

EMISSION FACTOR DOCUMENTATION FOR
AP-42 SECTION 3.4,
LARGE STATIONARY DIESEL &
ALL STATIONARY DUAL FUEL ENGINES

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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 that is the source of the emission (typically fuel consumption or energy output). Section 3.4 of the "Compilation of Air Pollutant Emission Factors" (AP-42) covers the emission factors for large (greater than 600 hp) stationary diesel and all stationary dual fuel engines. The inclusion of all dual fuel engines rather than only large dual fuel as was done previously is because there are few applications and only sparse data for moderate- sized engines. This small population does not warrant separate treatment elsewhere. Diesels, in contrast, have a large population and distinct designs in the mid-capacity range and are treated in Chapter 3.3.

The emission factors provide persons working in air pollution control with documented estimates of source emission rates. The emission factors, as stated in "Technical Procedures for Developing AP-42 Emission Factors and Preparing AP-42 Sections," can be used in determining: (1) estimates of areawide emissions; (2) emission estimates for a specific facility; and (3) evaluation of emissions in relation to ambient air quality. The intent of this emission factor document is to provide background information from all references used to support the revision of emission factors for Section 3.4 - Large Stationary Diesel and Dual Fuel Engines. Chapter 2 contains information on the combustion process used by large stationary diesel and all stationary dual fuel engines, as well as emission characterization and controls. Chapter 3 summarizes the procedures and data base available for this update and Chapter 4 presents the rationale for the changes and additions to the emission factors. Chapter 5 contains the revision of AP-42 Chapter 3.4 completed as a result of the analysis presented in this report.

The last update for Chapter 3.4 was in 1984. Previously, this Chapter contained only emission factors for total particulate, nitrogen oxides (NO_x), carbon monoxide (CO), hydrocarbons (grouped as methane or nonmethane), and sulfur oxides (SO_x). This revision will include emission factors for those species as well as for carbon dioxide (CO₂), total organic compounds (TOCs) speciation and air toxics, and particulate sizing. The overall scope of the current revision includes the following changes or additions:

- ! Updating of emissions factors for criteria pollutants during baseline, uncontrolled, operation using data available since the prior supplement;
- ! Inclusion of several noncriteria emission species for which data are available: organics speciation, air toxics, and greenhouse or ozone depletion gases (CO₂); and
- ! Inclusion of technical discussion and emissions data for engine operation with NO_x or CO control.

Section 3.4 deals with large compression ignition engines. 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 automatic ignition temperature of the fuel. The resulting high pressure products of combustion pushes the piston through the cylinder and is then converted from linear to rotary motion by a crankshaft. All diesel fueled engines are compression ignited. Although all natural gas fueled engines are spark ignited (spark of an electrical discharge initiates combustion), natural gas can be used in a compression ignition engine if a small amount of diesel fuel is injected into the compressed natural gas-air mixture in the cylinder to initiate combustion. Hence, the use of diesel with natural gas to fuel an engine is termed "dual fueled."

2. SOURCE DESCRIPTION

2.1 CHARACTERIZATION OF THE INDUSTRY

Stationary reciprocating internal combustion (IC) engines are used in a variety of applications where there is a requirement for mechanical work which can be derived from the power generated by a shaft. Installations range from within large urban centers to remote areas. Large engines can be found operating in a variety of applications such as electric power generation for both base and standby service, oil and gas pipeline pumping/transport, oil and gas exploration and production, irrigation, hoisting, and nuclear power plant emergency cooling water pump operation.

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 increases 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.^{1,2}

2.2 PROCESS DESCRIPTION¹

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

basic process, there are variations that classify engine types. Engines are generally classified by their: fuel burned, method of ignition, combustion cycle, and charging method.

2.2.1 Fuel Type

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

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

Natural gas is used more than any other fuel for large stationary reciprocating or turbine IC engines, typically operating pumps or compressors on gas pipelines.

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

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 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 automatic 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. Although all diesel fueled engines are compression ignited and all gasoline and gas fueled engines are spark ignited, gas can be used in a compression ignition engine if a small amount of diesel fuel is injected into the compressed gas/air mixture to burn any mixture ratio of gas and diesel oil, from 6 to 100 percent oil (based on heating value).

2.2.3 Combustion Cycle

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

In the four-stroke cycle, the sequence of events is 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; and
- ! Exhaust stroke -- expulsion of the exhaust gases from the cylinder by the upward movement of the piston.

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

ports are covered (and/or valves closed), and the fresh charge of air or air and fuel is again compressed for the next cycle.

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

2.2.4 Charging Method

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

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

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

In a CI engine, fuel is injected into the cylinder near the end of the compression stroke; whereas, in 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 engines) 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 hydrocarbons escape from the crankcase as a result of blowby (gases that 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 hydrocarbons 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 because of 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, 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. Other fugitive emissions may result from diesel fuel storage.

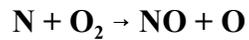
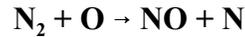
The primary pollutants from internal combustion engines are NO_x , hydrocarbons and other organic compounds (HC), CO, and particulates, which include both visible (smoke) and nonvisible emissions. Nitrogen oxide formation is directly related to high pressures and temperatures during the combustion process and to the nitrogen content 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. Oxides of sulfur also appear in the exhaust from IC engines. The sulfur compounds, mainly sulfur dioxide (SO_2), are directly related to the sulfur content of the fuel.²

2.3.1 Nitrogen Oxides

Nitrogen oxide formation occurs by two fundamentally different mechanisms. The principal mechanism with engines is thermal NO_x , which arises from the thermal dissociation and subsequent reaction of nitrogen (N_2) and oxygen (O_2) molecules in the combustion air. Most thermal NO_x is formed in the highest temperature region of the flame after most hydrocarbon oxidation has occurred. A component of thermal NO_x , called prompt NO_x , is formed from early reactions of nitrogen intermediaries and hydrocarbon radicals from the fuel. The second mechanism, fuel NO_x , stems from the evolution and reaction of fuel-bound nitrogen compounds with oxygen. Natural gas has negligible chemically bound fuel nitrogen (although some molecular nitrogen) and essentially all NO_x formed is thermal NO_x . Distillate oils have small levels of bound nitrogen. These levels are usually significant only for high degrees of NO_x controls where thermal NO_x has been suppressed to the level where fuel NO_x is significant. Residual oils contain fuel bound nitrogen; upon combustion, NO_x is formed by all mechanisms. Some large bore diesels fire residual oil, but the NO_x potential is usually a deterrent to this fuel. The formation of prompt NO_x is significant only in very fuel-rich flames and is not of major importance for reciprocating IC engines.

At high temperatures (thermal NO_x), both N_2 and O_2 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_x 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_x 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_x . The thermal NO_x generation decreases rapidly as the temperature drops below the adiabatic temperature (for a given stoichiometry). Thus, maximum reduction of thermal NO_x generation can be achieved by control of both the combustion temperature and the stoichiometry.

In fuel NO_x , the nitrogen compounds (primarily organic) contained in the fuels are evolved and react to form 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 NO_x conversion that reduces the effectiveness of NO_x controls which rely on temperature effects in the combustion of nitrogen-bearing fuels. Here, as with thermal NO_x , controlling excess O_2 (stoichiometry) is an important part of controlling NO_x formation.

The majority of NO_x formed is thermal NO_x because of the high flame temperatures and pressures of IC engines. As diesel fuel and natural gas are the predominate fuels for this source, little fuel 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 pre-mixed 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 ignition delay. When 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 NO_x emissions.

In general, engine load does not have a profound effect on the brake-specific (NO_x rate to power output ratio) 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 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 NO_x in the exhaust gas increases. However, the brake-specific NO_x emission rate remains roughly the same since power output also increases by the same factor.

Brake-specific 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 compression ignition 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 NO_x is produced.

Preignition chamber engines have lower baseline 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 pre-ignition chamber to the main combustion chamber, lowering peak temperatures.³

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 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 prior to, 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); and
- ! Low cylinder temperature due to excessive cooling 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.²

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 CO₂ 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.

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. They 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 these 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 drops 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 CO₂ if it is still at an elevated temperature. Since the temperature of the exhaust system is too low for this oxidation to occur, soot exiting the combustion chamber before it has had the opportunity to oxidize completely will be discharged as visible particles.

Because soot formation is very sensitive to the need for oxygen, its discharge is greatest when the engine is operating at rich air/fuel ratios, such as at rated power and speed. Therefore, naturally aspirated engines are likely to have higher smoke levels than

turbocharged engines, which operate at leaner air/fuel ratios.²

PM-10 is the name applied to the smaller sized particles in the air (less than 10 micrometers in aerodynamic diameter). Exposure to PM-10 can result in both short and long term reductions in lung function because the particles are too small to be trapped by the nose and large enough that some deposition in the lungs occurs.

PM-10 is the pollutant that causes most of the air-pollution-induced reduction in visibility.⁴

2.3.5 Sulfur Oxides

Sulfur oxide emissions are a function of the sulfur content in the fuel, only, rather than of any combustion variables. In fact, during the combustion process essentially all the sulfur in the fuel is oxidized to SO₂. The oxidation of SO₂ gives sulfur trioxide (SO₃), which reacts with water to give sulfuric acid (H₂SO₄), 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. The reduction of SO_x emissions would also minimize corrosion of the engine parts.^{2,4}

2.3.6 Carbon Dioxide

Concern about the increasing release of greenhouse gases such as CO₂ 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. The warming of the surface of the Earth and cooling of the stratosphere is achieved via molecular absorption and reradiation in the infrared spectrum. The Earth reemits the energy it absorbs back to space, thus maintaining an energy balance. Because the Earth is much colder than the Sun, the bulk of the Earth's emission takes place at longer wavelengths than those of incoming solar radiation. Although CO₂ is a relatively inefficient absorber of solar radiation, it is a strong absorber of longwave (infrared) radiation. Although some of the reemitted energy reaches space, some of the radiation is transmitted downward, leading to a net trapping of longwave radiation and a warming of the surface.⁵

2.4 CONTROL TECHNOLOGIES

The majority of control development and installation for large reciprocating engines has been for NO_x control. The implementation of control has occurred in a few isolated geographical areas, since there is no federal New Source Performance Standard (NSPS). Most control techniques have attempted to modify the combustion process to suppress NO_x formation. In stringent cases, post combustion control by catalytic or noncatalytic reduction has been partially developed, but installed in only rare cases due to cost and the previous lack of need for such levels of reduction. Some stationary engines are starting to be retrofitted with catalytic CO or volatile organic compound (VOC) control. There is insufficient information on CO and VOC add-on controls to formulate emission factors at this time.

2.4.1 Engine Controls^{1,4,6}

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-cycle 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_x emissions than the 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_x. Internal EGR can reduce NO_x emissions from 4 to 37 percent. External EGR (turbocharged models) can have reductions varying from 25 to 34 percent. These reductions were obtained with exhaust gas recirculation rates of 6.5 to 12 percent. At 6 percent EGR, NO_x reductions ranged from 10 to 22 percent. In general, fuel consumption remained unchanged for EGR rates less than 12 percent.

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_x 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. There is, therefore, a NO_x 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_x 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 Pre-Ignition Chamber Combustion - "Clean Burn" Technology. The use of a preignition chamber can improve fuel efficiency and reduce NO_x emissions. The system is designed to burn lean air/fuel mixtures. The fuel charge is introduced into the pre-chamber as a rich mixture and ignited by a spark-plug. Since it burns in the absence of excess oxygen, NO_x 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 NO_x formation occurs. This NO_x control is currently applied to 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 hydrocarbon emission increase. Carbon monoxide increases because carbon is not sufficiently oxidized to CO_2 . 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_2 . Nitrogen oxides first increase rapidly with the air/fuel ratio near stoichiometric, because of the excess oxygen and peak temperatures, then decreases 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/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_x 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_x 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 (A/F). 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. As with combustion turbines, water or steam injection can reduce NO_x emissions from IC engines. Water injection reduces NO_x emissions but usually increases HC emissions because of the lower peak temperatures achieved. Carbon monoxide appears to be unaffected by water injection. Wet control effectiveness correlates inversely with excess air levels. Since wet controls reduce peak temperature by increasing the charge mass and absorbing the latent heat of vaporization, the technique is more effective in a low excess air system than in one with much excess air. The lower thermal mass with low excess air will experience a greater temperature drop. Systems with high excess air absorb all the heat that can be transferred to a fluid in the short time between combustion and peak temperature. The application of this control to IC engines has been limited due to inaccessibility of water injection. A more plausible application of wet controls may be in the use of water-fuel emulsions.

2.4.1.6 Derating. An engine can be derated by restricting its operation to lower levels of power production than normal for the given application. Derating reduces cylinder pressures and temperatures and thus lowers NO_x formation rates. Although NO_x exhaust concentrations (i.e., moles of NO_x per mole of exhaust) are reduced, it is quite possible for this reduction to be no greater than the power decrease. In such a case, brake specific emissions (i.e., 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_x emissions are less responsive to derating for turbocharged engines. Derating also reduces the engine's operating temperature, which can result in higher CO and HC emissions.

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^{2,7,8}

2.4.2.1 Selective Catalytic Reduction. In the selective catalytic reduction (SCR) process, anhydrous ammonia (NH₃) gas, usually diluted with air or steam, is injected through a grid system into the exhaust gas stream upstream of a catalyst bed. On the catalyst surface, the NH₃ reacts with NO_x to form molecular nitrogen and water.

Depending on system design, NO_x removal of 80 to 90 percent and higher is achievable.

The global reactions that occur in the presence of the catalyst are as follows:



The reaction of NH₃ and NO_x is favored by the presence of excess oxygen (fuel-lean conditions). The primary variable affecting NO_x reduction is temperature. Optimum NO_x reduction occurs at catalyst bed temperatures between 600 and 750 °F for conventional (vanadium or titanium-based) catalyst types, and 470 to 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 approximately +/- 50 °F for applications where flue gas O₂ concentrations are greater than 1 percent. Below this optimum temperature range, the catalyst activity is greatly reduced, allowing unreacted ammonia to slip through. Above 850 °F, ammonia 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 due to thermal stress at temperatures in excess of 850 °F. It is important, therefore, to have stable operations and uniform flue gas temperatures for this process to achieve optimum NO_x control.

The optimal effectiveness of the catalytic process is also dependent on the NO₃/NO_x ratio. Ammonia injection rates must be controlled to give an optimum NH₃/NO_x mole ratio of about 1:1. As the mole ratio of NH₃/NO_x increases to approximately 1:1, the NO_x reduction increases. Operating with ammonia injection above this level with insufficient

catalyst volume will result in unreacted ammonia slipping through the catalyst bed. On-stream analyzers and quick feedback control are required to optimize the NO_x removal and minimize NH₃ emissions.

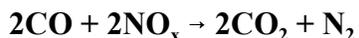
Another variable that affects NO_x reduction is space velocity, the ratio of flue gas flow rate to catalyst volume, or the inverse of residence time. For a given catalyst volume, increased flue gas rate decreases the conversion NO_x. Conversely, for a given flue gas flow rate, increased catalyst volume 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, NH₃ emissions result. In a properly designed and controlled system, NH₃ emissions should be less than 10 ppm. A major issue results when flue gas temperatures are outside the proper catalyst operating range. This problem can be aggravated by load changes. The correction may require retrofit of reheat heat exchange hardware. An increase in back pressure due to pressure drop across the catalyst results in a decrease to fuel efficiency. In addition, the formation of ammonium sulfate and bisulfate in the presence of SO₃ and unreacted ammonia 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_x, unburned hydrocarbon, and CO simultaneously. 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, the oxides of nitrogen are reduced by the CO, resulting in nitrogen and CO₂. Sulfur resistant catalysts supports of titanium, molybdenum or tungsten are available for SO₃-laden stream applications. Deposits are controlled by control of NH₃ slip to below 5 ppmv.

Nonselective catalytic reduction systems primarily utilize the following reaction in

reducing NO_x:



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_x 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, zinc) and inadequate control systems. Nonselective catalytic reduction is limited in application to IC engines with fuel-rich ignition systems.

2.4.3 Control Technology Applications

From an NO_x control viewpoint, the most important distinction among 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 O₂.

For natural gas/dual fuel engines, engine design in general and clean burn or pre-ignition chamber combustion have been the most commonly applied engine control technology for NO_x reduction in the past decade. 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, and dual fueled engines where the exhaust gas O₂ concentrations are high as the SCR reaction mechanisms require presence of oxygen. Application of NSCR requires fuel-rich engine operation or the addition of reducing agents in the flue gas upstream of the catalyst. Therefore, application of this technology has been limited to rich-burn engines.

The Manufacturers of Emission Controls Association (MECA), states that catalytic oxidation controls for CO emissions are achieving 90 to 99 percent in commercial applications. For NO_x control, SCR technology on lean-burn engines has achieved 90 to 95 percent control efficiency in limited applications. Nonselective catalytic reduction on rich-

burn engines has reached 90 to 99 percent control efficiency levels with more extensive experience, particularly in California. There is also commercial availability of VOC controls for diesel, lean-burn, and rich-burn IC engines, but with very little operating experience to data.

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, and large-scale carbon dioxide removal and disposal processes are very expensive.⁹

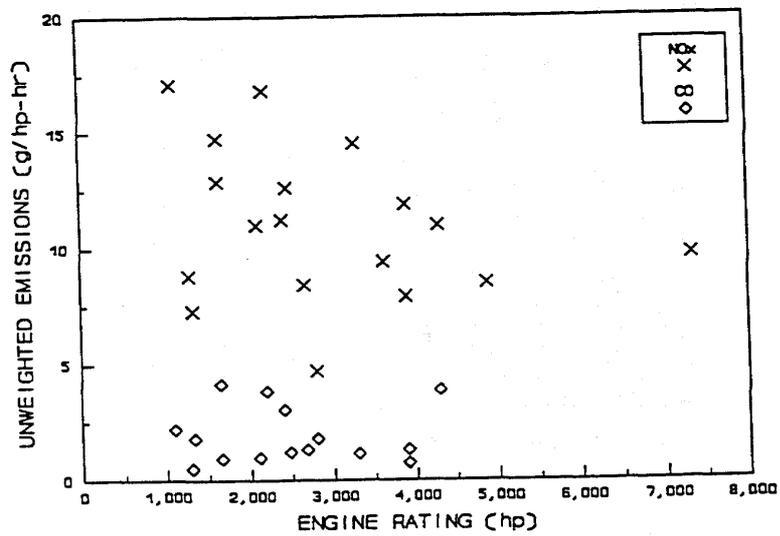


Figure 4-1. Unweighted NO_x and CO emissions versus engine ratings for diesel engines greater than 600 hp.

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2. **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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REFERENCES FOR CHAPTER 2 (Continued)

9. **"Limiting Net Greenhouse Gas Emissions in the United States, Volume I: Energy Technologies," DOE/PE-0101 Vol. I, Report for the Office of Environmental Analysis, Office of Policy, Planning and Analysis, Department of Energy, September 1991.**

3. EMISSION DATA REVIEW AND ANALYSIS PROCEDURES

This section reviews the literature search and evaluation procedures used to obtain and review documents (or other sources of information). It also presents data found among the sources, the data quality (indicating the level of confidence of the data in terms of method used to sample and reporting of results), and the guideline tables used to determine the data quality. All emissions data were reviewed and analyzed based on EPA guidelines which define criteria for rating both emissions test data and overall emission factors that were adopted in this update.¹ The criteria for rating individual emissions data were as follows:

- 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.**

The criteria for ranking emission factors are discussed in Chapter 4.

3.1 LITERATURE SEARCH AND EVALUATION

Several different approaches were followed to obtain literature/data to help update

the emission factors. The applicable references and sources listed in Table 3-1 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 extensive contacts with trade organizations, manufacturers, local, state, and federal air quality agencies. The references used and reviewed for this update are shown as References 2 through 14 at the end of this chapter.

A number of contacts were active when the emission factor documentation for this current update was produced. It is expected that some of these will produce new data after the current update is concluded. In particular, a letter was submitted to the Engine Manufacturers Association (EMA). Their responses to the request for information are in process. A complete list of contacts made can be found in Appendix B.

After reviewing all information provided by the sources, the data were initially evaluated according to EPA guidelines. A tabular rating checklist was used for each reference to determine whether or not data were presented in enough detail to be useful. These tables, presented in Chapter 4, also helped determine the data quality.

TABLE 3-1. EVALUATION OF REFERENCES

Reference	Used in AP-42 revision	Why/Why Not	Parameter of Interest	Raw Emission Factor Data
2	Yes	Used in previous revision/Source test data on diesel & dual fuels	Criteria	Yes
3	Yes	Review of combustion controls and table on percent NO _x reductions	NO _x	No
4	Yes	Review of NO _x control technology	NO _x	No
5	Yes	Review of criteria pollutant properties	Criteria	No
6	Yes	Technical results of complete source test on a diesel engine	Criteria, Non-Criteria, Organics, Particulate, PM-10	Yes
7	Yes	Data supplement of Reference No. 5	Criteria, Non-Criteria, Organics, Particulate, PM-10	Yes
8	Yes	Review of catalyst controls	NO _x , CO, Organics	No
9	Yes	Review of selective catalytic controls (on a natural gas engine)	NO _x	No
10	Yes	Source test data	CO ₂ , Organics	Yes
11	No	Source test data (engines too small) (belongs in AP-42 sect. 3.3)	Organics	Yes
12	Yes	Review of CO ₂ properties	CO ₂	No
13	Yes	Review of CO ₂ controls	CO ₂	No
14	No	Uses the same data from Reference No. 1	Criteria	Yes

REFERENCES FOR CHAPTER 3

1. **Technical Procedures for Developing AP-42 Emission Factors and Preparing AP-42 Sections, (Draft), Emission Inventory Branch, Technical Support Division, Office of Air and Radiation, Office of Air Quality Planning and Standards, U.S. Environmental Protection Agency, Research Triangle Park, NC, March 6, 1992.**
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 10. **"Pooled Source Emission Test Report: Oil and Gas Production Combustion Sources, Fresno and Ventura Counties, California," ENSR #7230-007-700, prepared by ENSR Consulting and Engineering for Western States Petroleum Association (WSPA), Bakersfield, CA, December 1990.**
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 13. **"Limiting Net Greenhouse Gas Emissions in the United States, Volume I: Energy Technologies," DOE/PE-0101 Vol. I, Report for the Office of Environmental Analysis, Office of Policy, Planning and Analysis, Department of Energy, September 1991.**
 14. **Shih, C.C., J.W. Hamersma, D.G. Ackerman, et al., "Emissions Assessment of Conventional Stationary Combustion Systems, Volume II: Internal Combustion Sources," EPA-600/7-79-029c, Report by TRW for Industrial Environmental Research Laboratory, Office of Energy, Minerals, and Industry, U.S. Environmental Protection Agency, February 1979.**

4. EMISSION FACTOR DEVELOPMENT

The unit(s) chosen for the emission factors was lb (of pollutant)/MMBtu (energy content of the fuel used), an input specific emission factor. Parallel SI units of ng/J were also tabulated. The emission factors are also presented in an output specific emission factor of g/hp-hr. To ensure the accuracy of the data for this revision, data without a brake specific fuel consumption (BSFC) number or without enough information to calculate the BSFC, were not used for calculations of the output specific emission factor in g/hp-hr. However, the input specific emission factor was still calculated (providing there were enough relevant data).

After all the data were ranked and entered into a table, the appropriate data were averaged to produce an emission factor. All emission factors were reviewed and analyzed based on EPA guidelines.¹ The significant difference in definitions between data ranking and emission factor ranking 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

A review of the previous criteria emission factors revealed that all data was taken from the Standard Support and Environmental Impact Statement (SSEIS).² This summary report determined emission factors for stationary reciprocating internal combustion engines fueled by diesel, dual fuel, or natural gas. The old data represented a computed weight average based on sales of engine horsepower during the mid-1970s. There is no documentation of the sales data nor any example calculations used in its derivation. The baseline NO_x and CO values are obtained from the SSEIS. For this update, the weighing factors were eliminated and the unweighted emissions data used. The new emission factors do not reflect any weighing factors.

In reviewing the raw data from the SSEIS, some of the CI engines used in the earlier calculations had some engines rated as "greater than 350 in³/cylinder" or medium bore. Since it is possible for an engine with >350 cubic inch displacement per cylinder (CID/CYL) to have engine ratings of less than 600 hp (thus categorizing it in Section 3.3 of AP-42, which includes medium bore engines), the decision was to not include these engines in the current revision of Section 3.4. This set of data has been rated a data quality rating of "C" because of the ambiguity of appropriate capacity designation.

Although about half of the dual fuel engines have engine ratings of >350 CID/CYL, the data were used in calculating the emission factor because there is no category for dual fuels in Section 3.3 of AP-42 and by default would belong to Section 3.4. Since there is no ambiguity of which section the data belong to in AP-42, the data were rated a quality of "B" quality.

The SO₂ emission factor is based on the assumption that all of the sulfur in fuel is converted to SO₂. Since the value was based on a valid assumption, it has been rated as "B" quality data.

After reviewing the raw data, it was decided that NO_x and CO from engines with BSFC data and actual engine horsepower ratings will be used in the new revision of emission factors for AP-42.

4.1.2 Review of New Data

There has been only two new data sets added to the current revision. One data set cannot be used to calculate the g/hp-hr value because it appears that the engine was not tested at full load.³ It could, however, be used to calculate the lb/MMBtu emission for CO. There is sufficient information to rate this source as of data quality "B." The other data point is of "A" data quality and was obtained from a report on a 2100 hp diesel engine.⁴

Although there are data for CO₂, an assumed value was used instead because the accuracy and method of measurement for CO₂ was not well defined. It was assumed that all of the carbon going into the engine as fuel will appear in the exhaust as CO₂. The contribution of carbon to other gases (such as CO and hydrocarbons [typically less than 0.1 percent]) is small. The emission factor for CO₂ will be a theoretical calculation of the carbon content of the fuel and 100 percent conversion of C into CO₂. Average carbon content of natural gas and diesel will be 70 percent by weight for natural gas (which is approximately 85 to 95 percent methane and the carbon content of methane is 75 percent by weight) and 87 percent by weight for diesel.

Since there is no new data for dual fuel engines, the data from the SSEIS were used (without the sales weighted factors). The old SO_x value in g/hp-hr (based on assumptions) was retained and an assumed BSFC value of 7163 Btu/hp-hr for dual fuel engines was used

to calculate the lb/MMBtu value.²

All formulas and assumptions used to make conversions and calculations are presented in Appendix C.

An attempt was made to examine the utility of AP-42 based on horsepower ranges in order to better estimate specific emissions for particular sources. Based on Figure 4-1, there are currently insufficient data to minimize the scatter in the available data for segregation of engines by horsepower to be meaningful.

4.1.3 Compilation of Baseline Emission Factors

Table 4-1a (Reference 2) and 4-1b (Reference 3 and 4) for diesel engines and Table 4-2 for dual fuel engines provide summaries of criteria pollutant and CO₂ emissions data.

4.1.4 Compilation of Controlled Emission Factors

There are many variables associated with the abatement of NO_x emissions. Table 4-3 shows the percentage of NO_x reduction and fuel consumption penalties for diesel and dual fuel engines.^{5,6,7}

4.2 TOTAL ORGANIC COMPOUNDS AND AIR TOXICS

4.2.1 Review of Previous Data

In the previous version of AP-42, the TOCs were divided into methane and nonmethane values. This classification was based on the assumptions that nonmethane organic compounds is 90 percent of TOC from diesel engines and only 25 percent of TOC emissions from dual fuel engines. Since the values were based upon assumptions, they should be replaced with measured data. Without new data, the emission factor quality rating based upon assumptions was rated as "E" quality.

4.2.2 Review of New Data

A new source was obtained for speciated TOC data, and it was determined to be of "B" data quality for lb/MMBtu values.³ This was from the source that did not appear to be running at full load and will, therefore, not be used to calculate g/hp-hr

values. With only one data point for speciated organic compounds, the emission factor quality rating of "D" was given.

Another new source did not speciate the TOC data into comparable compounds to the source mentioned above.⁴ It did, however speciate the TOC into "C₁," "C₂," "C₃," etc., and had a TOC value. By assuming C₁ to be methane (CH₄), there can be a separation of methane and nonmethane emission values based upon measured data. This one data point would upgrade the emission factor quality from "E" quality to "D." Since there is no new data for dual fuel engines, the emission factor (based upon assumptions stated in Section 4.2.1) will have a emission factor quality rating of "E."

The trace element data obtained did not have a mass balance from lube and diesel fuel analyses to the engine exhaust analysis.⁴ Therefore, there is no emission factor included for trace elements in this revision of AP-42. Appendix B contains the data table of the calculated trace metal throughput for future reference.

All formulas and assumptions used to make conversions and calculations are presented in Appendix C.

4.2.3 Compilation of Emission Factors

Table 4-4 shows a summary of the organic compounds data for diesel and dual fuel engines. Air toxics data for a diesel engine are summarized in Table 4-5.

4.3 PARTICULATE EMISSION AND SIZE DISTRIBUTION

4.3.1 Review of Previous Data

The previous particulate data in AP-42 were based on assumptions and a single telephone communication. Since the value was based upon assumptions, it should be replaced with real measured data. Without new data, the emission factor quality rating based upon assumptions should continue to be rated as "E" quality.

4.3.2 Review of New Data

There was only one new source on particulate emission and size distribution. This sole source is based upon a comprehensive study of one engine and is of "A" data quality.⁴ The particulate data were obtained using EPA Method 5, while the size distribution data were obtained using a Source Assessment Sampling System (SASS). The two methods

resulted in total particulate emissions having a difference of < 20 percent. This new data will replace the old "E" quality data point, but because of the lack of additional data, the emission factor rating must be "D" quality.

4.3.3 Compilation of Emission Factors

Particulate emission and size distribution emissions are shown on Table 4-6.

TABLE 4-1a. SUMMARY OF CRITERIA EMISSIONS DATA FOR DIESEL ENGINES^a

	Data Rating	Units	Rating	BSFC	HHV	NO _x	CO
Engine 24	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	610	6,930	19,300 137,030	8.70 379 2.77	8.50 371 2.70
Engine 62	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	711	7,040	19,300 137,030	4.80 206 1.50	3.90 167 1.22
Engine 43	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	1,100		19,300 137,030	17.07 ERR ^b ERR	2.20 ERR ERR
Engine 56	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	1,300	6,668	19,300 137,030	8.80 399 2.91	0.50 23 0.17
Engine 55	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	1,336	6,962	19,300 137,030	7.30 317 2.31	1.80 78 0.57
Engine 41	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	1,650		19,300 137,030	12.88 ERR ERR	0.93 ERR ERR
Engine 42	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	1,650		19,300 137,030	14.74 ERR ERR	4.14 ERR ERR

TABLE 4-1a. SUMMARY OF CRITERIA EMISSIONS DATA FOR DIESEL ENGINES^a (Continued)

	Data Rating	Units	Rating	BSFC	HHV	NO _x	CO
Engine 17	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	2,200	7,565	19,300 137,030	16.81 671 4.90	3.83 153 1.12
Engine 15	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	2,410	6,366	19,300 137,030	11.24 533 3.89	3.02 143 1.05
Engine 3	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	2,475		19,300 137,030	12.61 ERR ERR	1.19 ERR ERR
Engine 57	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	2,675	6,943	19,300 137,030	8.40 365 2.67	1.30 57 0.41
Engine 58	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	2,800	7,328	19,300 137,030	4.70 194 1.41	1.80 74 0.54
Engine 18	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	3,300	6,744	19,300 137,030	14.52 650 4.75	1.14 51 0.37
Engine 52	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	3,631	6,302	19,300 137,030	9.40 451 3.29	

TABLE 4-1a. SUMMARY OF CRITERIA EMISSIONS DATA FOR DIESEL ENGINES^a (Continued)

	Data Rating	Units	Rating	BSFC	HHV	NO _x	CO
Engine 19	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	3,900		19,300 137,030	11.87 ERR ERR	0.73 ERR ERR
Engine 59	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	3,900	6,888	19,300 137,030	7.90 346 2.53	1.30 57 0.42
Engine 5	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	4,300	6,677	19,300 137,030	10.99 497 3.63	3.85 174 1.27
Engine 51	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	4,880	6,650	19,300 137,030	8.50 386 2.82	
Engine 47	B	hp Btu/hp-hr Btu/lb Btu/gal g/hp-hr lb/1000gal lb/MMBtu	7,313	6,485	19,300 137,030	9.70 452 3.30	

- a. Reference 2. Data for engines 8-12, 26-27, 35-36, 60, 68, 72-75, and 77 were determined to be "C" quality, but were not used in emission factor development due to ambiguity of engine size.
- b. ERR = Insufficient information was available to compute emissions.

**TABLE 4-1b. SUMMARY OF CRITERIA AND CO₂ EMISSIONS DATA
FOR DIESEL ENGINES**

Data Type/ or Pollutant	Units	WSPA^a	TR-81-81^b
Data Rating		B	A
Rating	hp	850	2,100
wt % S (Fuel)		0.03	0.19
%O₂		10.2	13.7
F factor	dscf/MMBtu	9,157	9,220
BSFC	Btu/hp-hr		7,009
HHV	Btu/lb	19,479	18,943
	Btu/gal	138,300	134,497
NO_x	g/hp-hr		11.0
	lb/1000gal		465
	lb/MMBtu		3.46
CO	ppm	308	
	g/hp-hr		0.96
	lb/1000gal	55	41
	lb/MMBtu	0.40	0.30
SO_x	g/hp-hr		0.41
	lb/1000gal		17
	lb/MMBtu		0.13
CO₂	lb/1000gal	55	
	lb/MMBtu	0.40	

- a. Reference 3.
- b. Reference 4.

TABLE 4-2. SUMMARY OF CRITERIA EMISSIONS DATA FOR DUAL FUEL ENGINES^a

Data Type/ or Pollutant	Units	Engine 61	Engine 16	Engine 6
Data Rating		B	B	B
Rating	hp	711	2,410	8,658
BSFC	Btu/hp-hr	6,570	6,464	6,340
HHV	Btu/lb	19,300	19,300	19,300
	Btu/gal	137,030	137,030	137,030
NO _x	g/hp-hr	7.30	7.79	8.96
	lb/1000gal	336	364	427
	lb/MMBtu	2.45	2.66	3.12
CO	g/hp-hr	2.00	3.69	4.50
	lb/1000gal	92	172	214
	lb/MMBtu	0.67	1.26	1.56

- a. Reference 2. Data for engines 7, 13-14, 50, 76 and 78 were determined to be "B" quality, but were not used in emission factor development due to ambiguity of engine size.

TABLE 4-3. NO_x REDUCTION AND FUEL CONSUMPTION PENALTIES FOR DIESEL AND DUAL FUEL ENGINES

Control Approach		Diesel		Dual Fuel	
		%NO _x Reduction	ΔBSFC, ^a %	%NO _x Reduction	ΔBSFC, ^a %
Derate	10%			<20	4
	20%	<20	4		
	25%	5-23	1-5	1-33	1-7
Retard	2°	<20	4	<20	3
	4°	<40	4	<40	1
	8°	28-45	2-8	50-73	3-5
Air/Fuel	3%			<20	0
	±10%	7-8	3	25-40	1-3
Water Injection	50%	25-35	2-4		
(H ₂ O/fuel ratio)					
Selective Catalytic Reduction (SCR)		80-95	0	80-95	0

- a. BSFC = Brake Specific Fuel Consumption

TABLE 4-4. SUMMARY OF ORGANIC COMPOUND EMISSIONS DATA FOR DIESEL AND DUAL FUEL ENGINES

Data Source/ Engine	Fuel Type	Reference	Data Rating	Units	Methane	Non methane	HC: Total Exhaust (as CH₄)
TR-81-81	D	4	A	g/hp-hr lb/1000gal lb/MMBtu	0.03 1 0.01	0.33 14 0.10	0.36 15 0.11
WSPA	D	3	B	ppm lb/1000gal lb/MMBtu			34.5 3.25 0.0235
Engine 24	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.25 11 0.08
Engine 62	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.18 8 0.06
Engine 43	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.37 ERR^a ERR
Engine 56	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.10 5 0.03
Engine 55	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.30 13 0.09
Engine 41	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.42 ERR ERR
Engine 42	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.49 ERR ERR
Engine 17	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.42 17 0.12
Engine 15	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.46 22 0.16
Engine 3	D	2	B	g/hp-hr lb/1000gal lb/MMBtu			0.39 ERR ERR

TABLE 4-4. SUMMARY OF ORGANIC COMPOUND EMISSIONS DATA FOR DIESEL AND

DUAL FUEL ENGINES (Continued)

Engine	Fuel Type	Reference	Data Rating	Units	HC: Total Exhaust (as CH ₄)
Engine 57	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.30 13 0.10
Engine 58	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.20 8 0.06
Engine 18	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.42 19 0.14
Engine 52	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.27 13 0.09
Engine 19	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.37 ERR ERR
Engine 59	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.50 22 0.16
Engine 5	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.13 6 0.04
Engine 51	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.31 14 0.10
Engine 47	D	2	B	g/hp-hr lb/1000gal lb/MMBtu	0.17 8 0.06
Engine 61	DF	2	B	g/hp-hr lb/1000gal lb/MMBtu	1.50 69 0.50
Engine 16	DF	2	B	g/hp-hr lb/1000gal lb/MMBtu	5.67 265 1.93
Engine 6	DF	2	B	g/hp-hr lb/1000gal lb/MMBtu	5.16 246 1.79

Note: Reference 2: Data for engines 8-12, 26-27, 35-36, 60, 68, 72-75, and 77 were determined to be "C" quality, but were not used; D = Diesel; DF = Dual Fuel.
in emission factor development due to ambiguity of engine size.
a. ERR = Insufficient information was available to compute emissions.

TABLE 4-5. SUMMARY OF AIR TOXICS EMISSIONS DATA FOR A DIESEL ENGINE^a

Data Type or Pollutant	Units	Data
Rating	hp	850
wt % S (Fuel)		0.03
%O ₂		10.2
F factor	dscf/MMBtu	9,157
HHV	Btu/lb	19,479
	Btu/gal	138,300
Benzene	ppm	0.213
	lb/1000gal	0.1073
	lb/MMBtu	7.76E-04
Toluene	ppm	0.064
	lb/1000gal	0.0388
	lb/MMBtu	2.81E-04
Xylenes	ppm	0.040
	lb/1000gal	0.0268
	lb/MMBtu	1.93E-04
Propylene	ppm	1.433
	lb/1000gal	0.3863
	lb/MMBtu	2.79E-03
Formaldehyde	ppm	0.057
	lb/1000gal	0.0109
	lb/MMBtu	7.89E-05
Acetaldehyde	ppm	0.012
	lb/1000gal	0.0035
	lb/MMBtu	2.52E-05
Acrolein	ppm	0.003
	lb/1000gal	0.0011
	lb/MMBtu	7.88E-06
Naphthalene	μg/m ³	116.91
	lb/1000gal	0.0180
	lb/MMBtu	1.30E-04
Acenaphthylene	μg/m ³	8.27
	lb/1000gal	0.0013
	lb/MMBtu	9.23E-06
Acenaphthene	μg/m ³	4.19
	lb/1000gal	0.0006
	lb/MMBtu	4.68E-06

TABLE 4-5. SUMMARY OF AIR TOXICS EMISSIONS DATA FOR A DIESEL ENGINE^a (Continued)

Data Type or Pollutant	Units	Data
Fluorene	$\mu\text{g}/\text{m}^3$	11.49
	lb/1000gal	0.0018
	lb/MMBtu	1.28E-05
Phenanthrene	$\mu\text{g}/\text{m}^3$	36.58
	lb/1000gal	0.0056
	lb/MMBtu	4.08E-05
Anthracene	$\mu\text{g}/\text{m}^3$	1.11
	lb/1000gal	0.0002
	lb/MMBtu	1.23E-06
Fluoranthene	$\mu\text{g}/\text{m}^3$	3.62
	lb/1000gal	0.0006
	lb/MMBtu	4.03E-06
Pyrene	$\mu\text{g}/\text{m}^3$	3.33
	lb/1000gal	0.0005
	lb/MMBtu	3.71E-06
Benz(a)anthracene	$\mu\text{g}/\text{m}^3$	0.56
	lb/1000gal	0.0001
	lb/MMBtu	6.22E-07
Chrysene	$\mu\text{g}/\text{m}^3$	1.37
	lb/1000gal	0.0002
	lb/MMBtu	1.53E-06
Benzo(b)fluoranthene	$\mu\text{g}/\text{m}^3$	1
	lb/1000gal	0.0002
	lb/MMBtu	1.11E-06
Benzo(k)fluoranthene	$\mu\text{g}/\text{m}^3$	<0.2
	lb/1000gal	<3E-05
	lb/MMBtu	<2.18E-07
Benzo(a)pyrene	$\mu\text{g}/\text{m}^3$	<0.23
	lb/1000gal	<4E-05
	lb/MMBtu	<2.57E-07
Ideno(1,2,3-cd)pyrene	$\mu\text{g}/\text{m}^3$	<0.37
	lb/1000gal	<0.0001
	lb/MMBtu	<4.14E-07
Dibenz(a,h)anthracene	$\mu\text{g}/\text{m}^3$	<0.31
	lb/1000gal	<0.5E-05
	lb/MMBtu	<3.46E-07
Benzo(g,h,l)perylene	$\mu\text{g}/\text{m}^3$	<0.5
	lb/1000gal	<0.0001
	lb/MMBtu	<5.56E-07

a. Reference 3. All data are B quality.

**TABLE 4-6. SUMMARY OF PARTICULATE EMISSIONS DATA FOR
DIESEL ENGINES^a**

Data Type/ or Pollutant	Units	Emission Factor	
PM Size Distribution	<1μm	g/hp-hr	0.1520
		lb/1000gal	6.4302
		lb/MMBtu	0.0478
	1-3μm	g/hp-hr	0.0004
		lb/1000gal	0.0169
		lb/MMBtu	0.0001
	3-10μm	g/hp-hr	0.0054
		lb/1000gal	0.2284
		lb/MMBtu	0.0017
	>10μm	g/hp-hr	0.0394
		lb/1000gal	1.6668
		lb/MMBtu	0.0124
Total	g/hp-hr	0.1972	
	lb/1000gal	8.3424	
	lb/MMBtu	0.0620	
Total PM	Solids	g/hp-hr	0.2181
		lb/1000gal	9.2265
		lb/MMBtu	0.0686
	Condensables	g/hp-hr	0.0245
		lb/1000gal	1.0365
		lb/MMBtu	0.0077
	Total	g/hp-hr	0.2426
		lb/1000gal	10.2630
		lb/MMBtu	0.0763

a. Reference 4

REFERENCES FOR CHAPTER 4

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2. **"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.**
3. **"Pooled Source Emission Test Report: Oil and Gas Production Combustion Sources, Fresno and Ventura Counties, California," ESNR #7230-007-700, prepared by ENSR Consulting and Engineering for Western States Petroleum Association (WSPA, Bakersfield, CA, December 1990.**
4. **Castaldini, C., "Environmental Assessment of NO_x Control on a Compression Ignition Large Bore Reciprocating Internal Combustion Engine, Volume I: Technical Results," EPA-600/7-86/001a, prepared by Acurex Corp., for the Combustion Research Branch of the Energy Assessment and Control Division, Industrial Environmental Research Laboratory, Office of Research and Development, U.S. Environmental Protection Agency, Washington, DC, April 1984.**
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**5. AP-42 SECTION 3.4: LARGE STATIONARY DIESEL & ALL
STATIONARY DUAL FUEL ENGINES**

The revision to Section 3.4 of AP-42 is presented in the following pages as it would appear in the document.

3.4 LARGE STATIONARY DIESEL AND ALL STATIONARY DUAL FUEL ENGINES

3.4.1 General

The primary domestic use of large stationary diesel engines (greater than 600 hp) is in oil and gas exploration and production. These engines, in groups of three to five, supply mechanical power to operate drilling (rotary table), mud pumping and hoisting equipment, and may also operate pumps or auxiliary power generators. Another frequent application of large stationary diesels is electricity generation for both base and standby service. Smaller uses include irrigation, hoisting and nuclear power plant emergency cooling water pump operation.

Dual fuel engines were developed to obtain compression ignition performance and the economy of natural gas, using a minimum of 5 to 6 percent diesel fuel to ignite the natural gas. Large dual fuel engines have been used almost exclusively for prime electric power generation. This section includes all dual fuel engines.

3.4.2 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 3.4 deals only with compression ignition 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. Although all diesel fueled engines are compression ignited and all gasoline and gas fueled engines are spark ignited, gas can be used in a compression ignition engine if a small amount of diesel fuel is injected into the compressed gas/air mixture to burn any mixture ratio of gas and diesel oil (hence the name dual fuel), from 6- to 100-percent diesel oil.

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.

3.4.3 Emissions and Controls

Most of the pollutants from IC engines are emitted through the exhaust. However, some hydrocarbons 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 hydrocarbons from diesel compression ignition (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.

The primary pollutants from internal combustion engines are oxides of nitrogen (NO_x), organic compounds (hydrocarbons), carbon monoxide (CO), and particulates, which include both visible (smoke) and nonvisible emissions. The other pollutants are primarily the result of incomplete combustion. Ash and metallic additives in the fuel also contribute to the particulate content of the exhaust. Oxides of sulfur (SO_x) also appears in the exhaust from IC engines.

The primary pollutant of concern from large stationary diesel and all stationary dual fuel engines is NO_x , which readily forms in the high temperature, pressure, nitrogen content of the fuel, and excess air environment found in these engines. Lesser amounts of CO and organic compounds are emitted. The sulfur compounds, mainly SO_2 , are directly related to the sulfur content of the fuel. SO_x emissions will usually be quite low because of the negligible sulfur content of diesel fuels and natural gas.

Tables 3.4-1 (English units) and 3.4-2 (Metric units) contain gaseous emission factors.

Table 3.4-3 shows the speciated organic compound emission factors and Table 3.4-4 shows the emission factors for polycyclic aromatic hydrocarbons (PAH). These tables do not provide a complete speciated organic compound and PAH listing since they are based only on a single engine test; they are to be used for rough order of magnitude comparisons.

Table 3.4-5 shows the particulate and particle sizing emission factors.

Control measures to date have been directed mainly at limiting NO_x emissions because NO_x is the primary pollutant from diesel and dual fuel engines. Table 3.4-6 shows the NO_x reduction and fuel consumption penalties for diesel and dual fueled engines based on some of the available control techniques. All of these controls are engine control techniques except for the selective catalytic reduction (SCR) technique, which is a post-combustion control. The emission reductions shown are those which have been demonstrated. The effectiveness of controls on an particular engine will depend on the specific design of each engine and the effectiveness of each technique could vary considerably. Other NO_x control techniques exist but are not included in Table 3.4-6. These techniques include internal/external exhaust gas recirculation (EGR), combustion chamber modification, manifold air cooling, and turbocharging.

**TABLE 3.4-1. (ENGLISH UNITS) GASEOUS EMISSION FACTORS FOR LARGE STATIONARY DIESEL
AND ALL STATIONARY DUAL FUEL ENGINES^a
(Source Classification Codes)**

Pollutant	Diesel Fuel (SCC 20200401)			Dual Fuel ^b (SCC 20200402)		
	[grams/hp-hr] (power output)	[lb/MMBtu] (fuel input)	Emission Factor Rating ^c	[grams/hp-hr] (power output)	[lb/MMBtu] (fuel input)	Emission Factor Rating ^c
NO _x	11	3.1	C	9.2	3.1	D
CO	2.4	0.81	C	2.3	0.79	D
SO _x ^d	3.67S ₁	1.01S ₁	B	0.184S ₁ + 4.34S ₂	0.05S ₁ + 0.895S ₂	B
CO ₂ ^e	524	165	B	350	110	B
TOC, ^f (as CH ₄)	0.32	0.09	C	2.4	0.8	D
Methane	0.03	0.01	E ^g	1.8	0.6	E ^h
Nonmethane	0.33	0.10	E ^g	0.6	0.2	E ^h

^aData are based on uncontrolled levels for each fuel from references 4, 5, and 6. When necessary, the average heating value of diesel was assumed to be 19300 Btu/lb with a density of 7.1 lb/gal. The power output and fuel input values were averaged independently from each other due to the use of actual Brake Specific Fuel Consumption values for each data point and the use of data that may have enough information to calculate only one of the two emission factors (e.g., if there was enough information to calculate lb/MMBtu, but not enough to calculate the g/hp-hr). The emission factors are based on averages across all manufacturers and duty cycles. The actual emissions from a particular engine or manufacturer could vary considerably from these levels.

^bDual fuel is based on a mixture of 95 percent natural gas and 5 percent diesel fuel.

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

^dEmission factors are based on the assumption that all sulfur in the fuel is converted to SO₂. S₁ = percent sulfur in diesel fuel; S₂ = percent sulfur in gas.

^eBased on assumed 100 percent conversion of carbon in fuel to CO₂ with 87 weight percent carbon in diesel, 70 weight percent carbon in natural gas, dual fuel mixture of 5 percent diesel with 95 percent natural gas, average brake specific fuel consumption of 7000 Btu/hp-hr, diesel heating value of 19,300 Btu/lb, and natural gas heating value of 23,900 Btu/lb.

^fTotal Organic Compounds.

^gBased on emissions data from one engine.

^hBased on the assumption that nonmethane organic compounds are 25 percent of TOC emissions from dual fuel engines. Molecular weight of nonmethane gas stream is assumed to be that of methane.

TABLE 3.4-2. (METRIC UNITS) GASEOUS EMISSION FACTORS FOR LARGE STATIONARY DIESEL AND ALL STATIONARY DUAL FUEL ENGINES^a
(Source Classification Codes)

Pollutant	Diesel Fuel (SCC 20200401)			Dual Fuel ^b (SCC 20200402)		
	[g/kW-hr] (power output)	[ng/J] (fuel input)	Emission Factor Rating ^c	[g/kW-hr] (power output)	[ng/J] (fuel input)	Emission Factor Rating ^c
NO _x	14	1,322	C	12.3	1,331	D
CO	3.2	349	C	3.1	340	D
SO _x ^d	4.92S ₁	434S ₁	B	0.25S ₁ + 4.34S ₁	21.7S ₁ + 384S ₂	B
CO ₂ ^e	703	70,942	B	469	47,424	B
TOC, ^f (as CH ₄)	0.43	38	C	3.2	352	D
Methane	0.04	4	E ^g	2.4	240	E ^h
Nonmethane	0.44	45	E ^g	0.8	80	E ^h

^aData are based on uncontrolled levels for each fuel from references 4, 5, and 6. When necessary, the average heating value of diesel was assumed to be 19300 Btu/lb with a density of 7.1 lb/gal. The power output and fuel input values were averaged independently from each other due to the use of actual Brake Specific Fuel Consumption values for each data point and the use of data that may have enough information to calculate only one of the two emission factors (e.g., if there was enough information to calculate lb/MMBtu, but not enough to calculate the g/hp-hr). The emission factors are based on averages across all manufacturers and duty cycles. The actual emissions from a particular engine or manufacturer could vary considerably from these levels.

^bDual fuel is based on 95 percent natural gas and 5 percent diesel fuel.

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

^dEmission factors are based on the assumption that all sulfur in the fuel is converted to SO₂. S₁ = percent sulfur in fuel oil; S₂ = percent sulfur in gas.

^eBased on assumed 100 percent conversion of carbon in fuel to CO₂ with 87 weight percent carbon in diesel, 70 weight percent carbon in natural gas, dual fuel mixture of 5 percent diesel with 95 percent natural gas, average brake specific fuel consumption of 7000 Btu/hp-hr, diesel heating value of 19,300 Btu/lb, and natural gas heating value of 23,900 Btu/lb.

^fTotal Organic Compounds.

^gBased on emissions data from one engine.

^hBased on the assumption that nonmethane organic compounds are 25 percent of TOC emissions from dual fuel engines. Molecular weight of nonmethane gas stream is assumed to be that of methane.

**TABLE 3.4-3. (ENGLISH AND METRIC UNITS) SPECIATED ORGANIC COMPOUND
EMISSION FACTORS FOR LARGE STATIONARY DIESEL ENGINES^a
(Source Classification Code: 20200401)**

(Emission Factor Rating: E)^b

Pollutant	[lb/MMBtu] (fuel input)	[ng/J] (fuel input)
Benzene	7.76 E-04	3.34 E-01
Toluene	2.81 E-04	1.21 E-01
Xylenes	1.93 E-04	8.30 E-02
Propylene	2.79 E-03	1.20 E-00
Formaldehyde	7.89 E-05	3.39 E-02
Acetaldehyde	2.52 E-05	1.08 E-02
Acrolein	7.88 E-06	3.39 E-03

^aData based on the uncontrolled levels of one diesel engine from reference 5. There was enough information to compute the input specific emission factors of lb/MMBtu, but not enough to calculate the output specific emission factor of g/hp-hr. There was enough information to compute the input specific emission factors of ng/J, but not enough to calculate the output specific emission factor of g/kW-hr.

^b"E" rating for 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.

**TABLE 3.4-4. (ENGLISH AND METRIC UNITS) POLYCYCLIC AROMATIC
HYDROCARBON (PAH) EMISSION FACTORS FOR LARGE STATIONARY DIESEL
ENGINES^a**

(Source Classification Code: 20200401)

(Emission Factor Rating: E)^b

Pollutant	[lb/MMBtu] (fuel input)	[ng/J] (fuel input)
Polycyclic Aromatic Hydrocarbons (PAH)		
Naphthalene	1.30 E-04	5.59 E-02
Acenaphthylene	9.23 E-06	3.97 E-03
Acenaphthene	4.68 E-06	2.01 E-03
Fluorene	1.28 E-05	5.50 E-03
Phenanthrene	4.08 E-05	1.75 E-02
Anthracene	1.23 E-06	5.29 E-04
Fluoranthene	4.03 E-06	1.73 E-03
Pyrene	3.71 E-06	1.60 E-03
Benz(a)anthracene	6.22 E-07	2.67 E-04
Chrysene	1.53 E-06	6.58 E-04
Benzo(b)fluoranthene	1.11 E-06	4.77 E-04
Benzo(k)fluoranthene	< 2.18 E-07	< 9.37 E-05
Benzo(a)pyrene	< 2.57 E-07	< 1.10 E-04
Indeno(1,2,3-cd)pyrene	< 4.14 E-07	< 1.78 E-04
Dibenz(a,h)anthracene	< 3.46 E-07	< 1.49 E-04
Benzo(g,h,l)perylene	< 5.56 E-07	< 2.39 E-04
Total PAH	2.12 E-04	9.09 E-02

^aData are based on the uncontrolled levels of one diesel engine from reference 5. There was enough information to compute the input specific emission factors of lb/MMBtu and ng/J but not enough to calculate the output specific emission factor of g/hp-hr and g/kW-hr.

^b"E" rating for emission factors is 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.

**TABLE 3.4-5. (ENGLISH AND METRIC UNITS) PARTICULATE AND PARTICLE SIZING
EMISSION FACTORS FOR LARGE STATIONARY DIESEL ENGINES^a
(Source Classification Code: 20200401)**

(Emission Factor Rating: E)^b

Pollutant	Power Output		Fuel Input	
	[grams/hp-hr]	[grams/kW-hr]	[lb/MMBtu]	[ng/J]
Particulate Size Distribution				
<1 µm	0.1520	0.2038	0.0478	20.56
1-3 µm	0.0004	0.0005	0.0001	0.05
3-10 µm	0.0054	0.0072	0.0017	0.73
>10 µm	0.0394	0.0528	0.0124	5.33
Total PM-10 (≤10 µm)	0.1578	0.2116	0.0496	21.34
TOTAL	0.1972	0.2644	0.0620	26.67
Particulate Emissions				
Solids	0.2181	0.2925	0.0686	29.49
Condensables	0.0245	0.0329	0.0077	3.31
TOTAL	0.2426	0.3253	0.0763	32.81

^aData are based on the uncontrolled levels of one diesel engine from reference 6. The data for the particulate emissions were collected using Method 5 and the particle size distributions were collected using a Source Assessment Sampling System (SASS).

^b"E" rating for emission factors is 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.

**TABLE 3.4-6. NO_x REDUCTION AND FUEL CONSUMPTION PENALTIES FOR
LARGE STATIONARY DIESEL AND DUAL FUEL ENGINES^a
(Source Classification Codes)**

Control Approach	Diesel (SCC 20200401)		Dual Fuel (SCC 20200402)	
	Percent NO _x Reduction	ΔBSFC, ^b Percent	Percent NO _x Reduction	ΔBSFC, ^b Percent
Derate	10%		<20	4
	20%	<20	4	
	25%	5-23	1-5	1-33
Retard	2°	<20	4	<20
	4°	<40	4	<40
	8°	28-45	2-8	50-73
Air-to-Fuel	3%		<20	0
	±10%	7-8	3	25-40
Water Injection (H ₂ O/fuel ratio)	50%	25-35	2-4	
Selective Catalytic Reduction (SCR)		80-95	0	80-95

^aData are based on references 1, 2, and 3. The reductions shown are typical and will vary depending on the engine and duty cycle.

^bBSFC = Brake Specific Fuel Consumption.

References for Section 3.4

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3. Catalysts for Air Pollution Control, brochure by the Manufacturers of Emission Controls Association (MECA), Washington, DC, March 1992.
4. 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.
5. 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.
6. Castaldini, C., Environmental Assessment of NO_x Control on a Compression Ignition Large Bore Reciprocating Internal Combustion Engine, Volume I: Technical Results, EPA-600/7-86/001a, Combustion Research Branch of the Energy Assessment and Control Division, Industrial Environmental Research Laboratory, Office of Research and Development, U.S. Environmental Protection Agency, Washington, DC, April 1984.

APPENDIX A

SUMMARY OF COMMUNICATIONS ATTEMPTED/MADE

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

COMPANY/AGENCY	CONTACT PERSON
UTAH AIR QUALITY BOARD	DON ROBINSON, NORMAN ERICKSON
WAUKESHA	PAUL CANNESTRA
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 B

TRACE ELEMENT DATA

Trace Element Emissions Section 3.4
 (Based on 1 engine)
 05/14/92

BASED ON: BSFC = 7,009 Btu/hph
 HHV = 134,497 Btu/gal
 Engine= 2,100 hp

|-----RAW DATA-----|

|-----Lube Oil-----| |-----Diesel Fuel-----| |-----Exhaust-----| Lube Oil Fuel Exhaust % Out 0-50% Out >50-150% Out
 g/hp-hr lb/Mgal lb/MMBtu g/hp-hr lb/Mgal lb/MMBtu g/hp-hr lb/Mgal lb/MMBtu ug/s ug/s ug/s -----

Aluminum	4.29E-07	1.81E-05	1.35E-07	>3.60E-03	>1.52E-01	>1.13E-03	>3.94E-03	>1.67E-01	>1.24E-03	0.25	> 2100	> 2300	110%	110% Aluminum	
Antimony	1.20E-08	5.08E-07	3.77E-09	-----	-----	-----	9.94E-07	4.21E-05	3.13E-07	0.007	0.58	8286%			
Arsenic	6.00E-07	2.54E-05	1.89E-07	-----	-----	-----	5.49E-07	2.32E-05	1.73E-07	0.35	0.32	91%	91% Arsenic		
Barium	3.09E-05	1.31E-03	9.71E-06	1.34E-04	5.66E-03	4.21E-05	4.97E-06	2.10E-04	1.56E-06	18	78	2.9	3%	3% Barium	
Beryllium	<6.86E-09	<2.90E-07	<2.16E-09	-----	-----	-----	<5.49E-08	<2.32E-06	<1.73E-08	< 0.004	< 0.032	800%			
Bismuth	-----	-----	-----	-----	-----	-----	<5.49E-08	<2.32E-06	<1.73E-08	< 0.032					
Boron	<6.86E-09	<2.90E-07	<2.16E-09	3.43E-06	1.45E-04	1.08E-06	6.17E-05	2.61E-03	1.94E-05	< 0.004	2	36	1796%		
Bromine	4.29E-07	1.81E-05	1.35E-07	6.69E-05	2.83E-03	2.10E-05	2.57E-05	1.09E-03	8.09E-06	0.25	39	15	38%	38% Bromine	
Cadmium	4.29E-08	1.81E-06	1.35E-08	-----	-----	-----	<1.05E-05	<4.42E-04	<3.29E-06	0.025	< 6.1	24400%			
Calcium	>6.00E-05	>2.54E-03	>1.89E-05	8.57E-03	3.63E-01	2.70E-03	3.94E-03	1.67E-01	1.24E-03	>	35	5000	2300	46% 46% Calcium	
Cerium	-----	-----	-----	-----	-----	-----	4.97E-07	2.10E-05	1.56E-07	0.29					
Cesium	6.86E-09	2.90E-07	2.16E-09	-----	-----	-----	<5.49E-07	<2.32E-05	<1.73E-07	0.004	< 0.32	8000%			
Chlorine	3.26E-05	1.38E-03	1.02E-05	1.17E-04	4.93E-03	3.67E-05	4.97E-05	2.10E-03	1.56E-05	19	68	29	33%	33% Chlorine	
Chromium	4.11E-06	1.74E-04	1.29E-06	8.40E-06	3.55E-04	2.64E-06	1.49E-05	6.31E-04	4.69E-06	2.4	4.9	8.7	119%	119% Chromium	
Cobalt	4.29E-07	1.81E-05	1.35E-07	1.34E-05	5.66E-04	4.21E-06	1.47E-06	6.24E-05	4.64E-07	0.25	7.8	0.86	11%	11% Cobalt	
Copper	3.09E-06	1.31E-04	9.71E-07	1.01E-04	4.28E-03	3.18E-05	4.63E-04	1.96E-02	1.46E-04	1.8	59	270	444%		
Fluorine	1.22E-07	5.15E-06	3.83E-08	3.43E-05	1.45E-03	1.08E-05	2.57E-05	1.09E-03	8.09E-06	0.071	20	15	75%	75% Fluorine	
Gallium	3.09E-08	1.31E-06	9.71E-09	-----	-----	-----	2.40E-06	1.02E-04	7.55E-07	0.018	1.4	7778%			
Germanium	-----	-----	-----	-----	-----	-----	<5.49E-07	<2.32E-05	<1.73E-07	< 0.32					
Iodine	-----	-----	-----	<4.97E-06	<2.10E-04	<1.56E-06	1.10E-07	4.64E-06	3.45E-08	< 2.9	0.064	2%	2%	2% Iodine	
Iron	5.14E-05	2.18E-03	1.62E-05	1.68E-03	7.11E-02	5.28E-04	1.49E-03	6.31E-02	4.69E-04	30	980	870	86%	86% Iron	
Lanthanum	1.89E-08	7.98E-07	5.93E-09	-----	-----	-----	1.47E-06	6.24E-05	4.64E-07	0.011	0.86	7818%			
Lead	6.00E-07	2.54E-05	1.89E-07	3.43E-05	1.45E-03	1.08E-05	5.14E-06	2.18E-04	1.62E-06	0.35	20	3	15%	15% Lead	
Lithium	4.29E-08	1.81E-06	1.35E-08	1.01E-05	4.28E-04	3.18E-06	4.46E-07	1.89E-05	1.40E-07	0.025	5.9	0.26	4%	4% Lithium	
Magnesium	1.89E-06	7.98E-05	5.93E-07	3.43E-04	1.45E-02	1.08E-04	1.59E-04	6.74E-03	5.01E-05	1.1	200	93	46%	46% Magnesium	
Manganese	3.09E-07	1.31E-05	9.71E-08	3.43E-05	1.45E-03	1.08E-05	3.94E-05	1.67E-03	1.24E-05	0.18	20	23	114%	114% Manganese	
Mercury	-----	-----	-----	-----	-----	-----	<4.97E-04	<2.10E-02	<1.56E-04	< 290					
Molybdenum	2.40E-07	1.02E-05	7.55E-08	3.43E-04	1.45E-02	1.08E-04	5.14E-06	2.18E-04	1.62E-06	0.14	200	3	1%	1% Molybdenum	
Neodymium	-----	-----	-----	-----	-----	-----	1.47E-06	6.24E-05	4.64E-07	0.86					
Nickel	1.22E-06	5.15E-05	3.83E-07	1.17E-04	4.93E-03	3.67E-05	8.40E-05	3.55E-03	2.64E-05	0.71	68	49	71%	71% Nickel	
Niobium	-----	-----	-----	-----	-----	-----	<4.97E-07	<2.10E-05	<1.56E-07	< 0.29					
Phosphorus	3.60E-06	1.52E-04	1.13E-06	1.17E-03	4.93E-02	3.67E-04	>4.46E-03	>1.89E-01	>1.40E-03	2.1	680	> 2600	381%		
Potassium	>6.00E-05	>2.54E-03	>1.89E-05	7.37E-03	3.12E-01	2.32E-03	>8.06E-03	>3.41E-01	>2.53E-03	>	35	4300	> 4700	108%	108% Potassium
Rubidium	2.40E-08	1.02E-06	7.55E-09	-----	-----	-----	4.46E-06	1.89E-04	1.40E-06	0.014	2.6	18571%			
Samarium	-----	-----	-----	-----	-----	-----	5.49E-08	2.32E-06	1.73E-08	0.032					
Scandium	1.20E-08	5.08E-07	3.77E-09	<3.43E-06	<1.45E-04	<1.08E-06	4.97E-05	2.10E-03	1.56E-05	0.007	< 2	29	1445%		
Selenium	1.89E-07	7.98E-06	5.93E-08	-----	-----	-----	1.47E-04	6.24E-03	4.64E-05	0.11	86	78182%			
Silicon	5.83E-05	2.47E-03	1.83E-05	2.57E-03	1.09E-01	8.09E-04	2.06E-03	8.70E-02	6.47E-04	34	1500	1200	78%	78% Silicon	
Silver	6.86E-09	2.90E-07	2.16E-09	-----	-----	-----	6.34E-05	2.68E-03	2.00E-05	0.004	37	925000%			
Sodium	>6.00E-05	>2.54E-03	>1.89E-05	1.68E-04	7.11E-03	5.28E-05	>4.63E-03	>1.96E-01	>1.46E-03	>	35	98	> 2700	2030%	
Strontium	1.39E-05	5.87E-04	4.37E-06	1.68E-05	7.11E-04	5.28E-06	9.94E-06	4.21E-04	3.13E-06	8.1	9.8	5.8	32%	32% Strontium	
Sulfur	>5.83E-03	>2.47E-01	>1.83E-03	3.26E-01	1.38E+01	1.02E-01	>7.03E-03	>2.97E-01	>2.21E-03	>	3400	190000	> 4100	2%	2% Sulfur
Tellurium	-----	-----	-----	-----	-----	-----	2.74E-07	1.16E-05	8.63E-08	0.16					
Tin	1.22E-07	5.15E-06	3.83E-08	-----	-----	-----	3.43E-05	1.45E-03	1.08E-05	0.071	20	28169%			
Titanium	4.29E-06	1.81E-04	1.35E-06	3.43E-04	1.45E-02	1.08E-04	1.53E-04	6.45E-03	4.80E-05	2.5	200	89	44%	44% Titanium	
Tungsten	-----	-----	-----	<1.68E-05	<7.11E-04	<5.28E-06	-----	-----	-----	< 9.8	0%	0%			

Vanadium	2.40E-08	1.02E-06	7.55E-09	6.69E-06	2.83E-04	2.10E-06	4.97E-07	2.10E-05	1.56E-07	0.014	3.9	0.29	7%	7% Vanadium
Yttrium	-----	-----	-----	-----	-----	<4.97E-07	<2.10E-05	<1.56E-07		< 0.29				
Zinc	1.08E-05	4.57E-04	3.40E-06	1.51E-04	6.38E-03	4.75E-05	4.80E-04	2.03E-02	1.51E-04	6.3	88	280	297%	
Zirconium	-----	-----	-----	1.51E-05	6.38E-04	4.75E-06	6.17E-06	2.61E-04	1.94E-06	8.8	3.6	41%	41%	Zirconium

APPENDIX C

FORMULAS AND ASSUMPTIONS USED TO CONVERT AND CALCULATE EMISSION FACTORS

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 natural gas is 23900 Btu/lb

Density of diesel is 7.1 lb/gal

Weight % of carbon in diesel is 87%

Weight % of carbon in natural gas is 70%

For dual fuels, mixture of 5% diesel with 95% natural gas

Average brake specific fuel consumption (BSFC) = 7000 Btu/np-hr

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

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

$$\text{(heat input)} * (1 / (\text{engine rating in hp})) = \text{BSFC}$$

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

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

$$\text{(Btu/lb)} * (7.1 \text{ lb/gal}) = \text{Btu/gal}$$

$$\text{(Btu/gal)} / (7.1 \text{ lb/gal}) = \text{Btu/lb}$$

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

$$(\%/100) * (1000000/1000000) = (\% * 10000) / 1000000 = \% * 10000 \text{ ppmv}$$

e.g., 5.1% CO = 51000 ppmv

To convert µg/s to g/hp-hr

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

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

For Dual Fuel Engines only, to convert from g/hp-hr to lb/MMBtu

Assumptions: Average BSFC for dual fuel engines is 7163 Btu/hp-hr (see Reference No. ?)

$$(g/hp-hr) * (1/(7163 Btu/hp-hr)) * (1 lb/453.6 g) * (1000000 Btu/MMBtu) = lb/MMBtu$$

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 (ft³/MMBtu)

You know the % O₂ level

At STP conditions (293 K and 1 atm), 1 mole of gas occupies 385.3 ft³/lb-mol

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

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

$$(g/hp-hr) * (1.341 \text{ hp-hr/kW-hr}) = g/kW-hr$$

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

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

To calculate CO₂ emissions for diesel and dual fuel from weight % of carbon and BSFC (g/hp-hr):

Assumptions: 87 wt% carbon in diesel
 70 wt% carbon in natural gas
 Fuel mixture = 5% diesel and 95% natural gas
 Diesel heating value = 19300 Btu/lb
 Natural gas heating value = 23900 Btu/lb
 BSFC = 7000 Btu/hp-hr

For diesel fuel

$$(87 \text{ lbC}/100 \text{ lbFuel}) * (1 \text{ lb-molC}/12 \text{ lbC}) * (1 \text{ lb-molCO}_2/\text{lb-molC}) * (44 \text{ lbCO}_2/\text{lb-molCO}_2) * (\text{lbFuel}/19300 \text{ Btu}) * (1000000 \text{ Btu}/\text{MMBtu}) = \underline{165 \text{ lb/MMBtu}}$$

and

$$(165 \text{ lbCO}_2/\text{MMBtu}) * (1 \text{ MMBtu}/1000000 \text{ Btu}) * (453.6 \text{ g/lb}) * (7000 \text{ Btu/hp-hr}) = \underline{524 \text{ g/hp-hr}}$$

For natural gas

$$(70 \text{ lbC}/100 \text{ lbFuel}) * (1 \text{ lb-molC}/12 \text{ lbC}) * (1 \text{ lb-molCO}_2/\text{lb-molC}) * (44 \text{ lbCO}_2/\text{lb-molCO}_2) * (\text{lbFuel}/23900 \text{ Btu}) * (1000000 \text{ Btu}/\text{MMBtu}) = \underline{107.4 \text{ lb/MMBtu}}$$

For dual fuel (using lb CO₂/MMBtu emissions from the above calculations):

$$(\text{diesel emissions}) * (\text{diesel percentage}) + (\text{nat. gas emissions}) * (\text{nat. gas percentage}) \\ (165 \text{ lb/MMBtu}) * (5 \text{ lb diesel}/100 \text{ lb fuel}) + (107.4 \text{ lb/MMBtu}) * (95 \text{ lb nat. gas}/100 \text{ lb fuel}) \\ = \underline{110 \text{ lb/MMBtu}}$$

and

$$(110 \text{ lbCO}_2/\text{MMBtu}) * (1 \text{ MMBtu}/1000000 \text{ Btu}) * (453.6 \text{ g/lb}) * (7000 \text{ Btu/hp-hr}) = \underline{350 \text{ g/hp-hr}}$$

APPENDIX D

MARKED-UP PREVIOUS AP-42 SECTION 3.4

