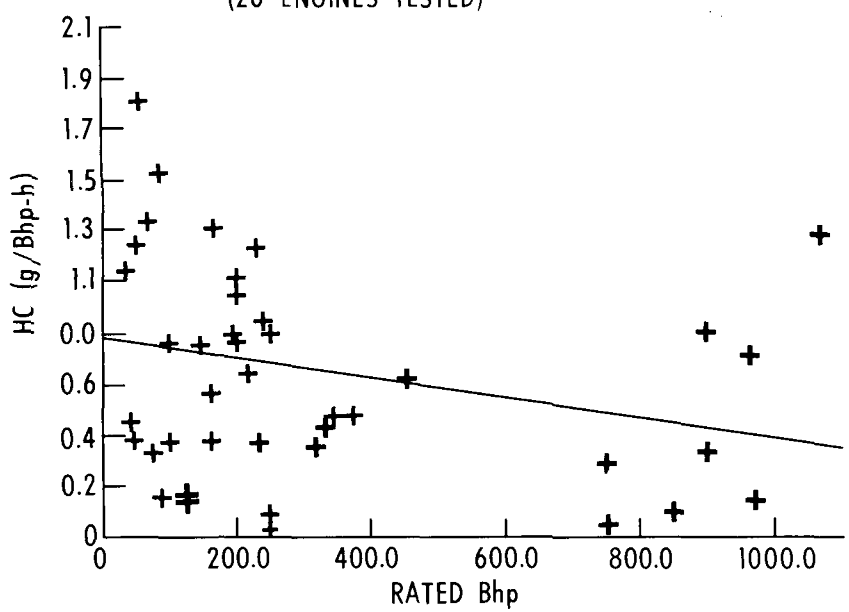


FIGURE 10 HC EMISSIONS FROM DIESEL ENGINES (44 ENGINES TESTED)

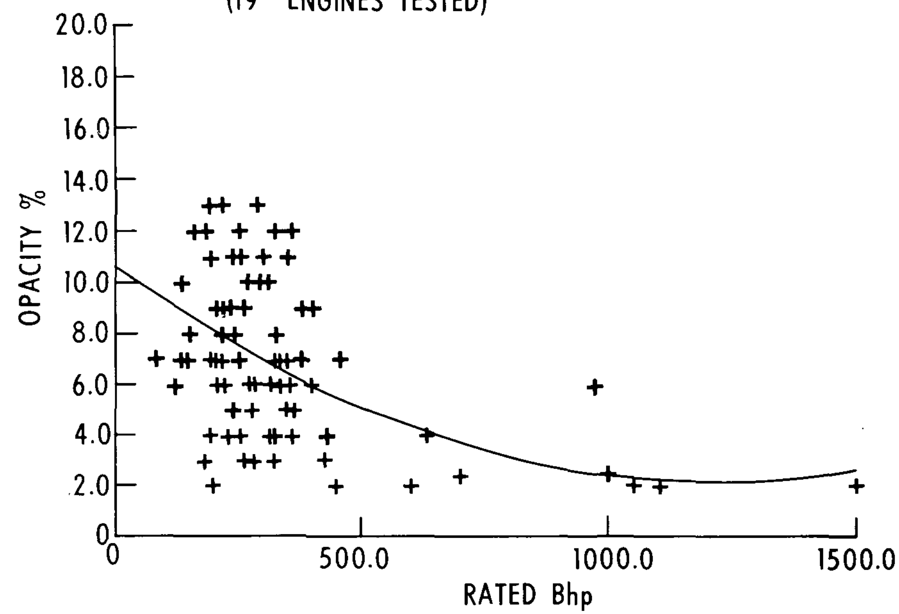


FIGURE 12 STEADY-STATE SMOKE OPACITY (%) FROM DIESEL ENGINES (94 ENGINES TESTED)

TABLE 2 EMISSION FACTORS RESULTING FROM THIS STUDY

Engine type	Emission factor $\frac{g}{kWh}$ NO$_x$	CO	HC	Emission factor $\frac{lb}{10^6\ Btu}$ NO$_x$	CO	HC	Particulates	Opacity %	Reference
Direct injection	13.13	6.59	0.91	3.25	1.64	0.22	–	7.2	
Gas engines	13.13	6.59	5.17	3.25	1.64	1.26	–	–	
Precombustion chamber diesel engines	6.24	1.82	0.91	1.33	0.39	0.22	–	7.2	Appendix
Gas turbines (oil)	5.35[x]	0.47[x]	0.05[x]	0.8	0.07	0.007	0.07	–	
Gas turbines (natural gas)	2.61[x]	–	–	0.39	–	–	–	–	

[x] Assumes: Specific fuel consumption 14.740 Btu/kWh

TABLE 3 EMISSION FACTORS (Comparison of different sources)

Engine type	Fuel	Emission factors lb/10^6Btu NO	CO	HC	Particulates	Source
Direct injection diesel engines	Oil	3.25	1.64	0.22	0.05*	Present study
		3.29	1.20	0.03	–	5
Gas engines	Natural Gas	3.25	1.64	1.26	–	Present study
		3.65	1.17	1.17	–	5
Gas turbines	Natural Gas	0.39	–	–	–	Present study
		0.34	–	–	–	5
		0.57	–	–	–	15 (1973)
		0.43	0.12	0.04	0.01	15 (1975)
Gas turbines	Oil	0.80	0.07	0.007	0.07	Present study
		0.84	0	0	0.06	15 (1973)
		0.23	0.04	0.005	0.03	16
		0.52	0.12	0.04	0.04	15 (1975)

* Corresponds to 10% opacity

TABLE 4 EMISSION FACTORS PROPOSED FOR EMISSION RATE ESTIMATES

Engine type		No_x		CO		HC		Particulates	
		g/kWh	lb/10^6 Btu	g/kWh	lb/10^6 Btu	g/kWh	lb/10^6 Btu	g/kWh	lb/10^6 Btu
4-cycle diesels	D.I. - N.A.								
	D.I. - T.C.								
2-cycle diesels	D.I. - S.C.					0.94	0.23	0.22	0.05
	D.I. - T.C.								
		13.40	3.32	6.60	1.64				
4-cycle gas engines	N.A.								
	T.C.								
2-cycle gas engines	S.C.					4.0	1.31	NA	NA
	T.C.								
4-cycle, dual-fuel engines	N.A.								
	T.C.								
4-cycle diesels	P.C. - N.A.								
	P.C. - T.C.								
2-cycle diesels	P.C. - S.C.	6.70	1.42	1.88	0.40	0.94	0.23	0.22	0.05
	P.C. - T.C.								
Gas turbines (natural gas)		4.00*	0.60	0.80*	0.12	0.27*	0.04	0.07*	0.01
Gas turbines (distillate fuel)		5.35*	0.80	0.80*	0.12	0.27*	0.04	0.47*	0.07

* Assumes: Specific fuel consumption 14740 Btu/kWh

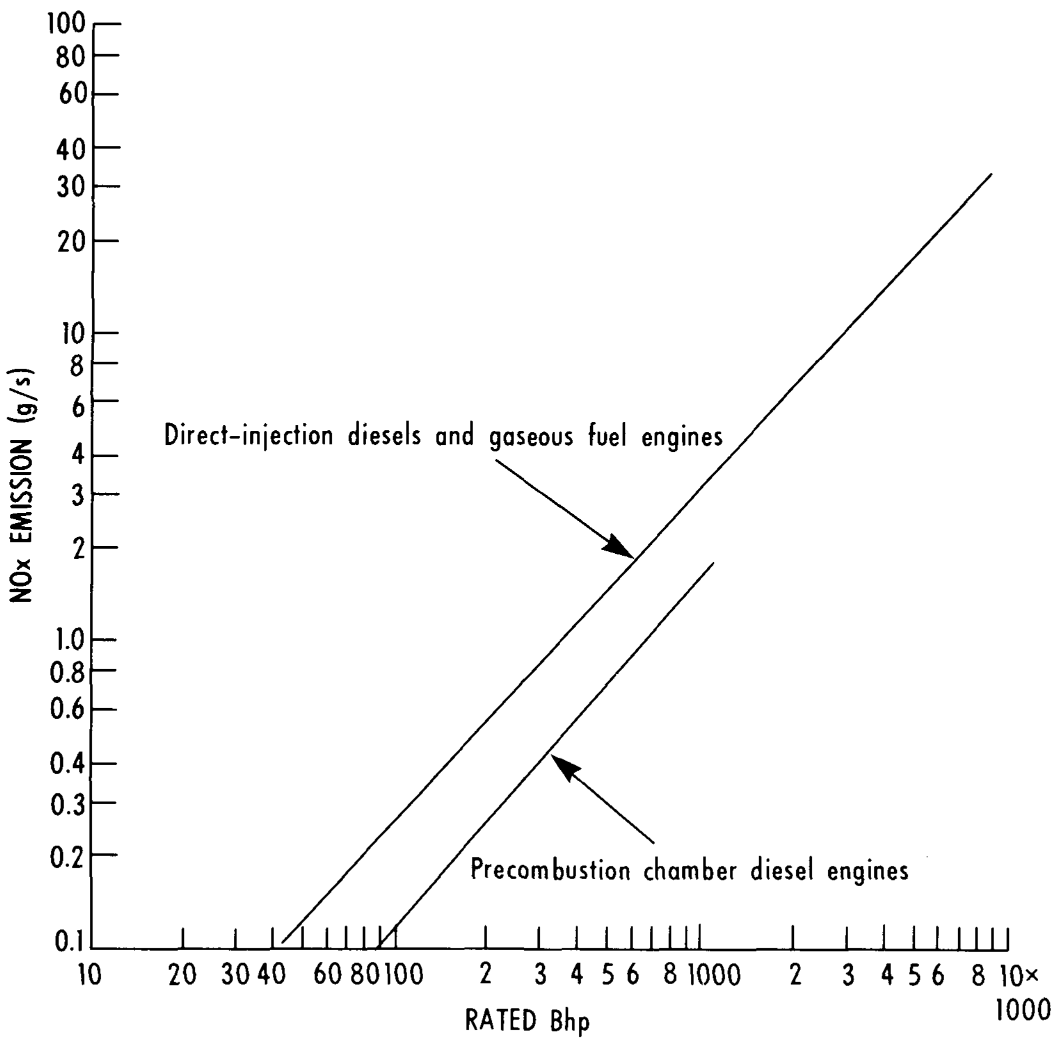


FIGURE 13 NOx EMISSION RATES

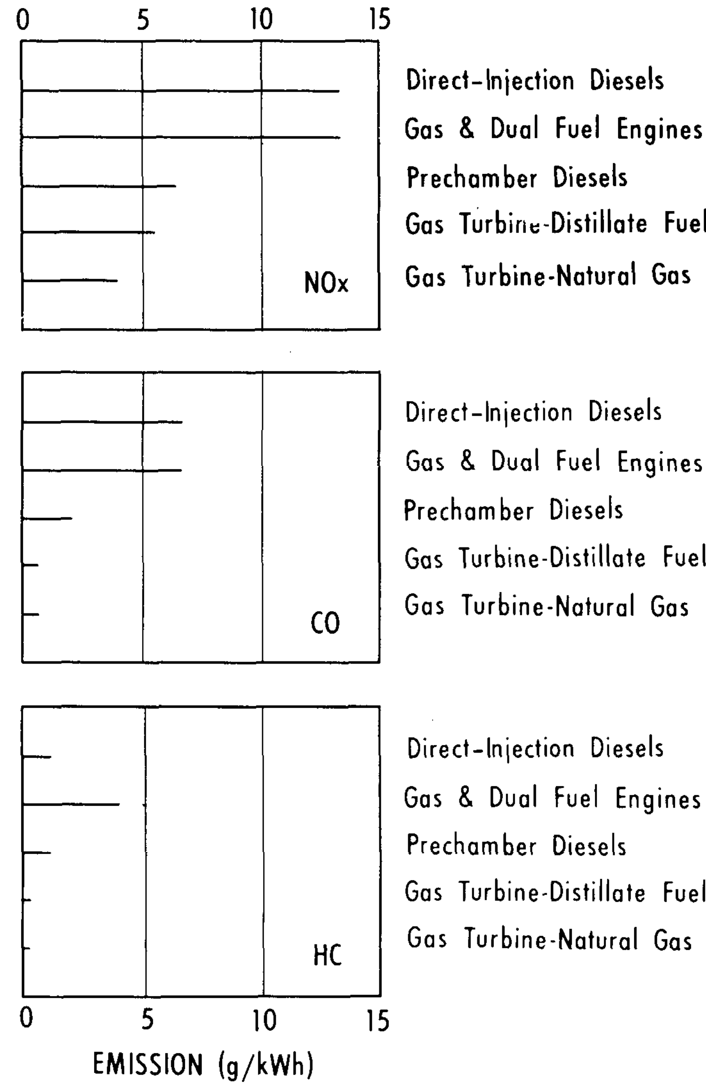


FIGURE 14 AVERAGE EMISSION RATES FROM STATIONARY INTERNAL COMBUSTION ENGINES

Figures 8 and 9. This is due to better mixing by stratified charge and higher excess air quantities trapped in the cylinder during combustion.

Gas engines have generally higher HC emissions than diesel engines, Figures 10 and 11. This is probably due to different air-fuel mixture preparation. In particular, the unburned mixture can escape more easily during valve overlap in four-cycle engines, and during scavenging in two-cycle engines.

As can be seen from Table 3, HC and CO emissions from gas turbines are generally not a problem. Secondary air dilutes the exhaust to approximately 300% excess air at high temperatures which effectively oxidizes any unburned HC and CO.

4.3.2 Proposed Emission Factors. In Figure 14, emission factors are compared graphically and the emission factors proposed for emission rate estimates are summarized in Table 4. It is believed that only minor changes will occur as more emission data become available. Soot emission data for gas engines are not available. In all probability, gas engine soot emission is markedly less than that from a diesel engine of the same power.

5 EMISSION CONTROL METHODS

In this section the available control methods for stationary reciprocating engines and gas turbines are summarized, and some general observations are made on those methods that appear to have the greatest potential for stationary engine application.

5.1 Reciprocating Engines

Three types of control techniques are available for reciprocating engines - operating condition changes, engine modification and exhaust treatment methods. These are defined further in Table 5.

Most engine modifications and changes in operating conditions cannot be used to control NO_x, CO and HC emissions simultaneously. Changes that reduce NO_x emissions generally have the reverse effect on emissions of CO and HC and on fuel consumption. This results from the fact that the conditions favoring NO_x formation - high temperatures and readily available oxygen - also favor combustion of CO and HC.

Exhaust treatment controls include exhaust thermal reactors, catalytic oxidation of CO and HC and catalytic reduction of NO_x. These controls can be added to new or existing engines with little or no effect on engine performance and fuel consumption.

Choosing the most effective emission control methods for stationary engines requires close examination of the effects of the various controls on fuel consumption, reliability, durability and engine life, and effectiveness of emission control. Fuel price increases will place a premium on maximizing fuel economy.

TABLE 5 EMISSION CONTROL METHODS FOR RECIPROCATING ENGINES

Change in operating conditions:	a)	Speed
	b)	Load
	c)	Equivalence ratio
	d)	Ignition timing
	e)	Injection timing
	f)	Intake air temperature
	g)	Intake air pressure
	h)	Exhaust back-pressure
Engine modification:	a)	Exhaust recirculation
	b)	Water injection
	c)	Valve timing
	d)	Combustion chamber configuration
	e)	Compression ratio
Exhaust treatment	a)	Exhaust thermal reactor
	b)	Stack gas scrubbing and solid sorption
	c)	Catalytic converters

On the basis of available information, which is summarized in the next three sections, the following emission control methods appear to have the greatest potential as short-, intermediate- and long-term solutions:

Engine	Short and Intermediate Terms		Long Term
Diesel	Water Injection	Precombustion Chamber	Catalytic Reduction
Gas	Water Injection	Increased Valve Overlap for 4-cycle N.A. Engines	Catalytic Reduction

5.1.1 ***Operating Conditions*** (10-14). For stationary engines it is worthwhile to investigate the effect of operating conditions on mass emissions and fuel consumption at constant power.

In diesel engines, NO_x emissions are essentially proportional to power output, for a given speed, and at constant power NO_x emissions tend to increase with speed. As pointed out previously, this behavior is fundamentally different from that of gas engines, (Figure 5). However, HC and CO emissions exhibit much more variability with speed in diesel engines.

In Table 6 the effects of various changes in standard operating conditions for the GMVA-8 engine are summarized. Retarding the ignition from 10 to 4 degrees BDTC reduces NO_x emissions by 16% but increases fuel consumption by 6%. Reduction of the air manifold temperature from 130° to 80°F reduces NO_x emissions by 47% and increases fuel consumption by 1% (probably due to decrease of the volumetric efficiency). The most impressive reduction of NO_x emissions occurs when speed is increased from 300 to 330 rpm: NO_x emissions are reduced by 58% to 6.4 g/Bhp-h and fuel consumption increases only 1.6%.

Simultaneous determination of the air/fuel ratio of the mixture trapped in the cylinder shows that many of the effects are attributable in part to a simultaneous change in air/fuel ratio. Reduction of the intake-air temperature and increase of the intake-air pressure each increase the air density in the air manifold, resulting in a leaner air-fuel mixture and lower NO_x emissions. Hence, the effect of a given parameter largely depends on the engine type. The same thing can be said for applicability of a given NO_x control technique. In turbocharged engines, for example, the exhaust back-pressure cannot be increased in order to dilute the mixture and speed cannot be increased easily in integral-generator engines.

5.1.2 Engine Modification. The second phase in the application of emission controls is engine modification. The methods include exhaust recirculation, water injection, valve-timing changes, combustion chamber redesign, and modification of compression ratio.

5.1.2.1 Exhaust Recirculation. Exhaust gas recirculation (EGR) is now being used by the automobile industry in some new cars to reduce NO_x emissions. The fundamental effect is that of charge dilution, leading to lower temperatures and lower NO_x emissions. Figure 15 illustrates the effect of EGR on emissions as a function of recirculation rate, and injection timing for the supercharged precombustion chamber diesel engine (15). At 15% EGR the peak NO_x emission is reduced by 75%. The graph also indicates that, at constant load, at least a small increase in CO emissions can generally be expected with EGR, together with a slight reduction in HC emissions. Since EGR reduces peak cycle temperature, it tends to reduce engine efficiency and therefore produces a rapidly increasing penality in fuel economy at full load.

The Caterpillar Tractor Company has also published EGR data for a precombustion chamber diesel engine (16), which are illustrated in Figure 16. At 15% EGR and 100% rated load, NO_x emissions decrease from 730 to 220 g/h, a reduction of about 73%. At higher speeds or lower load, the effectiveness is diminished.

There are many technical problems that must be overcome before EGR can be applied to stationary engines. A system is needed to accurately meter the amount of exhaust recirculated. An efficient heat exchanger must be developed to cool the exhaust without condensing the water vapour in

TABLE 6 EMISSION CONTROL BY MODIFICATION OF OPERATING CONDITIONS - COPPER BESSEMER GMVA-8 2-CYCLE SCAVENGED SPARK-IGNITION GAS ENGINE

Operating conditions	Mass Emissions (g/Bhp-h)			Exhaust Conc. (ppm_v)			Fuel Consumption (Btu/Bhp-h)	Change from base values (%)	
	NO_x	HC_T	CO	NO_x	HC_T	CO		NO_x Emissions	Fuel consumption
Base conditions[a]	15.23	1.94	.29	1079	395	34	7079	-	-
Retard ignition 10^o to 4^o BTDC	12.75	2.26	.35	918	466	42	7496	-16.2	+5.8
Increase air flow 161% to 201% displacement	14.66	2.14	.22	842	352	21	7223	-3.7	+2.0
Decrease air manifold temp. 130^o to 80^oF	8.09	2.19	.34	574	446	40	7169	-46.9	+1.2
Increase exhaust back-pressure 0" to 6" Hg	9.53	2.16	.30	686	447	36	7673	-37.4	+8.4
Increase speed at constant Bhp 300 to 330 rpm	6.41	2.24	.41	418	420	44	7192	-57.9	+1.6
Combination of: 4^o BTDC ignition 100^oF air manifold temp.	10.63	2.08	.32	760	426	38	7572	-30.2	+7.0
4^o BTDC ignition 100^oF air manifold temp. 182.0% displacement air	8.73	2.19	.31	549	395	32	7654	-42.7	+8.1
4^o BTDC ignition 100^oF air manifold temp. 182.1% displacement air 8.2" Hg exhaust back-pressure	5.26	2.28	.40	332	412	41	8702	-65.5	+22.9

a) Base conditions: Speed - 300 rpm
Power - 1080 Bhp
Torque - 82.5 BHEP
Ignition - 10^o BTDC
Air flow rate - 160% displacement
Air manifold temp. - 130^oF
Exhaust back-pressure - 0" Hg

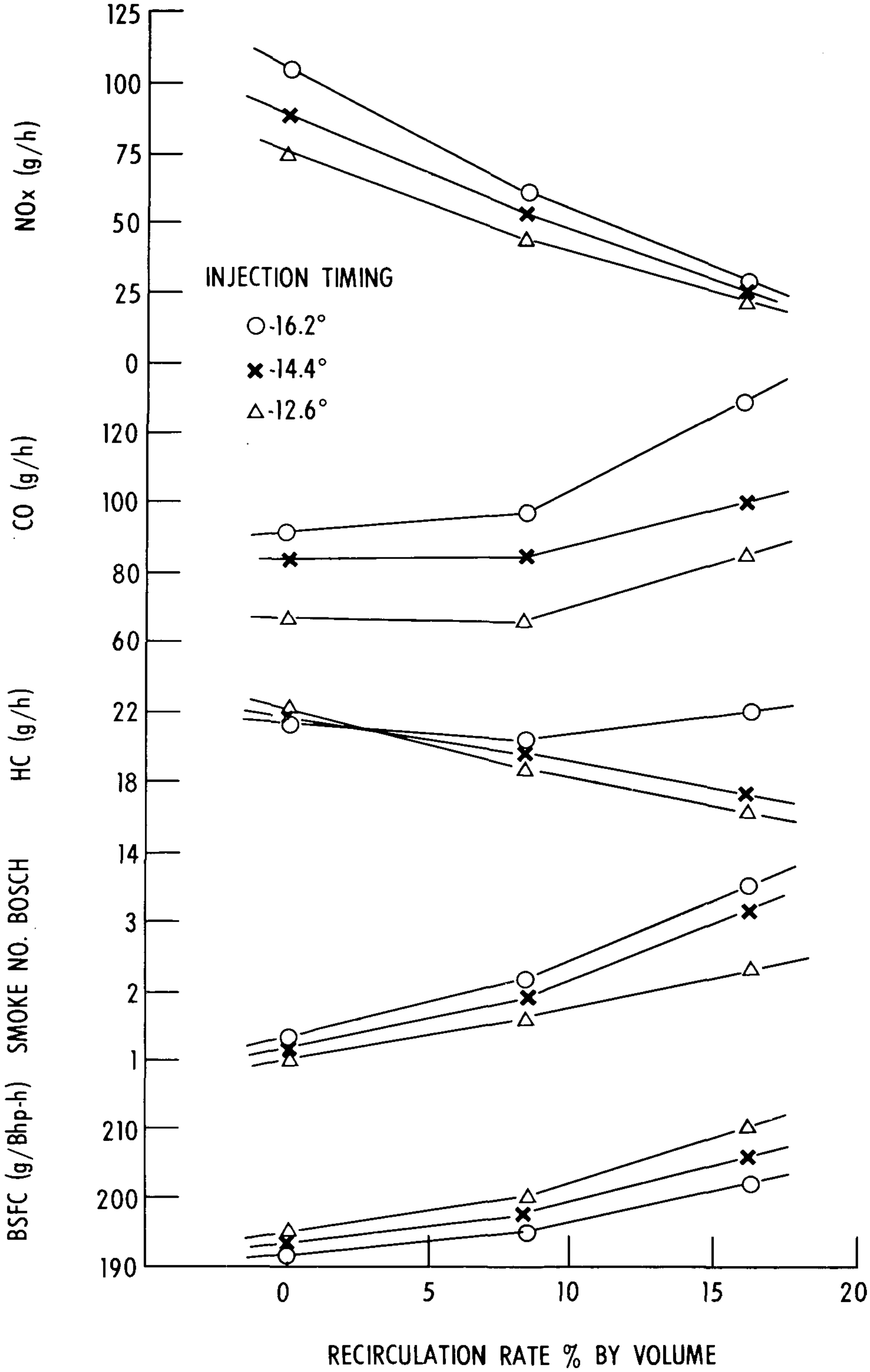
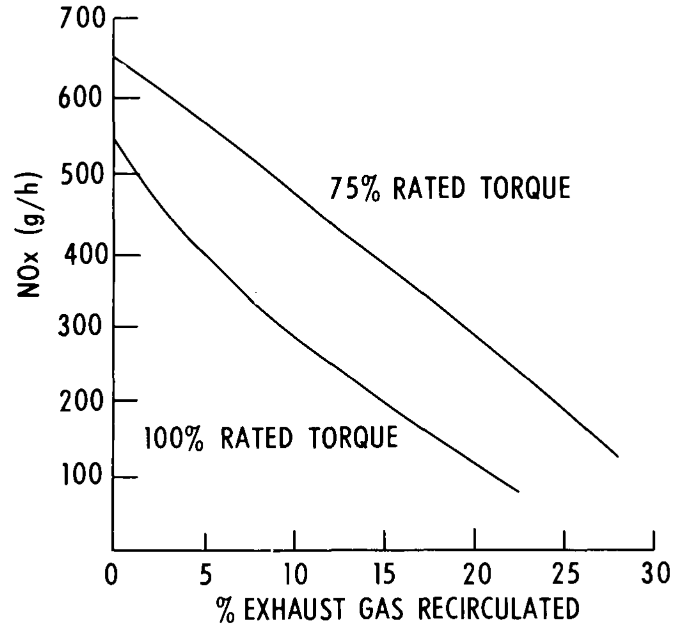
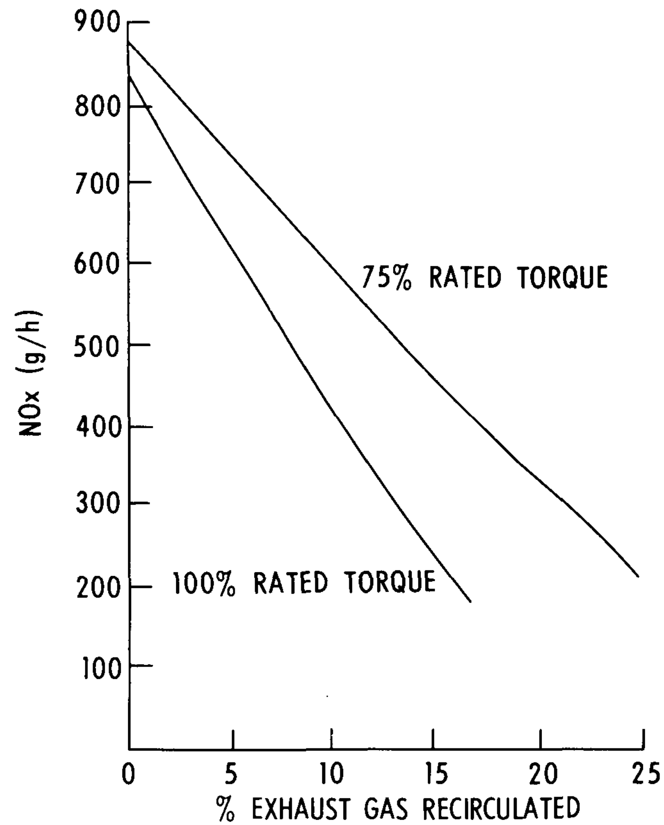


FIGURE 15 EFFECT OF EXHAUST RECIRCULATION AND INJECTION TIMING ON EMISSIONS AND FUEL ECONOMY OF A SUPERCHARGED PRECHAMBER DIESEL ENGINE



EXHAUST RECIRCULATION VERSUS NOx EMISSION
1600 RPM



EXHAUST GAS RECIRCULATION VERSUS NOx
EMISSION 2200 RPM

FIGURE 16 EFFECT OF EXHAUST RECIRCULATION ON NOx EMISSIONS - CATERPILLAR 4-CYCLE PRECOMBUSTION CHAMBER DIESEL ENGINE

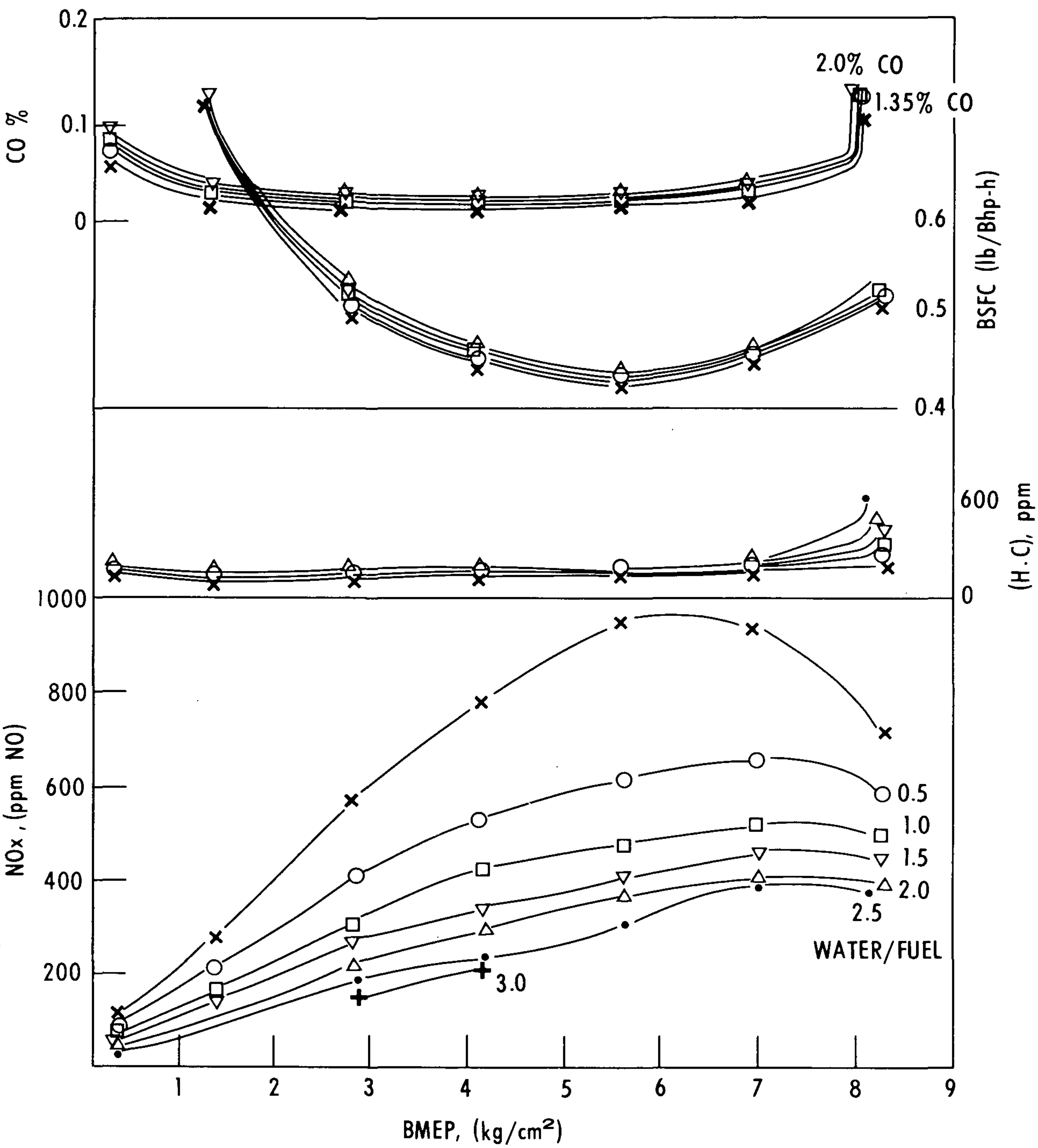


FIGURE 17 EFFECT OF WATER ADDITION ON THE EMISSIONS IN AN I.D.I. COMET V DIESEL ENGINE

the exhaust. This is particularly true for the four-cycle gas engines for which exhaust temperatures are in the range 600° to 650°C. For diesel engines, problems with fouling of intake manifolds, after-coolers, and other equipment by particulates must be overcome. Finally the long-term effect on lubricating oil and engine life must be assessed.

5.1.2.2 *Water Injection.* Water injection serves the same purpose as exhaust recirculation in reducing NO_x emissions, i.e., intake charge dilution (4). The primary function of the water, however, appears to be heat absorption and, consequently, peak temperature reduction in the combustion zone. This is accomplished by injecting distilled or deionized water either directly into each cylinder or at the intake valve of each cylinder. Injection of at least one pound of water for each pound of fuel burned will reduce NO_x emissions by 70% or more.

As the ratio of water to fuel is increased, the ignition delay increases, and fuel injection must be advanced to obtain peak power. It is believed that NO_x emission is affected by two factors: the ability of water to reduce maximum temperatures and oxygen concentration, and the effect of injection advance on the maximum temperatures reached. These two factors result in an increase in NO_x emissions with injection advance and a decrease in NO_x emissions with injected water.

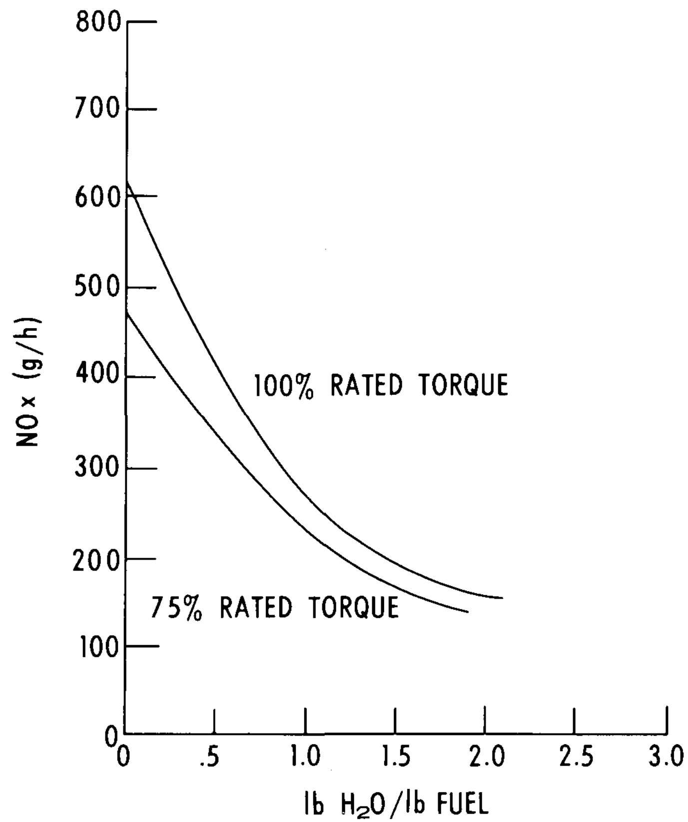
The combined effect of water addition and injection advance for optimum power also affects other emissions, especially at higher loads. The effect of water injection on a swirl-type precombustion chamber diesel engine, Comet V 11, is shown in Figure 17. The data indicate that CO and HC emissions increase significantly with water injection at higher loads.

In Figure 18 data are presented for the Caterpillar precombustion chamber diesel engine (5). At 1.5 lb of water/lb fuel and 100% of rated torque, NO_x mass emissions are reduced from 600 g/h to 200 g/h, a reduction of 67%.

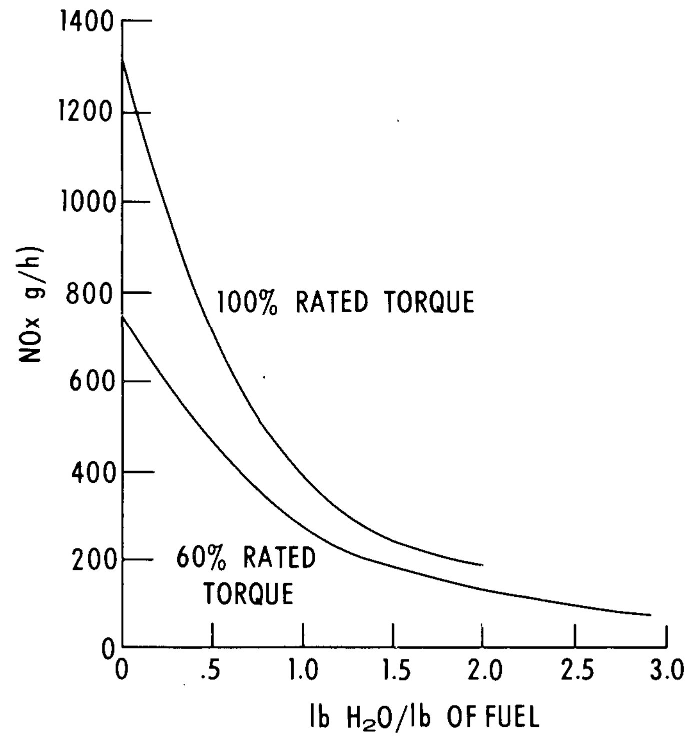
The results of water injection tests on an Ingersoll-Rand PKVGR-12, four-cycle, naturally aspirated gas engine (4) are shown in Figure 19. Water injection at 2 gpm (1.62 lb H_2O/lb fuel) reduces NO_x emissions by 83%, but increases fuel consumption by 10%, HC emissions by 106% and CO emissions by 260%. It is not known if, in this test, the ignition was advanced to obtain peak power.

Water injection is not without its own problems. Controls are needed to measure the water; high-pressure water is needed to overcome manifold pressure on turbocharged engines, and water-injection nozzles with acceptably long lives must be developed. A portion of the water finds its way into the engine crankcase, causing potential oil-sludging problems. The corrosion of valves and combustion chambers must be also taken into account. Before water injection can be widely applied, the long-term effects on engine life must be investigated. At 1.0 lb water/lb fuel, a plant using 10 000 Bhp will require about 400 gal/h of distilled water at full load.

5.1.2.3 *Valve Timing.* Valve timing is also known to affect emissions. In the case of four-cycle, naturally aspirated engines, increasing the valve overlap will produce the same effect as exhaust recirculation. At the end of the exhaust stroke, the intake and exhaust valve are open simultaneously (overlap). Exhaust gas can pass back into the cylinder due to the pressure difference between the intake



WATER INDUCTION VERSUS NOx EMISSION
1600 RPM



WATER INDUCTION VERSUS NOx EMISSION
2200 RPM

FIGURE 18 EFFECT OF WATER INJECTION ON NOx EMISSIONS - CATERPILLAR 4-CYCLE PRECOMBUSTION CHAMBER DIESEL ENGINE

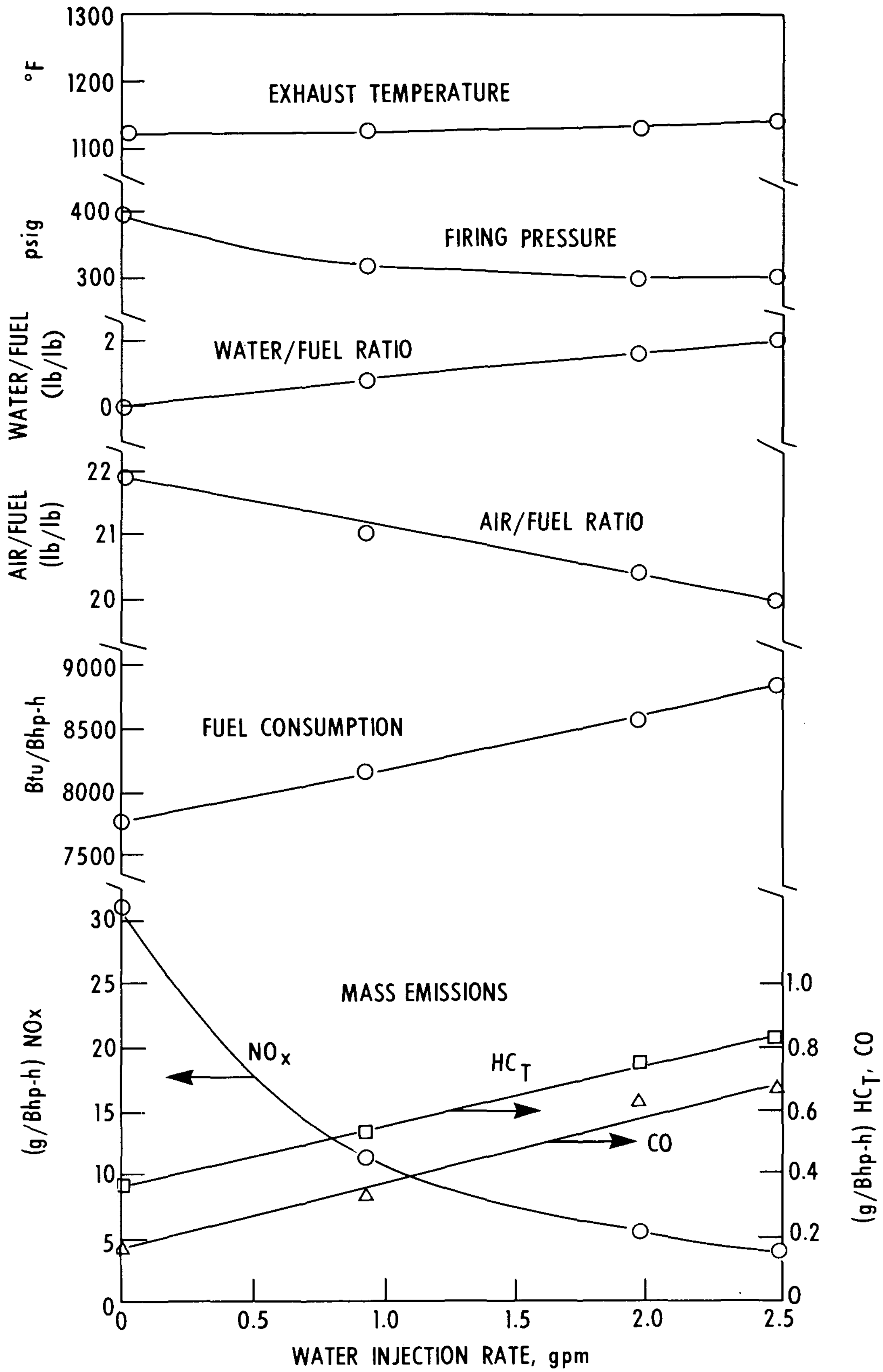


FIGURE 19 EFFECT OF WATER INJECTION - INGERSOLL-RAND PKVGR-12 4-CYCLE NATURALLY ASPIRATED SPARK-IGNITION GAS ENGINE

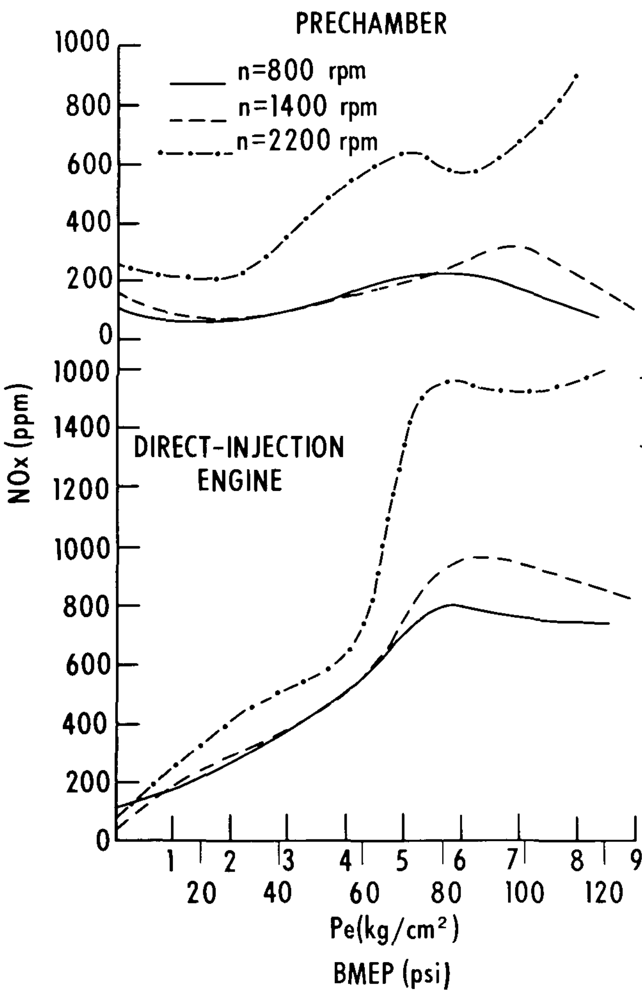


FIGURE 20
NOx CURVES FOR PRECHAMBER
AND DIRECT-INJECTION ENGINES

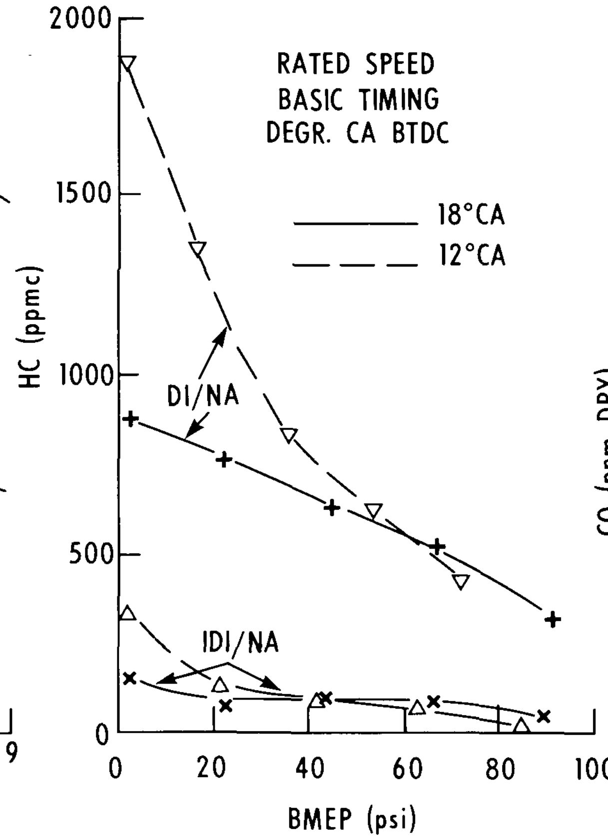


FIGURE 21
HC CONCENTRATIONS, NATURALLY
ASPIRATED DIRECT-INJECTION AND
PRECHAMBER ENGINES

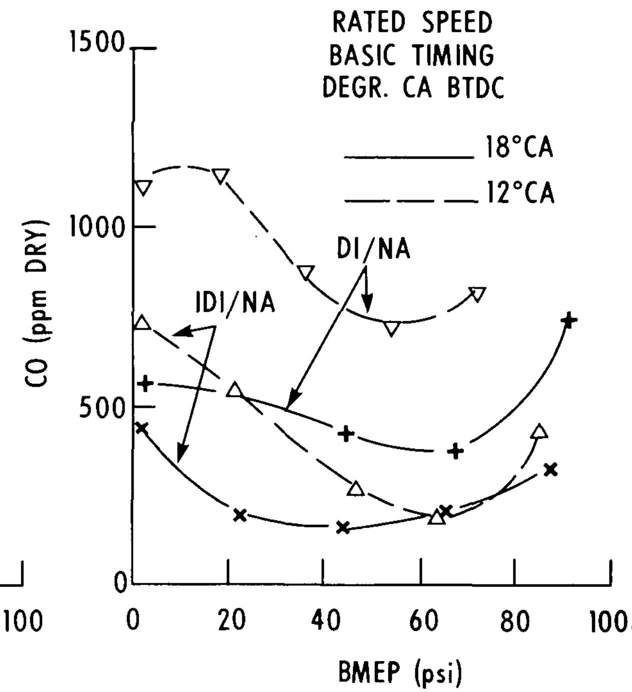


FIGURE 22
CO CONCENTRATIONS, NATURALLY
ASPIRATED DIRECT-INJECTION AND
PRECHAMBER ENGINES

and exhaust manifolds. As valve overlap is increased, the fraction of exhaust present in the fresh charge increases, resulting in an EGR effect.

An increase in valve overlap reduces NO_x emissions by about 60%. Hydrocarbon emissions are also reduced slightly, while CO emissions are not affected. It is probable, however, that fuel economy suffers as valve overlap increases. Increased valve overlap can be applied only to four-cycle, naturally aspirated engines.

For stationary-engine applications, valve-timing modification has some potential for emission control. It is a far simpler alternative than exhaust recirculation and water injection, and could probably be applied to existing as well as to new engines.

5.1.2.4 Combustion Chamber Configuration

a) *Precombustion Chamber.* Precombustion chambers were in use before air pollution became a public concern. Their principal advantages are smoother operation and therefore longer engine life. Disadvantages can be starting difficulties as well as higher fuel consumption compared to a direct-injection engine because of higher heat losses through the larger combustion chamber surface.

The combustion process in precombustion chamber engines is known as stratified-charge combustion. In such a process, ignition of the air-fuel mixture occurs under fuel-rich conditions, even though the air/fuel ratio of the overall mixture is lean. The system is analogous to two-stage air addition in boilers or gas turbines, and results in reduced NO_x emissions, due to a rich mixture present in the primary combustion zone. Hydrocarbon and CO emissions are also generally low due to the lean air/fuel ratio.

According to the literature(4), stratified-charge combustion is also possible for gas engines; however, it is difficult to control, and is sensitive to operating conditions. Stratified-charge combustion in gas engines may therefore not be practical.

The effectiveness of stratified charge combustion is illustrated in Figures 20, 21 and 22. Figure 20 illustrates the fact that the prechamber engine has lower NO_x emission (17). The upper part of the illustration shows the load-dependent prechamber engine NO_x curve for three different engine speeds. The lower part refers to the direct-fuel-injection diesel engine. This difference in combustion must be taken into consideration for an optimization of emission rate in the two types of engines. From Figure 21 (18) it can be seen that the prechamber engine is clearly superior in HC emission control. Also CO emissions from the prechamber engine are considerably lower than those from the direct-injection engine, as shown in Figure 22.

Precombustion models give lower emission values only when operated with standard injection timing of the engine as produced. When injection is retarded, lower NO_x concentrations are emitted from the direct-injection engines (18). In Figure 23 data is compared from 13-mode cycle tests on four engines; two having direct injection, and the other two precombustion chamber combustion systems, but identical in all other respects. From the graphs it can be seen that by retarding fuel injection timing of the direct-injection engine, even lower NO_x emission can be achieved than in the prechamber

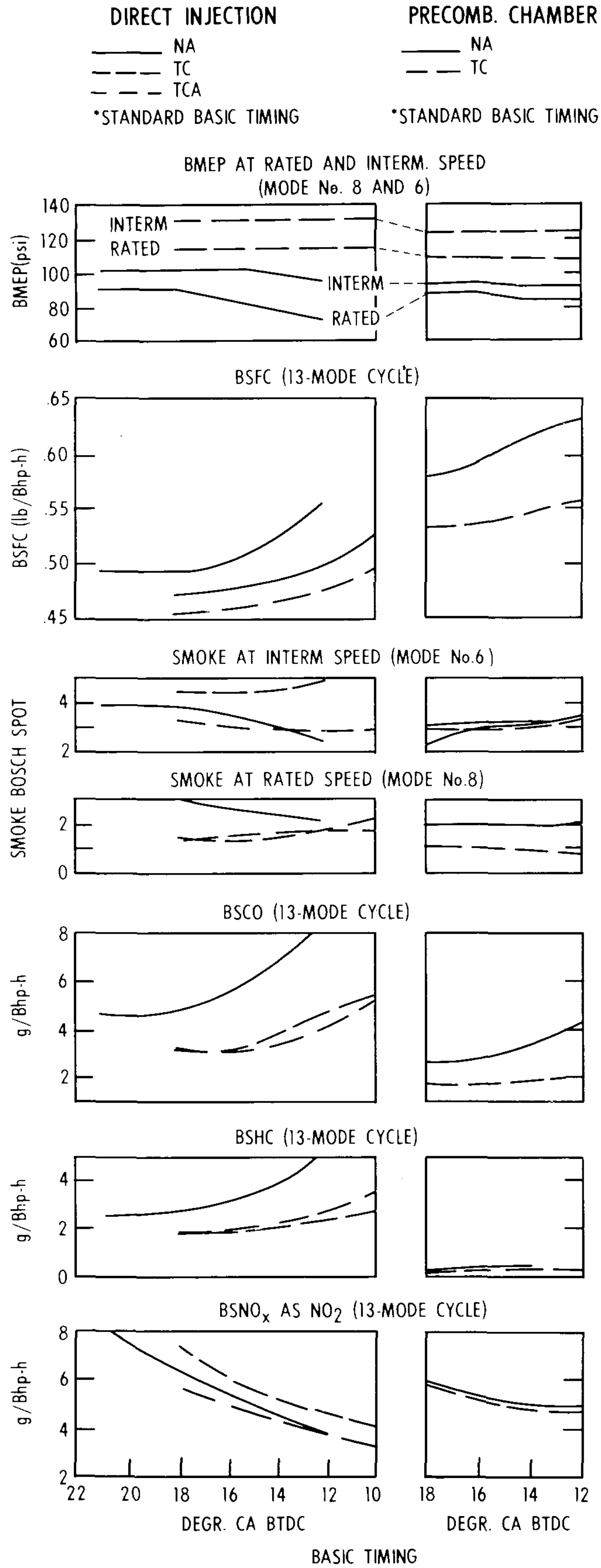


FIGURE 23 COMPARISON BETWEEN DIRECT-INJECTION AND PRECHAMBER ENGINES

combustion engine, and although fuel consumption increases in the retarded-timing engine, it is still less than that in the prechamber combustion engine. The graphs also show that, in the precombustion chamber engine, retarding injection causes fuel consumption to increase with only a small decrease in NO_x emissions, but HC and CO emissions are still lower in precombustion chamber engines.

b) *The M-System (M.A.N. Company).* In the M-system, the fuel is injected onto the temperature-controlled surface of a spherical chamber in the piston. The rate of evaporation is controlled by the wall temperature and the air swirl. After the fuel vapour mixes with the air, it is ignited by several ignition sources formed by the injection of a small percentage of the fuel into the chamber, away from the walls. Emissions from the M-system engine show the same values and trends as those from other direct-injection engines (14).

5.1.3 Exhaust Treatment. Exhaust emissions can either be removed or they can be converted to N_2, CO_2, H_2O and other harmless chemicals by devices located at the engine exhaust. These devices include exhaust-manifold thermal reactors, catalytic converters, stack-gas scrubbers, and solid sorbents. For reasons outlined in the following sections, including effectiveness, ease of installation, and no adverse effect on fuel economy, it can be seen that the catalytic converter may be the most practical exhaust treatment system for stationary engines in the future.

5.1.3.1 Exhaust Thermal Reactors. The exhaust thermal reactor is a modified exhaust manifold designed to maintain high enough temperatures, about 700° - 760°C, to burn unburned CO and HC in the exhaust. The automotive industry has found it necessary to operate engines on a rich mixture in order to provide enough CO and HC to maintain these temperatures. Rich-mixture operation is not felt to be practical for stationary diesels due to the resulting poor fuel economy and smoke emissions. Lean mixtures can also be used with thermal reactors; however, higher exhaust temperatures are required which necessitate burning fuel after the exhaust manifold.

5.1.3.2 Stack-Gas Scrubbing and Solid Sorption. Stack-gas scrubbing and solid sorption each create secondary pollution problems that must be solved; the former a liquid waste problem the latter a solid waste problem. Although these controls could be applied in the form of a single unit that treats all exhaust from engines, boilers, and other combustion equipment, they do not seem to have much potential for emission control of single engines. Some of these methods pass the exhaust flows through a water spray or bubble the exhaust through a tank of water. They are often employed in underground mine vehicles. Tests have shown little or no reduction of the gaseous emissions with which this report is concerned. They effectively cool the exhaust and thus condense heavy hydrocarbons and perhaps collect some particulate matter. Carbon monoxide and NO_x are practically insoluble in water and, therefore, scrubbers are of no value in reducing these important contaminants.

5.1.3.3 Catalytic Converters. Nitrogen oxides, CO and HC can be converted to harmless species in catalytic converters. The operation of a catalytic unit for a stationary engine would be less arduous than for an automotive engine, because it would not be necessary to meet the automotive requirements of minimum warmup time and operation over widely varying flow rates and temperatures.

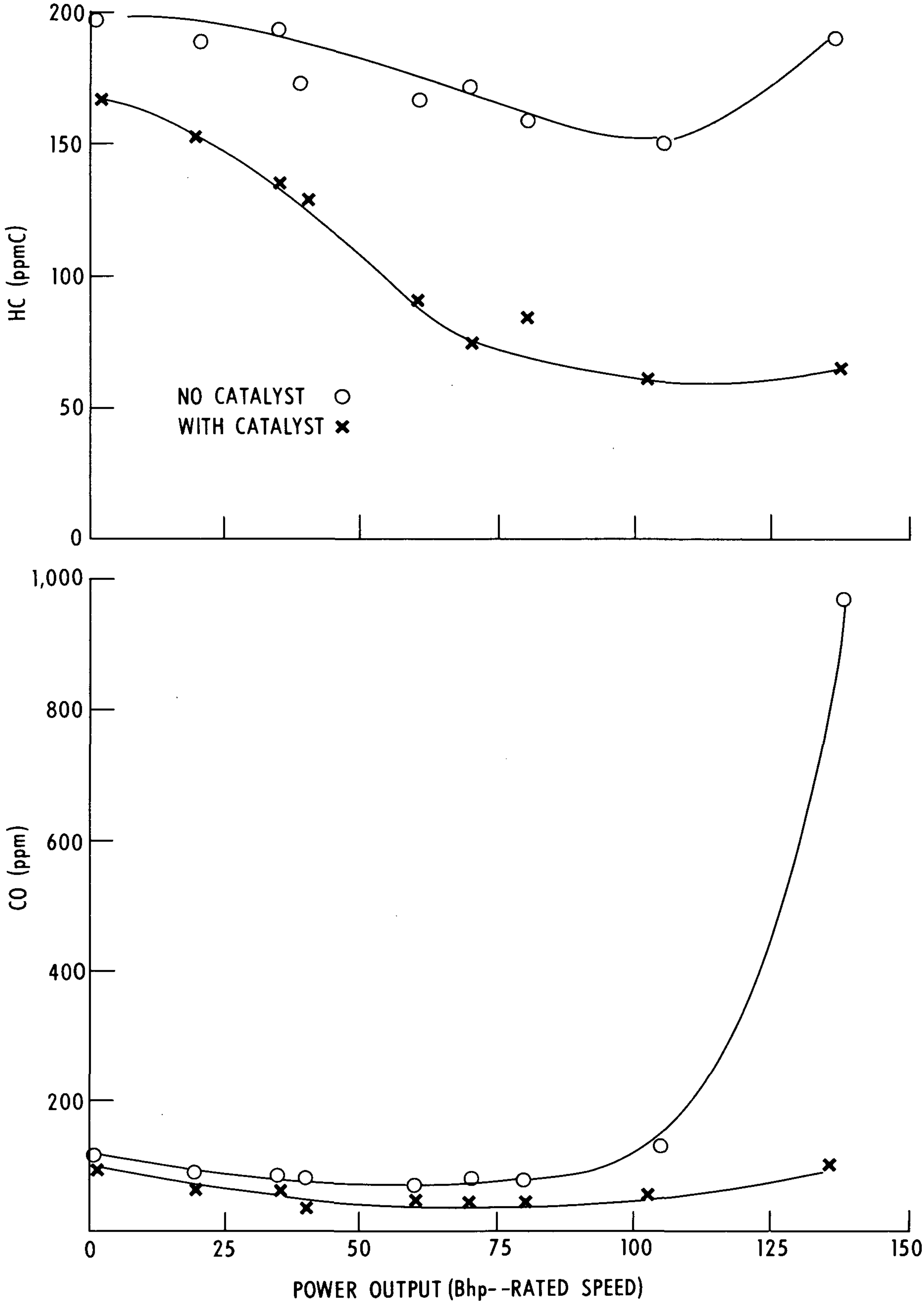


FIGURE 24 EFFECT OF PLATINUM CATALYST ON EMISSION LEVELS - 2-CYCLE, DIRECT-INJECTION, AIR-SCAVENGED DIESEL ENGINE

a) *Oxidation of CO and HC.* In the converter, CO and unburned hydrocarbons are removed by catalytic oxidation to CO_2 and H_2O. The catalyst allows the reactions to occur at lower temperatures than are required in a noncatalytic thermal reactor. Most four-cycle diesel engines operate with lean mixtures, and have sufficient oxygen present in the exhaust for the oxidation reaction (4%-5%). Two-cycle engines always have a large excess of oxygen in the exhaust as a result of dilution by scavenging air (15%). Four-cycle engines operating on a rich or stoichiometric mixture will, however, require the mixing of additional air with the exhaust.

In the case of diesel engines, where HC and CO emissions are very low, catalysts are available (5) which can reduce these even further; however, tests demonstrate that catalytic devices deteriorate with use, and must be replaced or regenerated at regular intervals if they are to maintain any degree of effectiveness. Oxides of nitrogen are not reduced because the oxidation catalyst helps promote complete combustion and NO_x is a normal product of combustion.

The catalyst has also been studied with two-cycle engines at standard operating modes. As shown in Figure 24, emissions of HC and CO were reduced (19). One major obstacle to the successful removal of all oxidizable materials in diesel exhaust by catalysts is the low exhaust temperature (particularly for a two-cycle, air-scavenged engine). To increase this temperature additional fuel consumption is necessary.

b) *Reduction of NO_x.* The catalysis of NO_x requires a chemical process different from the normal combustion process. This can be accomplished by introducing a reducing agent such as CO, H_2, ammonia (NH_3) or natural gas.

Hydrogen and CO will be present in sufficient amounts for NO_x reduction only in the case of four-cycle engines operating on rich mixtures. Hydrogen is produced via the water-gas shift reaction under rich conditions:

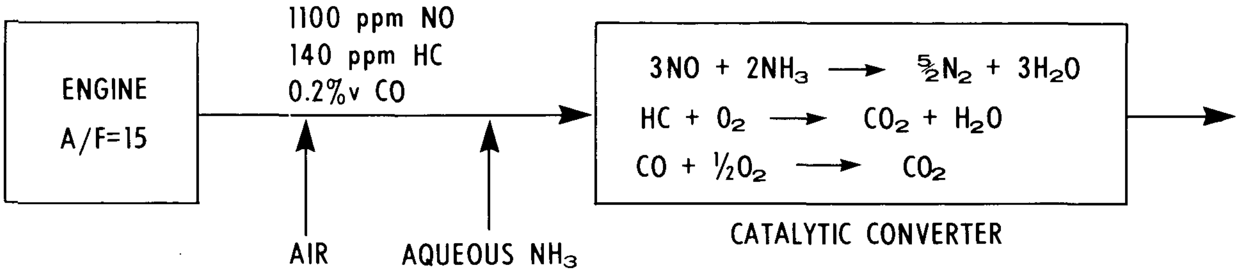
$$CO + H_2O \longrightarrow H_2 + CO_2$$

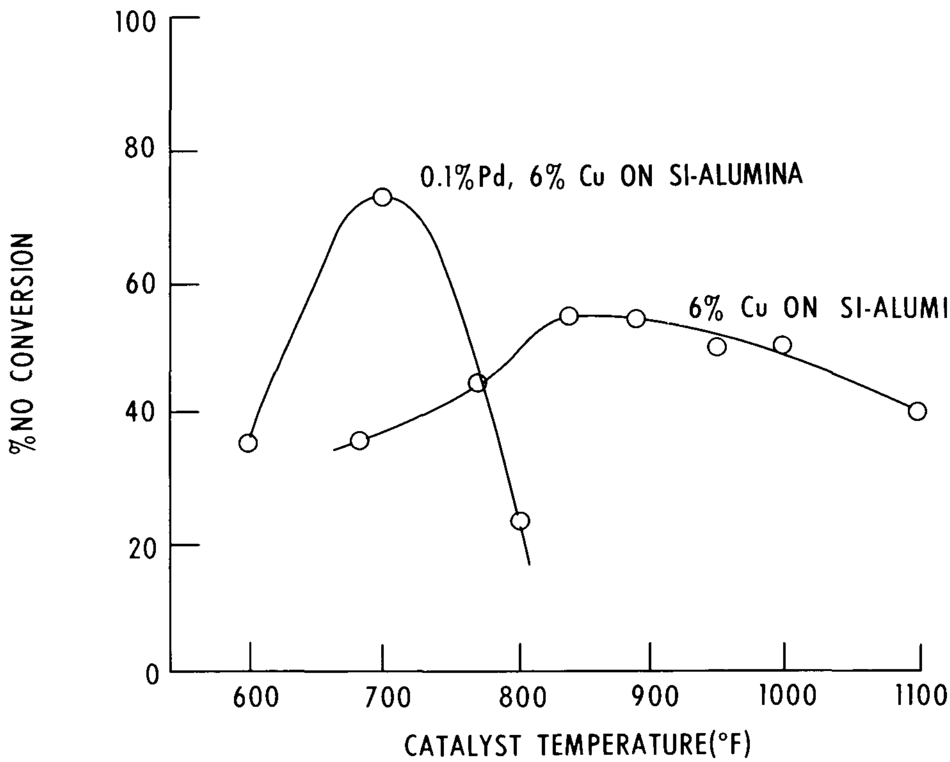
and is probably the primary reducing agent. The automotive industry has found that under certain conditions, low oxygen concentrations and low temperatures, the hydrogen can also reduce NO to NH_3 (4);

$$5H_2 + 2NO \longrightarrow 2NH_3 + 2H_2O$$

leading to an unwanted by-product.

Most stationary engines are operated at lean air-fuel settings for better fuel economy. Enough oxygen is present in the exhaust to make necessary the addition of a reducing agent such as H_2, natural gas, or NH_3 to the exhaust before catalytic reduction. Since CO is practically nonexistent in diesel exhaust, the CO or H_2 must be produced by a methane or propane burner in the exhaust stream. In addition, any excess CO or H_2 formed must be oxidized. In the case of natural gas, it is necessary to add enough gas to completely react with the oxygen before NO_x can be reduced. If the oxygen concentration is high, as in two-cycle engines, it is necessary to use multiple catalytic stages with interstage addition of natural gas in order to avoid burning up the catalyst.





(REFERENCE: U.S. PATENT 3,449,063)

FIGURE 25 CATALYTIC REDUCTION OF NO_x BY AMMONIA

It is worth emphasizing, however, that higher fuel cost, oxidation of excess reducing agents and maintenance of an active catalyst are complex problems which do not have practical solutions with today's state-of-the-art.

Ammonia will reduce NO_x even in the presence of oxygen. Figure 25 reproduces data reported in an Ethyl Corporation patent (4). Exhaust from an internal combustion engine was passed over a palladium-copper oxide catalyst and the conversion of NO_x was monitored as a function of temperature. An optimum temperature was found near 700°F (370°C) at which overall NO_x conversion reached a maximum of about 75%. Above this optimum temperature the ammonia reducing agent begins to oxidize to NO and H_2O:

$$4NH_3 + 5O_2 \longrightarrow 4NO + 6H_2O$$

Similar data are shown in Figure 26 for a platinum catalyst unit operating at space velocities between 10 000 and 90 000/h, where space velocity is exhaust flow rate divided by catalyst volume (4). The optimum temperature occurs near 428°F (220°C). Optimum removal of NO_x is above 90%, and is relatively insensitive to space velocities. These results were obtained using a synthetic mixture containing 3000 ppm NO, 3% O_2, 0.8% H_2O by volume, and 3000 ppm NH_3. The water vapour content of engine exhaust is closer to 15% for four-cycle engines. Consequently, the anticipated conversion would very likely be lower, since water vapour competes for catalytic sites.

A second advantage of the ammonia reduction system is that catalytic oxidation of CO and HC will occur simultaneously over the same catalyst. For the copper oxide catalyst identified in Figure 25, 46% of the CO and 38% of the HC had been removed at the optimum temperature for NO_x reduction.

To maintain a space velocity in the range 30 000 to 50 000/h a 1000 Bhp engine would require about two cubic feet of catalyst.

Of all the possible emission control methods, catalytic reduction by NH_3, natural gas or CO would seem to be the best long-term NO_x control method for stationary reciprocating engines. The method allows operation of the engine at conditions corresponding to maximum fuel economy or power, and is known to be effective for controlling emissions NO_x, CO and HC.

Significant development will be required, however, before wide-scale application is practical. Information on the optimum catalyst formulation and composition, catalyst durability, and resistance to catalyst poisons in the fuel must be sought in order to develop a practical catalytic converter for stationary engines. With regard to the ammonia reduction system for NO_x, it must be determined whether the system will work at the oxygen concentrations present in two-cycle engine exhaust.

5.2 Gas Turbines

The development of gas turbines during the last few years is characterized by a permanent increase in specific output (power/weight ratio) and in thermal efficiency.

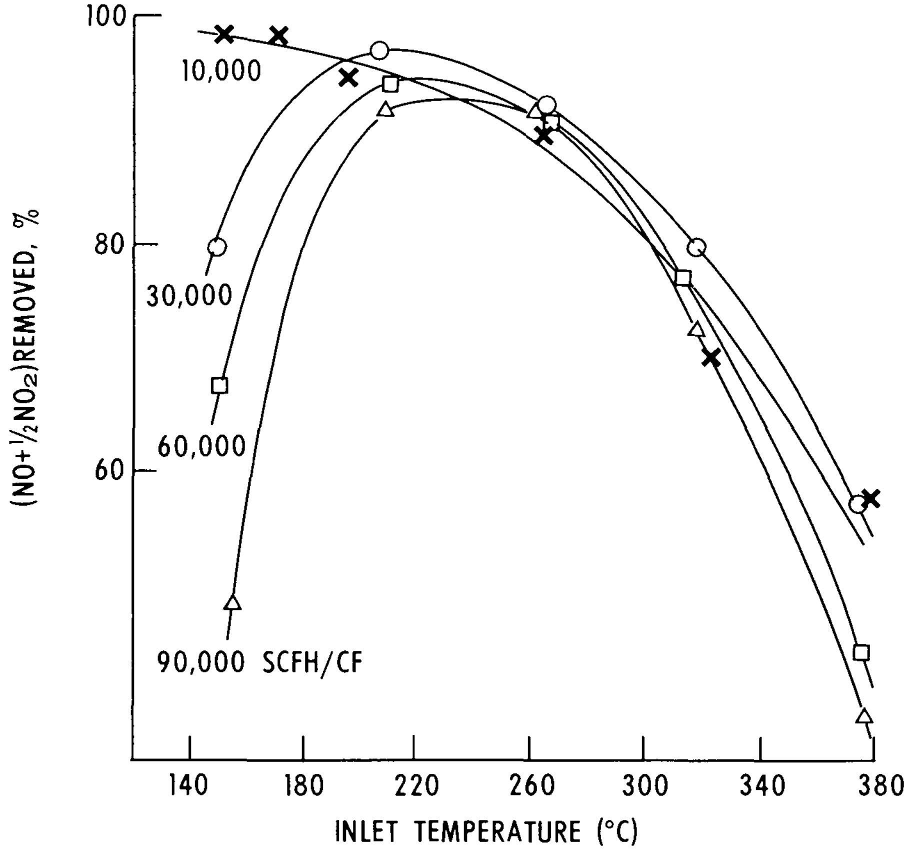


FIGURE 26 CATALYTIC REDUCTION OF NOx BY AMMONIA OVER A PLATINUM CATALYST

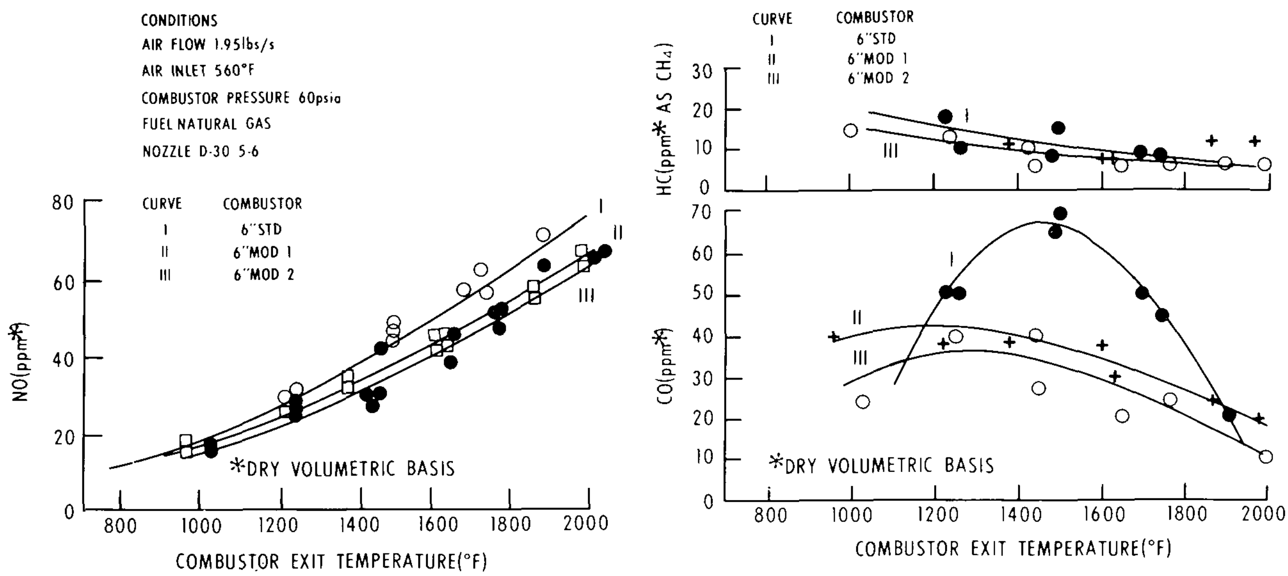


FIGURE 27 EFFECT OF COMBUSTOR EXIT TEMPERATURE AND PRIMARY ZONE LEANING ON EMISSIONS (NATURAL GAS) - STANDARD W-251AA WESTINGHOUSE GAS TURBINE

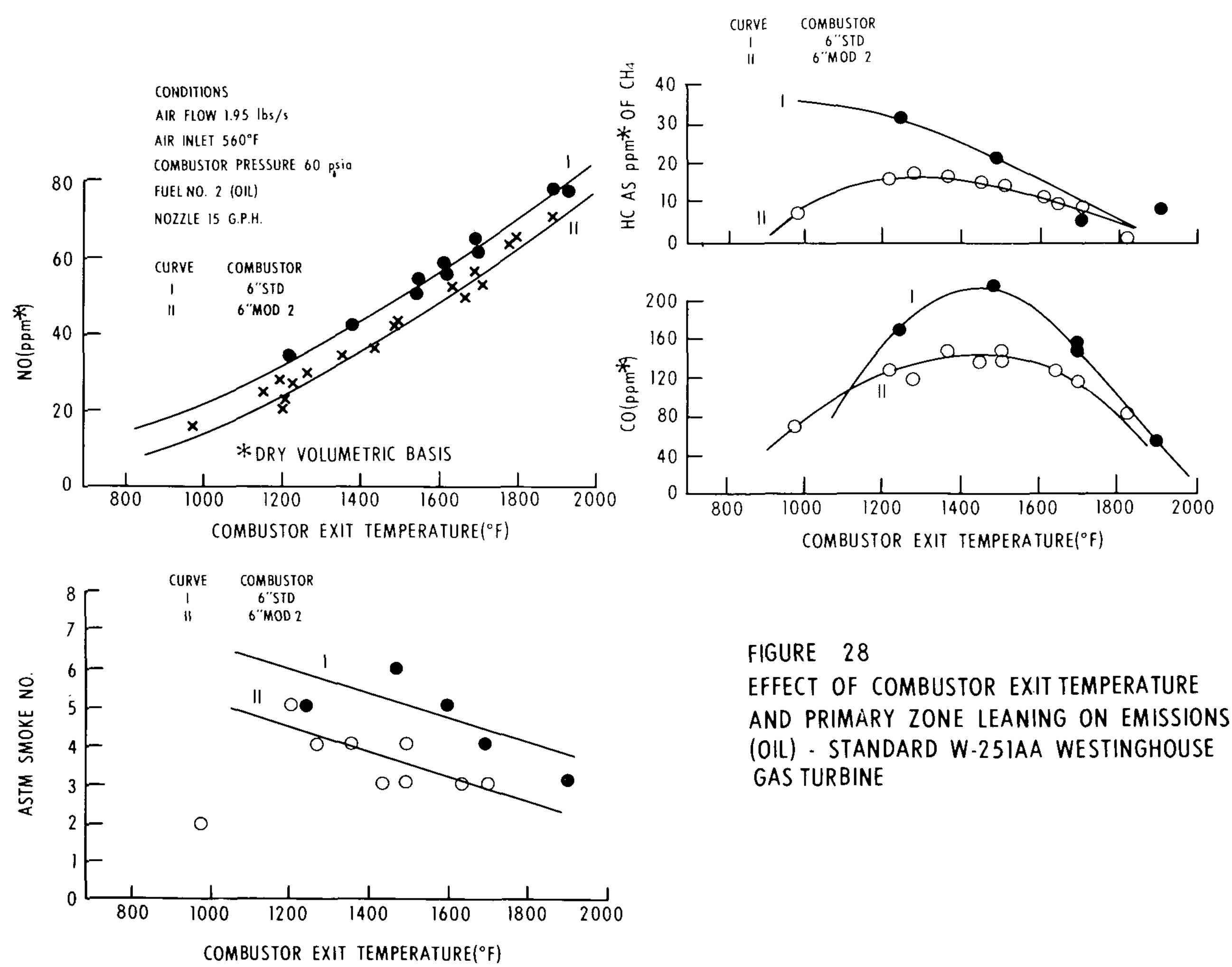


FIGURE 28
EFFECT OF COMBUSTOR EXIT TEMPERATURE AND PRIMARY ZONE LEANING ON EMISSIONS (OIL) - STANDARD W-251AA WESTINGHOUSE GAS TURBINE

The air pollutants, NO_x, CO, HC, SO_2 and particulates are produced. If we exclude SO_2 from consideration, as these emissions depend entirely upon the sulphur content of the fuel used, it is only *the harmful NO_x emissions which present a serious problem. The following sections deal with studies* performed to date on possible methods of reducing NO_x emissions, in conjunction with particulates and smoke.

5.2.1 Changes in Operating Conditions. A large gas turbine (33MW) with no NOx control device can emit up to 260 ppm of NO_x, equivalent to a mass emission rate of 426 lb/h. Although CO emissions can be very low (20 to 40 ppm) (20), in one instance, installation of equipment to control NO_x resulted in a substantial increase of CO emissions. In a second case, CO emissions decreased by 67% after installation of NO_x control (20).

Turbines fired with premium, low-sulphur fuels produce negligible SO_2 emissions. The trend, however, is toward the utilization of poorer, less refined fuels which have increasingly high sulphur values.

Visible emissions from gas turbines operating on premium fuels were below 20% opacity, in the few units for which this parameter was reported. The trend toward use of poorer fuels, however, may lead to increased particulate emissions.

Recent emphasis on the control of NO_x emissions has sparked considerable research into NO_x formation mechanisms and techniques for their control (20, 21, 22, 23, 24). Methods now used to reduce NO_x are designed to reduce the peak flame temperature, decrease the residence time of the gas at high temperatures, or both. These techniques fall into two broad classifications, 'dry' and 'wet'.

5.2.2 Dry Techniques. Dry methods can, generally, reduce gas turbine emissions up to 38%; however, higher rates of reduction have been reported by turbine manufacturers. Extensive research is underway to increase the effectiveness of dry methods in the hope of obviating the need for the more expensive wet-type control.

Dry combustion modifications fall into two general categories.

A. Techniques for lowering temperature:

- Increase air/fuel ratio
- Add cooled combustion products
- Modify combustor

B. Techniques for reducing residence time at high temperatures:

- Increase flow velocity through combustion chamber
- Add quench air earlier to terminate NO_x formation

5.2.2.1 Techniques for Lowering Temperature.

Increased Air/Fuel Ratio. Tests were performed by the Westinghouse Gas Turbine Laboratory at Lester, Penn.,U.S. Results are presented in Figures 27 and 28. In the modified combustor, changes were made (Mod 1), making the primary zone leaner. A further modification (Mod 2) of the standard combustor, caused further leaning of the primary zone, so that the Mod 2 configuration represents a very lean primary zone. It could be lit only with air heated to about 260°C. A 20% reduction in NO_x was noted between the standard combustor and the Mod 2 combustor at a combustor exit temperature of approximately 1000°C. Using oil as fuel, both the standard and Mod 2 combustors were sampled for NO_x, CO, HC, and smoke. In the case of oil, the reduction in NO_x is less than 11% at about 1000°C. It therefore appears that the maximum NO_x reduction attainable by primary zone leaning is between 10% and 20%. The effect of primary zone leaning on HC and CO emissions is also beneficial with both gas and oil as fuel. With gas, the observed smoke spot number is always zero for all combustors; however, for oil, it was found that primary zone leaning reduced the smoke number. One of the very serious disadvantages of lean mixtures is loss of stable operating range which results in poor ignition characteristics.

Addition of Cooled Combustion Products. The most readily available inerts for mixing with the combustion air are cooled exhaust products extracted from the combustor. Recirculated exhaust gas was cooled to about 300°C and mixed with the intake air at source temperature. Two series of tests were performed: one with No. 2 diesel oil, the other with natural gas.

The NO_x, CO and HC emissions are illustrated in Figure 29. Curve I corresponds to conditions with no recirculation, and forms a baseline for comparison with the results obtained with recirculation, shown by Curve II.

A combustor normally operating without recirculation and with a corresponding outlet temperature of 1000°C, was modified to include recirculation, of up to 26% of the normal air capacity. At this rate of recirculation, a 38% reduction in NO_x was noted with oil and 30% with natural gas.

Recirculation was also found to have an indirectly beneficial effect on CO and HC emissions at higher combustor exit temperatures. Recirculation will reduce the total stack discharge in proportion to the exhaust recirculated. The smoke spot numbers did not show any detectable change under conditions of recirculation compared to those with no recirculation. Similarly no degradation of the minimum combustor temperature rise was noted.

EGR is extremely effective in reducing NO_x emissions, and has the thermodynamic advantage over the use of excess air that stack loss is not increased.

Another independent factor influencing the effectiveness of EGR is the water content of the recirculated gas. It is known that air humidity affects NO_x production to the extent that addition of one percent of moisture by weight to combustion air will reduce NO_x production by about 20% (25).

Combustor Modification. When evaluating emission control methods involving engine modifications primary consideration was given to the following emission classes: CO, NO_x, HC, dry particulates and smoke. No specific estimates have been made for control of reactive hydrocarbons, odour, or aldehydes because control methods applicable to these emissions are not yet identified. Reduction of these emissions is expected to occur with reductions in HC emissions. Any of the modifications defined for existing turbine engines could be combined to achieve increased emission control effectiveness.

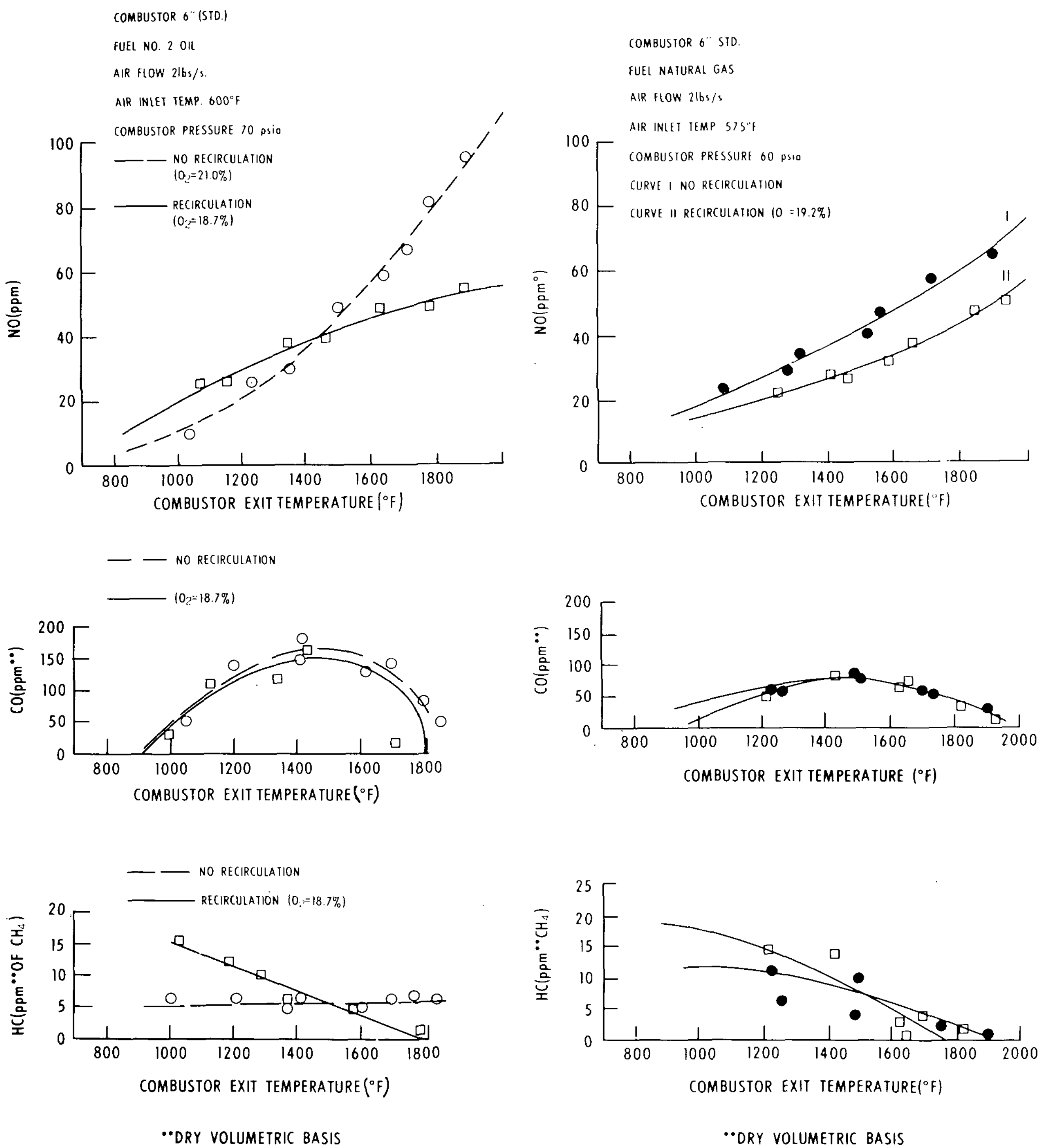


FIGURE 29 EFFECT OF COOLED EXHAUST GAS RECIRCULATION ON EMISSIONS FOR GASEOUS FUEL AND LIQUID FUEL COMBUSTOR - STANDARD W-251AA WESTINGHOUSE GAS TURBINE

To facilitate analyses of engine modification, gas turbines can be divided according to their power level. This is particularly useful for this analysis, since effectiveness factors and costs of control methods are similar for engine models within each class.

Two classes of turbine engines are defined:

Class I - Small turbine engines. These engines are considered as one class because the relatively small size of the combustor components makes control of certain emissions more difficult than in larger engines.

Class II - Medium to large turbine engines. The engine modification control methods considered feasible for turbine engines are:

A. Existing Engines

1) Minor combustion chamber redesign
2) Major combustion chamber redesign
3) Divided fuel supply system
4) Modified compressor air bleed rate

B. Future Engines

5) Variable-geometry combustion chamber
6) Staged-injection combustor
7) Externally vaporizing combustor.

The first four methods are, at least in principle, applicable to existing engines by retrofitting of new or modified parts, and to engines currently in production. The last three methods are considered applicable only to future engines of new design, since the modifications required are too extensive to be applied to engines already developed.

Minor Combustion Chamber Redesign. This method consists of simple modifications of the combustor and fuel nozzles to reduce all emission rates to the best levels currently attainable within each engine class, excluding extreme data points. In general, this method requires emission reduction to levels demonstrated by other engines of that model.

Major Combustion Chamber Redesign. This method consists of major modifications of the combustion chamber and fuel nozzle incorporating advanced fuel-injection concepts (carburation or prevaporization). This method appears to reduce smoke level, and presumably particulate emissions, by approximately half. Also, premixing of air and fuel can be used to give substantial NO_x reduction by decreasing residence time in the combustor.

Divided Fuel Supply System. This method results in control of the air/fuel ratio in the combustion zone. It appears to reduce HC and CO emissions only in idling and low output conditions.

Modified Compressor Air Bleed Rate. This method increases the air bleed rate from the compressor at low power operation to increase combustor air/fuel ratio. If maximum air bleed rate is 20%, CO and HC emission rates are reduced by 50% (26). The method is useful only at idle and shows no change at full load.

Variable-Geometry Combustion Chamber and Staged-Injection Combustor. These two methods, which can be applied only to future engines, seem to be the most efficient modifications for reducing emissions at full-load conditions. These methods use a variable airflow distribution to provide independent control of combustion zone air/fuel ratio, and an advanced combustor design concept involving a series of combustion zones with independently controlled fuel injection in each zone. This incorporates design characteristics that provide a good mixture in the combustion zone, and reduce NO_x emissions at full power by 75% and particulate emissions by 50% at all power levels (27).

Externally Vaporizing Combustor. Also very promising is the dry control method using an externally vaporizing combustor (26). As a result of injecting atomized fuel into a high-temperature air flow, vaporizing and premixing can occur but combustion is prevented. This premixed and vaporized homogeneous mixture permits maintenance of an optimum flame temperature, so that NO_x as well as CO and HC are kept low. By adjusting combustor geometry lean homogeneous combustion can be maintained throughout the range of operating conditions. No emission data are available for this control method.

Reductions in emissions, attainable through the use of a control method, vary with the pollutant considered, the engine class, and the operating conditions.

5.2.2.2 Techniques for Reducing Residence Time at High Temperatures.

a) One method for reducing residence time at high temperatures in the primary zone of combustion is to increase flow velocity through the combustion chamber. Typical data for the effect of primary zone residence time on NO_x and CO emissions (28) are shown in Figure 30. The effect on NO_x emissions is not linear; however, substantial reductions in CO, particularly at very lean conditions, are associated with the longer residence time.

b) The second method of reducing the flame residence time is to move secondary diluent holes upstream. Test data for this control method are not available; however, it is expected that the effect of this method will be similar to that of increased air/fuel ratio (Section 5.2.2.1).

5.2.3 Wet Techniques. In wet methods, water or steam is injected into the primary zone of the combustion chamber, as shown in Figure 31. In either form, water acts as a heat sink to absorb thermal energy, thereby reducing the peak combustion temperature. The degree of NO_x reduction achieved depends on the amount and effectiveness of the water used. To maximize effectiveness, some manufacturers have experimented with mixing the fuel and water prior to injection into a combustion chamber, others spray the water or steam into the primary air stream, and still others inject the fuel and water simultaneously so that the sprays impinge directly on each other. Dramatic reductions in NO_x emissions have been achieved with all these techniques. The NO_x reductions attainable with wet controls are illustrated in Figure 32, which is a compilation of information provided by three major manufacturers of gas turbines. The data points superimposed on the band represent results of tests by the U.S. Environmental Protection Agency and operators. Wet methods are reported to reduce NO_x emissions more than 80%.

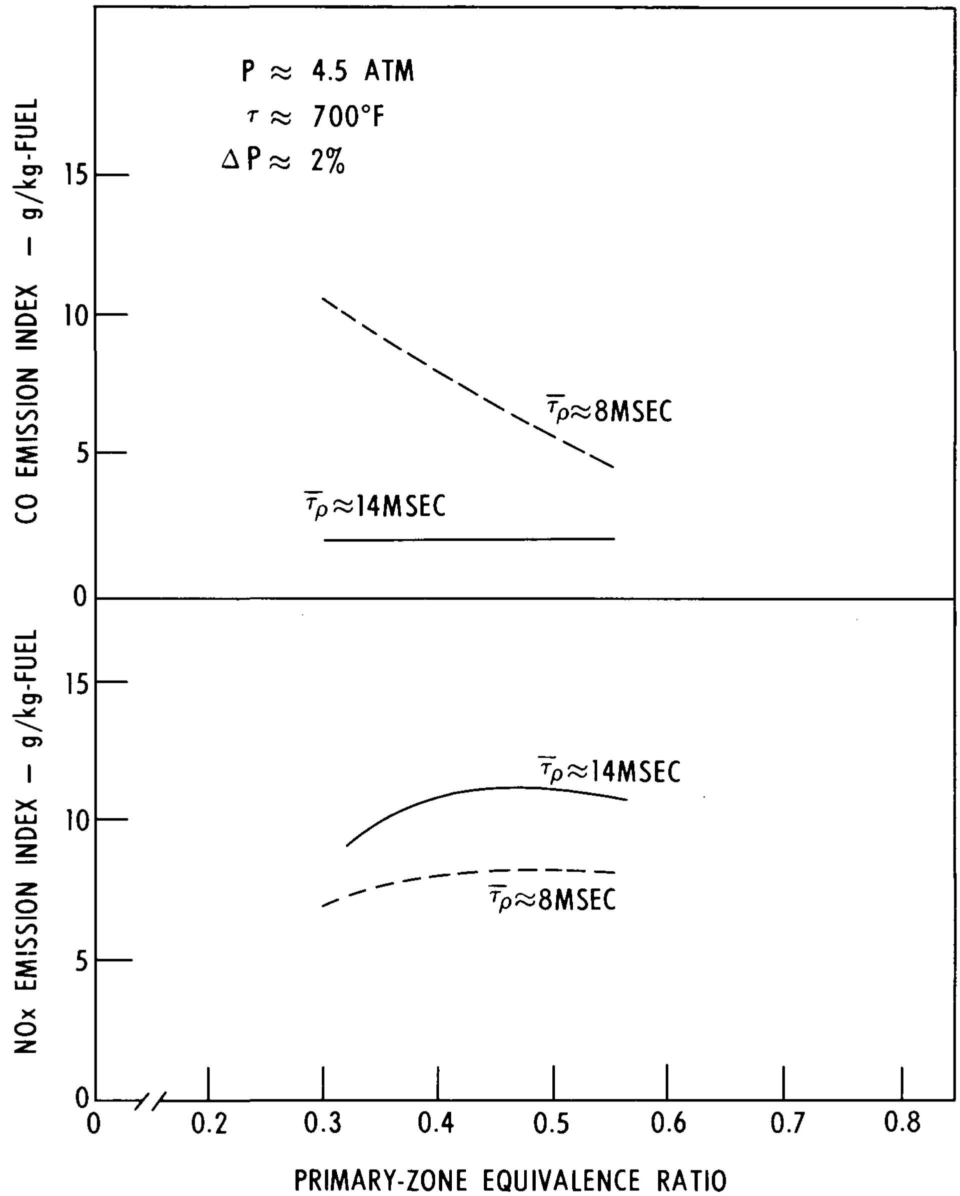


FIGURE 30 TYPICAL EFFECT OF RESIDENCE TIME ON NOx AND CO EMISSIONS IN AN AUTOMOTIVE GAS TURBINE COMBUSTOR. EQUIVALENCE RATIO WAS SPECIFIED AS

$$\frac{\text{A/F STOICHIOMETRIC}}{\text{A/F ACTUAL}}$$

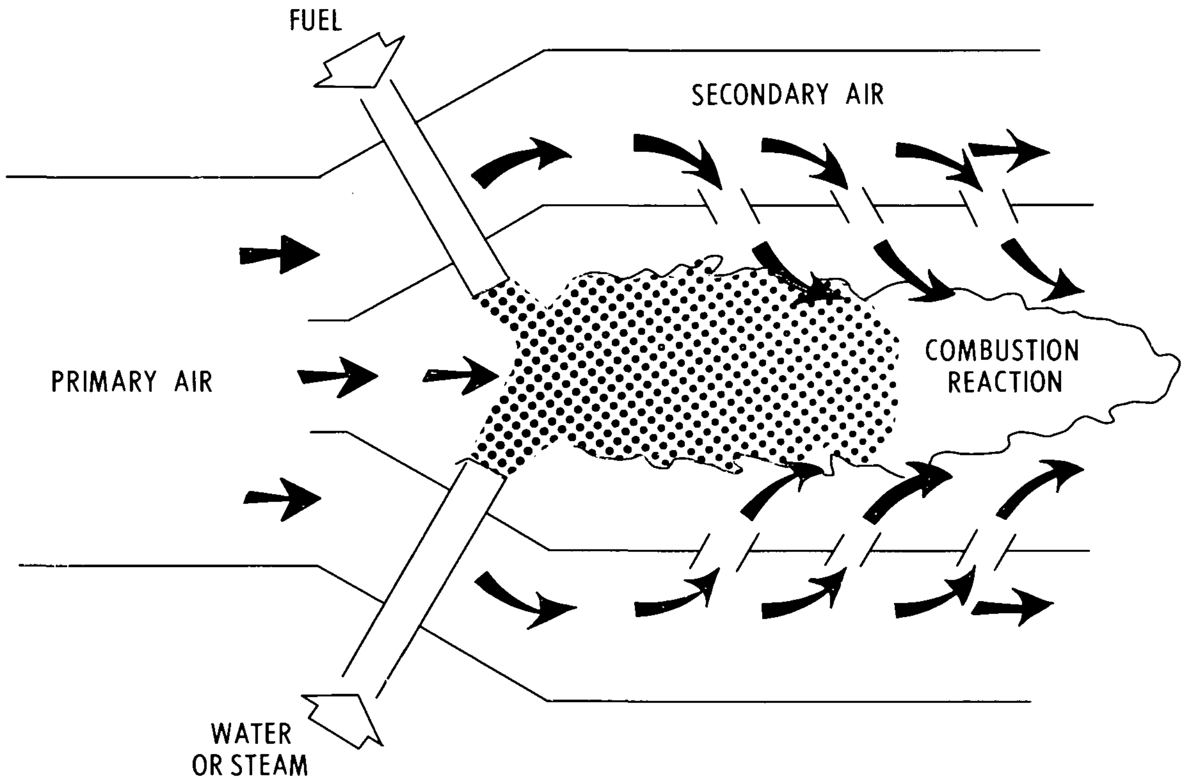


FIGURE 31 SCHEMATIC OF WET CONTROL TECHNIQUES (WATER OR STEAM INJECTION)

5.2.3.1 *Test Results.* It was observed during W-251 combustion laboratory tests (20), that peak-load control by water injection was very effective over a modest range of water/fuel ratios. With No. 2 distillate fuel, NO_x emissions were reduced to less than 140 lb/h (approximately 75 ppm at 15% excess oxygen) over the load range of the W-251 gas turbine by injection of water at rates up to 35 U.S. gal/min. Figure 33 illustrates NO_x levels without water injection, as well as the levels attained with water injection to satisfy the Los Angeles County Rules and limit the emission rate of NO_x to 140 lb/h. Further NO_x control to as low as 30 ppm was demonstrated by injection of water at rates up to 46 U.S. gal/min as shown in Figure 34.

The peak-load level of smoke was reduced from smoke spot No. 5 to No. 3 by the injection of water as shown in Figure 35.

Carbon monoxide emission levels remained unchanged at intermediate loads, and at high loads were reduced by as much as 25 ppm, to the 10 ppm level, by the injection of water. This variation with load can be seen in Figure 36.

Total HC levels were reduced slightly by the injection of water at high loads, remaining within the band between 0.70 and 0.96 ppm, as hexane, Figure 37.

With the injection of water, SO_2 showed a slight increase of 10-15 ppm to a maximum level of about 20 ppm. The apparent disappearance of 10-15 ppm SO_2 which accompanies the higher smoke levels when water injection is stopped suggests that significant quantities of sulphur may be absorbed by carbon smoke particles (20).

It is interesting to note that, when burning natural gas, considerably higher reduction factors for NO_x emissions were obtained with water injection (29). Results of this test are presented in Figure 38.

The 25-MW unit was also tested with natural gas and oil in the Port Mann Gas Turbine Generating Plant of British Columbia Hydro (30). Under water-injection conditions at full load, reduction in NO_x emissions was 83% and 93% for fuel oil and natural gas respectively.

Steam is the other commonly available inert vapour that can be added to air to lower flame temperatures. Various quantities of steam were added to the combustion air ahead of the combustor by the Westinghouse Gas Turbine Laboratory. A substantial decrease in NO_x emissions was noted even with small amounts of steam; however, the CO and HC emissions showed a substantial increase. This method does not affect smoke.

Information available from the literature on gas turbines indicates that NO_x is normally the most abundant pollutant and the most difficult to control. Furthermore, because large turbines are designed to operate at higher compression and temperature to improve fuel economy, NO_x emissions tend to vary directly with size in uncontrolled gas turbines.

Carbon monoxide and HC emissions are highly variable and appear to decrease with increasing turbine size. The addition of NO_x control can dramatically increase CO and HC emission;

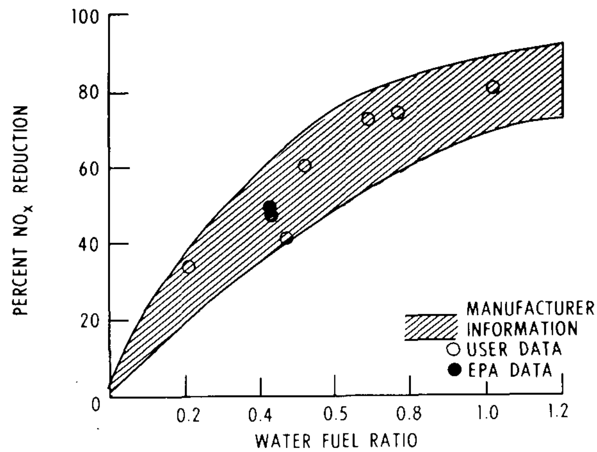


FIGURE 32 EFFECTIVENESS OF WATER OR STEAM INJECTION IN REDUCING NO_x FORMATION IN GAS TURBINE COMBUSTORS.

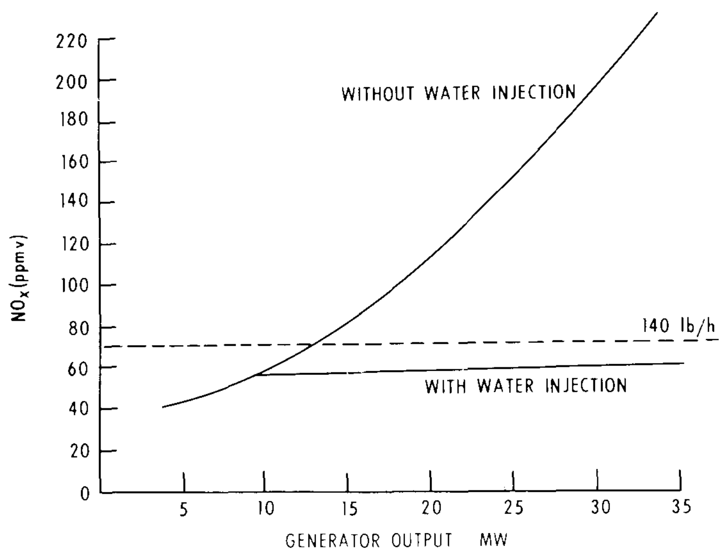


FIGURE 33 EFFECT OF WATER INJECTION ON NO_x EMISSIONS - STANDARD W - 251 WESTINGHOUSE GAS TURBINE (NO. 2 DIST. OIL), 32.8 MW.

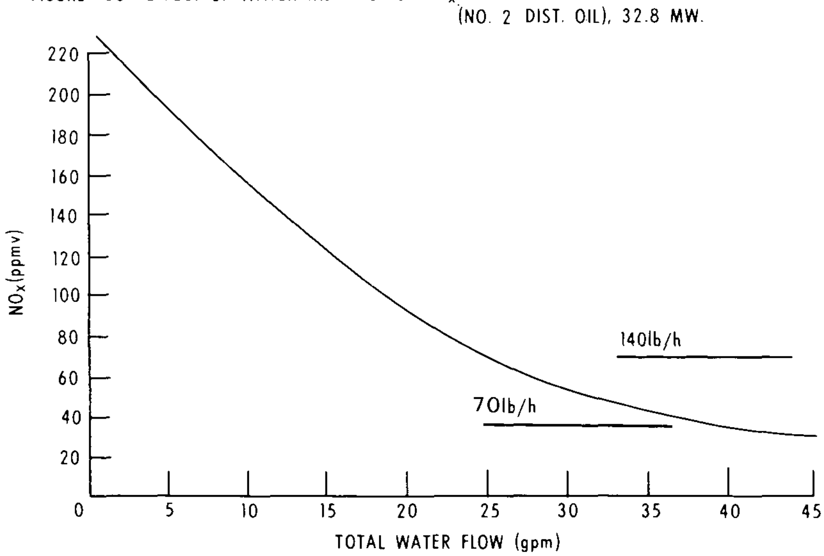


FIGURE 34 NO_x EMISSION CONTROL BY INJECTION OF WATER. W-251 WESTINGHOUSE GAS TURBINE (NO. 2 DIST. OIL), 32.8MW

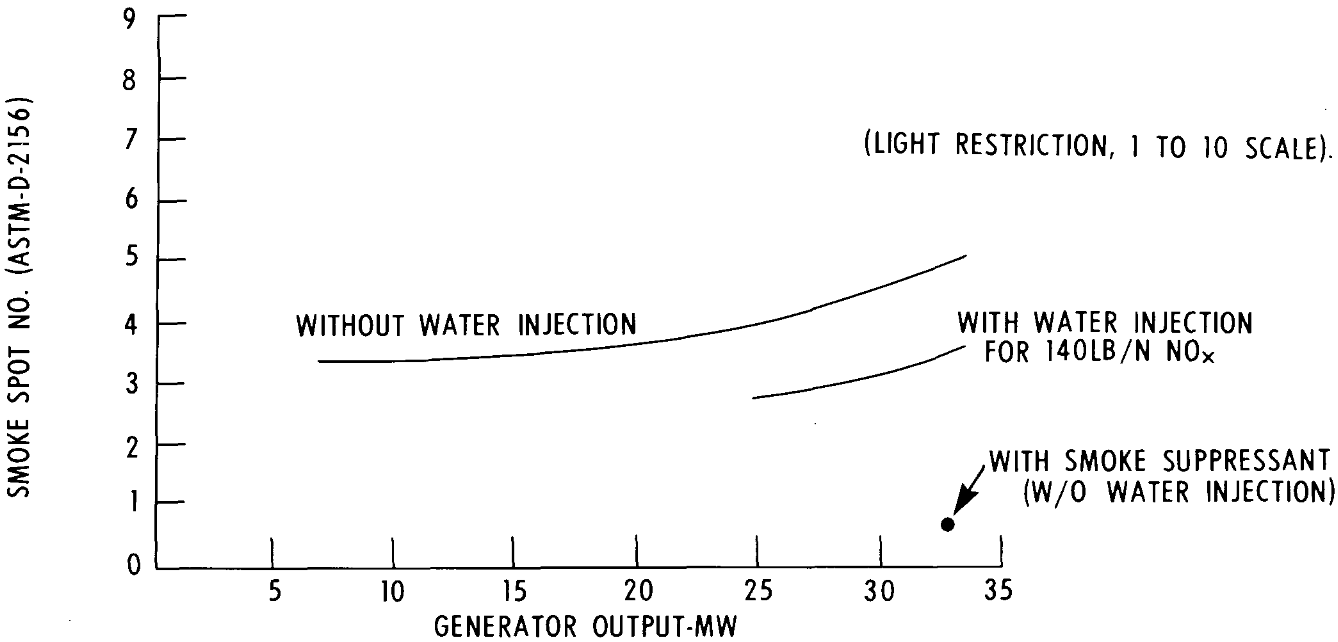


FIGURE 35 EFFECT OF WATER INJECTION ON SMOKE SPOT NUMBER
W-251 WESTINGHOUSE GAS TURBINE (NO. 2 DIST. OIL), 32.8 MW

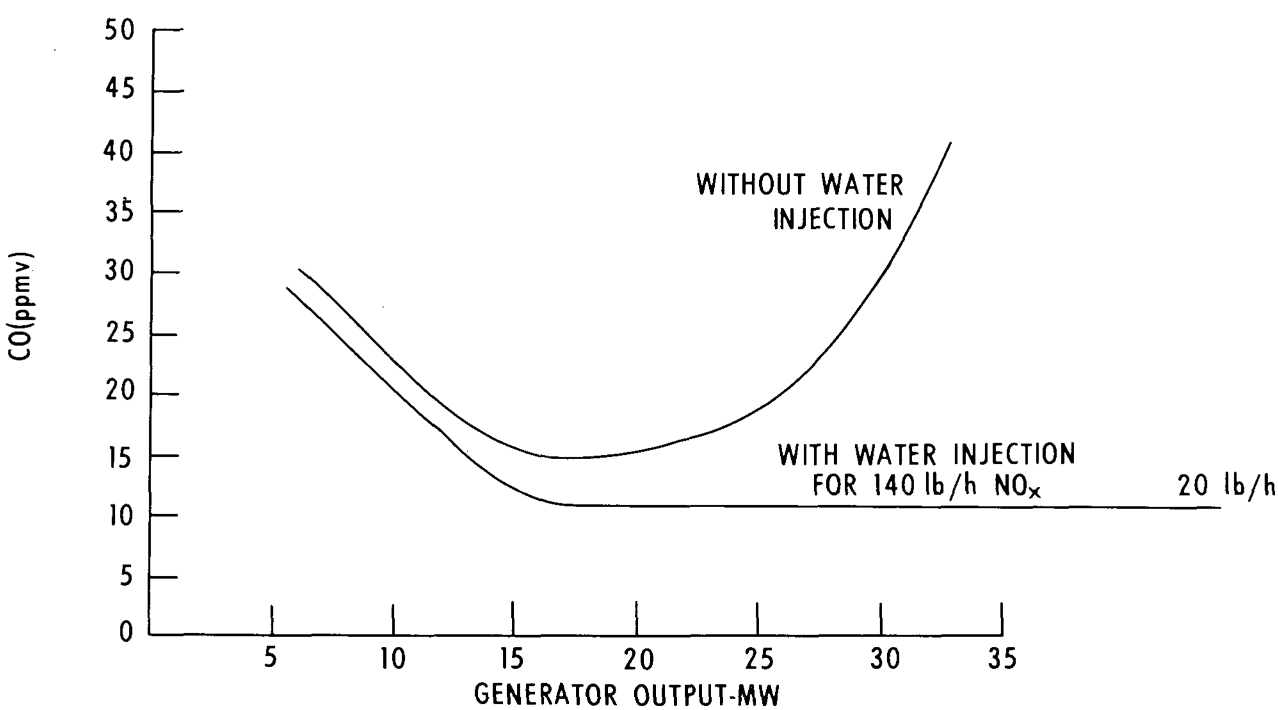


FIGURE 36 EFFECT OF WATER INJECTION ON CO EMISSION
W-251 WESTINGHOUSE GAS TURBINE (NO. 2 DIST. OIL), 32.8 MW

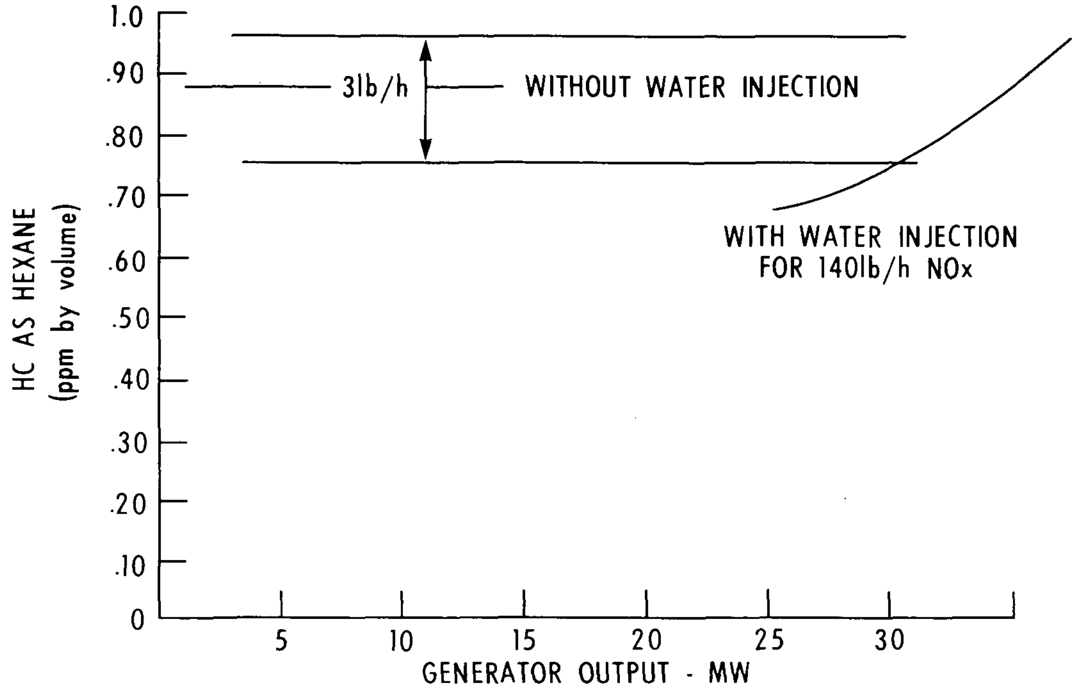
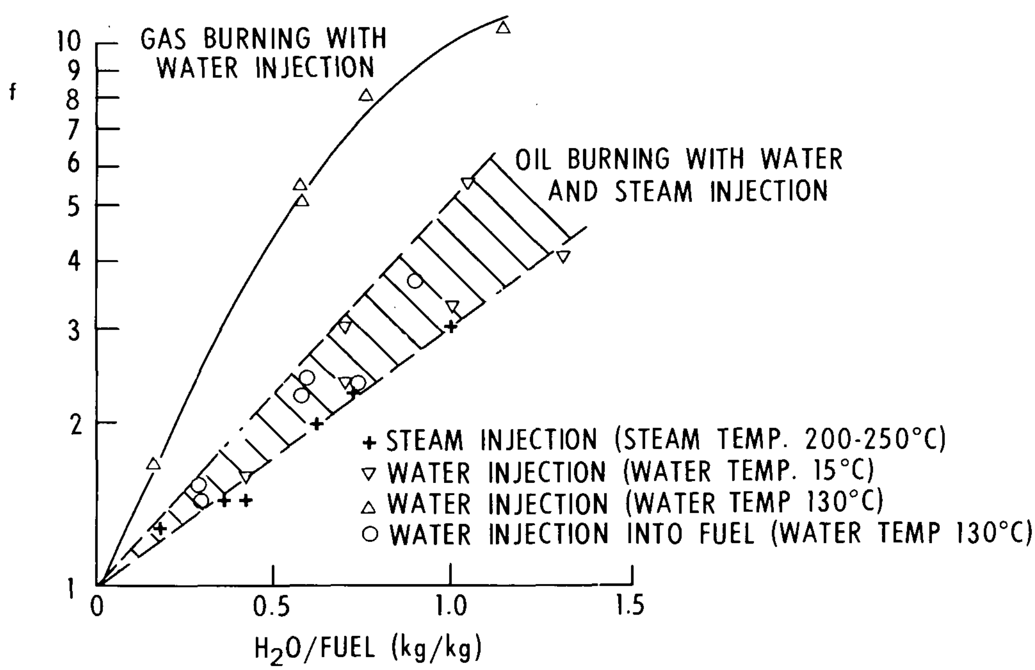


FIGURE 37 EFFECT OF WATER INJECTION ON HC EMISSION W-251 WESTINGHOUSE GAS TURBINE (NO. 2 DIST. OIL), 32.8 MW



$$f = \frac{NO_x(\text{WITHOUT } H_2O)}{NO_x(\text{WITH } H_2O)}$$

FIGURE 38 THE UPPER AND LOWER BAND NO_x REDUCTION OBTAINED OVER A RANGE OF WATER OR STEAM INJECTION. BROWN BOVERI-SULZER EXPERIMENTAL COMBUSTORS (NO. 2 DIST. OIL) however, evidence is available to show that HC, CO and NO_x can be simultaneously controlled to low levels.

In some cases manganese has been added to the fuel to decrease the opacity of the plume. This practice is questionable because the resulting smaller particles are reportedly a potential health hazard (6).

In summary, the test results obtained from 21 turbines ranging in size from 0.03 to 33 MW reveal the following:

a) NO_x emissions are consistently higher from oil-fired turbines than from the same units operated on gaseous fuel.

b) NO_x emissions from uncontrolled turbines of the same size can vary considerably, perhaps because of differences in design.

c) Although NO_x concentrations tend to increase with increasing turbine size, this is not necessarily a cause-effect relationship. Rather, it appears to result from the trend to operate larger turbines at higher temperatures and pressures to improve fuel economy.

d) The technology needed to achieve large reductions in NO_x emissions is available. Reductions in excess of 80% have been reported with wet control methods alone.

e) Reduction in NO_x emissions achieved with a given water/fuel injection ratio varies between turbines.

f) Reductions of 38% in NO_x emissions have been achieved with dry control methods.

g) CO and HC emissions tend to decrease with increasing turbine size.

Figure 39 is a compilation of results of tests by manufacturers, operators and the U.S. Environmental Protection Agency on six gas turbines operated with various NO_x control methods. Five units were tested on liquid fuels and three on gas. Emission levels with dry controls alone (plotted as control system A) represent average NO_x reductions of 37%.

Results of three tests by EPA show the reduction in NO_x emissions attainable with moderate levels (0.4 to 0.7 water/fuel ratio) of water or steam injection. These are presented as the results of tests on turbines 2 and 3.

Emissions of CO from a large oil-fired turbine operating with and without NO_x controls have been measured at about 11 and 34 ppm respectively.

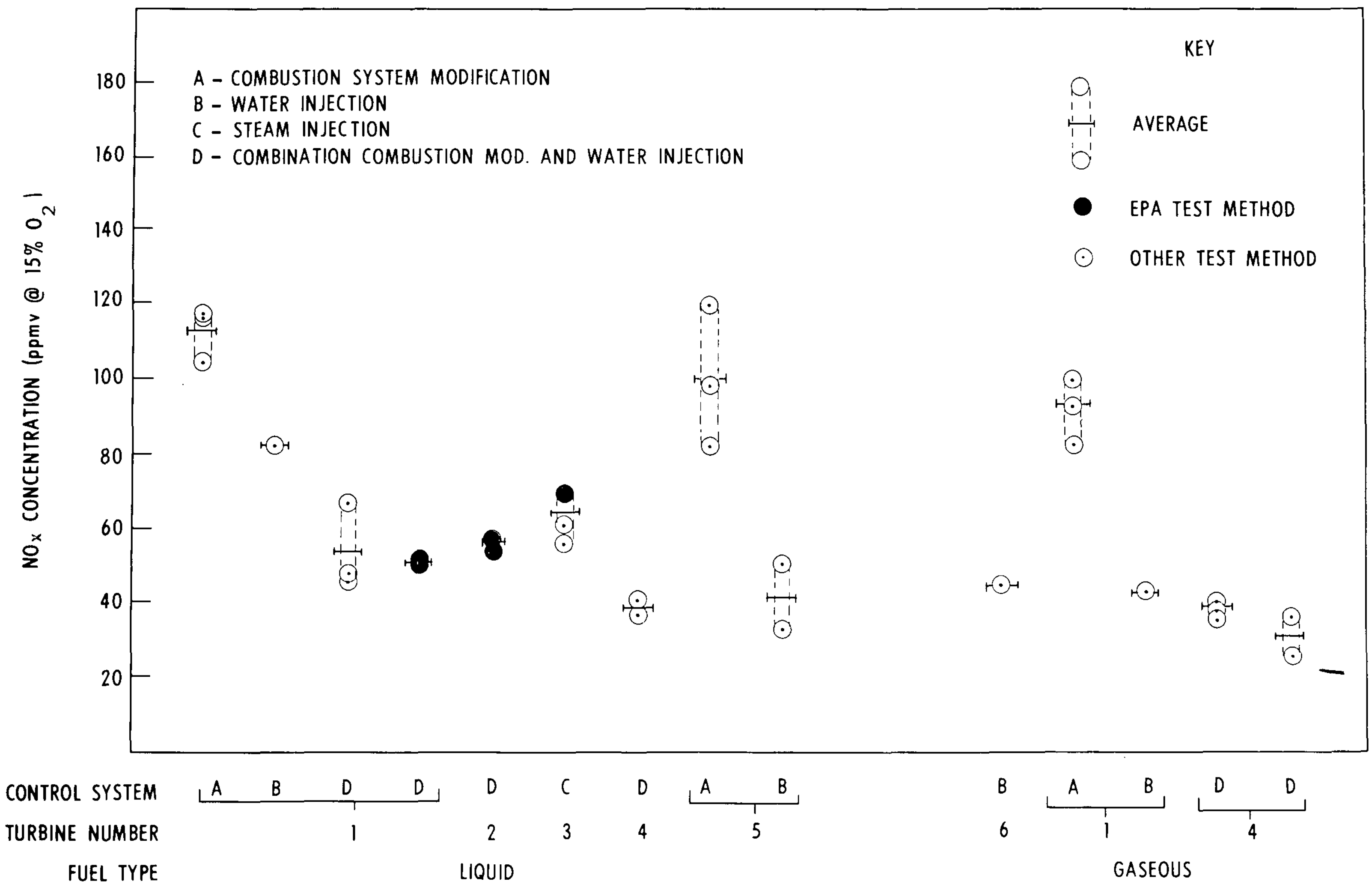


FIGURE 39 NO_x EMISSIONS FROM LARGE GAS TURBINES WITH NO_x CONTROLS

6 STATUS OF EXISTING EMISSION REGULATIONS AFFECTING DIESEL ENGINES AND GAS TURBINES

At the present time, there are no emission regulations for stationary internal combustion engines at either the provincial or federal level in Canada.

Elsewhere gas turbines are more heavily regulated for allowable emissions than diesel engines. The status of emission regulations affecting gas turbines in the United States and other nations can be found in references 31 and 32.

Most early emission measurement used observed concentration (ppm-volume) as a means of expressing emissions from diesel engines and gas turbines; however, this method underestimates emission levels because exhausts are diluted with the various amounts of excess air inherent in operation of unthrottled diesel engines (excess air approximately 100% at 100% load), and gas turbines (excess air approximately 250% at 100% load), which are not present in gasoline engines (excess air near zero at 100% load). Because of this, mass units are currently used by most testing laboratories for expressing diesel engine and gas turbine emissions.

The unit *grams/hour* is greatly affected by engine size, does not reflect the amount of work done, and would tend to encourage the use of smaller, less efficient engines. This can result in a net increase in air pollution.

The brake-specific emission concept, *grams/kilowatt hour,* seems to be the best basis for expressing mass emission. This basis encourages development of more efficient engines and relates emission rates to energy output, which is the primary cause of pollution. A separate study must be made, however, to solve problems related to the brake-specific emission concept; particularly, which power output should be used in the case of identical engines with identical fuel consumption, but different thermal efficiency (in the case of simple, regenerative and combined-cycle turbines, thermal efficiency varies from 25% to over 40%).

In the past, legislation against exhaust emissions, mainly CO and HC, has centred on the gasoline engine. This is because, compared with gasoline engines, diesel engines produce relatively small amounts of those gaseous pollutants currently of primary concern in North America. Diesel smoke is offensive to the senses and, although it presents a safety hazard by impairing visibility, public concern over it stems mainly from the ease with which it can be recognized compared with gaseous emissions. Most countries now have some form of legislation to deal with diesel smoke and restrictions may become tighter in the future.

In comparison with the effort expended on the gasoline engine, very little has been done regarding control of HC, CO, and NO_x emissions from diesel engines, gas engines and dual-fired engines.

7 EXHAUST EMISSION MEASUREMENT PROCEDURES

Both the size and operating conditions of stationary internal combustion engines are significantly different from those of engines used in motor vehicles. For example, vehicle load and speed demands on automotive engines are highly variable with time in comparison with the load-speed demands on stationary internal combustion engines used for power generation. Also special sample procurement techniques are required for measurement of emissions from stationary engines due to the required sample-line lengths and high oxygen content of the exhaust gas. Thus a separate measurement procedure is justified for these engines.

7.1 Intake Air or Exhaust Flow Rate Estimates

Instrument readings present the result in parts of gaseous pollutant emitted (volume) per million parts (volume) of exhaust gases. We wish to convert these volume readings to a mass reading, i.e., grams of specific pollutant emitted per kilowatt hour using measured or calculated values of fuel flow rate, air flow rate and power output.

The most difficult and complicated part of this technique is the accurate measurement of intake air or exhaust flow rate to determine mass emissions from measured concentrations.

The practice recommended (33) is use of a metering system and the associated equipment required to measure diesel engine gas flows at steady-state operating conditions. Commercial meters are available but, in many cases, especially outside of the manufacturer's testing laboratory, a special metering installation must be used because space is lacking for installation of the conventional model. Another difficulty arises from the necessity to use different sizes of flowmeters to measure different-size engines.

In the case of gas turbines, the use of an intake-air-metering system is especially impractical in the field.

Generally, the mass flow out of an engine equals the mass flow into the engine. Therefore, exhaust mass flow rate equals the intake-air mass flow rate plus the fuel mass flow rate. The basic equation is:

$$M_{exh} \leftrightarrows M_{air} + M_{fuel} \quad \text{.......................................(Equation 5)}$$

Using the equivalence ratio λ (Equation 1) and fuel consumption, the exhaust flow rate may be estimated from the equation:

$$M_{exh} \leftrightarrows M_{fuel} \times 14.5\lambda + M_{fuel} \text{ (lb/h)} \quad \text{..................(Equation 6)}$$

Typical equivalence ratios, λ, may be used as follows:

Four-stroke naturally aspirated diesel engines, dual-fuel engines and gaseous engines	$\lambda = 1.5$
Four-stroke supercharged diesel engines, dual-fuel engines and gaseous engines	$\lambda = 2.0$
Two-stroke scavenged and super-charged engines	$\lambda = 2.0$
Gas turbines	$\lambda = 3.5$

This calculation of exhaust gas flow rate is very often used when it is impossible to estimate flow rate by more accurate methods.

There are two ways to estimate air flow rate which are accepted by DEMA (6):

- By source testing using an intake-air metering system (orifice, venturs, pitot);
- By making a material balance across the entire process (using measured fuel rate, carbon dioxide concentration and oxygen concentration in exhaust, and fuel composition).

7.2 Gas Turbines

7.2.1 Sampling Techniques.

7.2.1.1 Large Gas Turbines. Emission measurement of most large gas turbines must be performed in the field, since manufacturing and test facilities are not normally equipped to dissipate the power generated by gas turbines operating at full load. Also, large gas turbines are usually assembled and performance tested on site.

Specific design and operating conditions of various turbines make sampling considerations difficult to generalize. The major characteristics affecting sampling methodology are:

- High stack temperature;
- Pollutant level concentrations below those associated with large steam stations;
- Stack-gas velocities in excess of 30 m/s;
- Velocity and concentration distributions which are nonuniform;
- Short stacks with internal baffling which limits locations where traverses can be performed;
- High flow turbulence;
- Large stack cross-sectional areas.

Gas turbine stacks are usually quite short and have relatively large cross sections (those of large gas turbines used for generating electrical power in Canada, for example, have a median height of 11m; the median area is 9m^2 in cross section).

In some gas turbine designs, cooling air is introduced into the stacks resulting in gas stratification.

To cope with these limitations, a traversing procedure has been devised (34) which representatively samples over the stack cross section. The equal-area approach of selecting test points is used. Multipoint sampling allows a better average concentration to be determined and also removes any distortion of the results due to dilution air.

The criteria used to determine the number of points required in the traverse were convenience of testing and the need to minimize the affect of stratified dilution air.

Instrumental methods were chosen because the low concentrations of pollutant encountered and the large number of sample points needed would be laborious and costly to obtain with manual techniques.

7.2.1.2 *Small Gas Turbines.* Options considered for emission measurements on small gas turbines were:

- Field measurements;
- In-plant emission measurements.

Because of cost, field tests are impractical for most small stationary gas turbines, and they are tested in the plant prior to delivery. These performance tests are conducted in specially instrumented test cells and the exhaust is normally vented to the outside in a 0.3· to 0.6m-diameter stack. An averaging device can be used to sample the exhaust gas in the stack, or, if the exhaust system is designed for good gas mixing, a single point sample can be used.

Due to the repetitive nature of the tests, and the fact that instrument technicians rather than chemists can be used to conduct these tests, instrumental procedures are less costly than manual techniques. The instrumentation is similar to that specified for jet-engine tests. Most turbine manufacturers have the equipment to make these measurements. If emission tests on small gas turbines are conducted in the field, the traversing procedures specified for large gas turbines are required and stack extensions must be installed for gas turbines without stacks.

7.2.1.3 *Measurment of SO_2.* Sulphur dioxide emissions can be estimated most accurately by a sulphur analysis of the fuel being burned; however, to remain consistent, an alternative emission test method using instrumental techniques is proposed for SO_2. Several instruments are commercially available which are sensitive enough to measure SO_2 at the 200 ppm level and below. These include nondispersive infrared (NDIR), nondispersive ultraviolet, (NDUV), and electrochemical methods.

7.2.1.4 *Measurement of CO.* Carbon monoxide concentration in emissions from small gas turbines is generally low (10-20 ppm) because of the good combustion conditions at full load. It is generally measured by nondispersive infrared techniques, which have been extensively researched. This methodology was, therefore, chosen. Manual techniques for CO measurement were rejected because their accuracy and repeatability are unacceptable at typical gas turbine exhaust concentrations.

7.2.1.5 *Measurement of NO_x.* Uncontrolled NO_x emissions from gas turbines can range from 20 to 250 ppm by volume. With NO_x control, instruments are required which can measure stack concentrations ranging from 20 to 100 ppm.

Experimental data, using the manual method, show that the method can be subject to excessive error at the low concentrations of NO_x normally associated with turbines. Therefore, instruments with instantaneous readout are the most desirable. The major manufacturers of gas turbines primarily use chemiluminescence and electrochemical instruments.

7.2.2 Suggested Procedure for Field Sampling.

a) A sampling site should be selected as close as is practical to the turbine exhaust. When possible, the sampling point should be selected before the introduction of dilution air into the stack. Sampling ports may be located before or after the upturn elbow in order to permit a complete stack diameter traverse. The sample ports should not be located within 5 ft or two diameters (whichever is less) from the turbine exhaust flange.

b) The minimum diameter of the sample ports should be 2.5 in.

c) The minimum number of sample points should be determined as:

- 8 for stacks up to 1.48 m^2
- 1 per 0.185 m^2 for stacks 1.48–9.3 m^2
- 1 per 0.37 m^2 for stacks 9.3 m^2 and greater

Figures 40 and 41 show the location of the traverse points for circular and rectangular stacks, respectively.

d) The turbine should be operated at the performance level specified in the applicable performance standard for a minimum of 10 min prior to the start of an emission test.

e) Samples should be taken at each point in the traverse for a minimum of 3 min, and the average steady-state concentration should be calculated for each point.

f) During the test period, a turbine operation record should be kept of time versus output energy of the turbine.

g) The sample port of small turbines should be located a minimum of 8 diameters downstream and two upstream of any obstruction.

h) For small turbines, a single point-sample may be sufficient, and should be located on the stack centreline.

i) The mass flow of gas through the turbine should be determined from the input flow of air, fuel and water into the turbine as determined by the manufacturer of the turbine, or measured by flow meter in the case of fuel and water. This total input flow should be adjusted to actual output conditions by correcting the flow to the average temperature measured at the gas sampling traverse points, which are believed to yield accurate results. Each traverse requires several hours to complete, with approximately 3 min allowed for readings at each traverse point. The single point tests are of 10 min duration.

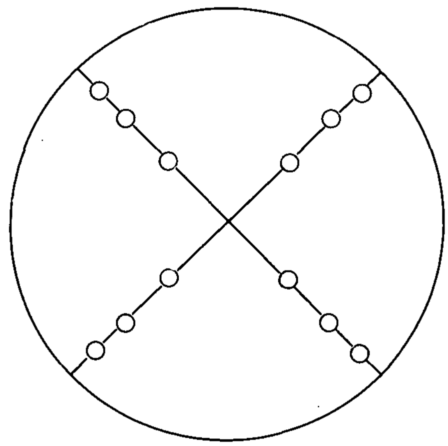


FIGURE 40 CROSS SECTION OF CIRCULAR STACK SHOWING LOCATION OF TRAVERSE POINTS ON PERPENDICULAR DIAMETERS

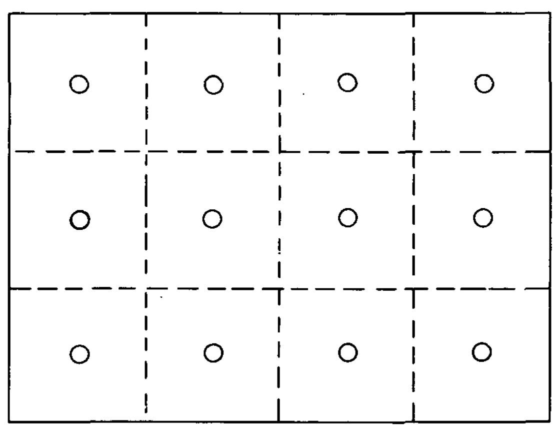


FIGURE 41 CROSS SECTION OF RECTANGULAR STACK DIVIDED INTO 12 SQUARE AREAS WITH TRAVERSE POINTS AT CENTROID OF EACH AREA

More detailed test and calculation procedures can be developed from aircraft and aircraft engine test procedures described in the U.S. Federal Register (35).

7.3 Reciprocating Engines

In developing the exhaust emission measurement procedure for stationary reciprocating engines, DEMA recognized that the existing SAE procedures, which were developed for vehicular engines, can be used only after substantial modification. The reasons are:

- Large diameter stacks require probes of an appropriate size, and sampling point locations can require sample lines 15 to 20 ft long.
- The mode of operation will not include variations of speed with load.

7.3.1 *Sampling Techniques.* The specified instrumentation was selected by DEMA:

- *Total Hydrocarbons* - measured by a heated flame ionization detector. The sample is to be kept in a hot, wet condition to prevent condensation of any of the hydrocarbons.
- *Oxides of Nitrogen* - A chemiluminescence analyzer was selected for the measurement of NO and NO_x.
- *Carbon Monoxide* - A nondispersive infrared analyzer was selected. Since the concentration would vary from 100 ppm to well over 1000 ppm for certain load conditions, two-cell nondispersive infrared analyzers were considered essential (each cell for a different range of concentration).
- *Oxygen* - Oxygen is measured by means of an instrument employing the paramagnetic effect.
- *Methane* - A gas chromatograph is used to measure the concentration of CH_4. Detection is by flame ionization.
- *Particulates* - The mass particulate measuring system consists of isokinetic sampling along a traverse of the exhaust stack. The weight of particulate collected over a period of time and the corresponding volume of exhaust gas flowing during the same period establishing the mass of particulate per unit volume of exhaust.
- *Opacity* - The opacity is to be measured with a light source, a detector, and a meter calibrated to read in percent light absorbed.

7.3.1.1 *Suggested Procedure for Testing.* The system is shown schematically in Figure 42.

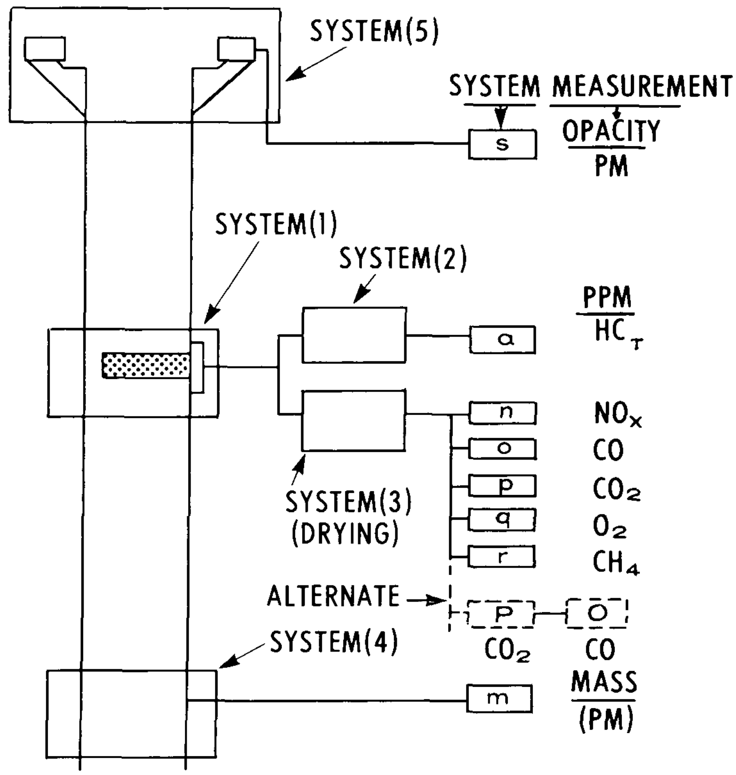


FIGURE 42 SCHEMATIC OF EXHAUST EMISSION MEASURING SYSTEM

The functions of the various sections are:

- System (1) is the sample collector for the analysis of all compounds except particulates. Its principal part is a probe installed in the exhaust stack and perforated along a length equal to 80% of the stack diameter.
- System (2) consists of the heated line and heated filter and delivers the sample in hot, wet, particulate-free condition to the heated flame ionisation detector for the measurement of HC.
- System (3) takes the sample from system (1) filters and cools the sample, removes the moisture, and delivers it to the various individual analysers, a,n,o,p,q, and r.
- System (4) and subsystem (m) cover the sampling and measurement of particulates.
- System (5) consists of a light source and detector for the measurement of opacity. System (s) is the readout.

The detailed test procedure for stationary reciprocating engines is published by DEMA(6).

8 NATIONAL EMISSION INVENTORY OF DIESEL AND GAS TURBINE ELECTRIC POWER GENERATION

The data discussed in this section are based on both published documents and on a survey of power generation stations conducted by questionnaire. The intention was to use these data to generate detailed information on total generating capacity by province, and to estimate emission rates and trends from the facilities.

The emission factors used to derive emission rates were discussed in Section 4.3 and are presented in Table 6.

8.1 Installed Capacity and Future Expansion

In Table 7 the installed capacity, number of units, the number of stations installed by province and total for Canada in 1973 are listed. The reported total megawatt output of reciprocating engines involved in electric power generation was 531.98 MW, and for gas turbines was 1163.02 MW.

In 1973 about 31% of the capacity for electric power generation using internal combustion engines was installed in the form of reciprocating engines, and 68% was installed as gas turbines. In 1973 about 78% of the reciprocating engine capacity was full diesel engines, about 15% was precombustion chamber engines, about 5% was dual-fuel engines, and about 2% gaseous fuel engines. About 65% of gas turbine capacity was oil fuel engines and 35% was gaseous fuel engines. Nearly all gas turbines are open-cycle installations with no heat recovery. Reported power generation figures indicate that reciprocating engines are running at about a 27% capacity factor on average, while gas turbines are running at about a 10% capacity factor. Capacity factors are low in electric power generation due to the use of engines and turbines primarily in peak shaving service.

The principal fuels are No. 2 fuel oil in the case of diesel engines and No. 2 fuel oil and natural gas in the case of dual-fuel engines and gas turbines. Only about 2% of reciprocating engine capacity, and less than 5% of gas turbine capacity, in Canada uses heavier fuels with a sulphur content greater than 0.8%.

Expansion of total electric generation beyond 1973, which includes units for service up to 1985, will add approximately 388 MW of diesel units and 1027 MW of gas turbine units, a total of 1415 MW, to Canada's generating capacity (see Table 8 and Figure 43).

8.2 Energy Generated and Emission Rates

The total energy generated in Canada and the estimated emission rates from electric power generation by internal combustion engines are presented in Table 8 and Figures 44 and 45.

In 1973, 16 640 tons of NO_x, 3850 tons of CO, 1740 tons of HC, 960 tons of particulate matter, and 2740 tons of SO_2 were discharged to the atmosphere from electric power generation by internal combustion engines.

TABLE 7 ELECTRIC POWER GENERATION INVENTORY, 1973

Province	Diesels		Turbines		Total	
	No. of Units	Capacity kW	No. of Units	Capacity kW	No. of Stations	Capacity kW
Newfoundland	227	61 190	2	28 300	69	89 490
P.E.I.	7	6 890	2	38 100	2	44 990
Nova Scotia	4	6 370	1	25 000	3	31 370
New Brunswick	8	6 850	1	23 380	4	30 230
Quebec	68	59 030	6	36 000	25	95 030
Ontario	27	34 520	35	472 800	17	507 320
Manitoba	21	33 150	2	27 800	5	60 950
Saskatchewan	16	33 150	6	88 880	7	122 030
Alberta	68	41 640	10	197 640	34	239 280
British Columbia	210	137 610	15	223 620	82	361 230
N.W. Territories	175	36 450	0	1 500	45	76 630
Yukon	48	36 450	0	0	15	36 450
TOTAL CANADA	879	531 980	81	1 163 020	309	1 695 000

In Table 9 emissions generated by electric power production in 1970, by internal combustion engines, are compared to emissions from all sources in Canada, by province, in the same year. It can be seen from the table, that HC and CO emissions from power generation are of minor importance relative to NO_x emissions which are more than ten times higher. The absolute magnitude of NO_x emissions from power generation, however, is still small in comparison to total emissions from all sources.

If the emission rates of projected units are the same as those of existing units, and no emission control is introduced, emissions to the atmosphere from Canada's electric power generation facilities, in 1985 will be approximately 28 000 tons of NO_x, 6000 tons of SO_2, 5500 tons of CO, 2000 tons of particulate matter, and 2000 tons of HC (see Figure 45).

TABLE 8 ELECTRIC POWER GENERATION, INSTALLED CAPACITY, ENERGY GENERATED AND EMISSIONS

	Year										
	1968	1969	1970	1971	1972	1973	1974	1975	1976	1977	1978
Energy generated - diesels kWh x 10^6	757.17	752.66	819.30	747.50	736.04	791.74	850.17	884.74	917.21	949.04	981.92
Energy generated - turbines kWh x 10^6	627.67	721.72	925.33	654.53	817.69	1000.77	1183.85	1290.80	1518.43	1713.22	1862.74
Energy generated - total kWh x 10^6	1384.84	1474.38	1744.63	1402.03	1553.73	1792.51	2034.02	2175.54	2435.64	2662.26	2844.66
Mass emission, tons/year											
NO_x	15256	15547	17825	15083	14766	16639	18371	19548	20575	21786	22499
CO	3655	3636	3980	3602	3479	3849	4141	4315	4453	4585	4721
HC	2274	2141	2228	1824	1644	1739	1798	1823	1829	1835	1838
Particulate matter	731	781	953	866	859	963	1091	1182	1247	1306	1363
SO_2	2128	2248	3291	2389	2365	2739	3335	3710	3937	4095	4258
Rating capacity - diesels MW	392.11	432.16	484.30	501.60	507.89	531.98	565.35	595.85	631.32	657.63	684.63
Rating capacity - turbines MW	847.75	841.59	898.59	983.20	994.50	1163.02	1268.82	1382.82	1625.12	1688.12	1688.15
Rating capacity - total MW	1239.86	1273.75	1382.89	1484.80	1502.39	1695.00	1834.17	1978.67	2256.44	2345.75	2372.78

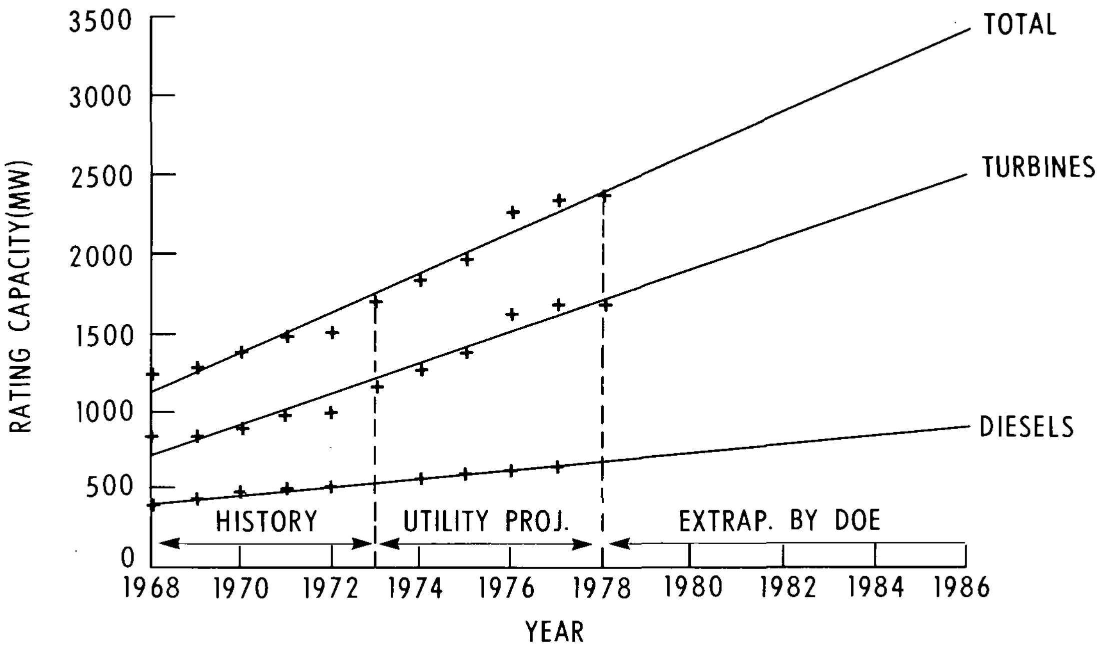


FIGURE 43 DIESEL AND GAS TURBINE GENERATION. RATING CAPACITY

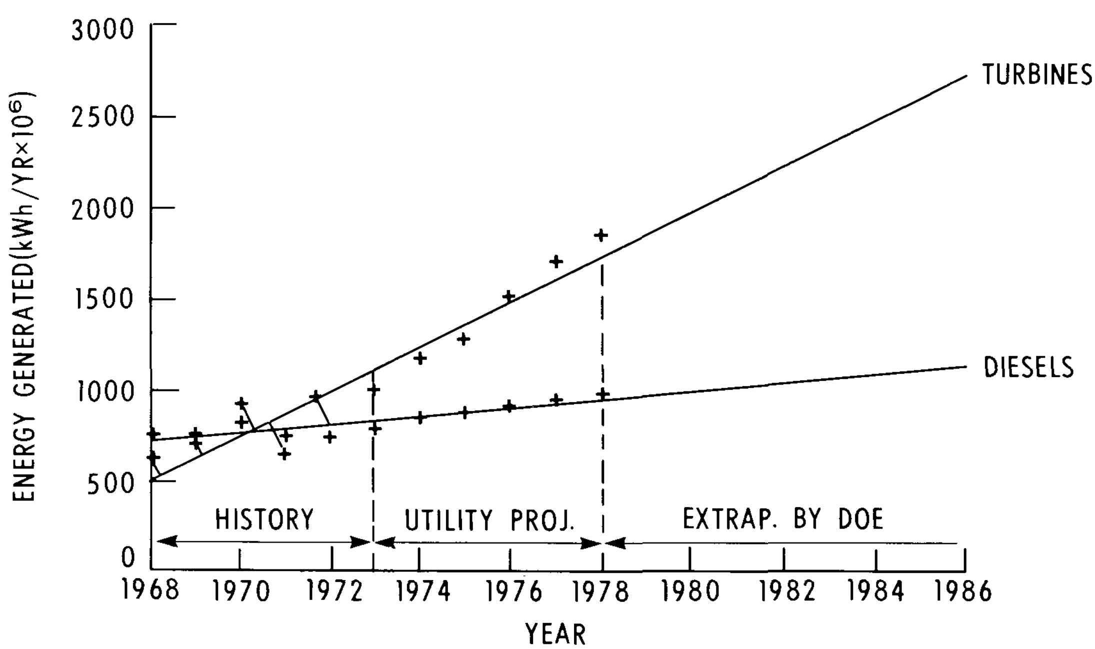


FIGURE 44 DIESEL AND GAS TURBINE GENERATION. ENERGY GENERATED IN CANADA

TABLE 9 EMISSION RATES FROM ELECTRIC POWER GENERATION BY INTERNAL COMBUSTION ENGINES VERSUS EMISSION RATES FROM ALL SOURCES IN CANADA, 1970 (30)

	Power Generation tons x 10^3			All Sources tons x 10^3			Percent of All Sources		
Province	NO_x	CO	HC	NO_x	CO	HC	NO_x	CO	HC
Newfoundland	0.45	0.13	0.03	25.38	63.46	314 66	1.77	0.20	0.01
P.E.I.	0.05	0.02	0.004	6.08	12.95	70.75	0.82	0.15	0.01
Nova Scotia	0	0	0	45.64	99.65	552.69	0	0	0
New Brunswick	0.22	0.07	0.02	36.52	78.91	456.79	0.60	0.09	0
Quebec	0.93	0.28	0.07	297.45	731.92	4217.51	0.31	0.04	0
Ontario	3.55	0.58	0.27	458.73	1009.69	5658.02	0.77	0.06	0.01
Manitoba	0.80	0.24	0.05	60.78	176.71	957.07	1.32	0.14	0.01
Saskatchewan	3.51	0.78	0.71	84.40	198.86	1110.86	4.16	0.39	0.06
Alberta	2.87	0.30	0.33	149.33	285.49	1595.18	1.92	0.11	0.02
British Columbia	3.98	1.15	0.62	112.81	334.28	2058.19	3.53	0.34	0.03
Yukon and N.W.T.	1.47	0.44	0.12	15.13	57.55	288.38	2.55	0.77	0.04
TOTAL CANADA	17.83	3.98	2.23	292.25	3049.47	17280.11	1.38	0.13	0.01

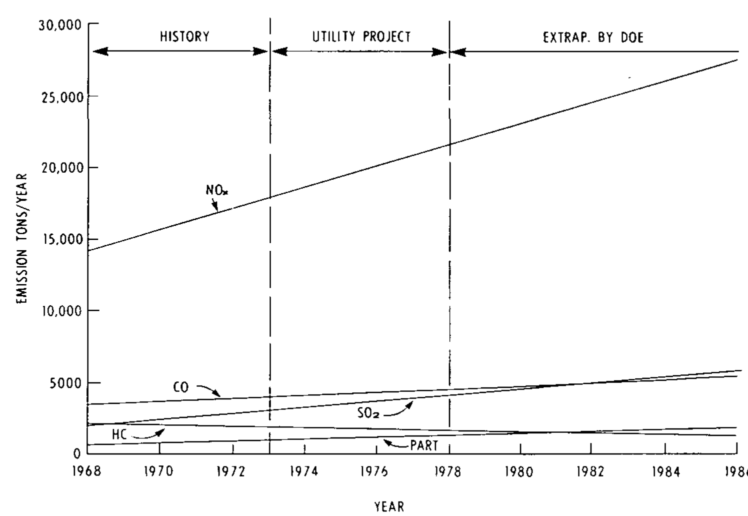


FIGURE 45 DIESEL & GAS TURBINE GENERATION. TOTAL EMISSION RATES

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APPENDIX – EMISSION DATA

EMISSION DATA

Direct Injection Diesels

Engine	BHP (Rated load)	BSFC (lb/Bhp-h)	Specific Emissions (g/Bhp-h) NO$_x$	CO	HC	Opacity (%)	Reference
1	1420	0.42	11.0	–	–	–	16
2	46.7	–	14.7	–	–	–	4
3	172	–	–	3.5	–	–	10
4	120	0.396	5.2	–	–	–	18
5	100	0.42	13.0	2.0	0.8	–	18
6	218	0.39	11.27	10.06	0.68	6	36
7	318	0.39	11.25	8.38	0.37	4	36
8	195	0.40	–	7.64	0.84	11	36
9	250	0.40	5.57	7.70	0.04		36
10	240	0.40	7.25	5.62	0.89	11	36
11	335	0.40	10.98	1.93	0.46	6	36
12	235	0.39	9.82	3.61	0.39	11	36
13	250	0.36	12.50	3.95	0.84	11	36
14	175	0.41	9.46	5.88	–	–	36
15	200	0.41	8.52	8.99	1.07	–	36
16	165	0.41	6.76	8.83	1.28	–	36
17	200	0.40	9.28	3.71	0.81	–	36
18	4354	–	11.34	3.82	–	–	37
19	4721	–	14.85	–	–	–	37
20	500	–	10.89	–	–	–	37
21	500	–	9.99	–	–	–	37
22	450	0.36	10.63	1.56	0.65	–	4
23	375	0.36	11.65	1.92	0.5	–	4
24	146	0.40	14.45	5.76	0.80	–	4
25	77.1	–	15.13	4.68		–	4
26	85.7	–	9.95	5.88	1.50	–	4
27	4296	0.36	10.98	3.85	0.13	–	38
28	230	0.42	15.78	1.09	1.19	–	10

Engine	BHP (Rated load)	BSFC (lb/Bhp-h)	Specific Emissions (g/Bhp-h) NO$_x$	CO	HC	Opacity (%)	Reference
29	162	0.376	9.0	0.50	0.4	–	11
30	232	–	9.58	–	–	–	39
31	900	–	12.45	0.9	0.35	–	40
32	900	–	12.15	1.7	0.35	–	40
33	900	–	13.51	5.1	0.85	–	40
34	190	0.43	9.35	5.28		–	41
35	230	–	15.78	1.09	1.19	–	41
36	150	–	8.1	–	–	–	18
37	232	–	9.58	–	–	–	42
38	200	0.375	9.58	3.0	1.0	–	12
39	34.5	–	7.4	12.0	1.1	–	4
40	53.6	–	5.7	5.04	1.8	–	4
41	57.1	–	–	1.7	–	5	4
42	58.1	–	15.20	5.88	1.8	–	4
43	68.1	–	8.08	6.72	1.30	–	4
44	73.0	–	5.85	6.84	0.35	–	4
45	162	–	11.9	1.0	0.6	–	43
46	46.7	–	12.5	7.32	0.4	–	4
47	148	–	11.65	–	–	–	41
48	50	–	14.6	2.5	1.2	–	4
49	148	–	11.65	–	–	–	10
50	400	–	7.01	6.60	–	6	44
51	160	–	6.38	7.91	–	12	44
52	185	–	7.23	10.74	–	12	44
53	210	–	6.80	7.35	–	9	44
54	210	–	5.10	6.22	–	7	44
55	280	–	5.95	2.83	–	3	44
56	325	–	5.95	3.39	–	7	44
57	165	–	5.31	5.40	–	12	44
58	220	–	7.20	8.0	–	6	44
59	365	–	11.25	4.8	–	5	44
60	350	–	10.39	4.0	–	5	44
61	400	–	9.90	5.6	–	6	44
62	295	–	7.65	8.8	–	10	44
63	350	–	7.65	3.2	–	5	44
64	320	–	7.65	3.2	–	3	44
65	240	–	7.65	5.6	–	8	44

Engine	BHP (Rated load)	BSFC (lb/Bhp-h)	Specific Emissions (g/Bhp-h) NO$_x$	CO	HC	Opacity (%)	Reference
66	350	–	11.70	4.0	–	5	44
67	400	–	9.45	4.80	–	9	44
68	450	–	9.81	3.20	–	2	44
69	149	–	8.33	15.20	–	8	44
70	202	–	9.90	12.0	–	2	44
71	216	–	8.10	8.0	–	13	44
72	240	–	8.55	4.0	–	5	44
73	635	–	9.90	4.0	–	4	44
74	315	–	11.25	4.8	–	6	44
75	230	–	8.55	4.8	–	4	44
76	600	–	8.55	1.6	–	2	44
77	350	–	7.65	4.8	–	7	44
78	230	–	6.75	6.40	–	9	44
79	136	–	10.88	13.69	–	10	44
80	210	–	11.31	9.52	–	6	44
81	250	–	10.01	17.26	–	12	44
82	219	–	6.96	9.52	–	9	44
83	333	–	10.44	11.31	–	7	44
84	309	–	7.83	10.71	–	10	44
85	456	–	9.57	9.82	–	7	44
86	280	–	9.14	5.36	–	5	44
87	258	–	6.96	7.14	–	9	44
88	182	–	8.99	4.76	–	3	44
89	198	–	9.57	5.36	–	2	44
90	262	–	6.96	2.38	–	3	44
91	350	–	9.57	4.76	–	6	44
92	130	–	9.57	5.95	–	7	44
93	285	–	8.70	16.66	–	13	44
94	380	–	10.88	10.12	–	9	44
95	322	–	9.57	3.57	–	4	44
96	430	–	9.57	4.76	–	4	44
97	360	–	7.40	2.98	–	4	44
98	190	–	8.85	3.96	–	13	44
99	350	–	9.15	2.60	–	11	44
100	215	–	10.64	1.80	–	8	44
101	190	–	10.64	1.40	–	7	44
102	180	–	9.55	7.00	–	12	44

Engine	BHP (Rated load)	BSFC (lb/Bhp-h)	Specific Emissions (g/Bhp-h) NO$_x$	CO	HC	Opacity (%)	Reference
103	250	–	10.79	3.0	–	4	44
104	235	–	9.86	2.75	–	9	44
105	280	–	11.21	4.0	–	6	44
106	322	–	10.51	3.0	–	12	44
107	300	–	7.52	1.78	–	11	44
108	355	–	6.23	2.0	–	12	44
109	80	–	7.65	3.9	–	7	44
110	120	–	5.5	3.9	–	6	44
111	145	–	6.0	1.8	–	7	44
112	190	–	9.35	3.0	–	4	44
113	215	–	10.63	3.6	–	6	44
114	155	–	8.25	10.32	–	12	44
115	205	–	9.78	6.0	–	6	44
116	134	–	6.80	3.60	–	10	44

EMISSION DATA

Precombustion Chamber Diesels

Engine	BHP (Rated load)	BSFC (lb/Bhp-h)	Specific Emissions (g/Bhp-h) NO$_x$	CO	HC	Opacity (%)	Reference
1	125	–	6.0	0.96	0.18	–	4
2	965	0.44	6.4	–	0.75	–	16
3	120	–	3.2	–	–	–	18
4	345	–	2.3	2.0	0.5	–	11
5	250	–	4.86	1.2	–	7	44
6	270	–	4.86	1.2	–	6	44
7	325	–	5.71	1.2	–	3	44
8	325	–	4.86	1.2	–	3	44
9	425	–	4.01	1.2	–	3	44
10	325	–	5.71	2.4	–	8	44
11	325	–	4.86	1.2	–	7	44
12	375	–	4.86	1.2	–	9	44
13	250	–	4.68	1.2	0.1	–	4
14	850	0.40	4.68	1.04	0.1	–	4
15	970	0.43	3.40	1.28	0.15	–	4
16	750	0.40	5.44	1.52	0.3	–	4
17	88.6	–	5.19	1.84	0.17	–	4
18	125	–	5.02	2.00	0.15	–	4
19	200	–	3.74	–	–	–	45
20	250	–	4.86	1.2	–	7	44
21	270	–	4.01	2.4	–	10	44
22	325	–	4.01	1.2	–	3	44
23	425	–	4.01	1.2	–	3	44
24	325	–	5.71	1.2	–	8	44
25	375	–	5.71	1.2	–	7	44
26	155	–	3.22	–	–	–	42
27	365	–	3.41	0.24	–	–	17
28	750	–	7.8	0.8–	0.05	–	4
29	100	–	5.6	1.6–	0.4	–	4
30	1070	0.42	–	–	1.25	–	16

Engine	BHP (Rated load)	BSFC (lb/Bhp-h)	Specific Emissions (g/Bhp-h) NO$_x$	CO	HC	Opacity (%)	Reference
31	140	-	5.2	-	-	-	18
32	140	-	5.6	-	-	-	18
33	200	-	3.74	-	-	-	46
34	42	0.42	2.4	1.6-	0.48	-	15

EMISSION DATA

Natural Gas and Dual-Fuel Engines

Engine	BHP (Rated load)	BSFC (lb/Bhp-h)	Specific Emissions (g/Bhp-h) NO_x	CO	HC	Reference
1	1600	6632	–	0.7	1.58	38
2	2140	7000	7.4	–	–	4
3	1940	7200	8.9	–	–	4
4	5233	6110	11.8	2.0	1.1	4
5	400	7177	7.6	–	–	4
6	800	7071	14.2	1.7	4.4	4
7	800	7071	9.4	6.0	–	4
8	1000	7067	–	7.4	–	4
9	2260	7500	9.6	–	–	4
10	1950	7500	12.1	–	–	4
11	1613	6099	9.7	3.8	4.5	4
12	1535	6409	10.9	3.0	3.2	4
13	3600	6123	9.1	1.8	4.1	4
14	3655	6108	7.4	2.5	4.4	4
15	2000	7067	8.5	4.6	5.4	4
16	225	8432	12.6	1.6	3.1	4
17	800	6509	9.1	0.5	–	4
18	1170	6509	14.4	1.3	2.5	4
19	496	6599	10.2	3.2	0.8	4
20	190	5975	12.8	2.4	9.3	4
21	800	–	10.9	–	2.0	4
22	310	–	12.1	6.3	3.5	4
23	1323	–	13.0	5.6	1.8	4
24	1218	–	13.42	0.2	–	37
25	5096	–	12.7	–	–	37
26	3810	–	12.75	–	–	37
27	7707	5760	7.7	0.61	1.9	4
28	4296	6340	8.96	4.50	5.16	38
29	669	–	5.12	1.5	–	37
30	4354	–	7.27	4.25	–	37

Engine	BHP (Rated load)	BSFC (lb/Bhp-h)	Specific Emissions (g/Bhp-h) NO$_x$	CO	HC	Reference
31	4721	-	7.12	-	-	37
32	1950	7300	14.1	-	-	4
33	4000	6500	10.4	-	-	4
34	925	7067	15.7	0.9	6.5	4
35	750	7063	12.5	1.1	-	4
36	1080	7070	15.23	0.29	1.94	4
37	1350	7700	10.0	-	-	4
38	400	7177	7.6	-	10.1	4

Emission Data

Gas Turbines (Natural Gas)

Engine	BHP (Rated load)	BSFC (BTU/Bhp-hr)	Specific emissions lb/10^6 Btu NO_x	CO	HC	PART	Reference
1	6 200	12 377	0.57				4
2	6 900	12 104	0.51				4
3	1 100	11 000	0.17				4
4	1 100	11 000	0.20				4
5	13 950	11 000	0.34				4
6	13 950	11 000	0.30				4
7	14 700	11 000	0.18				4
8	14 700	11 000	0.32				4
9	14 700	11 000	0.30				4
10	-	-	0.62				47
11	-	-	0.66				47
12	-	-	0.7				47
13	-	-	0.46				47
14	-	-	0.39				47
15	-	-	0.45				47
16	-	-	0.16				47
17	-	-	0.26				47

Emission Data

Gas Turbines (Distillate Fuel)

Engine	Engine Type	BSFC lb/hr	Specific emissions lb/10^6 Btu No_x	CO	HC	PART	Reference
1	JT-9D	16 641	1.0			0.30	48
2	JT-9D	17 052		0.03	0.009	0.01	13
3	JT-8D	8 400	1.01	0.05	0.009		48
4	JT-3D	10 835	0.74	0.06	0.02	0.04	13
5	JT-3D	9 420	0.63				48
6	JT-3C	10 183	0.64	0.05	0.005	0.03	13
7	JT-4A	15 511	0.83	0.07	0.002	0.07	13
8	JT-8D	8 755	1.23	0.25	0.005	0.02	13
9	Jumbo						
	CF 6	13 449		0.03	0.005	0.002	13
10	CJ-805	9 960	0.61	0.16	0.003	0.08	13
11	Rolls-						
	Royce						
	MK 511	7 625	1.09	0.10		0.11	13
12	All						
	T56-A 15	2 393	0.63	0.09	0.01	0.08	13
	T56-A 7	2 079	0.60	0.06	0.01	0.1	13
13	TPE	365	0.54	0.06	0.008	0.12	13
14	TP4-2		0.89				47
15	P & W		0.9				47
16	FT 45C			0.041	0.003		47
17	TP4-2		0.96				47
18	MS5001			0.038	0.003		47
19	MS5001		0.78	0.031			47
20	MS5001		0.55				47
21	MS7001-						
	SC		1.1				47
22	GG4A		0.43			0.02	47
23	GG4A		0.46			0.02	47
24	FT4		0.57				47
25	FT4SC			0.031	0.002		47
26	FT4SC		0.77				47

27	GG4A	0.33	0.038	47
28	MS5001	0.67		47
29	MS5001	0.95		47
30	MS5001-N	0.92		47
31	MS5001-N	0.71		47
32	W251-SC	0.94		47
33	W501-SC	0.84		47
34	W251-AA	0.65		47
35	W251	0.88		47
36	W501	0.88		47

GLOSSARY OF SYMBOLS AND ACRONYMS

A/F	Air/fuel ratio	–
ASTM	American Society for Testing Materials	–
BFC	Fuel Consumption	kg/h, lb/h
BMEP, Pe	Brake mean effective pressure	kg/cm^2, kp/cm^2, psi
BSFC	Brake specific fuel consumption	g/Bhp-h, lb/Bhp-h, g/kWh, cu.ft/Bhp-h, Btu/Bhp-h
BTDC	Before top dead centre	degrees
CA	Fuel injection advance (Crank angle)	degrees
CF	Conversion factor	–
D.F.	Dual fuel engine	–
DEMA	Diesel Engine Manufacturers Association	–
D.I.	Direct injection diesel engine	–
EF_1, EF_2	Emission factors	g/Bhp-h, lb/Bhp-h, g/kWh, $lb/10^6$ Btu, $lb/10^3 lb$ fuel
EGR	Exhaust gas recirculation	–
EPA	Environmental Protection Agency (USA)	–
F/A	Fuel/air ratio	–
G	Gas reciprocating engine	–
I.D.I.	Indirect injection diesel engine (Prechamber diesel engine)	–
LHV	Lower heating value of the fuel	cal/kg, cal/m^3, Btu/lb Btu/cu.ft, Btu/gallon
M_{air}	Air flow rate	kg/h, lb/h
M_{exh}	Exhaust flow rate	kg/h, lb/h
M_{fuel}	Fuel flow rate	kg/h, lb/h
N.A.	Naturally aspirated engine	–
NG	Natural gas	–
P.C.	Prechamber diesel engine	–
PPMC	Parts per million as carbon	ppm
Q, Q_1	Heat	cal, Btu
SAE	Society of Automotive Engineers	–
SC	Scavenged 2-stroke diesel engine	–

SME	Specific mass emission	g/Bhp–h, lb/Bhp–h, g/kWh
T.C.	Turbocharged engine	–
TCA	Turbocharged and aftercooled engine	–
V	Total cylinder displacement	cu.in, litres
υ	Volume	m^3, scf
λ	Equivalence ratio Ratio of the actual and stoichiometric air/fuel ratios.	–
tp	Time	secs., milliseconds