Air-Fuel Control and Emissions for Gas Engines

- White Paper -

by

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Abstract

The basics of air-fuel control for engines fuelled with gaseous fuels and effects on exhaust emissions are discussed. The sections on air-fuel control include data for specific fuels, the need for precise air-fuel control, and methods used for air-fuel control. The sections on emissions cover the effects of air-fuel control on emissions, methods to reduce emissions, and methods to calculate emissions. Appendices cover the combustion process, energy efficiency, and effects of altitude and humidity.

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Introduction

This paper covers a number of aspects of air-fuel[1] control and emissions for engines fueled with gaseous fuels such as natural gas, propane, and landfill gas[2]. The reasons for air-fuel control are covered followed by a discussion of the combustion process. Then, the evolution of air-fuel control techniques is discussed including the features needed by a control system. Engine emissions are discussed and the effect of air-fuel control on emissions is reviewed.

The objective of this paper is to provide an understanding of the topics. The basic principles of the internal combustion engine are well known; the detailed explanation of all of the processes is too complex to be covered here. Simplifying assumptions have been made with the intention of providing the reader with an insight of the processes involved. When mathematical expressions have been used as part of the explanations, associated diagrams and graphs have been provided to show the relationships. While a more detailed treatment of engines is provided by reference texts, in most cases the emphasis of these texts is for liquid fuelled engines (gasoline and diesel) rather than gaseous-fuelled engines. This paper deals specifically with gaseous-fuelled engines.

Why Control the Air-fuel mixture?

Oxygen necessary for the combustion comes from the air supplied to the engine. The term air fuel ratio is normally used in place of the more precise term oxygen-fuel control. Fundamentally, the ratio of air to fuel must be controlled within certain limits so the engine will run. If there is too much fuel relative to the oxygen present, the spark will not ignite the mixture in the cylinder and no combustion will occur. On the other hand, if there is not sufficient fuel, the spark will not start combustion, any combustion started by a spark will extinguish or the combustion will start too late to generate mechanical energy efficiently.

Why control the air fuel mixture precisely?

No automobile engine currently sold would be able to meet emissions standards and achieve a commercially acceptable fuel economy without precise air-fuel ratio control. With gas engines the same is true. Field experience shows that proper air fuel control affects:

- the fuel economy

[1] The term ‘air-fuel ratio’ is sometimes replaced by its inverse ‘fuel-air ratio’.
[2] Engines fueled by liquid gasoline are known as gasoline engines

- the maintenance needs
- the stability of operation
- emission rates for noxious and/or regulated exhaust gases
- ability to operate efficiently over a wide range of conditions
- ease of starting
- presence and degree of detonation

The need for control and regulatory compliance of emissions and the strong dependence of emissions on the air-fuel mixture is often a driving force for good and precise air-fuel control. Even without the need for emissions controls, significant savings in fuel economy have been demonstrated; for example see the REMVue quantified business results[3].

The Combustion Process

Gaseous fuel consists of combustible gases such as hydrocarbons (HCs), hydrogen (H_2) and carbon monoxide (CO). The fuel may also be mixed with non-combustible gases such as carbon dioxide (CO_2), nitrogen (N_2), helium (He), argon (A) and water vapor (H_20).

In the gas engine a spark normally ignites the compressed air-fuel mixture in the cylinder volume. The carbon portion of the fuel may be emitted as C (carbon), combines with the oxygen from the air to form CO (carbon monoxide) or CO_2 (carbon dioxide), or be contained in more complex compounds containing carbon. The hydrogen portion of the fuel is combined with oxygen to form H_2O (water vapor) or is contained in more complex compounds containing hydrogen, carbon and oxygen. The makeup of the combustion gases depends on the fuel type, temperature, pressure, the time after combustion starts, and the amount of oxygen present.

In a spark-ignited (SI) engine the objective is for the electrical discharge across the spark plug electrodes to start the combustion. The flame front should advance over the volume defined by the cylinder head and the piston to consume all of the air or fuel available. The details of the combustion process are shown in more detail in Appendix 1.
In the combustion process the fuel to air ratio affects

- the maximum cylinder pressure
- the rate of combustion (burn),
- the maximum temperature, and
- the concentration of unburned hydrocarbons, carbon monoxide, and nitrogen oxides in the exhaust gases.

These combustion parameters strongly influence

- the engine efficiency
- the spark plug life and

[3] Quantified business results relating to air-fuel control are available from REM Technology Inc.

- period between minor and major overhauls.

These factors all affect operational costs.

Air-fuel ratio control terminology

There are several terms to describe the ratio of air to fuel:

- **Stochiometric** – Here there is just sufficient air present for all the carbon and hydrogen in the fuel to be converted to CO_2 and H_2O respectively.
- **Rich** – There is insufficient oxygen to enable complete combustion.
- **Lean** – There is more oxygen is present than is required for complete combustion.

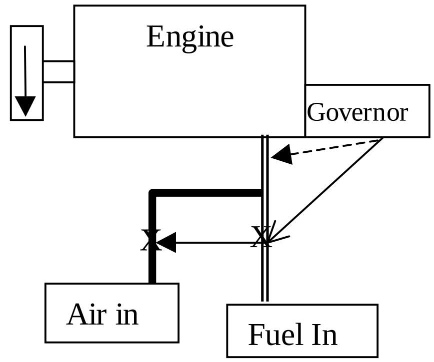
The mass and volume ratios of air and fuel for a stochiometric process depend on the chemical formula for the fuel. The table below shows the stochiometric ratios for a number of gaseous (g) and liquid (l) fuels. The volumetric ratio is shown only for gaseous fuels.

Table 1

Fuel	Formula	Air-fuel Ratio - Mass	Air-fuel ratio - Volume
Methane (g)	CH_4	17.2	9.5
Ethane (g)	C_2H_6	16.1	16.7
Propane (g)	C_3H_8	15.7	23.9
Gasoline (l)*	C_8H_{18} (octane)	14.6	
Light Diesel (l)*	$C_{16}H_{34}$ (cetane)	14.5	
Methanol (l)	CH_4O	6.5	
Carbon monoxide (g)	CO	2.47	2.39
Hydrogen (g)	H_2	34.3	2.39

* Gasoline and Diesel are normally a mixture of several different hydrocarbons.

Air-Fuel Control



To control the engine speed the engine governor controls the amount of fuel or air-fuel mixture reaching the engine. The amount of air is then controlled to ensure the desired air-fuel mixture ratio. Originally a mechanical governor was used for speed control. This was later replaced by the hydraulic governor. Air was typically controlled either by a carburetor or by a lever operating an airflow control valve.

Traditional problems with these systems have been the wear of the mechanical components, the difficulty with starting, and with achieving control for the full speed range and conditions ranging from starting to unloaded to overload conditions. It has proven to be almost impossible to meet modern emissions standards with these mechanical types of controls.

Electronic controls provide a much greater capability to control the engine speed and the air-fuel ratio not only at fixed conditions but also over the whole operating range of the engine. Furthermore, the electronic controls are not subject to wear and are consistent in their actions. The electronic control algorithms can be made to allow for many more factors that affect operating conditions such as

- Fuel heat content,
- Air temperature,
- Fuel temperature,
- Decreases in turbocharger efficiency,
- Changes in load, and
- Cylinder to cylinder variations

Air Fuel Control Factors

Lambda -The excess amount (or deficiency) of air relative to a stochiometric mixture is given by lambda (λ)[4]. It is the ratio of the actual air to fuel ratio (A/F) to the stochiometric air to fuel ratio $(A/F)_S$

$$\lambda = (A/F) / (A/F)_S$$

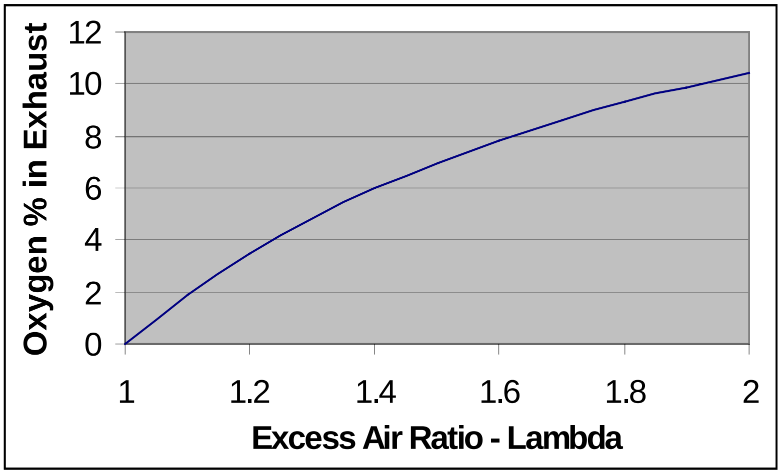
With lean conditions lambda can be measured from the percentage of oxygen in the exhaust gases.

[4] Often the fuel air equivalence ratio ϕ (phi) is used. This is the inverse of λ.

$\lambda = 20.9 / (20.9 - O_2\%)$

When all the oxygen is used by the fuel, $\lambda = 1.0$; as the amount of oxygen in the exhaust increases, the value of λ increases.

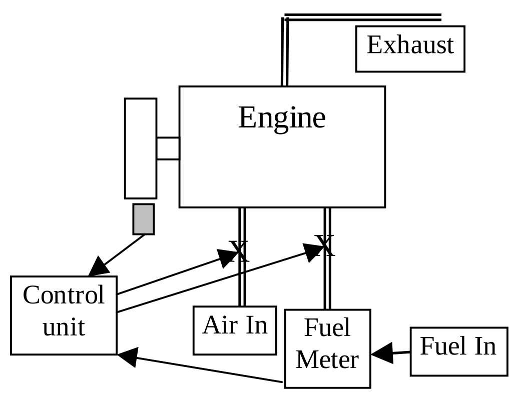
The air-fuel control system must be able to maintain the desired lambda value.



Air-Fuel Ratio Control Methods

The fuel provides the energy for operating the engine. Either the fuel is controlled manually or, more commonly, a governor controls the speed. The governor controls a fuel valve or a throttle valve such that when the speed drops below the desired speed (set point), more fuel is added and vice-versa. To maintain the desired air-fuel ratio either open loop or closed loop control may be used.

1. Open loop control – The fuel flow may be measured by volume or mass. Referring to Table 1, it is clear that if the fuel is measured by volume, the volume of air required depends strongly on the heat content of the fuel which may be a single component hydrocarbon or a mixture. For example, a given volume of propane fuel requires 2.5 times more air than the same volume of methane fuel. If the fuel is measured by mass flow (lbs or kg), then the mass of air is affected less by different fuel mixtures. If the fuel is measured by volume, it may be converted to mass if the fuel density and temperature are known.



   Once the fuel mass is known, the desired air mass may be calculated according to the desired λ value. This air mass can be converted to air volume by using the perfect gas law to calculate the desired air pressure

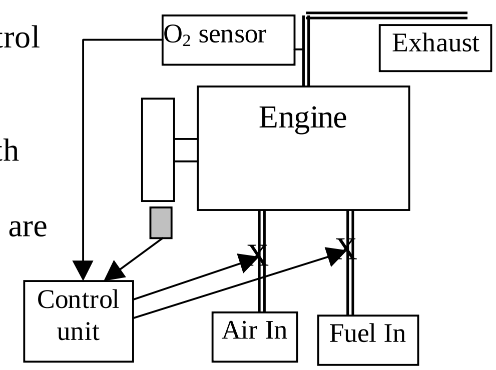
$$P_{air} = \lambda M_f R(T_a + 273) / F_a$$

Where M_f is the fuel mass flow rate,
R is the gas constant,
T_a is the air manifold temperature in degrees C,
P_{air} is the absolute air manifold pressure[5] and
F_a is the volumetric flow rate of the air.

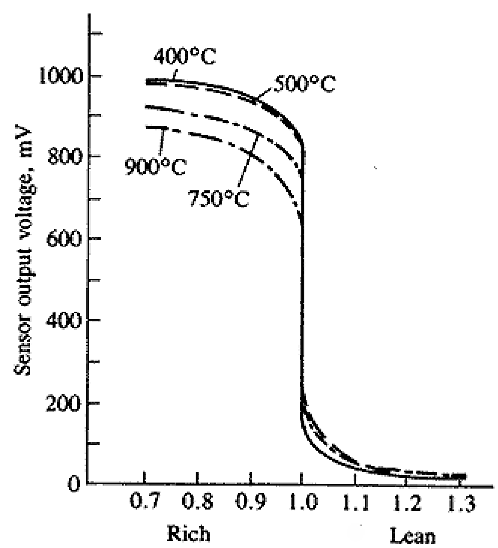
The volumetric flow rate, F_a, can be calculated from the engine displacement and the RPM.

If non-combustible gases dilute the fuel, an adjustment factor can be added to reduce the amount of air according to amount of fuel dilution.

2. Closed loop control – With closed loop control the amount of oxygen is measured in the exhaust and the amount of air is adjusted to maintain the desired amount of oxygen. With the closed loop control, no fuel flow measurement is required and no calculations are required to determine the amount of air.



A common sensor for measuring exhaust oxygen has a narrow response range limited to 0% oxygen ($\lambda = 1.0$) as shown at the right. Due to the step-wise response at stoichiometry the sensor is useful for stochiometric air-fuel control but has limited capability for other set points.



[5] Refer to Appendix 3 for effects of altitude and relative humidity

Efficiency

Efficiency of an engine is measured by the ratio of the rate of fuel energy in and the mechanical power out. The available fuel energy is defined by the lower heating value (LHV)[6]. Fuel measurement, while not required for closed loop control, is very useful to determine the operating efficiency of the engine. The energy flowing into the engine can be calculated and the brake power (mechanical energy out) can be measured. The result can be expressed either as thermal conversion efficiency or brake specific fuel consumption (BSFC).

The thermal conversion efficiency is rate of fuel energy in/mechanical power out. Typical efficiencies for fully loaded reciprocating engines range from 30 to 38%. The brake specific fuel consumption (BSFC) is the fuel energy in/unit time/brake power out. A typical BSFC (in English units) is 7000 to 8000 BTU/BHP-hr. In SI units typical values are 9890 to 11300 kJ/kW-hr.

The product of the fuel flow rate and the LHV of the fuel measure the rate of fuel energy going into an engine. Table 2 shows some typical values for the LHV.

Table 2

Fuel	LHV MJ/kg	LHV MJ/M^3 (gas fuels)	LHV BTU/scf
Methane (g)	49.8	35.8	966
Ethane (g)	44.2	60.0	1618
Ethylene (g)	44.1	55.6	1499
Propane (g)	45.5	92.0	2468
Gasoline (l)	43.0		
Light Diesel (l)	42.5		
Methanol (l)	20.05		
Carbon monoxide (g)	10.1	12.6	338
Hydrogen (g)	120.0	10.8	290

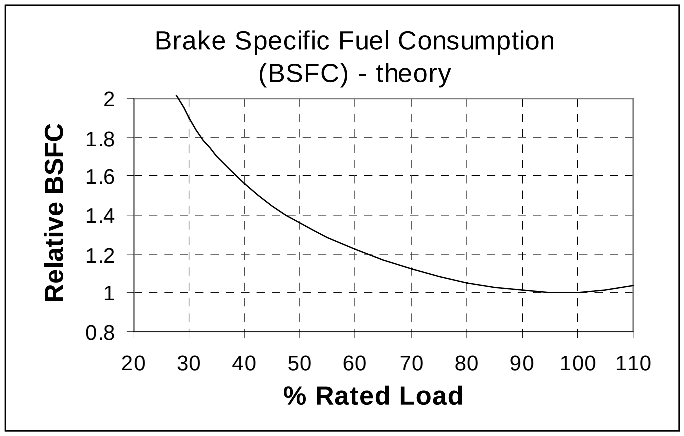
The mechanical power generated by the fuel combustion may be measured either by calculation of the indicated power (IP) from the engine cylinder pressures for each crankshaft position and the RPM or by measurement of the load (e.g. product of current and voltage for a generator, flow rate and compression ratio of a compressor) and adjustments for losses due to friction and engine parasitics such as oil pumps, fans etc.

[6] The lower heating value does not include the energy released from the condensation of the water vapor products of combustion, as is the case in engines. The HHV or higher heating value includes the heat released by water vapor condensation.

- The engine thermal efficiency, η = P/fuel energy rate, where P, the power, may be either the indicated power or the brake power[7].
- A more common measure of efficiency is the brake specific fuel consumption (BSFC).

BSFC = fuel energy rate/brake power.

The BSFC as a function of percentage load shows the typical behavior[8]. As the load increases to 100%, the BSFC decreases because a smaller fraction is used for the engine auxiliaries and friction. As the load increases past 100%, the BSFC increases due to incomplete fuel combustion and the increase of friction at high loads.

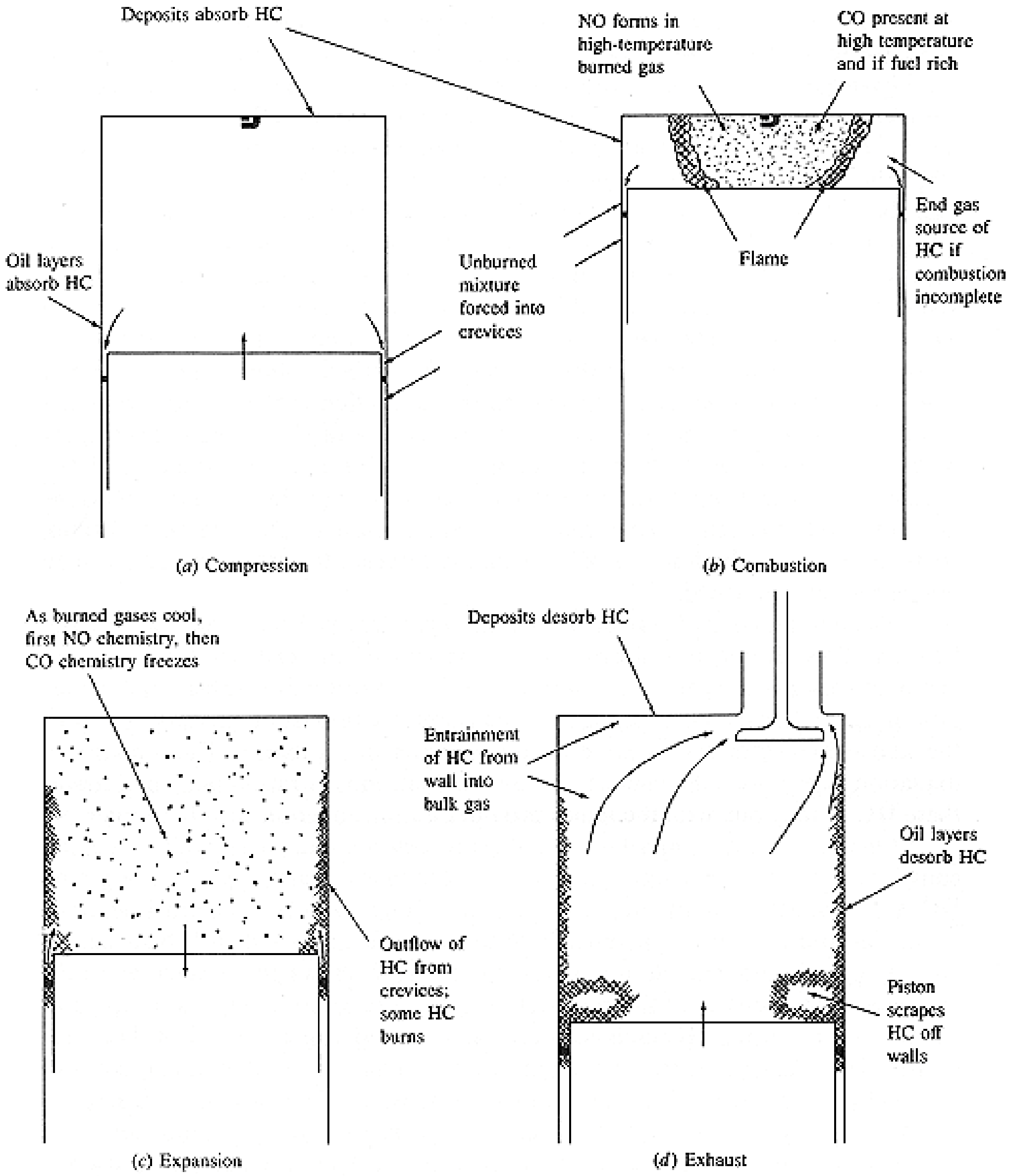


Each engine has a characteristic BSFC curve that depends on a number of factors. Deviation from the expected BSFC curve can be used as a warning of poor operation.

[7] For conversion – 1 HP = 2546 BTU/hr; 1 kW = 1000 joule/s
[8] See Appendix 2 for details.

Air-Fuel Ratio and Emissions and Relationship to Air-Fuel Ratio Control

Emissions refer to those gases that are produced by the engine in addition to the CO_2 and H_2O from combustion. The diagram[9] below shows the sources of emissions during the various stages in the combustion.



[9] Internal Combustion Engine Fundamentals – John B. Heywood

The emissions are generally placed into the following categories:

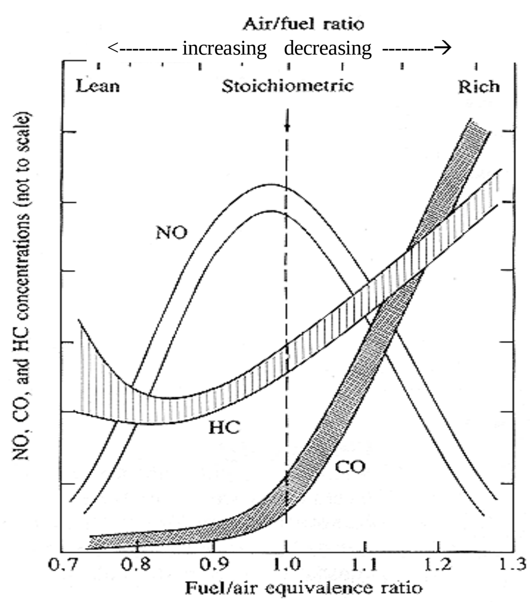
- Carbon monoxide – a poisonous gas
- NOX – the sum of NO and NO_2. This can generate ozone, a respiratory irritant. Normally the concentration of NO is several times that of NO_2 in SI engines; the NO gradually converts to NO_2 after leaving the engine.
- Unburned hydrocarbons (HCs) – any compounds with H and C that may include the original fuel gas. This can also include oil vapors. Sometimes the term volatile organic compound (VOC) is used.
- Particulates – normally particles of carbon from unburned fuel.
- Hazardous air pollutants (HAPs) – normally 90% formaldehyde (H_2CO) with about 10% other aldehydes and benzene type compounds.

Environmental agencies have generally placed restrictions on the CO and NOX emissions. In the USA there is a proposal to extend the regulations to HAPs. Emission limits generally depend on the regulations of the region (state, province, county, city). Normally the limits are related to the power output of the engine. Typical units of measure are g/BHP-hr and g/kW-hr.

NOX and CO is formed during combustion. The unburned hydrocarbons come from crevasses (e.g. between the cylinder wall and the piston above the top ring), desorption from the cylinder wall (oil film and deposits) and from regions of incomplete combustion within the main volume. The other source of emissions is the crankcase. While most automobiles divert the crankcase vapors to the intake air, this is less common for industrial engines.

When exactly the correct amount of oxygen is present the fuel is fully burned. This demands perfect mixing, a condition that in practice does not occur. In fact, with 0% oxygen in the exhaust, the combustion conditions are slightly rich.

When there is too little oxygen, the fraction of CO and unburned hydrocarbons (HCs) increases. As the amount of oxygen increases, the fraction of CO and HCs decreases. This is shown schematically by the graph[7].



As the air to fuel ratio increases above 1.0 (amount of excess air increases), the nitrogen oxide concentration first increases and then drops while the carbon monoxide concentration becomes very low. The concentration of unburned hydrocarbons initially decreases and then increases. This increase as the mixture becomes leaner is due to no combustion events or regions in the combustion chamber where the combustion is incomplete.

Why does the NOX decrease with excess air?

To understand this it is necessary to review the chemistry of NO production. NO_2 will be neglected as it normally is about 1/10th the concentration of NO and originates from NO.

The chemical reactions that produce NO are

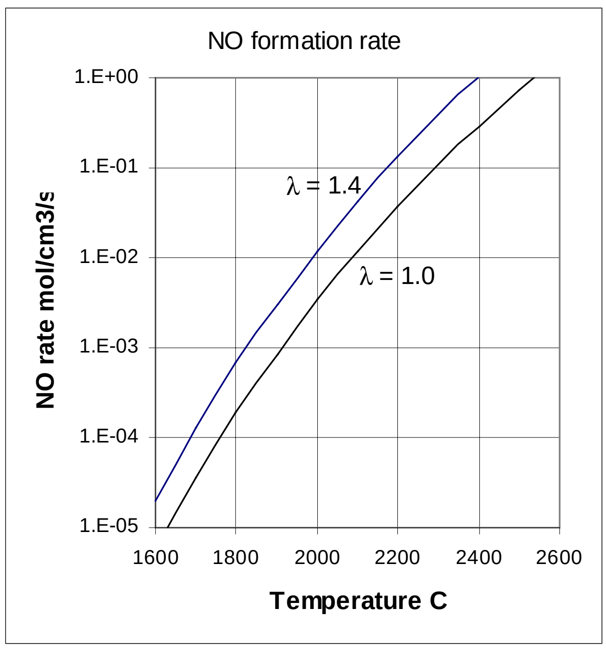
- $O + N_2 = NO + N$
- $N + O_2 = NO + O$
- $N + OH = NO + H$

Each of these has a chemical rate constant K that depends on temperature. Following Taylor[10] the rate of production of NO is given by

$$d[NO]/dt = (6 \times 10^{16} / T^{1/2}) \exp(-69000/T) [O_2]^{1/2} [N_2]$$

where [NO] is the molar concentration of NO,
T is the absolute temperature
$[O_2]$ is the equilibrium molar concentration of oxygen, and
$[N_2]$ is the equilibrium molar concentration of nitrogen.

The behavior of NO production with temperature is shown schematically below.
Note the strong temperature dependence and the dependence on oxygen concentration. The oxygen concentration depends on the excess air amount while the nitrogen concentration is almost constant. A graph showing the variation in the rate of NO production with temperature is shown here. An increase of only 200 degrees in temperature increases the NO formation rate by more than 10 times.

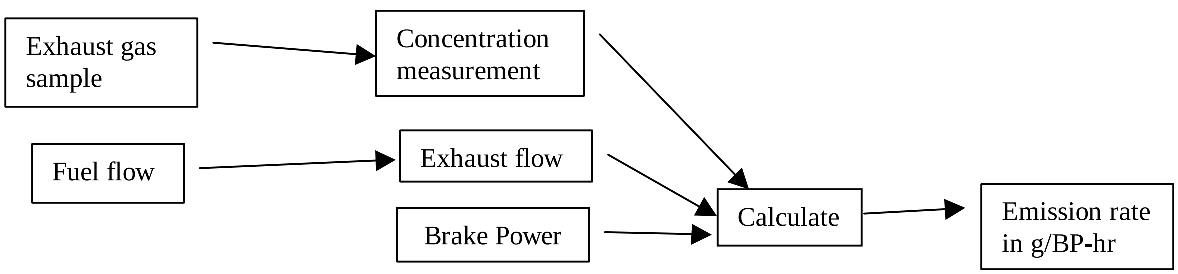


[10] The Internal Combustion Engine in Theory and Practice by Charles Fayette Taylor

Initially as the excess air (oxygen) increases, the production rate of NO increases; adding excess air reduces the maximum temperature, which reduces the rate of NO production.

Measurement of Emissions

Exhaust emissions are normally measured with an exhaust gas analysis unit. This device samples the exhaust gas, removes the water, and uses various electrochemical and other techniques to measure the oxygen percentage, the CO, CO_2, NO, NO_2 and hydrocarbons (HC). The gases are measured as a concentration expressed either as a percentage or in ppm (parts per million). To determine the total emissions the total exhaust flow must be measured or calculated. A method of calculating flow, accepted by the US Environmental Protection Agency (EPA), is known as Method 19 of 40CFR Part 60. The calculation is based on the measured fuel flow and the oxygen percentage in the exhaust. For more details refer to the REM Technology technical note "Calculation of Emissions using Method 19 of 40CFR Part 60".



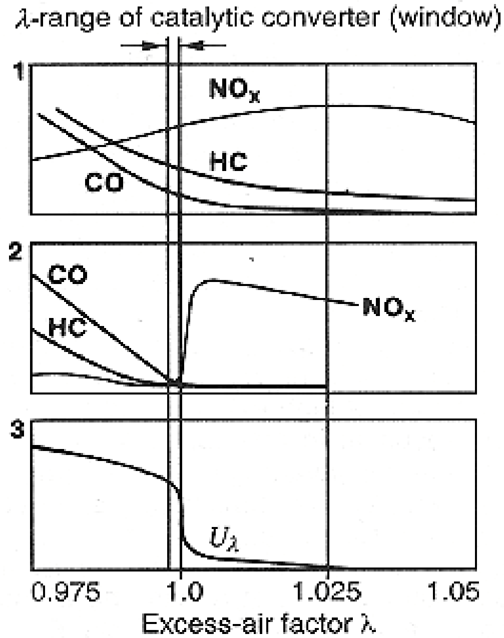
Methods to reduce CO, NOx and HC emissions

There are three commonly applied techniques for reducing the concentrations of these gases in the exhaust:

1. Use of a catalytic convertor
2. Addition of excess air
3. Recirculation of exhaust gases

1. Catalytic convertor

In a catalytic convertor a chemical reaction is promoted by the presence of catalysts such as Pt, Rh, Pd. The most common type, the three-way catalyst requires operation of the engine very close to the stochiometric region. The diagram[11] here shows the efficiency of the catalyst as a function of air-fuel ratio. Note in particular how rapidly the NOX increases as the excess air factor exceeds 1.0 by more than 0.5% (5 parts per thousand)



Catalytic-converter efficiency as a function of excess-air factor λ.
1 Exhaust emissions upstream of 3-way catalytic converter, 2 Exhaust emissions downstream of 3-way catalytic converter, 3 Electric signal from Lambda oxygen sensor, U_λ Sensor voltage.

The catalysts deteriorate with use due to loss of the catalysts, excess heating, or excess oxygen. One of the greatest problems is the effect of mis-fires (no combustion events) where the unburned fuel and unused oxygen leads to overheating and damage. Because of the precious metal content, the cost of replacement is substantial.

The three way catalytic convertor promotes the following reactions to consume the CO and NO:

$NO + CO \rightarrow \frac{1}{2} N_2 + CO_2$

$2NO + 5CO + 3 H_2O \rightarrow 2 NH_3 + 5 CO_2$

$2 NO + CO \rightarrow N_2O + CO_2$

$NO + H_2 \rightarrow \frac{1}{2} N_2 + H_2O$

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

$2 NO + H_2 \rightarrow N_2O + H_2O$

The production of ammonia (NH_3), which is an unpleasant gas, can be reduced but not eliminated, by the choice of catalysts.

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[11] Automotive Handbook, 4th Edition 1996, Robert Bosch Gmbh

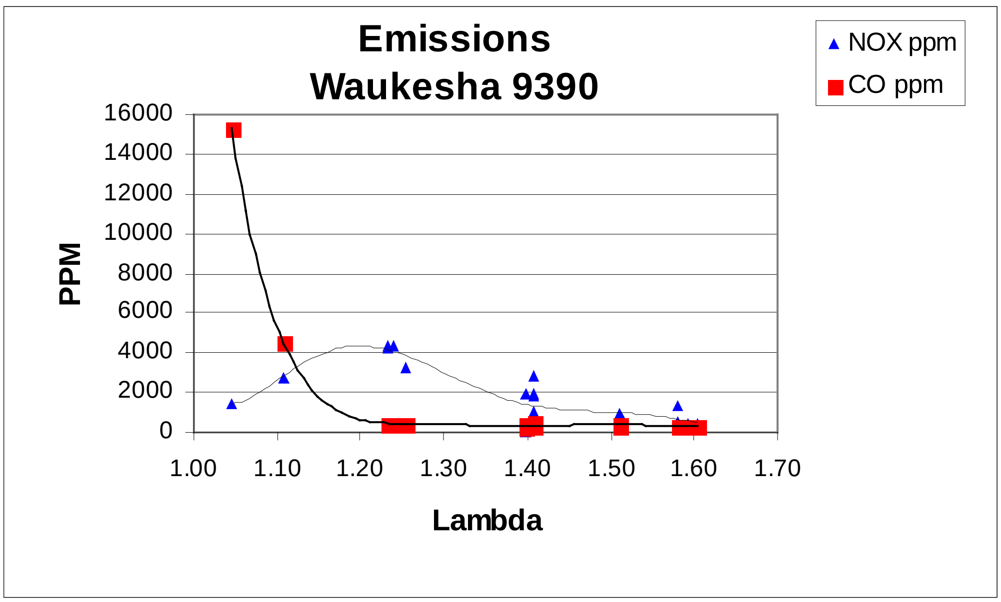
2. Addition of excess air

The diagram earlier (page 9) shows that using a very lean mixture reduces the NOX concentration. Unfortunately, as the mixture becomes leaner, it becomes more difficult to ignite the mixture with a spark plug. Four methods have been devised to increase ensure good ignition:

- Increased spark gap, higher voltages and multiple spark discharges (multi-strike),
- A stratified fuel charge so the fuel mixture is richer near to the spark plug,
- A precombustion chamber where the air-fuel mixture is richer than the main combustion chamber, and
- Use of pilot fuel – the injection of diesel fuel, which undergoes compression ignition to start the combustion of the gaseous fuel.

A supercharger (engine driven compressor) or a turbocharger (exhaust driven compressor) usually supplies the excess air. The effect of increased air on emissions is shown by the experimental results on emissions for a Waukesha engine.

Advantages of using excess air are more efficient fuel consumption and the elimination of the costs of a catalytic convertor. One disadvantage is the increase in the peak pressure in the cylinder during combustion[12]. In some cases the maximum power rating of the engine compared to operation with a stochiometric mixture must be reduced. A typical reduction is 10%.

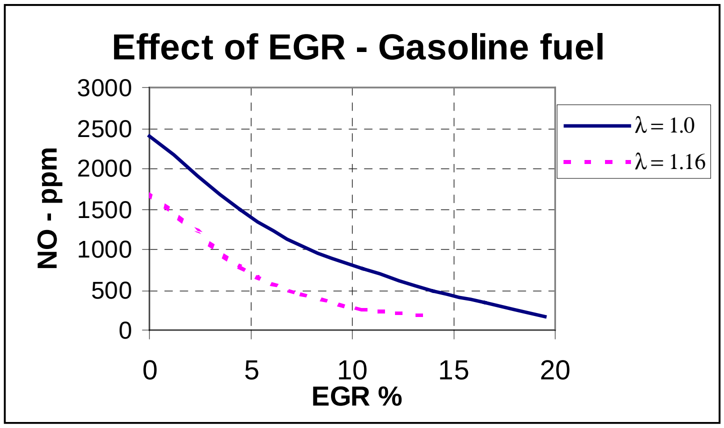


[12] See Appendix B

3. Exhaust Gas Recirculation (EGR)

By diverting a fraction of the exhaust gases to the engine intake the NOX emissions fall. The exhaust gases add mass to the intake gases and cause a temperature reduction during combustion similar to the effect of increased air. The maximum amount of EGR is limited by the ability to ignite the air-fuel mixture. EGR can be used to reduce the NOX emissions with a stochiometric air-fuel mixture, so can be used with a 3-way catalytic converter. EGR is used widely for automobile engines, but is not common with gas industrial engines where turbochargers are more common.

The graph here shows the effect of EGR with a spark ignited gasoline engine. For both a stochiometric air-fuel mixture and a 16% lean mixture the nitrous oxide (NO) concentration drops as the percentage of exhaust gas added to the intake air-fuel mixture increases.



Summary

This paper has outlined the basics of air-fuel control and the issues involved for gas engines.

The ratio of air to fuel has a large effect on combustion and hence on the operation of the engine. Precise air-fuel control ensures that an engine operates at the desired conditions over a wide range of operating conditions.

Control of regulated emissions requires air-fuel control. The choice of alternatives depends on various economic and operational trade-offs. A machine audit process can evaluate these trade-offs where the operational and technical options can be stated in economic terms.

For further information on specific circumstances, contact REM Technology Inc.

Appendix 1

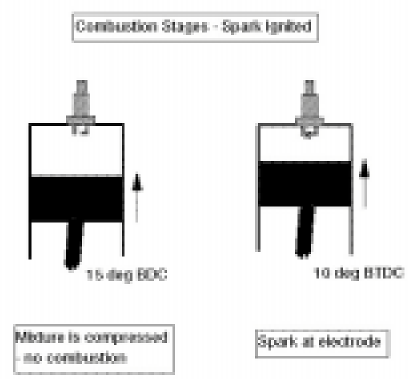
The Combustion Process

Combustion Stages

When the upward motion of the piston compresses the air-fuel mixture, the combustion proceeds in definite stages, which are discussed below for the spark-ignited engines. The stages are similar for both the 2 stroke and 4 stroke engines. The actual timing in degrees depends on a number of factors including engine speed, design, temperature, and type of fuel. The diagrams are schematic and the number of degrees indicated is typical for an engine operating at about 300 RPM.

Normal Combustion

Before the spark plug discharges, the compressed fuel and air mixture becomes hot due to the compression process, but does not reach the temperature of self-ignition. The ideal mixture is homogeneous, but in fact there are variations in the ratio of air and fuel due to incomplete mixing[13].

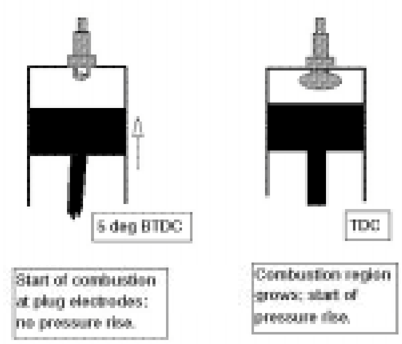


The spark discharge should ignite the fuel air mixture. The likelihood of ignition depends on the size of the spark, the duration, and the composition of the fuel-air mixture in the spark gap. The spark gap dimension is a compromise between the ability of the applied voltage to generate a spark (the larger the gap, the higher the voltage required) and the volume of the mixture available for ignition. The many designs of spark plug electrodes are an attempt to optimize these factors and the electrode life.

Assuming that the combustion has been started, the flame front expands outward at a rate dependent on several factors.

- the air-fuel mixture,
- the cylinder dimensions,
- the temperature,
- the turbulence and
- the type of fuel.

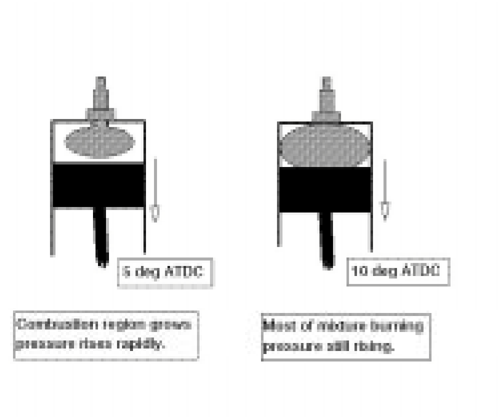
Because of the time required, the spark must occur earlier in high-speed engines



[13] The importance of mixing is demonstrated by reductions in combustion variability with high-pressure gas injection for 2 stroke gas engines.

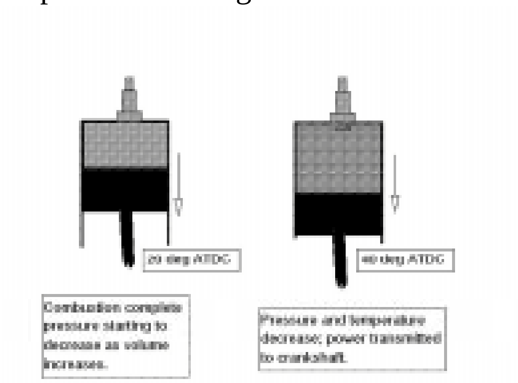
than with lower speed engines.

As the flame front grows, heat is released by the combustion causing the pressure in the cylinder to rise. To extract maximum mechanical energy the majority of the pressure rise should occur after TDC (top dead center).



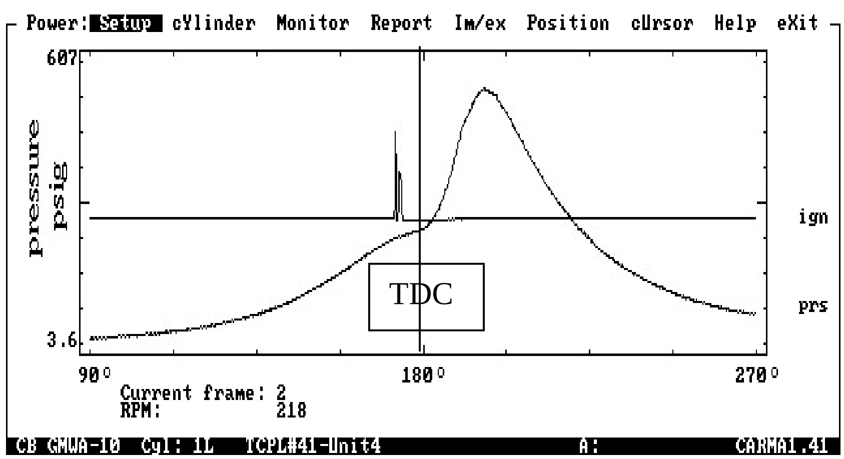
The flame front continues to grow until it reaches all the volume. The chemical reactions of the combustion continue quickly until the fuel or the oxygen is consumed. The heat generated causes the gas temperature and pressure to rise, generating a force on the piston. As the piston moves downward the volume increases which causes the pressure and temperature of the gases to decrease. After the combustion is complete the pressure and volume of the gases in the cylinder follow the relationship PV^n = constant. The force and motion of the piston becomes the power transmitted to the crankshaft.

Power from the expanding mixture is transferred to the crankshaft until the opening of an exhaust valve(s) or the opening of an exhaust port releases the pressure.

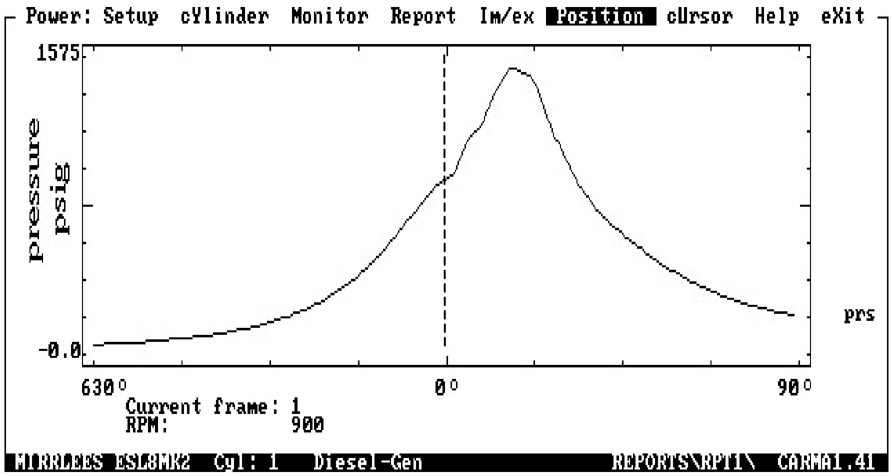


The stages of the combustion process are illustrated by:

1. A pressure curve from a **two stroke**, spark ignited engine operating at about 220 RPM using natural gas (methane) fuel. The pressure is shown from 90 degrees before TDC to 90 degrees after



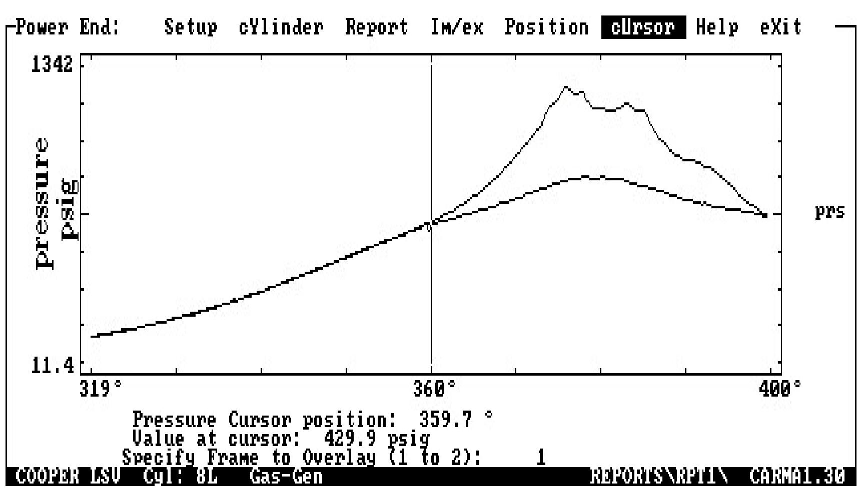
2. A pressure curve from a spark ignited **four-stroke** engine operating at 900 RPM using natural gas fuel. The pressure curve is shown from 90 degrees before TDC to 90 degrees after.



Abnormal combustion

Abnormal combustion results when the combustion process does not occur, is incomplete or occurs at the wrong time. The causes and effects are discussed.

- **Pre-ignition** - Here the combustion starts before the spark discharge. This results in the cylinder pressure rising before TDC and places excessive forces on the engine components.
- **Early Combustion** - Here the combustion occurs too early, resulting in excessive pressure rise before TDC and excessive forces on the engine components. This can be caused by ignition too early, excess fuel, or cylinder overheating.
- **Detonation** - Detonation is the uncontrolled combustion of the mixture. Excess fuel, a high compression pressure and/or temperature, and excessive ignition advance can cause this. Detonation can result in excessive pressures and temperatures. The excessive pressures cause excessive forces on the engine components and the excessive temperatures cause high NOX production and thermal damage to head components. It normally occurs when the mixture ignites at locations other than the spark plug electrodes; either from hot spots (e.g. carbon build-up) or self-ignition from the heating of the mixture in regions the flame front has not yet



reached. An uneven and abnormally high cylinder pressure characterizes detonation. An example of a normal combustion and one with detonation is shown here.

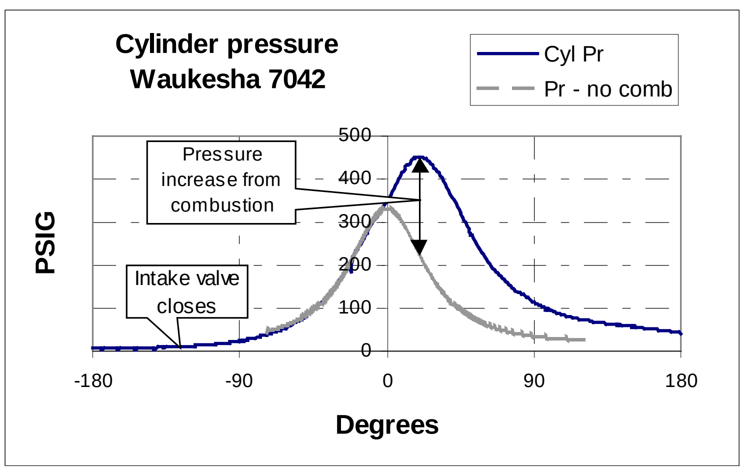
- **Late Combustion** - Here the combustion starts late, resulting in a peak pressure coming well after the optimum crankshaft angle. Causes are late ignition, a lean mixture, a slow burning fuel or poor air and fuel mixing.
- **No Combustion** - Here there is no cylinder pressure rise due to combustion and the pressure rise before TDC is mirrored by the pressure fall after TDC. Causes are lack of ignition, insufficient spark to start combustion, a fuel-air mixture that does not ignite.
- **Poor (Light) Combustion** - The combustion that occurs is either late or incomplete. The possible causes are lack of oxygen for the fuel, too little fuel, poor ignition.

Note that the degree of air-fuel mixing is a serious problem for 2-stroke ported engines where the fuel is admitted by a fuel valve into the cylinder. It has been shown recently that high-pressure fuel injection significantly improves the air-fuel mixing in such engines[14].

In summary, the air fuel ratio and air-fuel mixing have a strong effect on the type of combustion that can occur.

Effect of adding excess air or recirculation of exhaust gases

When excess air or exhaust gases are added to reduce the emissions, changes occur in the combustion conditions. The details of these changes are discussed below using cylinder pressure data from a Waukesha engine.



With a compression ratio of 8.5 to 1, a typical cylinder pressure versus crank rotation is shown below. The pressure from a no combustion event was calculated and is also shown.
The pressure increase after TDC is due to the combustion process. The heat energy generated in the combustion process is the product of the mass of fuel, m_f, times the

[14] "Relative Performance of High-Pressure Fuel Gas Delivery on Large Bore, Two stroke Natural Gas Engines" by G. Hutchison, B. Wilson, S. Hawley, and K. Willet; Gas Machinery Conference Proceedings, October 1997, Austin, Texas

lower heating value (LHV). This energy heats the gases in the combustion chamber. Making the simplifying assumption that the combustion occurs quickly and all the fuel is burned, then for a constant volume adiabatic process

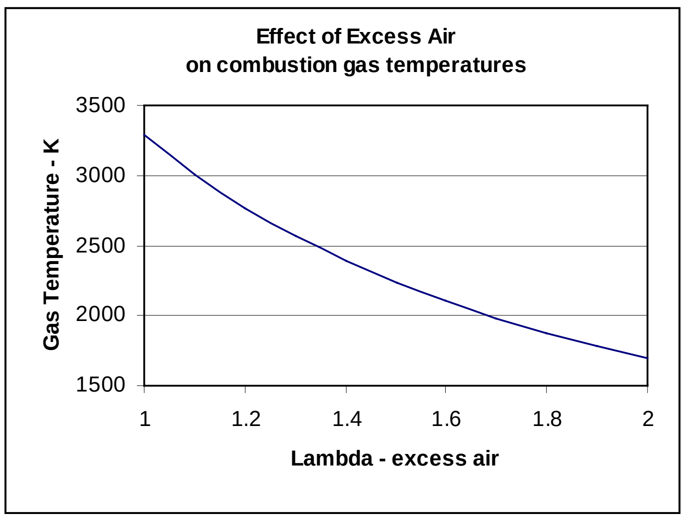
$$(m_a + m_f)c_v (T_b – T_u) = m_f Q_{LHV}$$

where m_a is the mass of the gases in the combustion volume,
c_v is the specific heat of the gases at constant volume (approximately the same for unburned and burned gases),
T_b is the temperature of the gases before combustion,
T_u is the temperature of the gases before combustion,
m_f is the mass of fuel in the combustion chamber and
Q_{LHV} is the lower heating value of the fuel

Taking the stochiometric ratio of air to fuel for methane as 17 and substituting the excess air factor λ, then

$$T_b – T_u = Q_{LHV} / (1 + 17\lambda) c_v$$

This expression shows that the combustion gas temperature decreases as λ, the excess air amount increases. This is shown by the attached graph.



When extra gases are added in the form of excess air (lean burn) or exhaust gases (EGR) the mass of the gases, m, in the combustion chamber increases while the mass of fuel is unchanged. Referring to the expression above, the increase in temperature, $T_b – T_u$, becomes less. For example, with a $\lambda = 1.5$, the temperature increase due to combustion will be 68% of that with a stochiometric mixture.

Adding extra air does, however, increase the pressure of the compressed gases before combustion. This is shown by the table below for a Waukesha engine (bore = 9.375 inches, stroke = 8.5 inches, compression ratio = 8.5).

Maximum compression Pressure at TDC

	Boost Pr – psig					
lambda	0	2	4	6	8	10
1	235	267	299	330	362	394
1.1	258	293	328	364	399	434
1.2	282	320	358	397	435	473
1.3	305	347	388	430	471	513
1.4	329	373	418	463	507	552
1.5	352	400	448	496	544	592
1.6	376	427	478	529	580	631
1.7	399	453	508	562	616	670
1.8	422	480	537	595	652	710
1.9	446	507	567	628	689	749

The temperature increase due to compression depends only on the amount of compression between when the intake valve closes and TDC. The swept volume compression ratio is 8.5. However in this engine the intake valve closes after bottom dead center so the actual compression ratio of the gases is less. Using an intake valve closing crank angle of 60 degrees after bottom dead center (from analyzer measurements) the effective compression ratio (volume when intake valve closes/clearance volume = V_{in}/V_θ) is 7.78. Using the adiabatic expression for temperature

$$T_\theta = T_{in} (V_{in}/ V_\theta)^{\gamma - 1}$$

where T_θ = the absolute temperature at crank angle θ
T_{in} = the absolute temperature of the intake air
V_θ = the volume at crank angle θ,
V_{in} = the volume when the intake valve closes, and
γ = the specific heat ratio for air (= 1.35),

gives a maximum compression temperature at TDC of 390 degrees C for an intake temperature of 50 degrees C, independent of the intake pressure. For every degree increase in intake temperature, the temperature at TDC increases by about 2.0 degrees C.

The net result of adding extra air is a small increase in peak cylinder pressure during combustion. Data[15] taken with an engine analysis unit where the peak pressures were measured for slightly lean (λ=1.1) and for a leaner (λ=1.5) air to fuel ratios showed about a 10% increase in peak pressure for the same engine load.

If there is a need to limit the maximum cylinder pressures due to mechanical or wear constraints, then the use of excess air should be accompanied by a reduction in the maximum rated power. For the example shown, a derating of 10 % would be sufficient.

[15] Peak pressures

Appendix 2   Energy Efficiency

The mechanical power coming from the engine crankshaft (brake HP or brake kW) originates from the heat energy released by the combustion. The indicated power is the net power delivered to the piston by the compression and expansion of gases in the combustion chamber. The indicated power is divided between the brake power delivered at the crankshaft available to the load, the friction losses and power used by auxiliary engine components. The brake power is the difference between the indicated power and In English units

IHP = BHP + FP + AP
Where BHP = brake horsepower,
IHP = indicated HP,
FP = friction power, and
AP = auxiliary power.

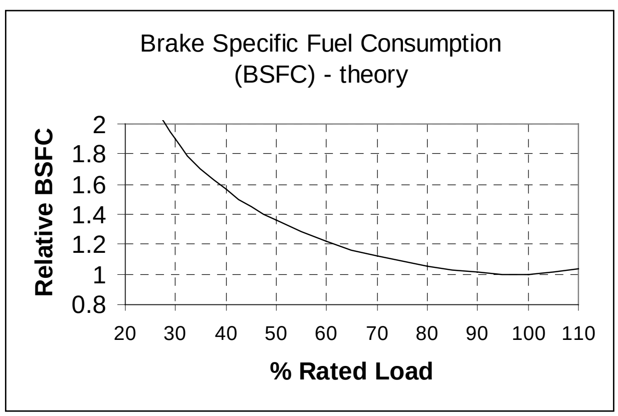
At no load where the BHP = 0, IHP = FP + AP. The frictional power has 2 main components – the rubbing friction, which increases with load, and, in a 4-cycle engine, the pumping losses, which decrease with load. Data for frictional power shows that it is generally constant over a wide range of loads[16], with a shallow minimum at about 70% load. The frictional losses increase with RPM.

The auxiliary power includes power used by the water and oil pumps, ignition drive, cam and valve assemblies, and the fan. All of these, except for the fan, are generally independent of load but increase linearly with RPM. The fan load increases as the cube of the RPM.

The energy efficiency is often expressed as brake specific fuel consumption (BSFC).

BSFC = heat energy in / brake power out.

A lower BSFC means a more efficient engine. The BSFC can vary from infinite (no load) to a best value. Assuming the thermal efficiency (ratio of indicated power to heat energy) is constant versus load up to 90% of rated load, and decreases as the engine approaches full load and overload, the BSFC behaves as shown.



In this example at a 30% load the BSFC increases to almost twice the lowest and best value.

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[16] Internal Combustion Engine Fundamentals – John B. Heywood

Appendix 3

Oxygen available

While the term air-fuel ratio is used, only the oxygen in the air is used for combustion. Normal air is composed of the following gases (molar or volume %):

Nitrogen	78.08%
Oxygen	20.95%
Argon	0.93%
Carbon dioxide*	0.03%
Other gases*	0.0027%
Total	100%
Water vapor**	depends on temperature and relative humidity

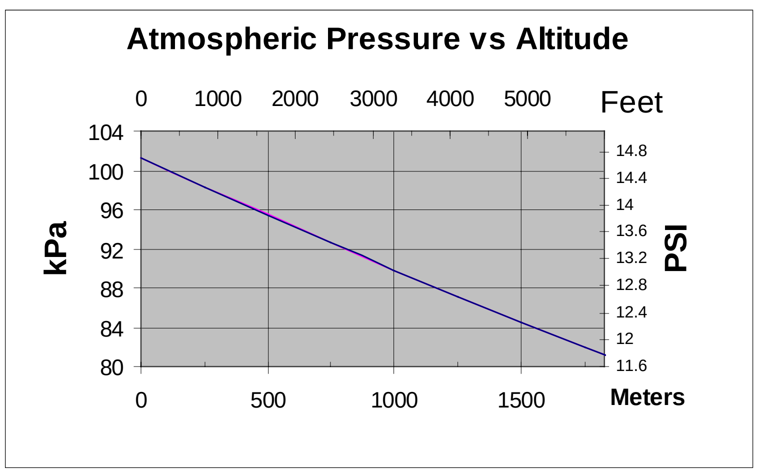
* Note that in enclosed or poorly vented environments these percentages may vary.

** The percentages are quoted for dry air due to the variability of the water vapor content.

A volume, mass or flow of a gas is normally stated at STP - standard pressure and temperature (0 deg. C and 1.00 atmosphere). In the gas industry, the more usual standard conditions are 15 deg. C and 1.00 Atm. = 101.3 kPa (59.6 deg. F and 14.7 PSI).

The actual amount of oxygen available for an engine depends on:

- The atmospheric pressure – depends on the altitude and the weather. The graph shown here shows the variation of air pressure with altitude. Weather changes can alter the atmospheric pressure by a maximum of +/- 5%.

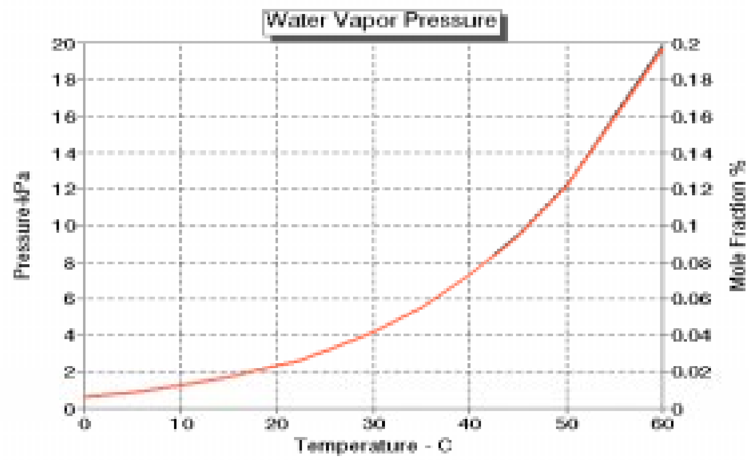


- and the percentage of water vapor present.

The more water vapor present, the less air is available for combustion. The effect of relative humidity follows.

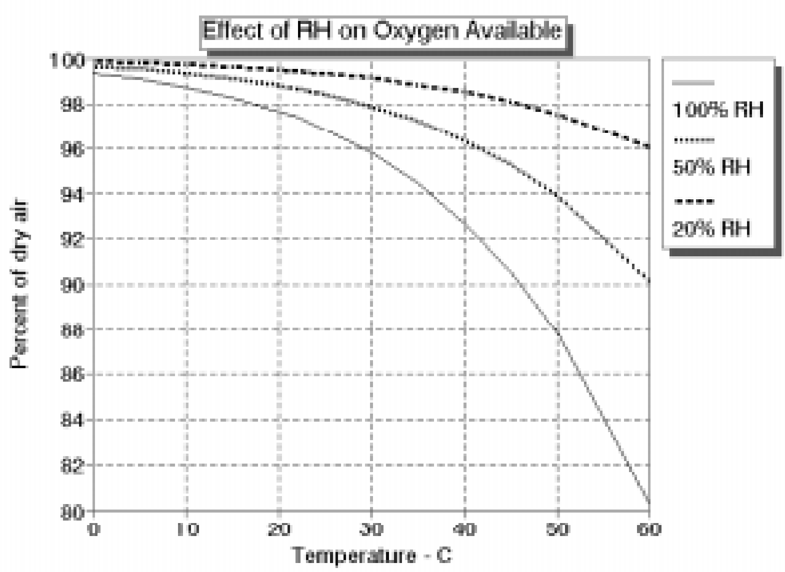
Appendix 3 contd. Effect of Humidity

As the temperature of air increases, it can retain more water vapor. The graph below shows the maximum vapor pressure of water vapor in air; this corresponds to 100% relative humidity (RH). At lower levels of relative humidity, the partial pressure of the water vapor is that shown by the graph times the relative humidity percentage. The partial pressure or the dry air is the total atmospheric pressure minus the partial pressure of the water vapor.



The water vapor in the air displaces the other elements of the air, so at high levels of RH and high temperatures, the amount of oxygen available for combustion decreases.

A graph showing the percentage effect of relative humidity on available oxygen is shown here. Note that the oxygen available for combustion is less when the relative humidity and temperature increase.



To compensate for the decrease in oxygen available, the air-fuel ratio needs to be increased to compensate for the decrease in available oxygen.

Water vapor is known to reduce the combustion velocity and the propensity for detonation. A spark advance can compensate the reduction in combustion velocity.

In practice the effects of humidity are not compensated for in open loop systems. In some closed loop systems where exhaust oxygen is measured, some compensation does occur.