

# REPORT ON REVISIONS TO

## 5TH EDITION AP-42

### Section 3.3

## Gasoline and Diesel Industrial Engines

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## 1.0 INTRODUCTION

This report supplements the Emission Factor (EMF) Documentation for AP-42 Section 3.3, Gasoline And Diesel Industrial Engines, dated April, 1993. The EMF describes the source and rationale for the material in the most recent updates to the 4th Edition, while this report provides documentation for the updates written in both Supplements A and B to the 5th Edition.

Section 3.3 of AP-42 was reviewed by internal peer reviewers to identify technical inadequacies and areas where state-of-the-art technological advances need to be incorporated. Based on this review, text has been updated or modified to address any technical inadequacies or provide clarification. Additionally, emission factors were checked for accuracy with information in the EMF Document and new emission factors generated if recent test data were available.

If discrepancies were found when checking the factors with the information in the EMF Document, the appropriate reference materials were then checked. In some cases, the factors could not be verified with the information in the EMF Document or from the reference materials, in which case the factors were not changed.

Four sections follow this introduction. Section 2 of this report documents the revisions and the basis for the changes. Section 3 presents the references for the changes documented in this report. Section 4 presents the revised AP-42 Section 3.3, and Section 5 contains the EMF documentation dated April, 1993.

## 2.0 REVISIONS

### 2.1 General Text Changes

Information in the EMF document was used to enhance text concerning emissions and controls. Also, at the request of the EPA, the metric units were removed.

### 2.2 Emission Factors

All emission factors (NO<sub>x</sub>, CO, SO<sub>x</sub>, PM-10, TOC, organic compounds, etc.) were checked against information in the EMF Document and no changes were necessary.

### 2.3 Carbon Dioxide, CO<sub>2</sub>

CO<sub>2</sub> emission factors in Table 3.3-2 were originally calculated assuming 100% conversion of fuel carbon content to CO<sub>2</sub>; however; 1% of liquid fuels typically pass through the combustion process unoxidized.<sup>(1-6)</sup> The CO<sub>2</sub> factors in Table 3.1-1 were modified to reflect 99% conversion.

### 3.0 REFERENCES

1. G. Marland and R. M. Rotty, *Carbon Dioxide Emissions From Fossil Fuels: A Procedure For Estimation And Results For 1951-1981*, DOE/NBB-0036 TR-003, Carbon Dioxide Research Division, Office of Energy Research, U. S. Department of Energy, Oak Ridge, TN, 1983.
2. A. Rosland, *Greenhouse Gas Emissions in Norway: Inventories and Estimation Methods*, Oslo: Ministry of Environment, 1993.
3. *Sector-Specific Issues and Reporting Methodologies Supporting the General Guidelines for the Voluntary Reporting of Greenhouse Gases under Section 1605(b) of the Energy Policy Act of 1992* (1994) DOE/PO-0028, Volume 2 of 3, U.S. Department of Energy.
4. G. Marland and R. M. Rotty, *Carbon Dioxide Emissions From Fossil Fuels: A Procedure For Estimation And Results For 1950-1982*, Tellus 36B:232-261, 1984.
5. *Inventory Of U. S. Greenhouse Gas Emissions And Sinks: 1990-1991*, EPA-230-R-96-006, U. S. Environmental Protection Agency, Washington, DC, November 1995.
6. *IPCC Guidelines For National Greenhouse Gas Inventories Workbook*, Intergovernmental Panel on Climate Change/Organization for Economic Cooperation and Development, Paris, France, 1995.

#### 4.0 REVISED SECTION 3.3

This section contains the revised Section 3.3 of AP-42, 5th Edition.

## 5.0 EMISSION FACTOR DOCUMENTATION, APRIL 1993

This section contains the Emission Factor Documentation for Section 3.3 dated April 1993.

**EMISSION FACTOR DOCUMENTATION FOR**  
**AP-42 SECTION 3.3,**  
**GASOLINE AND DIESEL INDUSTRIAL ENGINES**

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### **Disclaimer**

**This report has been reviewed by the Office of Air Quality Planning and Standards, U.S. Environmental Protection Agency, and approved for publication. Mention of trade names or commercial products does not constitute endorsement or recommendation for use.**

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## **1. INTRODUCTION**

**An emission factor is an estimate of the rate at which a pollutant (in terms of its mass) is released to the atmosphere divided by the level of activity from the emission source. Section 3.3 of the "Compilation of Air Pollutant Emission Factors" (AP-42) covers the emission factors for gasoline (up to 250 hp) and diesel (up to 600 hp) industrial engines. The emission factors provide persons working in air pollution control with documented estimates of source emission rates. Uses of emission factors reported in AP-42 include:**

- ! Estimates of area-wide emissions;**
- ! Emission estimates for a specific facility; and**
- ! Evaluation of emissions in relation to ambient air quality.**

**The intent of this emission factor document is to provide background information used to support the revision of emission factors for AP-42 Chapter 3.3 - Gasoline and Diesel Industrial Engines.**

**The last update of AP-42 Chapter 3.3 was in 1975 and contained only emission factors for carbon monoxide (CO), volatile organic compounds (VOCs) [i.e., exhaust hydrocarbons (HC), evaporative HC, crankcase HC], nitrogen oxides (NO<sub>x</sub>), aldehydes, sulfur oxides (SO<sub>x</sub>), and particulate matter (PM) for baseline (uncontrolled) operation. This revision includes emission factors for those species as well as for carbon dioxide (CO<sub>2</sub>) and speciated organic compounds. The overall scope of the current revision includes the following changes or additions:**

- ! Review of existing criteria pollutant emission factors for uncontrolled baseline operation using data available since the prior supplement;**

- ! **Inclusion of several non-criteria emission species for which data are available: organics speciation, air toxics, particle sizing, and greenhouse gases (CO<sub>2</sub>); and**
- ! **Inclusion of technical discussion and emission control factors for engines operating with NO<sub>x</sub>, CO, VOC or diesel particulate controls.**

**AP-42 Chapter 3.3 deals with both types of reciprocating internal combustion engines, namely spark and compression ignition. The chapter treats industrial-sized compression ignition diesel engines and the industrialized spark ignition engines fired with gasoline. Larger diesel engines are addressed in AP-42 chapter 3.4. Larger spark ignition engines are fired with natural gas and are covered in Chapter 3.3. In compression ignition engines, the combustion air is compression heated in the cylinder before the diesel fuel oil is injected into the cylinder to produce spontaneous combustion. Spontaneous ignition occurs because the air is above the auto-ignition temperature of the fuel. In spark ignition engines, the gasoline uses the spark on an electrical discharge to initiate combustion.**

## **2. SOURCE DESCRIPTION**

### **2.1 CHARACTERIZATION OF THE INDUSTRY**

**Stationary (nonroad) reciprocating internal combustion (IC) engines are found in a variety of applications where there is a requirement for mechanical work which can be derived from the power generated by a shaft. The engine category addressed by Chapter 3.3 covers industrial applications by both gasoline and diesel internal combustion power plants, such as fork lift trucks, mobile refrigeration units, generators, pumps, heavy-duty farm and construction engines, and portable well-drilling equipment.**

**Nonroad engines cover a wide variety of equipment from lawn mowers and chain saws, to recreational equipment, to agricultural and construction machinery, and industrial equipment. There is an area of ambiguity in defining which engines are mobile or stationary because the engine designs can be used in either application. Accordingly, information for stationary engines may be contained in mobile source documents.**

**Nonroad engines are not regulated for emissions, and very few currently use emission control technology. Because of the diversity of nonroad equipment, characterization of the emissions from nonroad engines is a complex task.**

**Compression-ignition engines can operate at a higher compression ratio (ratio of cylinder volume when the piston is at the bottom of its stroke to the volume when it is at the top of its stroke) than spark-ignited engines because fuel is not present during compression; thus, there is no danger of premature automatic ignition. Since the thermal efficiency of an engine rises with increasing pressure ratio (and pressure ratio varies directly with compression ratio), compression-ignited engines are more efficient than spark-ignited engines. This increased efficiency is gained at the expense of poorer**

acceleration (response to load changes) and a heavier structure to withstand the higher pressures.<sup>1,2,3</sup>

## **2.2 PROCESS DESCRIPTION<sup>4,5</sup>**

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

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

### **2.2.1 Fuel Type**

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

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

Natural gas is the dominant fuel for large stationary IC engines, which typically operate pumps or compressors on gas pipelines.

Other fuels are burned in stationary IC engines, but their use is limited. Some larger engines fire heavy fuel oils, and a few fire waste gaseous or liquid fuels. Gaseous fuels such as sewer gas are sometimes used at wastewater treatment plants. Stationary IC

engines can be modified to burn almost any liquid or gaseous fuel if the engine is properly designed, adjusted and maintained.

### **2.2.2 Method of Ignition**

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

In CI engines, combustion air is first compression heated in the cylinder, and diesel fuel oil is then injected into the hot air. At this point, the temperature of the air is high enough to cause the fuel to ignite spontaneously (automatic ignition). SI engines initiate combustion by the spark of an electrical discharge. Usually the fuel is mixed with the air in a carburetor (for gasoline) or at the intake valve (for natural gas), but occasionally the fuel is injected into the compressed air in the cylinder. Although all diesel fueled engines are compression ignited and all gasoline and gas fueled engines are spark ignited, gas can be used in a CI engine if a small amount of diesel fuel is injected into the compressed gas/air mixture to burn any mixture ratio of gas and diesel oil, from 6 to 100 percent oil (based on heating value).

In SI engines, fuel and air are drawn into the cylinder together and are intended to form a homogeneous mixture of air and vapor by the time of the electrical discharge (spark) to initiate ignition, toward the end of the compression stroke. After the passage of a spark, the flame then progresses through the mixture until all of the fuel is consumed. If the compression ratio of a gasoline engine is significant enough to make the air and fuel mixture temperature too high, then some of the mixture will autoignite and burn so quickly that it will rattle the engine parts. This engine noise is called knock (or detonation).

### **2.2.3 Combustion Cycle**

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

In the four-stroke cycle, the sequence of events can be summarized as follows:

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

Two-stroke engines need only two strokes of the piston or one revolution to complete a cycle. Thus, there is a power stroke during every revolution instead of every two revolutions as with four-stroke engines. As the piston moves to the top of the cylinder, air or an air and fuel mixture is compressed for ignition. Following ignition and combustion, the piston delivers power as it moves down through the cylinder. Eventually the piston uncovers the exhaust ports (or exhaust valves open). As the piston begins the next cycle, exhaust gas continues to be purged from the cylinder, partially by the upward motion of the piston and partially by the scavenging action of the incoming fresh air. Finally, all ports are covered (and/or valves closed), and the fresh charge of air or air and fuel is again compressed for the next cycle.

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

#### **2.2.4 Charging Method**

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

**A naturally aspirated engine uses the vacuum created behind the moving piston during the intake stroke to suck in the fresh air charge. This process tends to be somewhat inefficient, however, since the actual amount of air drawn into the cylinder is only about 50 to 75 percent of the displaced volume. A more efficient method of charging is to pressurize the air or air and fuel mixture and force it into the cylinder with either a turbocharger or a supercharger. The turbocharger is powered by a turbine that is driven by the energy in the relatively hot exhaust gases, while a supercharger is driven off the engine crankshaft. Air pressurization increases the power density, or power output per unit weight (or volume) of the engine, since more air mass can be introduced into the cylinder. As air pressure increases, its temperature also rises because of the action of the compressor on the air. Therefore, the pressurized air is often cooled before entering the cylinder to further increase power by allowing more air mass to be introduced into the cylinder. This process is called intercooling or aftercooling. Two stroke engines are often aircharged by a blower, which also aids in purging the exhaust gases. Such systems are called blower-scavenged. This method is less efficient than turbocharging because the blower produces less pressure than a turbine. However, high volumetric flow rates are achieved, effectively purging the cylinder of exhaust gases.**

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

**Two methods of injection are commonly used. Direct injection places the fuel directly into the cylinder and the principal combustion chamber. These units are also**

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

### **2.3 EMISSIONS**

Most of the pollutants from IC engines are emitted through the exhaust. However, some HCs escape from the crankcase as a result of blowby (gases which are vented from the oil pan after they have escaped from the cylinder past the piston rings) and from the fuel tank and carburetor because of evaporation. Nearly all of the HCs from diesel (CI) engines enter the atmosphere from the exhaust. Crankcase blowby is minor because hydrocarbons are not present during compression of the charge. Evaporative losses are insignificant in diesel engines due to the low volatility of diesel fuels. In general, evaporative losses are also negligible in engines using gaseous fuels because these engines receive their fuel continuously from a pipe rather than via a fuel storage tank and fuel pump. In gasoline-fueled engines, however, 20 to 25 percent of the total hydrocarbon emissions from uncontrolled engines come from crankcase blowby and another 10 to 15 percent from evaporation of the fuel in the storage tank and the carburetor. However, crankcase blowby emissions can be virtually eliminated through the simple expedient use of the positive crankcase ventilation (PCV) valve. Additional fugitive emissions are possible from fuel storage and transport. These emissions are covered in AP-42 Chapter 4.

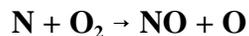
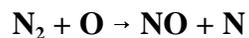
The primary pollutants from internal combustion engines are oxides of nitrogen ( $\text{NO}_x$ ), hydrocarbons and other organic compounds, CO, and particulates, which include both visible (smoke) and nonvisible emissions. Nitrogen oxide formation is directly related to high pressures and temperatures during the combustion process and to the nitrogen content, if any, of the fuel. The other pollutants, HC, CO, and smoke, are primarily the result of incomplete combustion. Ash and metallic additives in the fuel also contribute to the particulate content of the exhaust. Sulfur oxides also appear in the exhaust from IC

engines. The sulfur compounds, mainly sulfur dioxide (SO<sub>2</sub>), are directly related to the sulfur content of the fuel.<sup>2</sup>

### 2.3.1 Nitrogen Oxides

Nitrogen oxide formation occurs by two fundamentally different mechanisms. The predominant mechanism with internal combustion engines is thermal NO<sub>x</sub> which arises from the thermal dissociation and subsequent reaction of nitrogen (N<sub>2</sub>) and oxygen (O<sub>2</sub>) molecules in the combustion air. Most thermal NO<sub>x</sub> is formed in the high temperature region of the flame from dissociated molecular nitrogen in the combustion air. Some NO<sub>x</sub>, called prompt NO<sub>x</sub>, is formed in the early part of the flame from reaction of nitrogen intermediary species, and HC radicals in the flame. The second mechanism, fuel NO<sub>x</sub>, stems from the evolution and reaction of fuel-bound nitrogen compounds with oxygen. Natural gas, gasoline, and most distillate oils have no chemically-bound fuel N<sub>2</sub> and essentially all NO<sub>x</sub> formed is thermal NO<sub>x</sub>. Residual oils and many liquid wastes have fuel-bound N<sub>2</sub> and when these are fired in engines, NO<sub>x</sub> is formed by both mechanisms. The formation of prompt NO<sub>x</sub> is only significant in very fuel-rich flames and is of no significant importance with relation to reciprocating IC engines.

At high temperatures (thermal NO<sub>x</sub>), both N<sub>2</sub> and O<sub>2</sub> molecules in the combustion air absorb the heat energy up to the point where they are dissociated into their respective atomic states, N and O. The subsequent reaction of these atoms to create thermal NO<sub>x</sub> is described by the Zeldovich mechanism:



The rates of these reactions are highly dependent upon the stoichiometric ratio, combustion temperature, and residence time at the combustion temperature.

The maximum thermal NO<sub>x</sub> production occurs at a slightly lean fuel mixture ratio because of the excess availability of oxygen for reaction. The control of stoichiometry is critical in achieving reductions in thermal NO<sub>x</sub>. The thermal NO<sub>x</sub> generation decreases rapidly as the temperature drops below the adiabatic temperature (for a given

stoichiometry). Maximum reduction of thermal  $\text{NO}_x$  generation can thus be achieved by control of both the combustion temperature and the stoichiometry.

With fuel  $\text{NO}_x$ , the nitrogen compounds (primarily organic) contained in the fuels are evolved and react to form  $\text{NO}_x$ . The degree of oxidation of the nitrogen in the fuel is strongly dependent upon the stoichiometric ratio and fuel nitrogen concentration, and weakly dependent upon the flame temperature and the nature of the organic nitrogen compound. It is the weak influence of temperature on gas-phase  $\text{NO}_x$  conversion that reduces the effectiveness of  $\text{NO}_x$  controls which rely on temperature effects in the combustion of nitrogen-bearing fuels. Here, as with thermal  $\text{NO}_x$ , controlling excess  $\text{O}_2$  (stoichiometry) is an important part of controlling  $\text{NO}_x$  formation.

Because of the high flame temperatures and pressures of IC engines, the majority of  $\text{NO}_x$  formed is thermal  $\text{NO}_x$ . As diesel fuel and natural gas are the predominate fuels for this source, little fuel  $\text{NO}_x$  is formed, except in engines that fire residual and/or crude oils.

When fuel is injected into the cylinder, it undergoes a series of reactions that lead to ignition. The time between the start of injection of the fuel and the start of combustion (as measured by the onset of energy release) is called the ignition delay. Initial combustion occurs around the periphery of the fuel jet, where the air/fuel ratio is close to the stoichiometric ratio. During ignition delay, some of the fuel is premixed with air and evaporates. After ignition occurs, the premixed charge burns extremely rapidly, thereby quickly releasing energy. Most of the burning takes place as a diffusion flame after the premixed charge has burned.

Nitrogen oxide emissions are directly affected by the amount of premixing which, in turn, is a function of the ignition delay. When the ignition delay is large, there is more premixing and a greater energy release rate at the start of combustion. This generally leads to higher temperatures and, accordingly, higher  $\text{NO}_x$  emissions.

In general, engine load does not have a profound effect on the brake-specific ( $\text{NO}_x$  rate to power output ratio)  $\text{NO}_x$  emission rates for diesel-fueled engines, although the total mass emission rates increase as the engine load increases. At very low engine loads, almost all of the energy is released during the premixed stage. Consequently, brake-specific

emissions under these conditions are relatively high. As load increases, the amount of premixed burning remains relatively constant while the amount of diffusion burning increases linearly. The amount of  $\text{NO}_x$  produced during this stage is proportional to the amount of fuel consumed because most of the diffusion burning takes place at stoichiometric conditions. Thus, as engine load increases, the concentration of  $\text{NO}_x$  in the exhaust gas increases. However, the brake-specific  $\text{NO}_x$  emission rate remains roughly the same since power output also increases by the same factor.

Brake-specific  $\text{NO}_x$  emission rates for dual-fuel compression ignition engines are sensitive to load. Emission rates are greatest at high loads. Dual-fuel engines generally burn a homogeneous charge of fuel. A CI engine is unthrottled; the air/fuel ratio of the charge decreases as engine load increases. At high loads, combustion occurs closer to the point where maximum  $\text{NO}_x$  is produced.

Preignition chamber engines have lower baseline  $\text{NO}_x$  emissions than direct fuel injection engines. Shorter ignition delay combined with the generally richer combustion conditions in the preignition chamber results in smoother combustion and lower peak temperatures. In addition, there are significant heat transfer losses as the combustion gas goes from the preignition chamber to the main combustion chamber, lowering peak temperatures.<sup>6</sup>

### **2.3.2 Total Organic Compounds (Hydrocarbons)**

The pollutants commonly classified as hydrocarbons are composed of a wide variety of organic compounds. They are discharged into the atmosphere when some of the fuel remains unburned or is only partially burned during the combustion process. Most unburned hydrocarbon emissions result from fuel droplets that were transported or injected into the quench layer during combustion. This is the region immediately adjacent to the combustion chamber surfaces, where heat transfer outward through the cylinder walls causes the mixture temperatures to be too low to support combustion.

Partially burned hydrocarbons can occur for a number of reasons:

- ! Poor air and fuel homogeneity due to incomplete mixing, before or during combustion.

- ! Incorrect air/fuel ratios in the cylinder during combustion due to maladjustment of the engine fuel system.
- ! Excessively large fuel droplets (diesel engines).
- ! Low cylinder temperature due to excessive cooling (quenching) through the walls or early cooling of the gases by expansion of the combustion volume caused by piston motion before combustion is completed.

All of these conditions can be caused by either poor maintenance or faulty design. Therefore, the lowest emissions will be achieved only by proper maintenance of engines designed specifically for low emissions.<sup>2</sup>

### **2.3.3 Carbon Monoxide**

Carbon monoxide is a colorless, odorless, relatively inert gas formed as an intermediate combustion product that appears in the exhaust when the reaction of CO to CO<sub>2</sub> cannot proceed to completion. This situation occurs if there is a lack of available oxygen near the hydrocarbon (fuel) molecule during combustion, if the gas temperature is too low, or if the residence time in the cylinder is too short. The oxidation rate of CO is limited by reaction kinetics and, as a consequence, can be accelerated only to a certain extent by improvements in air and fuel mixing during the combustion process.

Carbon monoxide is a primary (directly emitted) pollutant, unlike ozone and other secondary pollutants which are formed in the atmosphere by photochemical reactions (reactions that require light). Carbon monoxide combines with the hemoglobin in blood, preventing it from carrying needed oxygen, and adversely affects the ability to perform exercise.<sup>2,7</sup>

### **2.3.4 Smoke, Particulate Matter, and PM-10**

White, blue, and black smoke may be emitted from IC engines. Liquid particulates appear as white smoke in the exhaust during an engine cold start, idling, or low load operation. These are formed in the quench layer adjacent to the cylinder walls, where the temperature is not high enough to ignite the fuel. The liquid particulate consist primarily of raw fuel with some partially burned hydrocarbons and lubricating oil. White smoke emissions are generally associated with older gasoline engines and are rarely seen in the

exhaust from diesel or gas-fueled units. They cease when the engine reaches its normal operating temperature and can be minimized during low demand situations by proper idle adjustment.

Blue smoke is emitted when lubricating oil leaks, often past worn piston rings, into the combustion chamber and is partially burned. Proper maintenance is the most effective method of preventing blue smoke emissions from all types of IC engines.

The primary constituent of black smoke is agglomerated carbon particles (soot). These form in a two-step process in regions of the combustion mixture that are oxygen deficient. First the hydrocarbons decompose into acetylene and hydrogen in the high temperature regions of the cylinder. Then, when the local gas temperature decreases as the piston moves down and the gases expand, the acetylene condenses and releases its hydrogen atoms. As a result, pure carbon particles are created. This mechanism of formation is associated with the low air/fuel ratio conditions that commonly exist at the core of the injected fuel spray, in the center of large individual fuel droplets, and in fuel layers along the walls. The formation of particles from this source can be reduced by designing the fuel injector to provide for an even distribution of fine fuel droplets such that they do not impinge on the cylinder walls.

Once formed, the carbon will combine with oxygen to form CO and CO<sub>2</sub> if it is still at an elevated temperature. Since the temperature of the exhaust system is too low for this oxidation to occur, soot that leaves the combustion chamber before it has had the opportunity to oxidize completely will be discharged as visible particles. Discharge is greatest when the engine is operating at rich air/fuel ratios, such as at rated power and speed, because soot formation is very sensitive to the need for oxygen. Therefore, naturally aspirated engines are likely to have higher smoke levels than turbocharged engines, which operate at leaner air/fuel ratios.<sup>2</sup>

Exposure to particulate matter less than 10 micrometers in aerodynamic diameter (PM-10) can result in both short and long term reductions in lung function because they are too small to be trapped by the nose and large enough that some deposition in the lungs

occurs. Also, PM-10 is the pollutant that causes most of the air pollution-induced reduction in visibility.<sup>7</sup>

### **2.3.5 Sulfur Oxides**

Sulfur oxide emissions are a function of only the sulfur content in the fuel rather than any combustion variables. In fact, during the combustion process, essentially all the sulfur in the fuel is oxidized to SO<sub>2</sub>. The oxidation of SO<sub>2</sub> gives sulfur trioxide (SO<sub>3</sub>), which reacts with water to give sulfuric acid (H<sub>2</sub>SO<sub>4</sub>), a contributor to acid precipitation. Sulfuric acid reacts with basic substances to give sulfates, which are fine particulates that contribute to PM-10 and visibility reduction. Sulfur oxide emissions also contribute to corrosion of the engine parts.<sup>2,7</sup>

### **2.3.6 Carbon Dioxide**

Concern about the increasing release of greenhouse gases such as CO<sub>2</sub> has grown out of research that documents the buildup of gases in the atmosphere and estimates the implications of continued accumulations. Carbon dioxide is largely transparent to incoming solar radiation, but can absorb infrared radiation reemitted by the Earth. Because of this energy trapping property, such a gas is referred to as a greenhouse gas.<sup>8</sup>

## **2.4 CONTROL TECHNOLOGIES**

The control development and regulation has been less extensive for industrial engines than for boilers because industrial engines are a relatively small emission source compared to boilers. Controls for hydrocarbons and CO have been partly adopted from mobile sources. Controls for NO<sub>x</sub> have mostly focused on modifications to the combustion process. Postcombustion catalytic reduction is becoming available, but its use is limited because of cost.

### **2.4.1 Engine Controls**<sup>1,6,9</sup>

**2.4.1.1 Combustion Cycle.** Reciprocating IC engines may be either two- or four-stroke cycle. During combustion, emissions from either type are essentially identical. However, during the charging of a two-stroke engine, several events take place. On noninjected engines, the scavenging air, which purges the cylinder of exhaust gases and provides the combustion air, can also sweep out part of the fuel charge. Thus, carbureted

two-cycle engines often have higher organic compound emissions in the form of unburned fuel than fuel injected engines.

The two-stroke engine can also have lower NO<sub>x</sub> emissions than a four-stroke engine. If the cylinder is not completely purged of exhaust gases, the result is internal exhaust gas recirculation (EGR). The remaining inert exhaust gases absorb energy from combustion, lowering peak temperatures, and thereby lowering NO<sub>x</sub>. Internal EGR can reduce NO<sub>x</sub> emissions by 4 to 37 percent. External EGR (turbocharged models) can have reductions varying from 25 to 34 percent. These reductions are obtained with exhaust gas recirculation rates of 6.5 to 12 percent. At 6 percent EGR, NO<sub>x</sub> reductions range from 10 to 22 percent. In general, fuel consumption remains unchanged for EGR rates less than 12 percent.

**2.4.1.2 Injection Timing Retard.** Ignition in a normally adjusted IC engine is set to occur shortly before the piston reaches its uppermost position [top dead center (TDC)]. At TDC, the air or air and fuel mixture is at maximum compression. The timing of the start of injection or of the spark is given in terms of the number of degrees that the crankshaft must still rotate between this event and the arrival of the piston at TDC.

Retarding the timing beyond TDC, the point of optimum power and fuel consumption, reduces the rate of NO<sub>x</sub> production. Retarding causes more of the combustion to occur later in the cycle, during the expansion stroke, thus lowering peak temperatures, pressures, and residence times. The efficiency loss is identifiable by the increase in fuel flow needed to maintain rated power output. This practice carries with it a fuel consumption penalty of 5 to 8 percent and the potential of excessive smoke. Typical retard values range from 2° to 6° depending on the engine. Beyond these levels, fuel consumption increases rapidly, power drops, and misfiring occurs. Also, TOC, CO, and visible emissions increase, and elevated exhaust temperatures shorten exhaust valves and turbocharger service lives. Increasing the fuel injection rate has been used on some diesel systems to partially mitigate the CO and TOC emissions and fuel consumption effects of retarded injection timing. A high injection rate, however, results in increased mixing of air and fuel and a subsequently hotter flame at the initiation of combustion. Therefore, there

is a NO<sub>x</sub> trade-off with this modification. Injection timing retard is an applicable control with all IC engine fuels.

The reported level of control is in the range of 0.6 to 8.5 percent reduction for each degree of retard. On the average, diesel engines reduce NO<sub>x</sub> by 25 percent for 4° of retard and 40 percent for 8° of retard. Fuel usage increases approximately 2 percent at 4° retard, whereas 8° of retard raises fuel usage by about 6 percent.

**2.4.1.3 PreIgnition Chamber Combustion - "Clean Burn" Technology.** The use of a preignition chamber can improve fuel efficiency and reduce NO<sub>x</sub> emissions. The system is designed to burn lean air/fuel mixtures. The fuel charge is introduced into the prechamber as a rich mixture and ignited by a sparkplug. Since it burns in the absence of excess oxygen, NO<sub>x</sub> formation is inhibited. This "torch" of burning fuel expands into the power cylinder where it thoroughly ignites a lean mixture at reduced temperatures. Therefore, combustion is completed in an overall lean mixture at temperatures that are adequate for combustion but below those where high NO<sub>x</sub> formation occurs. This NO<sub>x</sub> control has currently been developed for natural gas-fired engines only.

**2.4.1.4 Air to Fuel Ratio.** In injection type engines, which include all diesel and many dual fuel and gas varieties, the air/fuel ratio for each cylinder can be adjusted by controlling the amount of fuel that enters each cylinder. These engines are therefore operated lean where combustion is most efficient and fuel consumption is optimum.

At air/fuel ratios below stoichiometric (rich), combustion occurs under conditions of insufficient oxygen and thus unburned HC emission increase. Carbon monoxide increases because carbon is not sufficiently oxidized to CO<sub>2</sub>. Nitrogen oxides decrease both because of insufficient oxygen and lower temperatures.

At air/fuel ratios above stoichiometric (lean), combustion occurs under conditions of excess oxygen, thus essentially all carbon is oxidized to CO<sub>2</sub>. Nitrogen oxides first increase rapidly with the air/fuel ratio near stoichiometric, because of the excess oxygen and peak temperatures, then decrease rapidly with increasing air/fuel ratio as the excess air cools peak combustion temperatures. Hydrocarbons stay at a low level, then begin to increase as the air/fuel ratio is increased because the lower temperatures inhibit combustion.

The choice of lean or rich operation often depends on engine use. Rich operating (meaning close to stoichiometry) engines give quicker response to changing conditions and/or loads, and also produce maximum power. The most practical use of air/fuel ratio adjustment as a control technique is to change the setting toward leaner operation. The oxygen availability will increase but so will the capability of the air and combustion products to absorb heat. Consequently, the peak temperature will fall, resulting in lower NO<sub>x</sub> formation rates. The limiting factor for lean operation is the increased emissions of hydrocarbons at the lower temperatures. Small changes in the air/fuel ratio, approximately 10 percent, can reduce NO<sub>x</sub> by about 30 percent with a fuel penalty of about 5 percent.

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

**2.4.1.5 Water Injection.** Water injection has extensive application to NO<sub>x</sub> control with combustion turbines. Water injection reduces NO<sub>x</sub> emissions but may increase HC emissions because of the lower peak temperature and the increased possibility that burnout reactions will be quenched before burnout occurs. Carbon monoxide appears to be less affected by water injection. Wet control effectiveness correlates inversely with excess air levels. Since wet controls reduce peak temperature by increasing the charge mass (and through absorption of the latent heat of vaporization), the technique is more effective in a low excess air system than in one with much excess air and hence, much thermal mass. At high excess air, the incremental temperature reduction is less although the initial temperature may also be less because of the larger thermal mass. The application of this control to IC engines has been limited because of inaccessibility of water injection. Some applications of wet controls have been made in the development of water-fuel emulsions.

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

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

## **2.4.2 Post-Combustion Control<sup>6,10,11</sup>**

**2.4.2.1 Selective Catalytic Reduction.** In the selective catalytic reduction (SCR) process, anhydrous ammonia (NH<sub>3</sub>) gas, usually diluted with air or steam, is injected through a grid system into the exhaust gas stream upstream of a catalyst. On the catalyst surface, the NH<sub>3</sub> reacts with NO<sub>x</sub> to form molecular nitrogen and water. Depending on system design, NO<sub>x</sub> removal of 80 to 90 percent and higher are achievable under idealized conditions. The global reactions that occur in the presence of the catalyst are the following:



The reaction of NH<sub>3</sub> and NO<sub>x</sub> is favored by the presence of excess oxygen (fuel lean conditions). The primary variable affecting NO<sub>x</sub> reduction is temperature. Optimum NO<sub>x</sub> reduction occurs at catalyst bed temperatures between 600 and 750 °F for conventional (vanadium or titanium-based) catalyst types, and between 470 and 510 °F for platinum catalysts. Performance for a given catalyst depends largely on the temperature of the exhaust gas being treated. A given catalyst exhibits optimum performance between a

temperature range of about  $\pm 50$  °F for applications where flue gas  $O_2$  concentrations are greater than 1 percent. Below this optimum temperature range, the catalyst activity is greatly reduced, allowing unreacted  $NH_3$  to slip through. Above 850 °F,  $NH_3$  begins to be oxidized to form additional  $NO_x$ . The ammonia oxidation to  $NO_x$  increases with increasing temperature. Depending on the catalyst substrate material, the catalyst may be quickly damaged because of thermal stress at temperatures in excess of 850 °F. It is important to have stable operations and uniform flue gas temperatures for this process in order to achieve optimum  $NO_x$  control.

The optimal effectiveness of the catalytic process is also dependent on the  $NO_3/NO_x$  ratio. Ammonia injection rates must be controlled to give an optimum  $NH_3/NO_x$  mole ratio of about 1:1. As the mole ratio of  $NH_3/NO_x$  increases the level of approximately 1:1, the  $NO_x$  reduction increases. Operating with ammonia injection above this level or with insufficient catalyst volume will result in unreacted  $NH_3$  slipping through the catalyst bed. On-stream analyzers and quick feedback control are required to optimize the  $NO_x$  removal and minimize  $NH_3$  emissions.

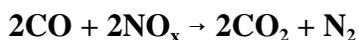
Other variables which affect  $NO_x$  reduction are space velocity, the ratio of flue gas flow rate to catalyst volume, or the inverse of residence time. For a given catalyst volume, increased flue gas rate decreases the conversion  $NO_x$ . Conversely, for a given flue gas flow rate, increased catalyst volume improves the  $NO_x$  removal effectiveness.

Site-specific factors including operating temperatures and fuel type affect the performance and emission rates achievable with SCR. There are a number of operating considerations with SCR. First, potential catalyst poisoning by either metals, acid gases, or particulate entrainment is detrimental. The potential loss of catalyst activity due to these fuel effects results in the use of an excess of catalyst to maintain the required process efficiency over an extended period of time. Second,  $NH_3$  emissions result. In a properly designed and controlled system,  $NH_3$  emissions should be less than 10 ppm. Also, flue gas temperatures may not be in the proper operating range for optimum  $NO_x$  reduction. This problem may be aggravated by load changes or air/fuel ratio changes, and may necessitate costly heat exchange equipment for adequate  $NO_x$  reduction or acceptable efficiency. An

increase in back pressure due to pressure drop across the catalyst results in a decrease in fuel efficiency. In addition, the formation of ammonium sulfate and bisulfate in the presence of SO<sub>3</sub> and unreacted NH<sub>3</sub> presents corrosion and plugging concerns.

**2.4.2.2 Nonselective Catalytic Reduction.** Nonselective catalytic reduction (NSCR) systems are often referred to as three-way conversion catalyst systems since they reduce NO<sub>x</sub>, unburned hydrocarbon, and CO simultaneously. When the overall mixture supplied to the engine is weak, catalysis will favor oxidation of hydrocarbons and CO to CO<sub>2</sub> and water vapor but will not affect NO<sub>x</sub>. To operate properly, the combustion process must occur with an air/fuel ratio slightly fuel-rich of stoichiometric. Under this condition, in the presence of the catalyst, NO<sub>x</sub> is reduced by the CO, resulting in nitrogen and CO<sub>2</sub>. Sulfur resistant catalyst supports of titanium, molybdenum or tungsten are available for SO<sub>3</sub>-laden stream applications. Deposits are controlled by control of NH<sub>3</sub> slip to below 5 ppmv.

Nonselective catalytic reduction systems primarily utilize the following reaction in reducing NO<sub>x</sub>:



The catalyst used to promote this reaction is generally a mixture of platinum and rhodium. The catalyst operating temperature limits are 700 to 1,500 °F, with 800 to 1,200 °F being the most desirable. Temperatures above 1,500 °F result in catalyst sintering.

Typical NO<sub>x</sub> conversion ranges from 80 to 95 percent with corresponding decreases in CO and HC. Potential problems associated with NSCR applications include catalyst poisoning by oil additives (e.g., phosphorous and zinc) and inadequate air/fuel ratio controllers. Nonselective catalytic reduction is currently limited to IC engines with fuel-rich ignition systems.

**2.4.2.3 Diesel Particulate Traps.**<sup>12,13</sup> The particulate trap consists of a filter positioned in the exhaust stream designed to collect a significant fraction of the particulate emissions while allowing the exhaust gases to pass through the system. The operating principle of the trap is based on the capture (the volume of particulate matter emitted is sufficient to fill up and plug a reasonably sized filter over time) and periodic incineration

(or regeneration) of the carbonaceous exhaust particulates. The regeneration is often achieved through the use of heat provided by a variety of sources (fueled burners, electric heaters, engine intake and throttling). Regeneration is usually triggered after reaching a preestablished pressure drop measured across the trap which is often made of a wallflow honeycomb structure. Diesel traps have been developed most extensively for mobile source applications because of greater regulatory activity in that sector. Application to stationary engines has lagged mobile source applications.

#### **2.4.3 Control Technology Applications**

From a  $\text{NO}_x$  control viewpoint, the most important distinction between different engine models and types for reciprocating engines is rich-burn versus lean-burn. Exhaust from rich-burn engines has little or no excess air while the exhaust from lean burn engines is characterized with medium to high levels of  $\text{O}_2$ .

In diesel oil-fueled engines, the most common engine control techniques employed include injection timing retard and Clean Burn. Selective catalytic reduction technology has been applied to lean-burn reciprocating, diesel engines where the exhaust gas  $\text{O}_2$  concentrations are high as the SCR reaction mechanisms require presence of oxygen. Concerns persist over engine air-fuel controllability, catalyst durability, and ammonia slip. Application of NSCR requires fuel-rich engine operation or the addition of reducing agents in the flue gas upstream of the catalyst. Therefore, efficient application of this technology is limited to rich-burn engines (gasoline).

The Manufacturers of Emission Controls Association (MECA) state that catalytic oxidation controls for CO emissions are achieving 90 to 99 percent reduction for commercial applications.<sup>10</sup> For  $\text{NO}_x$  control, limited experience with SCR technology on lean-burn engines has shown potential for 90 to 95 percent control, but long term experience under field conditions is sparse. Nonselective catalytic reduction on rich-burn engines has achieved 90 to 99 percent control efficiency levels, largely in response to regulations in California. There is also commercial availability of VOC controls for diesel, lean burn, and rich burn IC engines, mostly adapted from mobile sources.

Several commercial processes currently exist to remove carbon dioxide. However, currently, there is no regulatory or economic incentive for utilities or private industry to remove carbon dioxide. Large scale carbon dioxide removal and disposal processes are very expensive.<sup>14</sup>

As on-road engines have become cleaner to meet increasingly stringent emissions requirements, the off-road engines will become a relatively more significant contributor to nonattainment of air quality goals. The application of on-road engines experience and hardware can be used to accelerate the development process of improving emissions from off-road engines. Improvements in the emissions of engines in off-road service can be made by appropriate application of on-road technology (such as fuel injector tip geometry, injection timing, and charge air temperature).<sup>15</sup>

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### **3. EMISSION DATA REVIEW AND ANALYSIS PROCEDURES**

**This section reviews the literature search and data evaluation procedures used to identify and review documents or other sources of test data. It also summarizes which types of data sources were identified for specific pollutant types and summarizes the criteria of data quality to rate the level of confidence of the data in terms of method used to sample and reporting of results. Emissions data were reviewed and analyzed based on EPA guidelines.<sup>1</sup> Data from all sources were entered into summary tables to help calculate average emission factors and identify data gaps. Quality ratings were applied to both individual data sets and overall emission factors. The criteria for rating individual emissions data were as follows:**

#### **Definition of Data Rankings:**

- A - When tests are performed by a sound methodology and are reported in enough detail for adequate validation. These tests are not necessarily EPA reference method tests, although such reference methods are preferred and certainly to be used as a guide.**
- B - When tests are performed by a generally sound methodology, but they lack enough detail for adequate validation.**
- C - When tests are based on an untested or new methodology or are lacking a significant amount of background data.**
- D - When tests are based on a generally unacceptable method, but the method may provide an order-of-magnitude value for the source, or no background data is provided at all.**

**Emission factor criteria are discussed in Chapter 4.**

### **3.1 LITERATURE SEARCH AND EVALUATION**

**The literature search started with reviewing the documents used in the previous revision of AP-42 Chapter 3.3. The prior background document showed that the previous emission factors were essentially based on one report, "Exhaust Emissions from Uncontrolled Vehicles and Related Equipment Using Internal Combustion Engines, Final Report, Part 5: Heavy-Duty Farm, Construction, and Industrial Engines."<sup>2,3</sup> The emission factors found in this report were based on the emissions testing of eight diesels and four gasoline engines. After reviewing the background document, an internal and external literature search was conducted.**

**Several different approaches were undertaken to obtain literature and data to facilitate update of the emission factors. The applicable references and sources (listed at the end of this chapter) were obtained and reviewed along with documents found through a "Dialogue" computer abstract search, an in-house data search, an EPA library search, an Electric Power Research Institute (EPRI) library search, periodicals, and contacts with trade organizations, manufacturers, and local, state, and federal air regulatory agencies. A complete list of contacts made can be found in Appendix B. Table 3-1 shows an evaluation of the references found.**

**TABLE 3-1. EVALUATION OF REFERENCES**

<b>Reference</b>	<b>Used in AP-42 revision</b>	<b>Why/Why Not</b>	<b>Parameter of Interest</b>	<b>Useable Raw Emission Factor Data</b>
3	Yes	Used in previous revision/source test data on diesel/gasoline engines; confirmatory checks made	Criteria	Yes
4	Yes	Shows that prior AP-42 data is representative	Criteria	No
5	Yes	Evaluates which data set is currently best available	Criteria	No
6	Yes	Compilation of emission databases/population & duty-cycles	All	Yes
7	Yes	Good small engine (<20 hp) data	Criteria	Yes
8	Yes	Emissions data	Criteria	Yes
9	Yes	Emissions data	Criteria	Yes
10	Yes	Review of non-road emissions	All	Yes
11	No	Controlled emissions. Insufficient data to convert units	Criteria	No
12	Yes	Compilation of emissions data and control performance	Criteria	No
13	Yes	Summary of combustion control performance	NO <sub>x</sub>	No
14	Yes	Summary of NO <sub>x</sub> control performance	NO <sub>x</sub>	No
15	Yes	Review of criteria pollutant properties	Criteria	No
16	Yes	Review of large diesel engine properties and exhaust	Criteria, Non-Criteria	No
17	Yes	Data on selective catalytic reduction performance (gas)	NO <sub>x</sub>	No
18	Yes	Air toxics source test data	CO <sub>2</sub> , Organics	Yes
19	Yes	Air toxics source test data	CO <sub>2</sub> , Organics	Yes
20	No	Uses the same data from Reference 12	Criteria	Yes
21	Yes	Summary of test results. Vol. II data supplement needed.	Criteria	Yes
22	Yes	Review of non-road emissions of air toxics	Air toxics	Yes

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#### **4. EMISSION FACTOR DEVELOPMENT**

The prior AP-42 data and the new data identified in Chapter 3 were compiled, evaluated, and ranked using evaluation tables. The data judged as acceptable within AP-42 criteria were then averaged, and in some cases weighted according to market share, to produce an emission factor. All emission factors were reviewed and analyzed based on EPA guidelines.<sup>1</sup> The significant difference in definitions between data ranking criteria and emission factor ranking criteria should be noted.

##### **Definition of Emission Factor Rankings:**

- A - Developed only from A-rated source test data taken from many randomly chosen facilities in the industry population. The source category is specific enough to minimize variability within the source population.**
- B - Developed only from A-rated test data from a reasonable number of facilities. Although no specific bias is evident, it is not clear if the facilities tested represent a random sample of the industries. As with the A rating, the source is specific enough to minimize variability within the source population.**
- C - Developed only from A- and B-rated test data from a reasonable number of facilities. Although no specific bias is evident, it is not clear if the facilities tested represent a random sample of the industry. As with the A rating, the source category is specific enough to minimize variability within the source population.**
- D - The emission factor was developed only from A- and B-rated test data from a small number of facilities, and there may be reason to suspect that these**

facilities do not represent a random sample of the industry. There also may be evidence of variability within the source population.

E - The emission factor was developed from C- and or D- rated test data, and there may be reason to suspect that the facilities tested do not represent a random sample of the industry. There also may be evidence of variability within the source category population.

#### **4.1 CRITERIA POLLUTANTS AND CARBON DIOXIDE**

##### **4.1.1 Review of Previous Data<sup>2</sup>**

The quality of the data used in the previous AP-42 Chapter 3.3 revision was judged to be of "B" quality using the current criteria specifications. The test procedures used were similar to the Federal 13-mode tests or the EMA California 13-mode test, except for a few that utilized 21 mode (diesel) or 23 mode (gasoline) tests. The eight diesels and four gasoline engines ranged in size from 15 bhp to 210 bhp. The emission factors would have been designated as "C" quality but because of the limited range of engine horsepower tested, only a "D" rating was possible. The scope of Chapter 3.3 for diesels is up to 600 bhp, and the data cover only engines less than 210 bhp.

The major assumptions used in the prior AP-42 update to weight the pollutant test data to develop an emission factor were:<sup>3,4</sup>

- ! Engine shipments as reported by the Bureau of the Census, the total value of such shipments, and the values of the engines shipped according to power output can be used to estimate the average power output of industrial engines;
- ! A high percentage of gasoline engines classified "industrial" in the Bureau of the Census statistics are actually in the light-duty engine category covered by an earlier report;
- ! Annual usage of industrial engines is approximately one-half that of construction engines of similar power output, and service life is 2,500 hours for gasoline engines and 5,000 hours for diesel engines. Population of

industrial engines can be estimated using the Bureau of the Census shipment figures and the service life and annual usage estimates; and

! Engine operating cycles can be estimated by considering the type of operation most industrial engines undergo in the field.

#### **4.1.2 Review of New Data**

References 2 and 4 through 10 contain both primary and secondary emissions data for gasoline and diesels engines. Many of the newer references rely in part on the prior AP-42 compilation or the data sources used for AP-42. Evaluation of the data quality for the newer data gave a lower quality rating than for the original AP-42 data in Reference 2 due primarily to insufficient information for the primary test engine design specifications, operating conditions, or test methods. The prior AP-42 data were accorded a "B" rating, whereas the new data identified in Chapter 3 were accorded a "C" or "D" rating. The ranges of the new data were, however, in the same range as the prior AP-42 emission factors. In view of the general agreement between the newer and prior data, and the prohibition against mixing data of differing quality rankings, the old weighted emission factors will be retained for criteria pollutants. Since the prior update contained data of "B" quality, the decision was made to retain these emission factors without incorporating the "C" and "D" quality data.

Although there are ORSAT data for CO<sub>2</sub>, a calculated value based on assumptions was used instead because it was felt that calculated values were more accurate than ORSAT measurements. It was assumed that all of the carbon going into the engine as fuel will appear in the exhaust as CO<sub>2</sub>. The contribution of carbon to other gases [such as CO and hydrocarbons (typically less than 0.1 percent)] is small. The emission factor for CO<sub>2</sub> will be a theoretical calculation of the carbon content of the fuel and 100 percent conversion of C into CO<sub>2</sub>. The average carbon content is 86 percent by weight for gasoline and 87 percent by weight for diesel.

#### **4.1.3 Compilation of Baseline Emission Factors**

Table 4-1 shows a summary of the raw emissions data and their conversions for criteria and nonorganic gaseous emissions and CO<sub>2</sub>.

#### **4.1.4 Compilation of Controlled Emission Factors**

**Fragmentary information on control efficiencies and operational or emission side effects of control systems is available in References 9 through 10. Insufficient data are available to develop controlled emission factors which are representative of both engine designs and control technologies. Chapter 5 contains a summary of qualitative information on control applicability for the industrial engine sector.**

**Most of the technologies developed for on-road engines can be directly applied to the off-road application since many engine designs are similar or identical. The actual demonstration on the durability of these technologies with stationary application has yet to be proven. Some on-road technologies that may be inappropriate for off-road use are:<sup>3</sup>**

- ! High pressure turbocharging results in an engine of given horsepower rating having poor low speed torque, but many off-road engines require good low end torque to pull against high hydraulic loads;**
- ! Air-to-air intercooling requires an extra heat exchanger, and heat exchange surface fouling is common in off-road environments. Many manufacturers believe that this technology is inappropriate in several equipment applications;**
- ! Electronic timing control may or may not survive in the harsh environment of off-road use. Manufacturers are reluctant to use this technology until its durability characteristics are well understood; and**
- ! Particulate traps are not yet proven in an on-road environment, and manufacturers lack information needed to evaluate traps in off-road equipment. Concerns center around the high-load duty cycle, with possible extended operation at full load on the "lug" line. This may aid in trap regeneration, but trap durability may be adversely affected. The exact nature of changes, and costs for retrofitting engines depend on the status of each individual engine model's emission level, and the hardware changes required to meet the changes. This varies substantially among engines and manufacturers.**

## **4.2 TOTAL ORGANIC COMPOUNDS AND AIR TOXICS**

### **4.2.1 Review of Old Data**

The quality of the data in the previous AP-42 Chapter 3.3 revision was designated to be of "B" quality using the current quality rating criteria. The test procedures used were similar to the Federal 13-mode tests or the EMA California 13-mode test, except for a few that utilized 21 mode (diesel) or 23 mode (gasoline) tests. The eight diesels and four gasoline engines ranged in size from 15 bhp to 210 hp. A "D" rather than "C" emission factor rating was given because the data did not span the capacity range to 600 bhp as needed for the Chapter 3.3 source classifications. The measured data from the prior AP-42 update applicable to this section are the exhaust hydrocarbons and aldehydes. These data were reviewed by the AP-42 evaluation criteria and were judged to remain applicable.

There was only fragmentary information available for evaporative and crankcase hydrocarbon emissions in terms of power output (g/hp-hr) or fuel input (lb/MMBtu). Hence, no emission factors were developed.

The major assumptions used in the prior AP-42 Chapter 3.3 to weight the engine test data in order to arrive at an emission factor were:<sup>2</sup>

- ! Engine shipments as reported by the Bureau of the Census, the total value of such shipments, and the values of the engines shipped according to power output can be used to estimate the average power output of industrial engines;
- ! A high percentage of gasoline engines classified "industrial" in the Bureau of the Census statistics are actually in the light-duty engine category covered by an earlier report;<sup>11</sup>
- ! Annual usage of industrial engines is approximately one-half that of construction engines of similar power output, and service life is 2,500 hours for gasoline engines and 5,000 hours for diesel engines. Population of industrial engines can be estimated using the Bureau of the Census shipment figures and the service life and annual usage estimates; and

**! Engine operating cycles can be estimated by considering the type of operation most industrial engines undergo in the field.**

#### **4.2.2 Review of New Data**

**The test data were reviewed for data quality for exhaust hydrocarbons, aldehydes, and speciated VOCs.<sup>4,5,8,11</sup> The data quality ranking for total hydrocarbons and aldehydes were "C" or "D" for the few data which were available. Accordingly, the emission factors in the prior AP-42 update were retained.**

**New sources of "B" quality data were obtained for speciated TOC and/or air toxics data in terms of lb/MMBtu factors<sup>6,7</sup>. The test reports did not provide the load at which the engines were running and therefore could not be used to calculate g/hp-hr values. All formulas and assumptions used to make conversions and calculations are presented in Appendix A.**

**There is currently little information on air toxic emissions from non-road sources. Nonroad emission factors for air toxics are inadequate and require more work. A comprehensive study to acquire the necessary data of air toxics from representative nonroad engines has been suggested.<sup>4</sup> The references used in recent reports adapted data from the old AP-42 Chapter 3.3 section and Volume II of AP-42.<sup>2,11</sup> This indicates some overlap in sources designated as stationary or mobile.**

**Factors for evaporative, crankcase, and refueling hydrocarbon emissions were based on sparse background documentation.<sup>5</sup> The data were therefore given a data quality rating of "D," which results in a emission factor quality rating of "E."**

#### **4.2.3 Compilation of Emission Factors**

**Tables 4-2 through 4-3 summarize the raw emissions data and their conversions for speciated organic compound and air toxic emissions.**

### **4.3 PARTICULATE**

#### **4.3.1 Review of Old Data**

**As previously mentioned, the quality of the data used in the previous AP-42 Chapter 3.3 revision was determined to be "B" quality. The test procedures used were similar to the Federal 13-mode tests or the EMA California 13-mode test, except for a few**

which utilized 21 mode (diesel) or 23 mode (gasoline) tests. The eight diesels and four gasoline engines ranged in size from 15 bhp to 210 hp. An emission factor quality rating of "D" was assigned for these tests. The test data from the prior AP-42 applicable to this section are the particulate emissions. The prior data were evaluated by the AP-42 criteria and judged to remain applicable.

The major assumptions used to weight the test data for use in emission factors for this section were the same as the assumptions used to evaluate the previous TOC and air toxics data (section 4.2 of this report).

#### **4.3.2 Review of New Data**

References 8, 5, 4 and 10 contain emissions data for diesel particulate emissions identified in the search discussed in Chapter 3. The few new data which were identified were in the same range as the emission factors from the prior AP-42 update. The quality of the newer data was generally rated to be low, "C" and "D," because of the lack of detail on the source or test procedures. In the absence of better quality data for particulate, the old weighted emission factors were retained.

#### **4.3.3 Compilation of Emission Factors**

The data for particulate emissions are included in Table 4-1.

**TABLE 4-1. SUMMARY OF EMISSIONS DATA FOR STATIONARY INTERNAL COMBUSTION ENGINES:  
CRITERIA AND NONORGANIC GASEOUS EMISSIONS**

Data Type	Units	Data Source									
		AB2588 <sup>a</sup>			EPA			SSEIS Vol. 1. <sup>b</sup>			
		WSPA Water Injection	Drilling Rig	WSPA Workover	Nonroad Engines <sup>c</sup>	Nonroad APTD-1494 Engines <sup>d</sup>	APTD-1494 Engines <sup>d</sup>	Engine 30	Engine 31	Engine 32	Engine 33
Data Rating		B	B	B	D	B	B	C	C	C	C
Fuel <sup>e</sup>		D	D	D	D,G	D	G	D	D	D	D
Rating <sup>f</sup>	hp	350	550	350							
wt % S (Fuel)	%	0.37	0.24	0.004							
%02		13.7	17.1	16.5							
Exit	dscfm	743	160								
F factor	dscf/MMBtu	8,915	8,902	9,194							
Heat Input	MMBtu/hr	1.72	0.19		7000	7000	7000				
BSFC <sup>f</sup>	Btu/hp-hr	4914	345								
HHV	Btu/lb Btu/gal	19,500 138,450	19,500 138,450	19,972 141,800							

**TABLE 4-1. SUMMARY OF EMISSIONS DATA FOR STATIONARY INTERNAL COMBUSTION ENGINES:  
CRITERIA AND NONORGANIC GASEOUS EMISSIONS (Continued)**

Pollutant	Units	Data Source									
		AB2588 <sup>a</sup>			EPA			SSEIS Vol. 1 <sup>b</sup>			
		WSPA Water Injection	Drilling Rig	WSPA Workover Rig	Nonroad Engines <sup>c</sup>	Nonroad APTD- 1494 Engines <sup>d</sup>	APTD- 1494 Engines <sup>d</sup>	Engine 30	Engine 31	Engine 32	Engine 33
NO <sub>x</sub>	g/hp-hr lb/MMBtu					14.0 4.41	5.16 1.63	6.60	4.60	4.00	5.50
CO	μ/m <sup>3</sup> g/hp-hr lb/1000gal lb/MMBtu			1,500 0.00 691 4.87		3.03 0.95	1.99 62.67	-1.90	4.10	1.60	1.30
SO <sub>x</sub>	g/hp-hr lb/1000gal lb/MMBtu	0.85 53 0.38	0.04 34 0.25			0.931 0.29	0.268 0.084				
CO <sub>2</sub>	μ/m <sup>3</sup> ppm g/hp-hr lb/1000gal lb/MMBtu	51,000 336.04 20,871 150.75	19,000 16.65 14,711 106.26	35,000 0.00 25,332 178.65							
PM	g/hp-hr lb/1000gal lb/MMbtu						0.327 0 0.10				

<sup>a</sup>Reference 6 and 7. All engines are industrial.

<sup>b</sup>Reference 8. All engines are medium bore.

<sup>c</sup>Reference 5. All engines are industrial.

<sup>d</sup>Reference 3 (NO<sub>x</sub>, CO, SO<sub>x</sub>); Reference 2 (PM). All engines are industrial.

<sup>e</sup>D = Diesel; G = Gasoline.

<sup>f</sup>Data may be invalid.

**TABLE 4-2a. SUMMARY OF EMISSIONS DATA FOR STATIONARY INTERNAL COMBUSTION ENGINES:  
SPECIATED ORGANIC COMPOUNDS**

Data Type or Pollutant	Units	Data Source									
		AB2588 <sup>a</sup>		EPA				SSEIS, Vol 1. <sup>b</sup>			
		WSPA Water Injection	WSPA Workover Rig	Nonroad Engines <sup>c</sup>	Nonroad Engines <sup>d</sup>	Nonroad Engines <sup>c</sup>	Nonroad Engines <sup>d</sup>	Engine 30	Engine 31	Engine 32	Engine 33
Data Rating		B	B	D	B	D	B	C	C	C	C
Fuel <sup>e</sup>		D	D	D	D	G	G	D	D	D	D
Rating <sup>f</sup>	hp	350	350								
HC: Crank	g/hp-hr lb/MMBtu			0.02 0.01		2.20 0.69					
HC: Vapor	g/hp-hr lb/MMBtu			0.00 0.00		0.30 0.09					
HC: Refuel	g/hp-hr lb/MMBtu			0.00 0.00		0.49 0.15					
HC: Total Exhaust (as CH <sub>4</sub> )	ppm lb/hr g/hp-hr lb/1000gal lb/MMBtu	107 0.20 0.26 16 0.117	67.8  16 0.115		1.12  0.35		6.68  2.10	6.60	0.60	0.30	0.20
Aldehydes	g/hp-hr lb/MMBtu				0.21 0.07		0.22 0.07				

**TABLE 4-2b. SUMMARY OF EMISSIONS DATA FOR STATIONARY INTERNAL COMBUSTION  
ENGINES: SPECIATED ORGANIC COMPOUNDS**

Data Type or Pollutant	Units	AB2588 <sup>a</sup>	
		WSPA Water Injection	WSPA Workover Rig
Benzene	ppm	95	0.147
	ppb		
	lb/hr		
	g/hp-hr		
	lb/1000gal		
	lb/MMBtu		
Toluene	ppm	40	0.050
	ppb		
	lb/hr		
	g/hp-hr		
	lb/1000gal		
	lb/MMBtu		
Xylenes	ppm	41	0.021
	ppb		
	lb/hr		
	g/hp-hr		
	lb/1000gal		
	lb/MMBtu		
Propylene	ppm	430	0.793
	ppb		
	lb/hp-hr		
	g/hp-hr		
	lb/1000gal		
	lb/MMBtu		
1,3 Butadiene	ppb	<10	
	g/hp-hr		
	lb/1000gal		
	lb/MMBtu		
Formaldehyde	$\mu\text{g}/\text{m}^3$	821	0.284
	ppm		
	ppb		
	lb/hr		
	g/hp-hr		
	lb/1000gal		
lb/MMBtu			

**TABLE 4-2b. SUMMARY OF EMISSIONS DATA FOR STATIONARY INTERNAL COMBUSTION ENGINES: SPECIATED ORGANIC COMPOUNDS (Continued)**

Data Type or Pollutant	Units	AB2588 <sup>a</sup>	
		WSPA Water Injection	WSPA Workover Rig
Acetaldehyde	$\mu\text{g}/\text{m}^3$	632	0.091
	ppm		
	ppb	340	
	lb/hr	1.74E-03	
	g/hp-hr	2.39E-03	
	lb/1000gal	0.1481	
	lb/MMBtu	1.07E-03	
Acrolein	$\mu\text{g}/\text{m}^3$	<31	0.020
	ppm		
	ppb	<13	
	lb/hr	<8.60E-05	
	g/hp-hr	<1.18E-04	
	lb/1000gal	<0.0073	
	lb/MMBtu	<5.30E-05	
		1.32E-04	

<sup>a</sup>Reference 6 and 7.

**TABLE 4-3. SUMMARY OF EMISSIONS DATA FOR STATIONARY INTERNAL COMBUSTION ENGINES: AIR TOXICS**

Data Type or Pollutant	Units	AB2588	
		WSPA Water Injection <sup>a</sup>	WSPA Workover Rig <sup>b</sup>
Data Rating		B	B
Fuel		D	D
Rating	hp	350	350
Naphthalene	$\mu\text{g}/\text{m}^3$	31.8	41.90
	lb/hr	8.84E-05	
	lb/1000gal		0.0166
	lb/MMBtu	5.25E-05	1.17E-04
Acenaphthylene	$\mu\text{g}/\text{m}^3$	<0.006	3.61
	lb/hr	<1.75E-08	
	lb/1000gal		0.0014
	lb/MMBtu	<1.03E-08	1.01E-05
Acenaphthene	$\mu\text{g}/\text{m}^3$	<0.006	1.01
	lb/hr	<1.75E-08	
	lb/1000gal		0.0004
	lb/MMBtu	<1.03E-08	2.83E-06
Flourene	$\mu\text{g}/\text{m}^3$	31.9	2.17
	lb/hr	8.84E-05	
	lb/1000gal		0.0009
	lb/MMBtu	5.24E-05	6.04E-06
Phenanthrene	$\mu\text{g}/\text{m}^3$	27.7	4.77
	lb/hr	7.68E-05	
	lb/1000gal		0.0019
	lb/MMBtu	4.55E-05	1.33E-05
Anthracene	$\mu\text{g}/\text{m}^3$	1.4	0.51
	lb/hr	3.89E-06	
	lb/1000gal		0.0002
	lb/MMBtu	2.30E-06	1.43E-06
Flouranthene	$\mu\text{g}/\text{m}^3$	7.8	0.90
	lb/hr	2.17E-05	
	lb/1000gal		0.0004
	lb/MMBtu	1.27E-05	2.51E-06
Pyrene	$\mu\text{g}/\text{m}^3$	3.3	1.51
	lb/hr	9.09E-06	
	lb/1000gal		0.0006
	lb/MMBtu	5.35E-06	4.22E-06

**TABLE 4-3. SUMMARY OF EMISSIONS DATA FOR STATIONARY INTERNAL COMBUSTION ENGINES: AIR TOXICS (Continued)**

Data Type or Pollutant	Units	AB2588	
		WSPA Water Injection <sup>a</sup>	WSPA Workover Rig <sup>b</sup>
Benz(a)anthracene	$\mu\text{g}/\text{m}^3$	1.7	0.24
	lb/hr	4.60E-06	
	lb/1000gal lb/MMBtu	2.69E-06	0.0001 6.62E-07
Chrysene	$\mu\text{g}/\text{m}^3$	0.27	0.09
	lb/hr	7.52E-07	
	lb/1000gal lb/MMBtu	4.45E-07	0.00004 2.61E-07
Benzo(b)fluoranthene	$\mu\text{g}/\text{m}^3$	<0.006	0.07
	lb/hr	<1.65E-08	
	lb/1000gal lb/MMBtu	<1.03E-08	0.00003 1.88E-07
Benzo(k)fluoranthene	$\mu\text{g}/\text{m}^3$	<0.006	0.11
	lb/hr	<1.75E-08	
	lb/1000gal lb/MMBtu	<1.03E-08	0.00004 3.00E-07
Benzo(a)pyrene	$\mu\text{g}/\text{m}^3$	<0.006	<0.13
	lb/hr	<1.75E-08	
	lb/1000gal lb/MMBtu	<1.03E-08	<0.00005 <3.65E-07
Ideno(1,2,3-cd)pyrene	$\mu\text{g}/\text{m}^3$	0.17	<0.17
	lb/hr	4.63E-07	
	lb/1000gal lb/MMBtu	2.74E-08	<0.0001 <4.75E-07
Dibenz(a,h)anthracene	$\mu\text{g}/\text{m}^3$	0.25	<0.27
	lb/hr	7.02E-07	
	lb/1000gal lb/MMBtu	4.14E-07	<0.0001 7.51E-07
Benzo(g,h,l)perylene	$\mu\text{g}/\text{m}^3$	0.25	<0.20
	lb/hr	7.06E-07	
	lb/1000gal lb/MMBtu	4.16E-07	<0.0001 <5.61E-07

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<sup>b</sup>Reference 7.

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- 10. Lips, H.I., J.A. Gotterba, and K.J. Lim, "Environmental Assessment of Combustion Modification Controls for Stationary Internal Combustion Engines," EPA-600/7-81-127, Industrial Environmental Research Laboratory, Office of Environmental Engineering and Technology, Office of Air Quality Planning and Standards, U.S. Environmental Protection Agency, Research Triangle Park, NC, July 1981.**
- 11. "Feasibility of Controlling Emissions from Off-Road, Heavy-Duty Construction Equipment," Final Report, prepared by Energy and Environment Analysis for the California Air Resources Board, El Monte, CA, December 1988.**

## **5. AP-42 SECTION 3.3: GASOLINE AND DIESEL INDUSTRIAL ENGINES**

**The revision to Section 3.3 of AP-42 is presented in the following pages as it would appear in the document.**

### **3.3 GASOLINE AND DIESEL INDUSTRIAL ENGINES**

#### **3.3.1 General**

The engine category addressed by this section covers a wide variety of industrial applications of both gasoline and diesel internal combustion engines such as, aerial lifts, fork lifts, mobile refrigeration units, generators, pumps, industrial sweepers/scrubbers, material handling equipment (such as conveyors), and portable well-drilling equipment. The rated power of these engines covers a rather substantial range; up to 186 kW (250 hp) for gasoline engines and up to 447 kW (600 hp) for diesel engines. (Diesel engines greater than 600 hp are covered in Section 3.4: Large Stationary Diesel and All Stationary Dual Fuel Engines). Understandably, substantial differences in engine duty cycles exist. It was necessary, therefore, to make reasonable assumptions concerning usage in order to formulate some of the emission factors.

#### **3.3.2 Process Description**

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

There are two methods used for stationary reciprocating IC engines: compression ignition (CI) and spark ignition (SI). Section 3.3 deals with both types of reciprocating internal combustion engines.

In compression ignition engines, combustion air is first compression heated in the cylinder, and diesel fuel oil is then injected into the hot air. Ignition is spontaneous as the air is above the auto-ignition temperature of the fuel. Spark ignition engines initiate combustion by the spark of an electrical discharge. Usually the fuel is mixed with the air in a carburetor (for gasoline) or at the intake valve (for natural gas), but occasionally the fuel is injected into the compressed air in the cylinder. All diesel fueled engines are compression ignited and all gasoline fueled engines are spark ignited.

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

#### **3.3.3 Emissions and Controls**

The best method for calculating emissions is on the basis of "brake specific" emission factors (g/hp-hr or g/kW-hr). Emissions are calculated by taking the product of the brake specific emission

factor, the usage in hours (that is, hours per year or hours per day), the power available (rated power), and the load factor (the power actually used divided by the power available).

Once reasonable usage and duty cycles for this category were ascertained, emission values were aggregated to arrive at the factors presented in Tables 3.3-1 (English units) and 3.3-2 (Metric units) for criteria and organic pollutants. Emissions data for a specific design type were weighted according to estimated material share for industrial engines. The emission factors in this table are most appropriately applied to a population of industrial engines rather than to an individual power plant because of their aggregate nature. Table 3.3-3 shows unweighted speciated organic compound and air toxic emissions factors based upon only two engines. Their inclusion in this section is intended only for rough order of magnitude estimates.

Table 3.3-4 shows a summary of various diesel emission reduction technologies (some which may be applicable to gasoline engines). These technologies are categorized into fuel modifications, engine modifications, and exhaust after treatments. Current data are insufficient to quantify the results of the modifications. Table 3.3-4 provides general information on the trends of changes on selected parameters.

**TABLE 3.3-1. (ENGLISH UNITS) EMISSION FACTORS FOR UNCONTROLLED GASOLINE AND DIESEL INDUSTRIAL ENGINES<sup>a</sup>**  
(Source Classification Codes)

Pollutant [Rating] <sup>b</sup>	Gasoline Fuel (SCC 20200301, 20300301)		Diesel Fuel (SCC 20200102, 20300101)	
	[grams/hp-hr] (power output)	[lb/MMBtu] (fuel input)	[grams/hp-hr] (power output)	[lb/MMBtu] (fuel input)
NO <sub>x</sub> [D]	5.16	1.63	14.0	4.41
CO [D]	199	62.7	3.03	0.95
SO <sub>x</sub> [D]	0.268	0.084	0.931	0.29
Particulate [D]	0.327	0.10	1.00	0.31
CO <sub>2</sub> [B] <sup>c</sup>	493	155	525	165
Aldehydes [D]	0.22	0.07	0.21	0.07
<b>Hydrocarbons</b>				
Exhaust [D]	6.68	2.10	1.12	0.35
Evaporative [E]	0.30	0.09	0.00	0.00
Crankcase [E]	2.20	0.69	0.02	0.01
Refueling [E]	0.49	0.15	0.00	0.00

<sup>a</sup>Data based on uncontrolled levels for each fuel from References 1, 3 and 6.

When necessary, the average brake specific fuel consumption (BSFC) value was used to convert from g/hp-hr to lb/MMBtu was 7000 Btu/hp-hr.

<sup>b</sup>"D" and "E" rated emission factors are most appropriate when applied to a population of industrial engines rather than to an individual power plant, due to the aggregate nature of the emissions data.

<sup>c</sup>Based on assumed 100 percent conversion of carbon in fuel to CO<sub>2</sub> with 87 weight percent carbon in diesel, 86 weight percent carbon in gasoline, average brake specific fuel consumption of 7000 Btu/hp-hr, diesel heating value of 19300 Btu/lb, and gasoline heating value of 20300 Btu/lb.

**TABLE 3.3-2. (METRIC UNITS) EMISSION FACTORS FOR UNCONTROLLED GASOLINE AND DIESEL INDUSTRIAL ENGINES<sup>a</sup>**  
(Source Classification Codes)

Pollutant [Rating] <sup>b</sup>	Gasoline Fuel (SCC 20200301, 20300301)		Diesel Fuel (SCC 20200102, 20300101)	
	[grams/kW-hr] (power output)	[ng/J] (fuel input)	[grams/kW-hr] (power output)	[ng/J] (fuel input)
NO <sub>x</sub> [D]	6.92	699	18.8	1,896
CO [D]	267	26,947	4.06	410
SO <sub>x</sub> [D]	0.359	36	1.25	126
Particulate [D]	0.439	44	1.34	135
CO <sub>2</sub> [B] <sup>c</sup>	661	66,787	704	71,065
Aldehydes [D]	0.30	29	0.28	28
<b>Hydrocarbons</b>				
Exhaust [D]	8.96	905	1.50	152
Evaporative [E]	0.40	41	0.00	0.00
Crankcase [E]	2.95	298	0.03	2.71
Refueling [E]	0.66	66	0.00	0.00

<sup>a</sup>Data based on uncontrolled levels for each fuel from References 1, 3 and 6.

<sup>b</sup>"D" and "E" rated emission factors are most appropriate when applied to a population of industrial engines rather than to an individual power plant, due to the aggregate nature of the emissions data.

<sup>c</sup>Based on assumed 100 percent conversion of carbon in fuel to CO<sub>2</sub> with 87 weight percent carbon in diesel, 86 weight percent carbon in gasoline, average brake specific fuel consumption of 7000 Btu/hp-hr, diesel heating value of 19300 Btu/lb, and gasoline heating value of 20300 Btu/lb.

**TABLE 3.3-3. (ENGLISH AND METRIC UNITS) SPECIATED ORGANIC COMPOUNDS AND AIR TOXIC EMISSION FACTORS FOR UNCONTROLLED DIESEL ENGINES<sup>a</sup>**

(Source Classification Codes: 20200102, 20300101)

(ALL EMISSION FACTORS ARE RATED: E)<sup>b</sup>

Pollutant	[lb/MMBtu] (fuel input)	[ng/J] (fuel input)
Benzene	9.33 E-04	0.401
Toluene	4.09 E-04	0.176
Xylenes	2.85 E-04	0.122
Propylene	2.58 E-03	1.109
1,3 Butadiene <sup>c</sup>	< 3.91 E-05	< 0.017
Formaldehyde	1.18 E-03	0.509
Acetaldehyde	7.67 E-04	0.330
Acrolein	< 9.25 E-05	< 0.040
<b>Polycyclic Aromatic Hydrocarbons (PAH)</b>		
Naphthalene	8.48 E-05	3.64 E-02
Acenaphthylene	< 5.06 E-06	< 2.17 E-03
Acenaphthene	< 1.42 E-06	< 6.11 E-04
Fluorene	2.92 E-05	1.26 E-02
Phenanthrene	2.94 E-05	1.26 E-02
Anthracene	1.87 E-06	8.02 E-04
Fluoranthene	7.61 E-06	3.27 E-03
Pyrene	4.78 E-06	2.06 E-03
Benz(a)anthracene	1.68 E-06	7.21 E-04
Chrysene	3.53 E-07	1.52 E-04
Benzo(b)fluoranthene	< 9.91 E-08	< 4.26 E-05
Benzo(k)fluoranthene	< 1.55 E-07	< 6.67 E-05
Benzo(a)pyrene	< 1.88 E-07	< 8.07 E-05
Indeno(1,2,3-cd)pyrene	< 3.75 E-07	< 1.61 E-04
Dibenz(a,h)anthracene	< 5.83 E-07	< 2.50 E-04
Benzo(g,h,l)perylene	< 4.89 E-07	< 2.10 E-04
<b>Total PAH</b>	<b>1.68 E-04</b>	<b>7.22 E-02</b>

<sup>a</sup>Data are based on the uncontrolled levels of two diesel engines from References 6 and 7.

<sup>b</sup>"E" rated emission factors are due to limited data sets, inherent variability in the population and/or a lack of documentation of test results. "E" rated emission factors may not be suitable for specific facilities or populations and should be used with care.

<sup>c</sup>Data are based on one engine.

**TABLE 3.3-4. DIESEL EMISSION CONTROL TECHNOLOGIES<sup>a</sup>**

Technology	Affected Parameter <sup>b</sup>	
	Increase	Decrease
<b>Fuel Modifications</b>		
Sulfur Content Increase	PM, Wear	
Aromatic Content Increase	PM, NO <sub>x</sub>	
Cetane Number		PM, NO <sub>x</sub>
10 percent and 90 percent Boiling Point		PM
Fuel Additives		PM, NO <sub>x</sub>
Water/Fuel Emulsions		NO <sub>x</sub>
<b>Engine Modifications</b>		
Injection Timing	NO <sub>x</sub> , PM, BSFC, Power	NO <sub>x</sub>
Fuel Injection Pressure	PM, NO <sub>x</sub>	
Injection Rate Control		NO <sub>x</sub> , PM
Rapid Spill Nozzles		PM
Electronic Timing & Metering		NO <sub>x</sub> , PM
Injector Nozzle Geometry		PM
Combustion Chamber Modifications		NO <sub>x</sub> , PM
Turbocharging	PM, Power	NO <sub>x</sub>
Charge Cooling		NO <sub>x</sub>
Exhaust Gas Recirculation	PM, Power, Wear	NO <sub>x</sub>
Oil Consumption Control		PM, Wear
<b>Exhaust After Treatment</b>		
Particulate Traps		PM
Selective Catalytic Reduction		NO <sub>x</sub>
Oxidation Catalysts		HC, CO, PM

<sup>a</sup>Reference 4.

<sup>b</sup>NO<sub>x</sub> = Nitrogen oxides; PM = Particulate matter; HC = Hydrocarbons; CO = Carbon monoxide; BSFC = Brake specific fuel consumption.

### References for Section 3.3

1. Hare, C.T. and K.J. Springer, Exhaust Emissions from Uncontrolled Vehicles and Related Equipment using Internal Combustion Engines, Part 5: Farm, Construction, and Industrial Engines, U.S. Environmental Protection Agency, Research Triangle Park, NC, Publication APTD-1494, October 1973, pp. 96-101.
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3. Standards Support and Environmental Impact Statement, Volume I: Stationary Internal Combustion Engines, EPA-450/2-78-125a, Emission Standards and Engineering Division, Office of Air, Noise, and Radiation, Office of Air Quality Planning and Standards, U.S. Environmental Protection Agency, Research Triangle Park, NC, July 1979.
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5. Nonroad Engine and Vehicle Emission Study-Report, EPA-460/3-91-02, Certification Division, Office of Mobile Sources, Office of Air & Radiation, U.S. Environmental Protection Agency, Research Triangle Park, NC, November 1991.
6. Pooled Source Emission Test Report: Oil and Gas Production Combustion Sources, Fresno and Ventura Counties, California, Report prepared by ENSR Consulting and Engineering for Western States Petroleum Association (WSPA), Bakersfield, CA, December 1990, ENSR 7230-007-700.
7. Osborn, W.E., and M.D. McDannel, Emissions of Air Toxic Species: Test Conducted Under AB2588 for the Western States Petroleum Association, Report prepared by Carnot for Western States Petroleum Association (WSPA), Glendale, CA, May 1990, CR 72600-2061.

## **APPENDIX A**

### **SAMPLE CALCULATIONS**

**Convention used:            1/hp/hr = 1/hp-hr**

**Assumptions:    If needed (see Appendix E):**  
**Heating value of diesel is 19300 Btu/lb**  
**Heating value of gasoline is 20300 Btu/lb**  
**Density of diesel is 7.1 lb/gal**  
**Weight percent of carbon in diesel is 87percent**  
**Weight percent of carbon in gasoline is 86percent**  
**Average brake specific fuel consumption (BSFC) = 7000 Btu/hp-hr**

**To convert from heat input (MMBtu/hr) to BSFC (Btu/hp-hr)**

**Assumptions: Engine is running at full load (in hp)**

$$\begin{aligned} &(\text{heat input}) * (1/(\text{engine rating in hp})) = \text{BSFC} \\ &(\text{MMBtu/hr}) * (1000000 \text{ Btu/MMBtu}) * (1/\text{hp}) = \text{Btu/hp/hr} = \text{Btu/hp-hr} \end{aligned}$$

**To convert from g/hp-hr to lb/MMBtu**

**Assumptions: You have BSFC (Btu/hp-hr)**

$$(\text{g/hp-hr}) * (1/(\text{Btu/hp-hr})) * (1 \text{ lb}/453.6 \text{ g}) * (1000000 \text{ Btu/MMBtu}) = \text{lb/MMBtu}$$

**To calculate heating value (Btu/lb) from BSFC (lb/hp-hr) and fuel rate (Btu/hp-hr)**

$$(\text{Btu/hp-hr}) * (1/(\text{lb/hp-hr})) = \text{Btu/lb}$$

**To convert heating values of Btu/lb to Btu/gal and visa versa**

**Assumptions: Density = 7.1 lb/gal**

$$\begin{aligned} &(\text{Btu/lb}) * (7.1 \text{ lb/gal}) = \text{Btu/gal} \\ &(\text{Btu/gal}) / (7.1 \text{ lb/gal}) = \text{Btu/lb} \end{aligned}$$

**To convert from g/hp-hr to lb/MMBtu**

**Assumptions: You have the heat input value (MMBtu/hr)**  
**Engine is running at full load (in hp)**

$$(\text{g/hp-hr}) * (1/(\text{MMBtu/hr})) * (\text{hp}) * (1 \text{ lb}/453.6 \text{ g}) = \text{lb/MMBtu}$$

**To convert from g/hp-hr to lb/Mgal (where Mgal = 1000 gal)**

**Assumptions: You have BSFC (Btu/hp-hr) and heating value (Btu/gal)**

$$(g/hp-hr) * (1/(Btu/hp-hr)) * (1 lb/453.6 g) * (Btu/gal) * (1000 gal/Mgal) = lb/Mgal$$

**To convert from lb/MMBtu to lb/Mgal**

**Assumptions: You have the heating value (Btu/gal)**

$$(lb/MMBtu) * (1 MMBtu/1000000 Btu) * (Btu/gal) * (1000 gal/Mgal) = lb/Mgal$$

**To convert from lb/MMBtu to g/hp-hr**

**Assumptions: You have the BSFC value (Btu/hp-hr)**

$$(lb/MMBtu) * (1 MMBtu/1000000 Btu) * (453.6 g/lb) * (Btu/hp-hr) = g/hp-hr$$

**To convert % by volume to ppmv**

**% = parts per 100, therefore**

$$(%/100) * (1000000/1000000) = (% * 10000) / 1000000 = % * 10000 ppmv$$

**e.g., 5.1% CO = 51000 ppmv**

**To convert µg/s to g/hp-hr**

**Assumptions: Engine is running at full load (in hp)**

$$(\mu g/s) * (1 g/1000000 \mu g) * (3600 s/hr) * (1/hp) = g/hp-hr$$

**To convert from ppm of a gas to lbm/MMBtu:**

**Assumptions: You have the molecular weight of the gas (in lb/lb-mol)**

**The concentration of the gas (in ppm)**

**You know the fuel F-factor (ft<sup>3</sup>/MMBtu)**

**You know the percent O<sub>2</sub> level**

**At STP conditions (293 K and 1 atm), 1 mole of gas occupies 385.3 ft<sup>3</sup>/lb-mol**

$$(ppm/1000000) / (385.3 ft^3/lb-mol) * (lb/lb-mol) * (ft^3/MMBtu) * (20.9 / (20.9 - \%O_2)) = lb/MMBtu$$

To convert from g/hp-hr (english units) to g/kW-hr (metric units):

$$(g/hp-hr) * (1.341 hp-hr/kW-hr) = g/kW-hr$$

To convert from lb/MMBtu (english units) to ng/J (metric units):

$$(lb/MMBtu) * (453.6 g/lb) * (1 MMBtu/1000000 Btu) * (1 Btu/1055 J) * (1000000000 ng/g) = ng/J$$

To calculate CO<sub>2</sub> emissions for diesel and gasoline from weight percent of carbon and BSFC (g/hp-hr):

Assumptions: 87 wt% carbon in diesel  
76 wt% carbon in gasoline  
Diesel heating value = 19300 Btu/lb  
Gasoline heating value = 20300 Btu/lb  
BSFC = 7000 Btu/hp-hr

For diesel fuel

$$(87 lbC/100 lbFuel) * (1 lb-molC/12 lbC) * (1 lb-molCO_2/lb-molC) *$$

$$(44 lbCO_2/lb-molCO_2) * (lbFuel/19300 Btu) * (1000000 Btu/MMBtu) = \underline{165 lb/MMBtu}$$

and

$$(165 lbCO_2/MMBtu) * (1 MMBtu/1000000 Btu) * (453.6 g/lb) * (7000 Btu/hp-hr) = \underline{524 g/hp-hr}$$

hr

For gasoline fuel

$$(86 lbC/100 lbFuel) * (1 lb-molC/12 lbC) * (1 lb-molCO_2/lb-molC) *$$

$$(44 lbCO_2/lb-molCO_2) * (lbFuel/20300 Btu) * (1000000 Btu/MMBtu) = \underline{155.3 lb/MMBtu}$$

and

$$(155.3 lbCO_2/MMBtu) * (1 MMBtu/1000000 Btu) * (453.6 g/lb) * (7000 Btu/hp-hr) = \underline{493 g/hp-hr}$$

hr

**APPENDIX B**

**SUMMARY OF COMMUNICATIONS ATTEMPTED/MADE**

**TABLE B-1 SUMMARY OF COMMUNICATIONS ATTEMPTED/MADE**

COMPANY/AGENCY	CONTACT PERSON
KANSAS DEPT. OF HEALTH AND ENVIRONMENT	HARISH AGARWAL, MR. HINTHER
MANUFACTURERS OF EMISSION CONTROLS ASSOCIATION (MECA)	CAROLYN GILLESPIE, RAYMOND CONNOR
EPA OFFICE OF MOBILE SOURCES, ANN ARBOR, MI	GREG JANSSEN, KEVIN GREEN
AMERICAN PETROLEUM INSTITUTE (API)	JIM WILLIAMS
INDUSTRIAL GAS CLEANING INSTITUTE (IGCI)	JEFF SMITH
MONTEREY BAY AIR POLLUTION CONTROL DISTRICT	LARRY BORELLI
SOUTH COAST AIR QUALITY MANAGEMENT DISTRICT (SCAQMD)	BILL DENNISON
ENGINE MANUFACTURERS ASSOCIATION (EMA)	GLENN KELLER
AMERICAN HONDA	VIA EMA
AMERICAN SUZUKI MOTOR CORPORATION	VIA EMA
BRIGGS & STRATTON CORPORATION	VIA EMA
CATERPILLAR INC.	DON DOWDALL
CUMMINS ENGINE COMPANY	MIKE BRAND
DEERE & COMPANY	VIA EMA
DETROIT DIESEL CORPORATION	VIA EMA
DEUTZ CORPORATION	VIA EMA
FORD NEW HOLLAND	VIA EMA
FORD POWER PRODUCTS DIVISION	VIA EMA
GENERAL ELECTRIC	VIA EMA
GENERAL MOTORS CORPORATION	VIA EMA
ISUZU MOTORS AMERICA, INC.	VIA EMA
KAWASAKI MOTORS CORP.	VIA EMA
KOHLER COMPANY	VIA EMA
KOMATSU LTD.	VIA EMA
KUBOTA CORPORATION	VIA EMA
LISTER-PETTER, INC.	VIA EMA
MITSUBISHI ENGINE NORTH AMERICA, INC.	VIA EMA
ONAN CORPORATION	VIA EMA
TECUMSEH PRODUCTS COMPANY	VIA EMA
TELEDYNE TOTAL POWER	VIA EMA
YANMAR DIESEL AMERICA	VIA EMA
COOPER AJAX/SUPERIOR DIVISION	BRUCE CHRISMANN
TEXAS AIR CONTROL BOARD	RANDY HAMILTON

**TABLE B-1 SUMMARY OF COMMUNICATIONS ATTEMPTED/MADE (Continued)**

COMPANY/AGENCY	CONTACT PERSON
UTAH AIR QUALITY BOARD	DON ROBINSON, NORMAN ERICKSON
WAUKESHA	PAUL CANNESRA
FLORIDA DEPARTMENT OF ENVIRONMENTAL REGULATIONS	CLAIR FANCY
COOPER BESSEMER	BILL HEATER
BAY AREA AIR QUALITY MANAGEMENT DISTRICT	STEVE HILL, BOB NISHIMURA, MARK NASH
KANSAS EPA, AIR TOXICS DIVISION	ED BUCKNER
NEW YORK EPA	FRANK JON
FAIRBANKS MORSE ENGINE DIVISION	PAUL DANYLUK
MISSOURI NATURAL RESOURCE AIR POLLUTION CONTROL PROGRAM	TODD CRAWFORD
DIESEL ENGINE MANUFACTURERS ASSOCIATION (DEMA)	
VENTURA COUNTY AIR POLLUTION CONTROL DISTRICT	TERRI THOMAS
WESTERN STATES PETROLEUM ASSOCIATION (WSPA)	MIKE EMANUEL
FRESNO COUNTY AIR POLLUTION CONTROL DISTRICT	ROBERT DOWELL
NORTH COAST UNIFIED AIR QUALITY MANAGEMENT DIST.	BOB CLARK
SACRAMENTO METROPOLITAN AIR QUALITY MANAGEMENT DISTRICT	BRUCE NIXON
SAN JOAQUIN COUNTY AIR POLLUTION CONTROL DISTRICT	LAKHMIR GREWAL
CALAVERAS COUNTY AIR POLLUTION CONTROL DISTRICT	ROBERT MARSHALL
LAKE COUNTY AIR QUALITY MANAGEMENT DISTRICT	ROBERT REYNOLDS
LASSEN COUNTY AIR POLLUTION CONTROL DISTRICT	KENNETH SMITH

**APPENDIX C**

**MARKED-UP PREVIOUS AP-42 SECTION**

1. Lips, H. I., J. A. Gotterba, and K. J. Lim, Environmental Assessment of Combustion Modification Controls for Stationary Internal Combustion Engines, Report prepared by Acurex Corporation for Industrial Environmental Research Laboratory, Office of Environmental Engineering and Technology, Office of Air Quality Planning and Standards, EPA, Research Triangle Park, NC, July 1981, EPA-600/7-81-127.
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4. Lips, H. I., et al., Reference 1, pp. 3-1 to 3-7.
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