INTERNAL COMBUSTION ENGINES

Fernando Salazar

Department of Aerospace and Mechanical Engineering
University of Notre Dame
Notre Dame, IN 46556

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Chapter 1

Introduction

Internal combustion engines are seen every day in automobiles, trucks, and buses. The name internal combustion refers also to gas turbines except that the name is usually applied to reciprocating internal combustion (I.C.) engines like the ones found in everyday automobiles. There are basically two types of I.C. ignition engines, those which need a spark plug, and those that rely on compression of a fluid. Spark ignition engines take a mixture of fuel and air, compress it, and ignite it using a spark plug. Figure 1.1 shows a piston and some of its basic components. The name 'reciprocating' is given because of the motion that the crank mechanism goes through. The piston-cylinder engine is basically a crank-slider mechanism, where the slider is the piston in this case. The piston is moved up and down by the rotary motion of the two arms or links. The crankshaft rotates which makes the two links rotate. The piston is encapsulated within a combustion chamber. The bore is the diameter of the chamber. The valves on top represent induction and exhaust valves necessary for the intake of an air-fuel mixture and exhaust of chamber residuals. In a spark ignition engine a spark plug is required to transfer an electrical discharge to ignite the mixture. In compression ignition engines the mixture ignites at high temperatures and pressures. The lowest point where the piston reaches is called bottom dead center. The highest point where the piston reaches is called top dead center. The ratio of bottom dead center to top dead center is called the compression ratio. The compression ratio is very important in many aspects of both compression and spark ignition engines, by defining the efficiency of engines.

Compression ignition engines take atmospheric air, compress it to high pressure and temperature, at which time combustion occurs. These engines

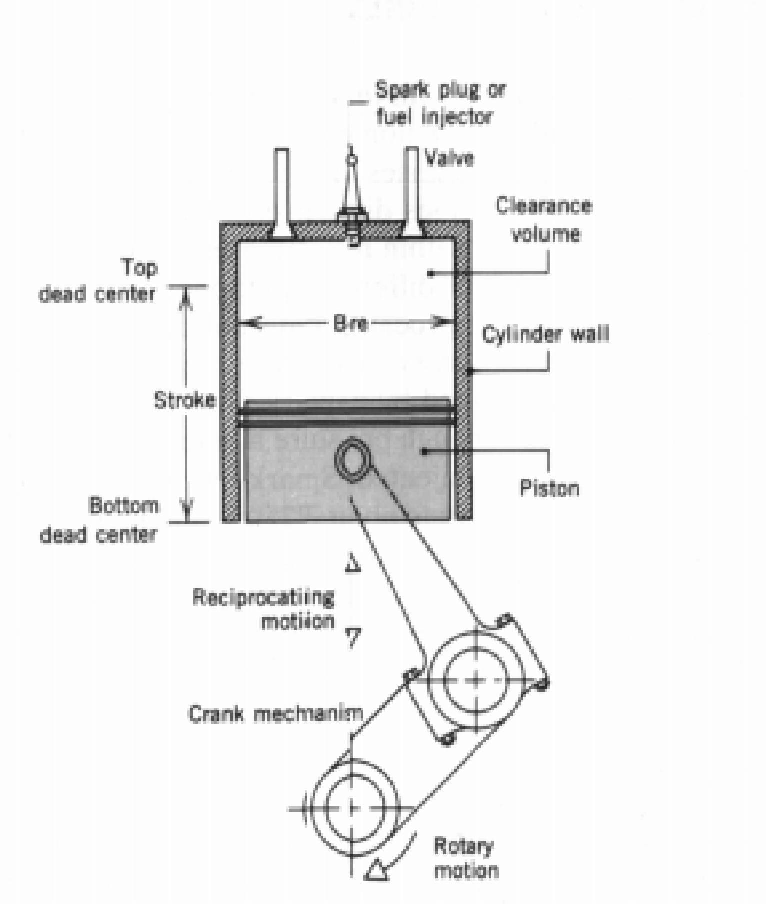

Figure 1.1: Piston

are high in power and fuel economy. Engines are also divided into four stroke and two stroke engines. In four stroke engines the piston accomplishes four distinct strokes for every two revolutions of the crankshaft. In a two stroke engine there are two distinct strokes in one revolution. Figure 1.2 shows a *p*-*v* diagram for the actual process of a four stroke internal conbustion (IC) engine. When the piston starts at bottom dead center (BDC) the intake valve opens. A mixture of fuel and water then is compressed to top dead center (TDC), where the spark plug is used to ignite the mixture. This is known as the compression stroke. After hitting TDC the air and fuel mixture have ignited and combustion occurs. The expansion stroke, or the power stroke, supplies the force necessary to drive the crankshaft. After the power stroke the piston then moves to BDC where the exhaust valve opens. The exhaust stroke is where the exhaust residuals leave the combustion chamber. In order for the exhaust residuals to leave the combustion chamber the pressure needs to be greater than atmospheric. Then the piston preceeds to TDC where the exhaust valve closes. The next stroke is the intake stroke. During the intake stroke the intake valve opens which permits the air and fuel mixture to enter the combustion chamber and repeat the same process.

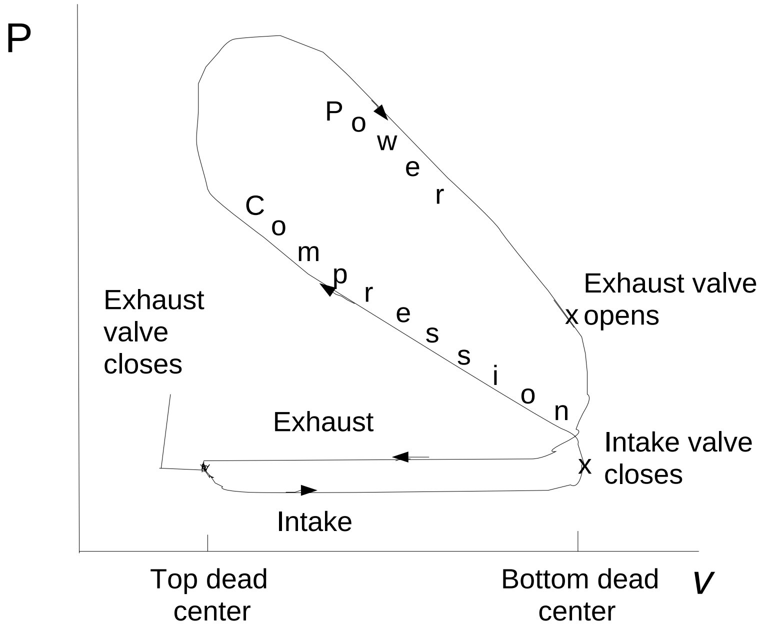

Figure 1.2: Actual cycle

Chapter 2

Ideal Engine Cycles

2.1 Otto Cycle

The Otto cycle is a model of the real cycle that assumes heat addition at top dead center. The Otto cycle consists of four internally reversible cycles, that describe the process of an engine. Figure 2.1, shows the p-v and T-s diagram for the Otto cycle.

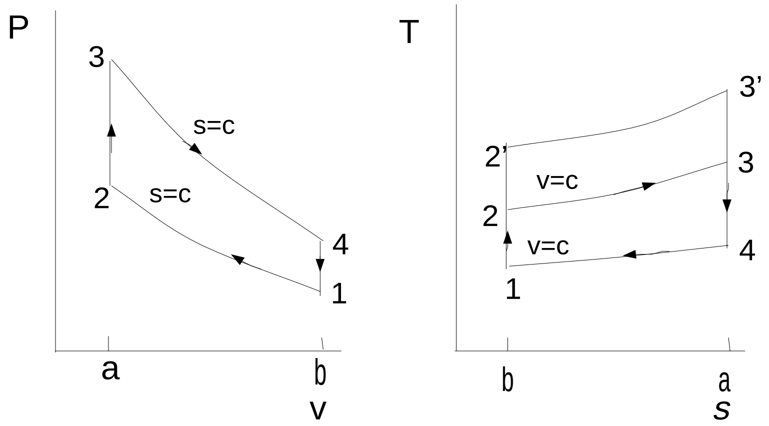

Figure 2.1: Otto cycle

Process 1-2 is an isentropic compression of air and fuel, which occcurs when the piston moves from bottom dead center (BDC) to TDC. In this process air and fuel are compressed and ready for the second process. Process 2-3 is a constant volume heat addition process where the air to fuel mixture is ignited. Process 3-4 is an isentropic expansion, where work is done on the piston, but no heat is added. This process is referred to as the power stroke. The final process, 4-1, is a constant volume heat removal that ends at BDC.

Work and heat are important aspects of engines, that can be represented by Figure 2.1. On the T-s diagram the area 1-4-a-b-1 corresponds to the heat rejected per unit of mass. Area 2-3-a-b-2 corresponds to the heat added per unit of mass. The enclosed area shown represents the net heat added during the process. The area 1-2-a-b-1 in the p-v diagram corresponds to the work input per unit mass and area 3-4-b-a-3 corresponds to work output per unit mass. The net work done is interpreted by the enclosed region in Figure 2.1, in the T-s diagram. In the Otto cycle there are therefore two processes that involve work but no heat transfer and two different processes that involve heat transfer but no work. The energy transfer can be expressed in the following form:

$$\frac{W_{12}}{m} = u_2 - u_1 \tag{2.1}$$

$$\frac{W_{34}}{m} = u_3 - u_4 \tag{2.2}$$

$$\frac{Q_{23}}{m} = u_3 - u_2 \tag{2.3}$$

$$\frac{Q_{41}}{m} = u_4 - u_1 \tag{2.4}$$

The efficiency for the engine can be expressed as the net work done over the heat added. The net work per unit mass is expressed as:

$$\frac{W_{cycle}}{m} = \frac{W_{34}}{m} + \frac{W_{12}}{m} \tag{2.5}$$

Therefore the efficiency for the engine is:

$$\eta = 1 - \frac{u_4 - u_1}{u_3 - u_2} \tag{2.6}$$

The thermal efficiency of the otto cycle increases with increasing compression ratio. When the Otto cycle is analyzed on a cold air standard basis an expression relating the compression ratio, r, temperature and pressure is obtained from isentrropic properties. The compressin ratio is a ratio of the volume displaced by the piston. From figure 2.1 it can be seen that the compression ratio is equal to $\frac{V_1}{V_2}$ and $\frac{V_4}{V_3}$. The expressions for the otto cycle, at constant k, for the isentropic processes are:

$$\frac{T_2}{T_1} = (\frac{V_1}{V_2})^{k-1} = r^{k-1} \tag{2.7}$$

$$\frac{T_4}{T_3} = (\frac{V_3}{V_4})^{k-1} = \frac{1}{r^{k-1}} \tag{2.8}$$

The specific heat ratio is defined as the ratio of specific heat at constant pressure divided by the specific heat at constant volume. Equation 2.9 defines the specific heat ratio.

$$k = \frac{c_p}{c_v} \tag{2.9}$$

From a cold air standard basis equation 2.6 can be rewritten to the following form,

$$\eta = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)} \tag{2.10}$$

or when rearranging to the form,

$$\eta = 1 - \frac{T_1}{T_2}(\frac{T_4/T_1 - 1}{T_3/T_2 - 1}) \tag{2.11}$$

From equations 2.7 and 2.8 it is shown that $\frac{T_4}{T_1} = \frac{T_3}{T_2}$. By using this relationship in equation 2.11 the following expression for the efficiency is obtained.

$$\eta = 1 - \frac{T_1}{T_2} \tag{2.12}$$

Finally using the relationship in equation 2.7 the relationship between the compression ratio, r, and the efficiency is shown by equation 2.13.

$$\eta = 1 - \frac{1}{r^{k-1}} \tag{2.13}$$

where k=1.4 for air at ambient temperature.

2.2 Diesel Cycle

The diesel cycle is similar to the Otto cycle, except that heat addition and rejection occur at different conditions. The diesel cycle is also an ideal cycle meaning that it does not give an exact representation of the actual process. The diesel cycle consists of four internally reversible processes. Process 1-2 is an isentropic compression. Process 2-3 is a constant pressure heat addition. This process makes the first part of the power stroke. Process 3-4 is an isentropic expansion, which makes up the rest of the power stroke. Process 4-1 finishes the cycle with a constant volume heat rejection with the piston at BDC. Figure 2.2 shows the p-v and T-s diagram for the diesel cycle.

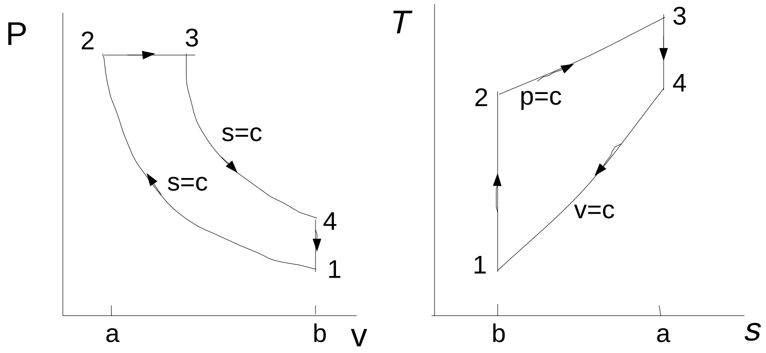

Figure 2.2: Diesel cycle

Since the diesel cycle consists of four internally reversible processes, areas on the p-v and T-s diagram represent work and heat. On the T-s diagram the area 2-3-a-b-2 represents the heat added to the system. Area, on the T-s curve, 1-4-a-b-1 represents the heat rejected. The enclosed area shown represents the net heat added during the process. On the p-v diagram the area 1-2-a-b-1 represent the work input and area 2-3-4-b-a-2 represents the work done as the piston moves to the BDC. The net work done is interpreted by the enclosed region in the p-v diagram. The efficiency for the engine is expressed as the net work done over the heat added. The efficiency is therefore:

$$\eta = \frac{W_{cycle}/m}{Q_{23}/m} = 1 - \frac{u_4 - u_1}{h_3 - h_2} \tag{2.14}$$

The compression ratio of a diesel engine plays a greater significance than in a spark ignition engine. The thermal efficiency of a compression ignition (CI) engine increases as the compression ratio increases. The *cutoff ratio*, r_c, is defined as:

$$r_c = \frac{V_3}{V_2} \tag{2.15}$$

Since V_4=V_1, the volume ratio for the isentropic process is expressed as:

$$\frac{V_4}{V_3} = \frac{V_4}{V_2}\frac{V_2}{V_3} = \frac{V_1}{V_2}\frac{V_1}{V_3} = \frac{r}{r_c} \tag{2.16}$$

Just as in the spark ignition engine a cold air standard analysis, with constant specific heat ratio, for an isentropic process for the diesel cycle results in the the following expressions.

$$\frac{T_2}{T_1} = (\frac{V_1}{V_2})^{k-1} = r^{k-1} \tag{2.17}$$

$$\frac{T_4}{T_3} = (\frac{V_3}{V_4})^{k-1} = \frac{{r_c}^{k-1}}{r} \tag{2.18}$$

Using equation 2.16, the above expressions, and the same approach used for the otto cycle the efficiency for the diesel cycle can be expressed in terms of the compression ratio and cutoff ratio. Equation 2.19 gives the expression for the efficiency.

$$\eta = 1 - \frac{1}{r^{k-1}}[\frac{r_c^k - 1}{k(r_c - 1)}] \tag{2.19}$$

2.3 Dual Cycle

The dual cycle is a better description of the actual pressure variation in the engine. There are several differences though with the Otto and Diesel cycle. In the dual cycle there are five processes. Process 1-2 is an isentropic compression where there is no heat transfer but there is work done. Process 2-3 is a constant volume heat addition process where there is no work done. Process 3-4 is another heat addition process but with constant pressure. This process is also know as the power stroke. Process 4-5 is an isentropic expansion that finishes with the remainder of the power stroke. Finally, process 5-1, is a constant volume heat rejection process. Figure 2.3 shows a p-v and T-s diagram of the dual cycle.

Since the dual cycle is composed of the same processes that the Otto and Diesel cycle the efficiency is equal to the net work done divided by the heat input. The efficiency therefore can be expressed as:

$$\eta = 1 - \frac{Q_{51}}{Q_{23} + Q_{34}} \tag{2.20}$$

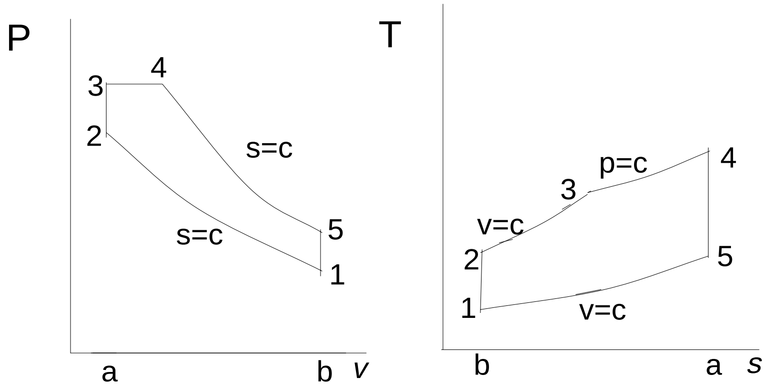

Figure 2.3: Dual cycle

or

$$\eta = 1 - \frac{q_{out}}{q_{in}} \tag{2.21}$$

Chapter 3

Spark Ignition and Compression Ignition Engines

3.1 Spark Ignition Engines

Internal combusiton engines are divided into spark ignition engines and compression ignition engines. Almost all automobiles today use spark ignition engines while trailers and some big trucks use compression ignition engines. The main difference between the two is the way in which the air to fuel mixture is ignited, and the design of the chamber which leads to certain power and efficiency characteristics.

Spark ignition engines use an air to fuel mixture that is compressed at high pressures. At this high pressure the mixture has to be near stoichiometric to be chemically inert and able to ignite. Stoichiometric means that there is a one to one ratio between the air and fuel mixture. So the mixture in order to ignite needs not to be either with too much fuel or too much air but rather have an overall even amount. There are several components to the spark ignition engine. Chamber design, mixture and the injection system are some of the most important aspects of the spark ignition engine. The importance of the chamber design will be discussed. The four basic designs for combustion chambers are as follow:

- the distance travelled by the flame front should be minimised
- the exhaust valve and spark plug should be close together
- there should be sufficient turbulence

- the end gas should be in a cool part of the combustion chamber.

The first design requires that the distance between the end gas and the spark plug be close in order for combustion to progress rapidly. If combustion is sped up then, (i) the engine speed is increased and therefore power output is higher, and (ii) the chain reactions that lead to knock are reduced. From the second design criteria the exhaust valve, since it is very hot, should be as far from the end gas in order to prevent knock or pre-ignition. The third design criteria suggest that there should be enough turbulance in order to "promote rapid combustion", through mixing. (Stone, p.126) Too much turbulance, however, will lead to excessive heat transfer from the chamber and too rapid combustion which causes noise. Turbulance in combustion chambers is generated by squish areas or shrouded inlet valves. The fourth design requires that the end gas be in a cool part of the combustion chamber. The cool part of the combustion chamber forms between the cylinder head and piston. There are many types of designs for combustion chambers. Four common combustion chambers are

- wedge chamber
- hemispherical head
- bowl in piston chamber
- bath-tub head

The wedge design is simple giving good results. In the wedge design the "valve drive train is easy to install, but the inlet and exhaust manifold have to be on the same side of the cylinder head." (Stone, p.127) The second type of combustion chamber is the hemispherical head. The advantage of a hemispherical chamber is its angled valves which are used in high performance engines. This design is expensive with twin overhead camshafts. The design allows for crossflow from inlet to exhaust, with crossflow occuring at the end of the exhaust stroke and at the beginning of the induction stroke while both valves are open. The third combustion chamber is a cheaper design that has good performance. The last combustion chamber design has a "compact combustion chamber that might be expected to give economical performance." (Stone, p.128)

The process by which the air to fuel mixture is prepared and put in the combustion chamber is through carburetors and fuel injectors. Spark plugs are part of all spark ignition engines. In order to start one of these engines a spark has to ignite a mixture into a flame. The way in which this spark is first initiated is through the car battery and a circuit directly leading to the spark plug. The battery supplies the electic current to initiate a spark in the spark plug. The Spark then ignites the air and fuel mixture. The type of fuel injectors used divide into multi-point and single-point injection. Carburettors divide into fixed and variable jet carburettors. The air and fuel mixture is analysed as either a lean or rich mixture depending on the content of fuel. A stoichiometric mixture is one in which there is a perfect ratio of air and fual molecules. A lean mixture would be defficient in fuel where a rich one would be saturated with fuel. To achieve economic status and yet receive the maximum power the engine would have to use a lean mixture and a rich one at full throttle. When the throttle is fully opened and a lean mixture is used the power output is economical because of the weak fuel. When the throttle is opened the combustion chamber needs the air to fuel mixture. Since a stream of air is generated extra fuel is needed to compensate for the insufficient flow of fuel. In order to obtain maximum power a rich mixture is needed. For good fuel economy all the fuel should be burnt and the "quench area where the flame is extinguished should be minimised." (Stone, p.126)

3.2 Compression Ignition Engines

Compression ignition engines differ from spark ignition engines in a variety of ways but the most obvious one being the way in which the air and fuel mixture is ignited. As stated above a spark plug is used to create a spark in the combustion chamber which ignites the mixture. In a compression ignition engine there is no spark to create the flame but rather high temperatures and pressures in the combustion chamber cause a flame to initiate at different sites of the combustion chamber. Combustion increases with increasing pressure and temperature. Compression ignition engines are divided into direct and indirect ignition engines. Diesel engines require fuel injection systems to inject fuel into the combustion chamber. Fuel injection systems are either linear or rotary. Rotary fuel injectors are used in indirect ignition engines because of low pressures.

Direct injection engines use pressures of up to 1000 bars to inject fuel into the combustion chamber. High pressure is needed because the heat addition process takes place at a compressed state, so in order for the fuel to inject well the pressure has to be greater than the one that has been accumulated through compression. There are several engineered direct injection combustion chambers. This goes to show that the actual design of compression ignition engines is not as critical as the design considered for spark ignition engines. Swirl is the most important air motion in the Diesel engine. The importance of swirl is that it mixes the air and fuel so that combustion can increase. The direction of swirl is at a downward angle so that proper mixing can take place. The compression ratio for direct ignition engines is usually between 12 : 1 and 16 : 1.

Indirect ignition engines have a pre-combustion chamber where the air to fuel mixture is first stored. The purpose of the separate chamber is to speed up the combustion process in order to increase the engine output by increasing the engine speed. The two basic combustion systems are the swirl and pre-combustion chambers. Pre-combustion chambers depend on turbulance to increase the combustion speed and swirl chambers depend on the fluid motion to raise combustion speed. In divided chambers the pressure required is not as high as the pressure required for direct ignition engines. The pressure required for both type of divided chambers is only about 300 bars.

With all Diesel engines there is some type of aid to help combustion. Electrical components aid in the initiation of the combustion process by using an electrical source, such as a car battery, to heat themselves and transfer the energy to the mixture for combustion. Cold starting a Diesel engine is very difficult without the use of these tabs that conduct an electric current. When electrical elements heat up and the air to fuel mixture comes in close contact with the tab then a combustion occurs. The Diesel engine has high thermal efficiencies, and therefore low fuel consumption. The disadvantage of Diesel engines is their low power output, relative to their weight, as compared with spark ignition engines.

Chapter 4

Two Stroke Engine

The fundamental difference between the four stroke engine and the two stroke engine is the way in which the induction and exhaust process takes place. In the four stroke engine there are separate strokes for the induction and exhaust processes. In the two stroke engine however, both the induction and exhaust processes take place with the same stroke. The process that involves both induction and exhaust is called scavenging, or simply a gas exchange process.

The two stroke engine can be either made into a spark ignition or compression ignition engine. The smallest engines used in two stroke engines are compression ignition engines. The engines are usually used in models and their power output does not exceed 100 W. The other type of two stroke engine with power output of up to 100 kW is spark ignition engine. Some of these engines output high power relative to their weight and bulk. Some applications of these engines are in motorcycles, chain saws and small generators.

A two stroke engine is seen in Figure 4.1. Some of the important parts of this engine are the exhaust, inlet, and crankcase port, and spark plug. The deflector is also an important design of the engine. The inlet port is where the charge is drawn from. The charge is a mixture of mainly air and fuel but may contain some exhaust. The exhaust port is where the exhaust leaves the piston, and the crankcase port provides the mixture. The combustion process for the two stroke engine goes through various processes. Following are the steps for combustion:

1) At 60° before hitting BDC the piston uncovers the exhaust port (EO), and the exhaust leaves the cylinder chamber while attaining atmospheric

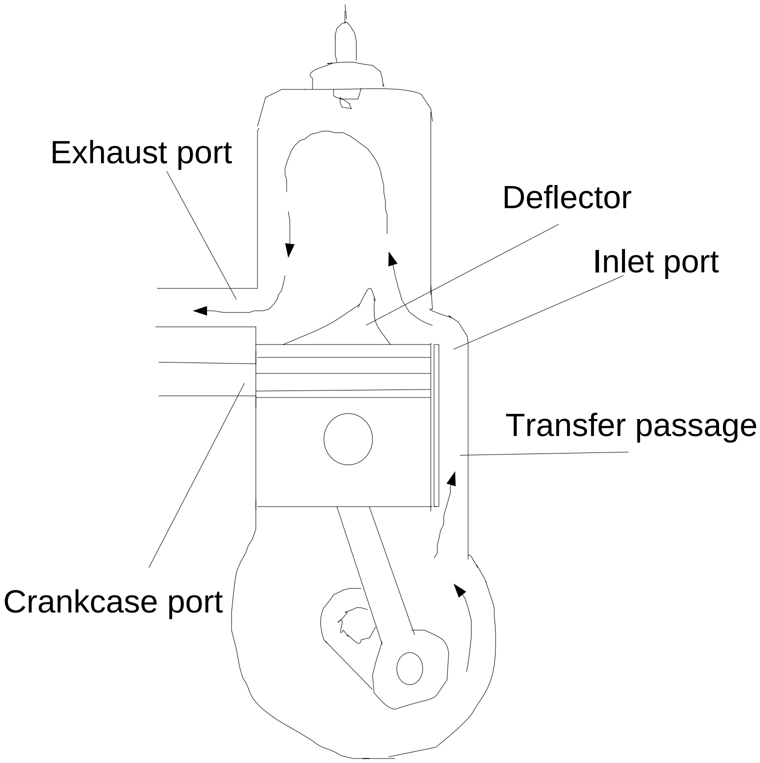

Figure 4.1: Two stroke engine

pressure. This is the end of the power stroke.

2) At 5-10° later the inlet port (IO) will open and the charge that was compressed by the crankcase will flow into the main chamber and mix with some exhaust residual. Some charge will leave the exhaust port. The deflector will aid in a way that it will divert the cross flow of charge from the inlet port into the exhaust port.

3) At about 55° after BDC, with the piston moving up, the inlet port will now close (IC). There will be some backflow of charge from the inlet port into the crankcase.

4) At 60° after BDC the exhaust port will close (EC) and the piston will now compress the charge through its upward movement.

5) At 60° before TDC the crankcase port will open (CO) and allow charge to flow into the crankcase. The charge will flow into the crankcase since the pressure in the crankcase is below the ambient pressure.

6) When the piston is within 10-40° before TDC the charge will be com-

pressed enough to be at a high temperature. Then combustion will follow with flame initiation from the spark plug. In this process work is done by the engine on the air and fuel mixture. The power stroke starts when the piston hits TDC and continous until the exhaust port opens in step (1). Figure 4.2 represents the steps talked above for the two stroke engine.

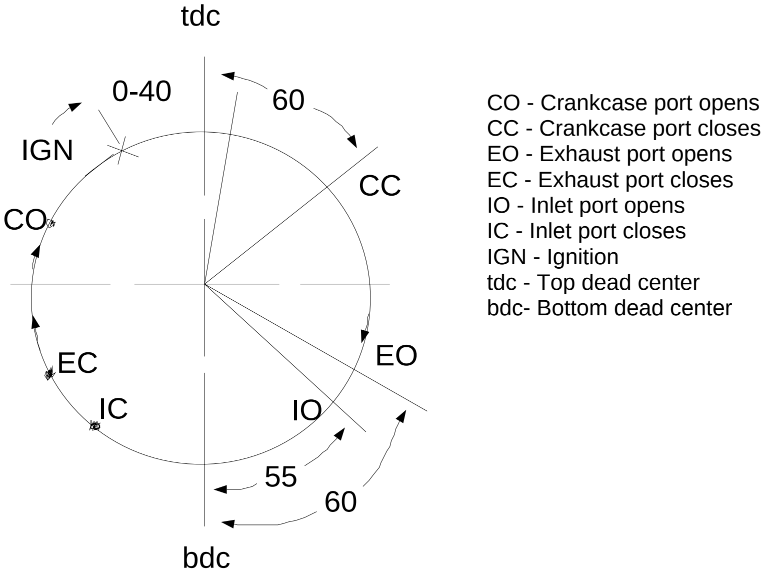

Figure 4.2: Timing diagram

Two undesirable aspects of the two stroke engine is the interfacing between the charge and the exhaust gas, and the passage of the charge into the exhaust system. Scavenging is the simultaneous process by which the charge enters the cylinder chamber and the exhaust leaves the chamber. Even though both gases will always interface with each other, there are ways of designing the piston and cylinder in order to minimize the cross flow from the inlet port to the exhaust port. Figure 4.3 shows various scavenging systems that will decrease the cross flow.

In the cross scavenge design the charge is diverted upwards by a deflector on the piston face. The second scavenge system is the loop scavenging design. This design has the inlet port just below the exhaust port, therefore creating a circular type gas flow throughout the cylinder. A modified design to the loope scavenging design is the Schnurle loop scaveng design. In this arrangement the ”inlet ports are located symmetrically around the exhaust ports” (Stone,

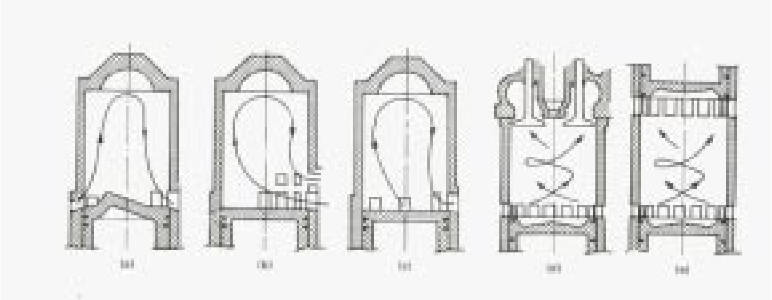

Figure 4.3: Scavenging designs

p.279). This arrangement will create a ‘U’ figure type flow in the cylinder. The last two arrangements are mainly for diesel engines. The inlet ports are at the bottom and the exhaust ports at top. The piston design is such that the formation of swirl is developed.

Chapter 5

Induction and Exhaust

In internal combustion engines the induction and exhaust processes give importance to the performance and efficiency of the engine. In the two stroke engine the flow is regulated by the piston covering and uncovering ports, but in the four stroke engine the induction and exhaust processes are controlled through valves. The four types of valves used are poppet, sleeve, rotary, and disc valves.

5.1 Valves

The most commonly used valve is the poppet valve. The poppet valve has a straight cylinder rod and its end has the shape of a mushroom. The advantages of the poppet valve are that it is cheap, it has good flow properties, good seating, it is easy to lubricate, and it has good heat transfer to the cylinder head. Rotary and disc valves are sometimes used, but contrary to poppet valves, they have heat transfer, lubrication, and clearance problems. The other type of valve is the sleeve valve. The sleeve valve has some advantages over the poppet valve, but its disadvantages discontinued the use of it. The use of sleeve valves was best suited for aerospace engines before the introduction of the gas turbine engine. The advantages of sleeve valves were that they eliminated the “hot spot associated with the poppet valve.” (Stone, p.232) Other advantages were that it produced higher outputs and higher efficiencies due to a higher compression ratio, which was due to the use of low octane fuel. The disadvantages of the sleeve valve were the cost and difficulty to manufacture, the lubrication and friction between the cylinder wall and sleeve valve, and the heat transfer from the piston through the sleeve and oil film to the cylinder.

A camshaft is used in the mechanism that operates the valves. Engines that use overhead poppet valves (ohv) use a camshaft that "is either mounted in the cylinder block, or in the cylinder head." (Stone, p.233) Overhead camshafts (ohc) use chain or toothed belts to provide its drive. Gear drives are also possible to use but they would be expensive, noisy, and cumbersome. The best belts are toothed belts because the rubber damps out torsional vibrations. The other type of valve system is the twin or double overhead camshaft. This mechanism is used when the need for two inlet and two exhaust valves are needed. High performance spark ignition engines or large compression ignition engines use the double overhead camshaft mechanism. One camshaft operates the inlet valves and the other camshaft operates the exhaust valves. The disadvantage of having two camshafts operate four valves is the cost of having a second camshaft, the more involved machining, and the difficulty of providing an extra drive. The British Leyland four valve pent-roof mechanism got rid of the problem of having two camshafts operating four valves by introducing a single camshaft operating all four valves with the aid of a rocker arm. Figure 5.1 shows the mechanism proposed by Leyland.

In this mechanism the camshaft comes in direct contact with the inlet valves and through a rocker the exhaust valves. The advantages of four valves per combustion chamber result in "larger valve throat areas for gas flow, smaller valve forces, and larger valve seat area." (Stone, p.236) The reason for having smaller valve forces is because a lighter valve with a lighter spring will reduce the hammering effect on the valve seat.

5.2 Valve Timing

Valve timing is characterized by the camshaft and valve mechanism. There are two timing processes. Both of these processes involve inlet flow and exhaust interaction, but the difference between the two is the time of interaction. In compression ignition engines and conventional spark igniton engines the valve overlap is only 15° of the crank angle but in high performance spark ignition engines the valve overlap is 65^o of the crank angle. Figure 5.2 shows the valve timing diagrams for small valve overlap and large valve overlap for the four stroke engine. In diagram (a), which corresponds to the small valve overlap, the inlet valve opens 5° before top dead center (btdc) and does not

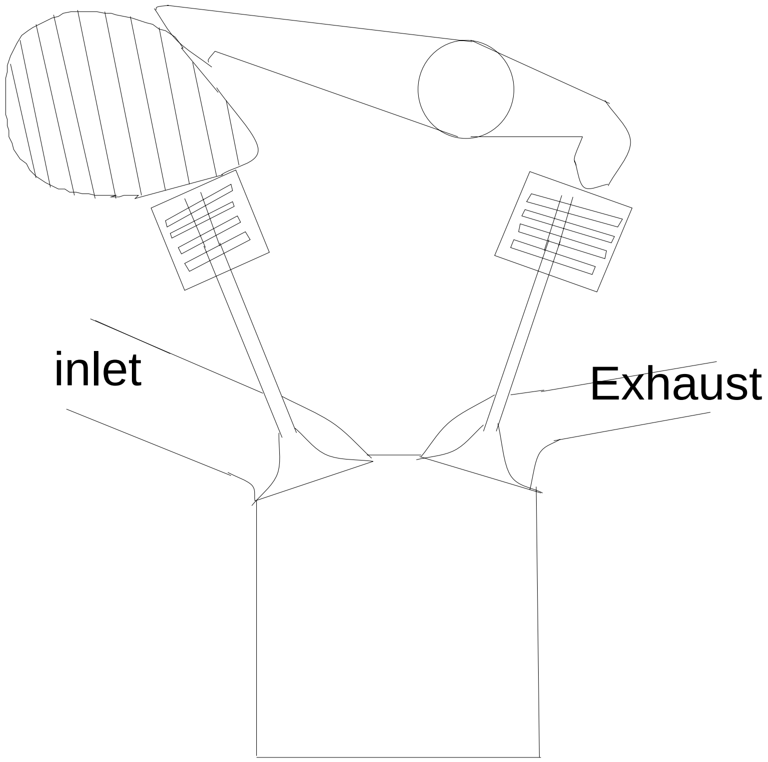

Figure 5.1: Poppet valve

close until 45° after bottom dead center (abdc). The reason for the rapid closure *abdc* is because the pressure needs to increase rapidly. If the inlet valve closed later in the compression stroke then there would not be enough pressure build up. Also after the piston moves to BDC following the induction stroke the the piston will start the compression stroke. If the inlet valve is open for to long of a time then the air and fuel mixture will be pushed out by the piston face. In compression ignition engines this will make cold starting the engine difficult. The compression stroke will then initiate until hitting top dead center, which will then lead to the power stroke. At about 40° before bottom dead center (bbdc) the exhaust valve will open, so that the combustion products have enough time to leave the chamber. By having the exhaust valve open 40° *bbdc*, the problem arises of wether or not opening the valve *bbdc* will cut away from the power stoke and therefore cause the engine to be less efficient. At 40° *bbdc* the power stroke is almost finished and as a matter of fact only 12 % of the stroke is lost. Another reason for opening the exhaust valve 40° *bbdc* is because the valve is not fully opened until 120° after starting to open. The cycle repeats again when the piston is 5° *btdc*.

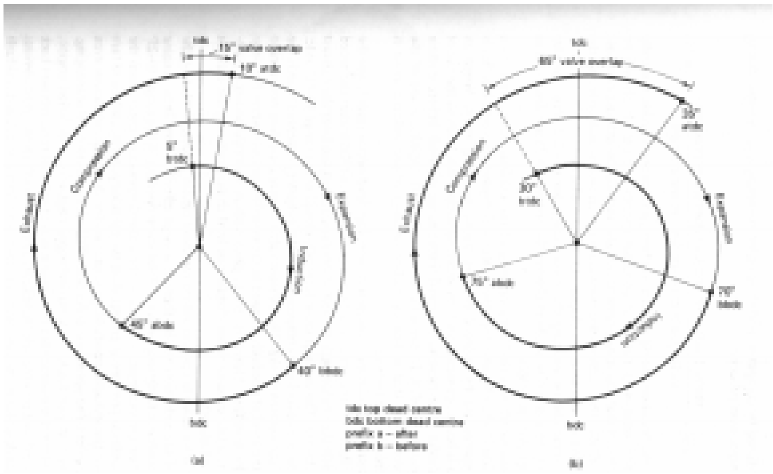

Figure 5.2: Overlap diagram for small (a) and large (b) valve overlap

The valve overlap of the high performance spark ignition engine is best described by diagram (b). The disadvantages of having a large overlap is due to the mixture caused by the exhaust and air and fuel mixture. Since there is a large valve overlap some unburnt fuel will cross flow from the inlet manifold into the exhaust valve and therefore the efficiency will be reduced.

Exhaust residuals are a concern in large valve overlap engines. An increase level of residual in the combustion chamber has a significant effect on the combustion process. With a high level of exhaust residual the efficiency of the engine will decrease since the engine will be doing work on both burned and unburned fuel instead of doing on as much unburnt fuel instead of doing work on as much unburnt fuel as possible. The level of residuals increase with a)decreasing absolute inlet manifold pressure, b)reducing compression ratio, c)increasing valve overlap, d)decreasing speed, and e)increasing exhaust back pressure.

Chapter 6

Thermochemistry and Fuels

6.1 Combustion Reactions

Internal combustion engines obtain their energy from the combustion of hydrocarbon fuel with air. The chemical energy stored in the fuel is converted to energy that the engine can use in the through hot gases within the chamber. The combustion process involves the chemical reaction of hydrocarbon fuel with oxygen to produce water vapor and CO_2. The maximum amount of chemical energy from the hydrocarbon fuel is when it reacts with stoichiometric oxygen. The meaning of *stoichiometric* oxygen is defined as the amount of oxygen that is needed to convert all of the carbon in the fuel to CO_2 and all of the hydrogen to H_2O. The simplest chemical reaction using the simplest hydrocarbon with stoichiometric oxygen is:

$$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O \tag{6.1}$$

For this reaction to be complete it would take two moles of oxygen to react with one mole of methane to produce one mole of carbon dioxide and two moles of water vapor. The hydrocarbon fuel used in engines is not a simple fuel like methane but rather consists of isooctane and various additives. The chemical reaction involving isooctane and oxygen is:

$$C_8H_{18} + 12.5O_2 - > 8CO_2 + 9H_2O \tag{6.2}$$

The above two chemical reactions involve the reaction of a hydrocarbon with oxygen. Since it would be extremely expensive to use pure oxygen the atmosphere is used as a rich source of oxygen. The hydrocarbon reacts with air which is composed of many substances. Nitrogen and oxygen are the two most found substances in air with a nitrogen composition of 78%, by mole, and oxygen composition of 21%. The stoichiometric combustion of isooctane with air is then:

$$C_8H_{18} + 12.5O_2 + 12.5(3.76)N_2 -> 8CO_2 + 9H_2O + 12.5(3.76)N_2 \quad (6.3)$$

Combustion can occur with an either lean or rich mixture. If the mixture is for example 150% of stoichiometric then there will be an excess amount of air and the products will involve excess oxygen. This is called a lean mixture since there is a deficiency is fuel. If on the other hand the mixture is 80% of stoichiometric then there will be excess fuel and carbon monoxide (CO) will be in the end product. This is a rich mixture since the misture has excess of fuel. Carbon monoxide is a colorless, odorless, poisonous gas which can be further burned to form CO_2. If there is a further deficiency in oxygen then more CO will go into the atmosphere as pollution.

6.2 Hydrocarbon Fuels

In SI engines the fuel used is gasoline. Gasoline on the other hand is composed of a mixture of many hydrocarbon fuels, which all come from crude oil. Crude oil is composed mainly of 85% carbon and 13% hydrogen. The carbon and hydrogen molecules combine to form thousands of hydrocarbon mixtures. Crude oil is seperated into components by cracking and or distillation using thermal or catalytic methods at oil refineries. The process of cracking involves the breaking up of large molecular components into smaller molecular components, which are then used for processing. It is important to break up the large strands because the smaller the molecular weight of the component the lower the boiling temperature will be. Fuels need to have components with low boiling points so that they can be readily vaporized. Crude oil in the U.S. is basically divided into Pennsylvania and Western crude. Pennsylvania crude has a high concentration of paraffins and Western crude has a high concentration of asphalt. Figure 1 shows the temperature vs. percent of fuel evaporated.

This figure shows the importance of having fuel consisting of a mixture of hydrocarbons. Figure 1 represents the typical gasoline mixture for SI engines. The mixture is composed of low and high molecular weight compounds. The low molecular weight compounds aid in the cold starting of the

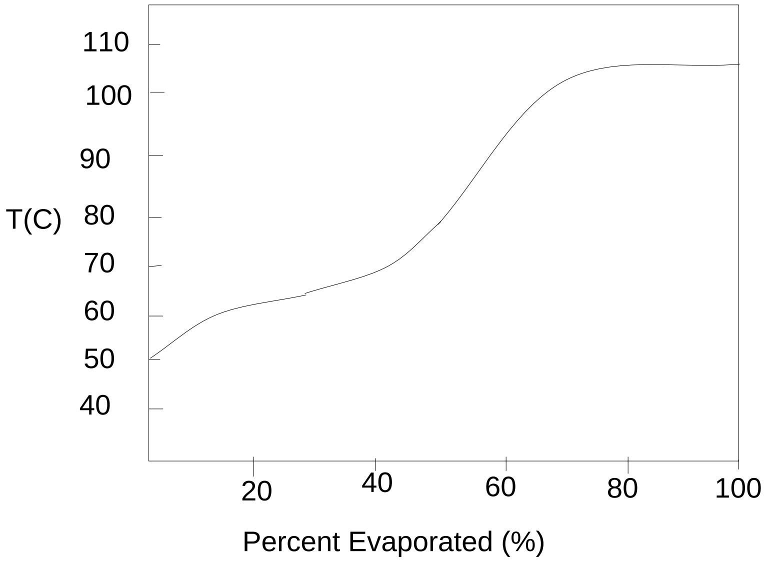

Figure 6.1: Temperature vs. Evaporation

engine while the high molecular weight compounds increase the efficiency by not vaporizing until late into the compression stroke. The low molecular weight compound is defined as *front-end volatility* while *high-end volatility* corresponds to high molecular weight compounds. One of the problems with front-end volatility is that the efficiency of the engine will be reduced if fuel vapor replaces air too early in the intake system. Another problem is *vapor lock*. Vapor lock occurs when fuel vaporizes in the fuel supply lines or carburettor. If this happens fuel is cut off and the engine will stop. One problem with high-end volatility is when too much fuel is supplied to the engine and not all the fuel is burnt during combustion. The unburnt fuel will then end up as pollution in the environment. Following are some basic hydrocarbon components.

6.2.1 Paraffins

The paraffin family, also called alkanes, are molecules with a carbon-hydrogen combination of C_nH_{2n+2}. The most stable paraffin is methane(CH_4), which is the main component of natural gas. The molecule structure is shown in Figure 6.2.

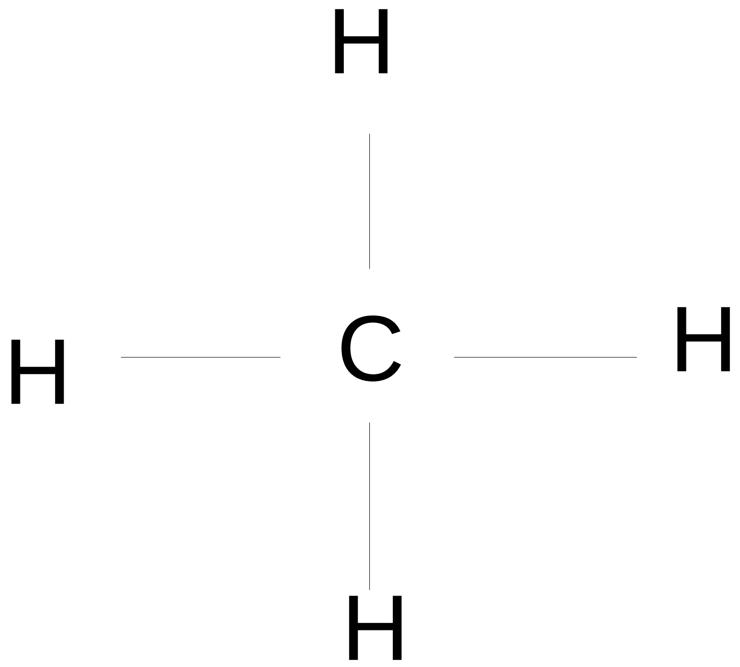

Figure 6.2: Methane chemical composition

Other paraffins include propane(C_3H_8), butane(C_4H_{10}), and isobutane(C_4H_{10}). The difference between the formulas for butane and isobutane is the chemical structure. Isooctane best matches the structure and thermodynamic properties of gasoline. The structure for isooctane(C_8H_{18}) is shown in Figure 6.3.

6.2.2 Olefins

The olefin family is made up of one double carbon-carbon bond. The structure of olefins is of the form C_nH_{2n}. Some olefins are ethene (C_2H_4), butene-1 (C_4H_8), and butene-2 (C_4H_8). Once again butene-1 and butene-2 have the same chemical formula but different chemical structure. These are called isomers.

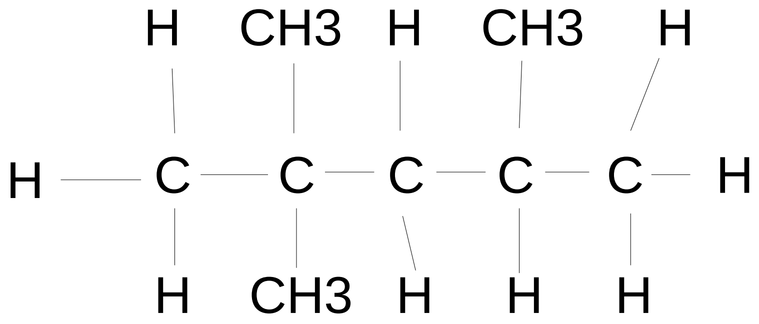

Figure 6.3: Isooctane chemical composition

6.2.3 Others

Cycloparaffins have a single-bond ring and a chemical formula of C_nH_{2n}. Some cycloparaffins are cyclobutane (C_4H_8) and cyclopentane (C_5H_{10}). Cycloparaffins are good gasoline components. Aromatics have double carbon-carbon bonds, and a chemical structure of C_nH_{2n-6}. The basic structure is the benzene ring. Aromatics usually are good gasoline components. They have high densities in the liquid state and therefore have high energy content per unit volume. Some other hydrocarbon components are alcohols, diolefins, and acetylene.

6.3 Octane Number

The octane number of a fuel describes how well it will or will not self ignite. The numerical scale is set by testing fuels. The fuel at question is compared to other fuels that have set standards. One fuel that is used for the test is isooctane (2,2,4 trimethylpentane), which is given the octane number (ON) of 100. The other fuel used for the test is n-heptane, which is given the ON of 0. The higher the octane number of the fuel the less likely it will self-ignite. In SI engines self-ignition will occur when the fuel ignites before the use of the spark due to high tempeartures. When self-ignition occurs in SI engines pressure pulses are genereted. This high pressure causes damage to the engine. This activity of self-ignition is called *knock*. Engines with low compression ratios can use low octane fuels since the temperatures and pressures are lower. High compression engines must use high octane fuel to avoid self-ignition and knock. Figure 4 shows the relationship between the compression ratio and the octane number.

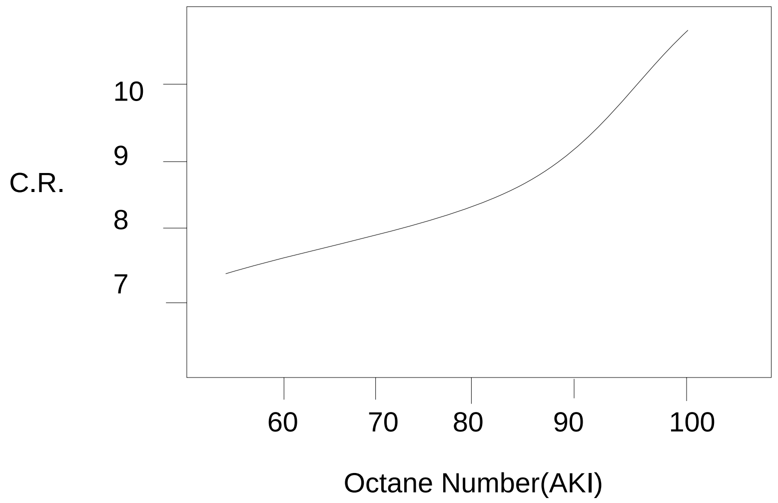

Figure 6.4: Compression ratio vs. Octane number

Fuels that were used earlier had low octane numbers so therefore engines with low compression ratios were used. As technology advanced the engine design advanced. Engines were designed with higher compression ratios so higher pressures and temperatures were attained. fuel had to be manufactured to have higher octane numbers. The structure of the fuel depicts the value of the octane number. For example hydrocarbon components that have long chains have low ON. On the other hand components with more side chains have higher ON. Also fuel components with ring molecules have high ON. One additive that was used to raise the ON of fuels was TEL, $(C_2H_5)_4Pb$. A few milliliters of TEL into several liters of fuel and the ON would rise several points in a very predictable manner. When TEL was first used it was an additive that was manually put into the fuel tank at the gas station. The turbulance created by the pouring was enough to create the mixture. Handling of the TEL by people at gas stations was harmful due to the toxic vapors and the harm that TEL could do the skin. Because of its harm to people, TEL was blended into the gasoline at refineries. This however created a need for more pumps and storage facilities for the new gasoline which was now divided into high-octane and low-octane gasoline. Figure 5 shows the relationship between the ON and the TEL added to the the gasoline.

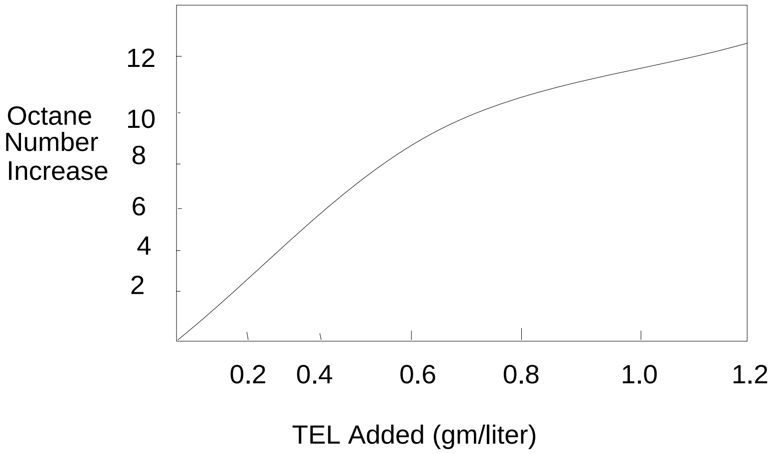

Figure 6.5: Octane number vs. TEL

The problem with TEL is the lead content that ends up in the exhaust. Lead is a very toxic engine emission, and its pollution to the atmosphere ended in the early 1990's. The elimination of leaded gasoline created problems for older cars and other older engines. When lead is introduced into combustion one of the results is lead deposited into the walls of the combustion chamber. This lead reacts with the hot walls and forms a very hard surface. When older engines were manufactured they were engineered to have softer steels in the walls, heads, and valve seats. When the engines were operated using leaded fuels the idea was that these parts would become heat treated and hardened during use. When these engines are operated with unleaded fuel the hardening process is not there and the parts wear through use. There are now TEL substitutes for older vehicles such as alcohols and organomanganese compounds.

6.4 Cetane Number

Diesel fuels are usually characterized by their molecular weight. There are low and high molecular weight fuels, each with different characteristics. Usually the greater the refining done on the fuel the less viscous, lower molecular weight, and higher cost of the fuel. The less refining done on the diesel fuel the more viscous, higher molecular weight, and lower cost of the fuel. Numerical scales exist that denote whether a diesel fuel is high or low in molecular weight. The scale ranges from one (1) to five (5) or six (6), with subcategories using alphabetical letters (e.g. 3B, 2D). The lower the number the lower the molecualr weight of the diesel fuel. Fuels with lower numbers are typically used in CI engines, while the high numbered diesel fuels are used in large, massive heating units. Diesel fuels can be divided into two extreme categories; light and heavy diesel fuel. Light diesel fuel has a molecular weight of about 170 and is approximated by the chemical formula $C_{12.3}H_{22.2}$. Heavy diesel fuel has a molecular weight of about 200 with approximately a chemical formula of $C_{14.6}H_{24.8}$.

In CI engines combustion starts with the self ignition of the air and fuel mixture. There are different fuels which have different ignition characteristics. Ignition delay is a property of CI engines that is dependant on the fuel used. The cetane number (CN) is a quantifiable number that gives a fuel the property of wether it will self ignite early or late. The higher the CN the shorter the ignition delay. On the other hand the lower the CN the longer the ignition delay. The CN ratings are established through testing. The two fuels used for the test are n-cetane (hexadecane), $C_{16}H_{34}$, and heptamethylnonane (HMN), $C_{12}H_{34}$. The n-cetane is given the cetane number of 100, while HMN is given the number of 15. The CN is then determined using equation 6.4.

$$CN = (\text{percent of n-cetane}) + (0.15)(\text{percent of HMN}) \tag{6.4}$$

The degree of CN of a fuel gives certain characteristics to the engine. Normal cetane numbers range on the order of 40 to 60. For a given engine, if the CN is too low then ignition delay will be too long. If ignition delay is prolongated then extra fuel "will be injected into th ecylinder" before the first fuel is ignited "causing a very large, fast pressure rise at the start of combustion." (Pulkrabek, 149) This fast pressure rise will cause low thermal efficiency. If the CN of the fuel is too high then combustion will start too soon in the compression stroke. Early combustion will cause a pressure rise before top dead center, and more work will be required.

Chapter 7

Combustion

7.1 Combustion In SI Engines

Combustion in SI engines is divided into three categories. Ignition and flame development is the first phase of combustion where only about 5% of the air-fuel mixture is consumed. During flame development combustion has barely started and there is very little pressure rise, so there is no significant work done. The second phase consists of the propagation of the flame. This phase consumes about 80-90% of the air-fuel mixture. During this phase there is significant pressure rise, which provides the force that produces the work in the expansion stroke. The third and final phase of the combustion process is the flame termination. This phase consumes only about 5% of the air-fuel mixture. During this phase the pressure quickly decreases and combustion ends. Figure 7.1 shows the pressure as a function of the crank angle. The maximum pressure is reached after TDC which supplies the force necessary in the expansion stroke to provide useful work.

7.1.1 Ignition and Flame Development

Ignition of the air-fuel mixture is initiated by an electrical discharge across the electrodes of a spark plug. This explosion usually occurs 10-30° degrees before TDC. When this ignition takes place, combustion reactions follow in an outward direction. The voltage necessary to cause the plug to spark is 25,000-40,000 volts with a maximum current o fabout 200 amps. This amount of energy dissipation only last about 10 nsec since the spark of the plug is instantaneous. This amount af current and voltage cause a maximum

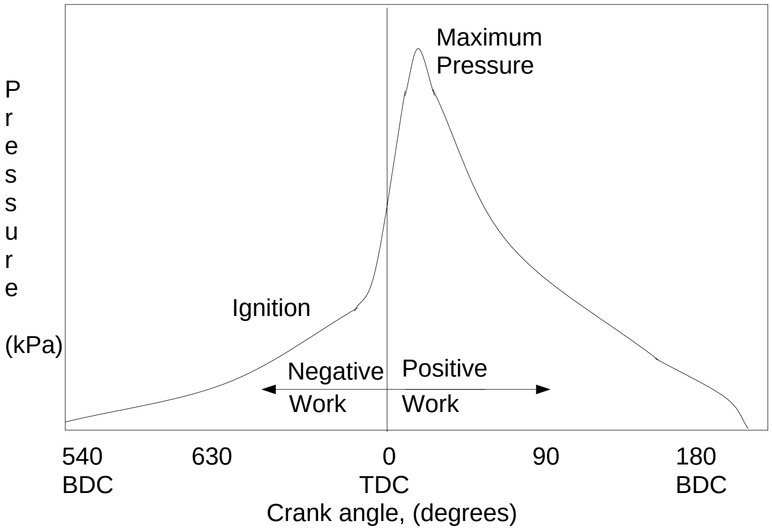

Figure 7.1: Pressure vs Crank angle

peak temperature of about 60,000 K. The actual temperature of the plug at time of ignition is only about 6000 K with spark discharge lasting around 0.001 second. The energy delivered by the spark plug is on the order of 30 to 50 mJ, which is sufficient to start combustion since only about 0.2 to 0.3 mJ of energy is necessary for stoichiometric mixtures and non-stoichiometric mixtures, respectively. One method to produce this high voltage is by using a coil in the battery. Most automobiles use a 12-volt battery that is not sufficient enough to cause the spark. A coil in combination with the battery is used to multiply the voltage many times and provide the necessay voltage.

The firing of the spark plug and flame propagation are related by how much pressure is in the cylinder. When the electrical discharge causes the air-fuel mixture to ignite the flame is very small and it travels very slow because of its size. Since the flame is very small and does not propage fast it does not generate enough energy, the pressure in the cylinder is not high enough to cause combustion. When 5-10% of the mixture is burnt and the rise in pressure due to the compression stroke is high then flame propagation starts. Having a rich air-fuel mixture around the electrodes of the spark plug speeds up ignition, gives a higher flame speed, and gives a "better start to the overall combustion process." (Pulkrabek, p.232) For this reason spark plugs are generally placed near the intake valve to assure a rich mixture.

7.1.2 Flame Propagation

Flame propagation causes the majority of the combustion in the cylinder. By the time that the first 5-10% of the air-fuel mixture has been burnt flame propagation has been well established. The flame has the speed to move quickly throughout the combustion chamber and cause the rest o fthe combustion to occur. Flame propagation is increased 10 times when induced turbulance, swirl, and squish are introduced, which always is because of engine design. If the propagation of the flame was laminar then the speed would be decreased. That is why characteristics like swirl and squish are desired; they induce the turbulent flow of the flame front.

When the gas mixture burns, the pressure and temperature rise. Burnt gases behind the flame front are hotter than the unburnt gases in front of the flame front. When this occurs the density of the burnt gases decreases while increassing the volume, and hence occupying a larger percentage of the volume of the combustion chamber. This can be seen from equation 7.1.

$$\rho = \frac{m}{V} \tag{7.1}$$

When the density decreases the volume increases. Figure 7.2 shows the relationship between the percent of mass burnt and the percent of volume burnt. When 30% of the gas mass is burnt the burnt gases occupy 60% of the total volume. This makes 70% of the total unburnt gas make only 40% of the total volume.

Since the volume of both burnt and unburnt fuel is kept constant in the combustion chamber, and the volume of burnt gases keeps increasing as the flame propagates the unburnt mixture is compressed, therefore incresing its temperature and hence the combustion rate. In addition to the rise in temperature from the compression of the fluid there is a rise in temperature due to radiative heat transfer. Radiation is emmitted from the flame reaction zone, which is at an approximate temperature of 3000 K, and transfered to the burnt and unburnt gases. As the flame propagates throughout the combustion chamber the temperature and pressure increase constantly. This rise in temperarure and pressure cause the chemical reaction to increase which causes the flame front to increase.

The effects on combustion not only come from turbulance, swirl, and squish, but also from the type of fuel used and the air-fuel ratio. Lean mixtures have slower flame speeds than rich mixtures. Figure 7.3 shows the relationship between the flame speed and air-fuel ratio. The flame speed will

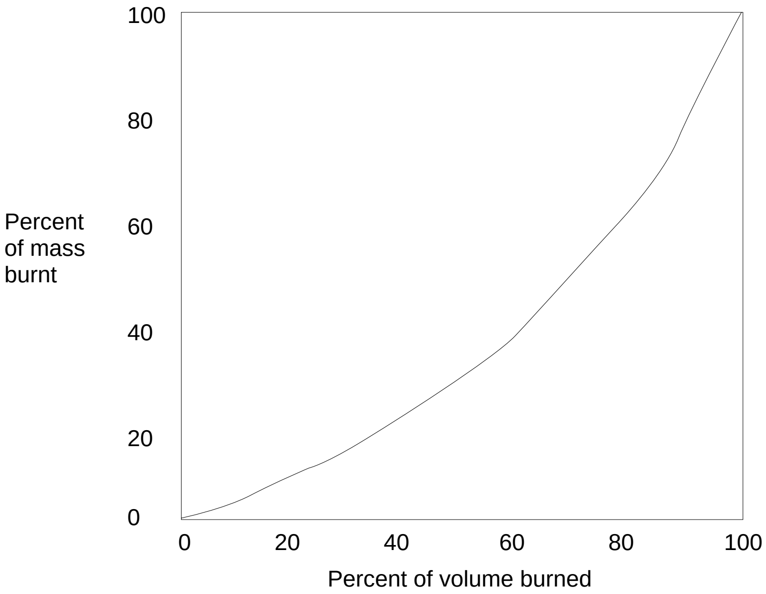

Figure 7.2: Percent of mass burnt vs Percent of volume burnt

be the greatest when there is a rich mixture but not too rich otherwise the flame speed will decrease again. The flame speed also increses with engine speed since turbulace, swirl, and squish are increased. Figure 7.4 gives an "average combustion chamber flame speed as a function of engine speed for a typical SI engine." (Pulkrabek, p.236) The burn angle is defined as the angle the crakshaft makes when combustion occurs. For most SI engines the crankangle is 25° before TDC. If ignition is too early then the pressure inside the cylinder will be high and the piston will do extra work. If however ignition is late then the pressure will be low, which will reduce the amount of work done on the engine. Actual ignition timing occurs anywhere from 10-30° before TDC. Figure 7.5 describes the relationship between the burn angle and the engine speed. Flame propagation starts at 5% and ends at 95% consumption. The typical crankangle during the main part of combustion is 25°.

7.1.3 Flame Termination

At about 15-20° after TDC 90-95% of the combustion process has taken place and the flame front has reached all corners of the combustion chamber.

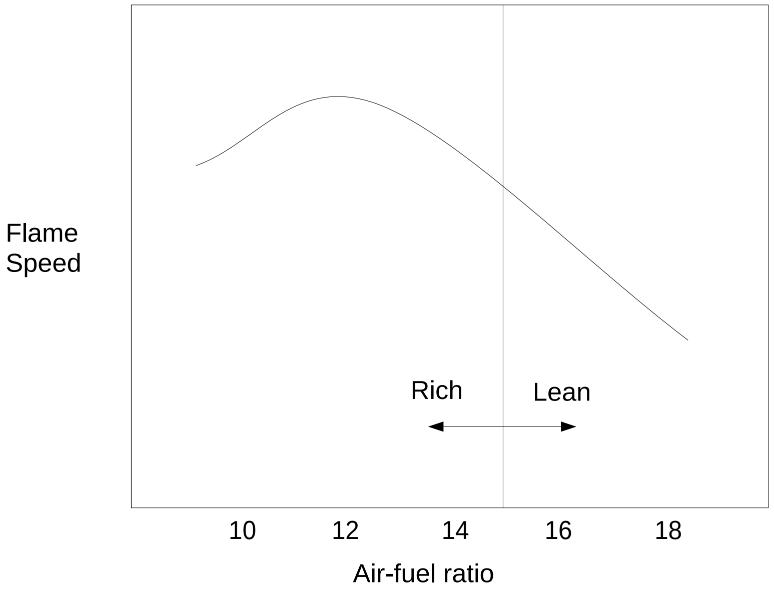

Figure 7.3: Flame speed vs Air-fuel ratio

The termination of the flame consumes roughly 5% of the air-fuel mixture. Figure 7.2 shows that when there is only about 5-10% of the gas mass left, it has been compressed into a small percent of the combustion chamber volume. The small volume left at the end of the combustion process makes the *endgas* combust with the combustion chamber walls and corners.

Because the end gas is so close to the cylinder walls the reactions are reduced in rate. Apart from being a slow reaction the large mass of the metal walls “acts as heat sink and conducts away much of the energy being released in the reaction flame.” (Pulkrabek, p.238) Since the final mixture has been minimized in energy content the rate of reaction and flame speed are reduced. This is wanted out of the last phase of combustion. Since the pressure is low at the last stage of combustion the forces exerted on the cylinder head are minimized and therefore a smooth ending to combustion is accomplished.

7.2 Combustion In CI Engines

The main difference between combustion in CI and SI engines is the way in which combustion occurs. In SI engines the combustion process first occurs

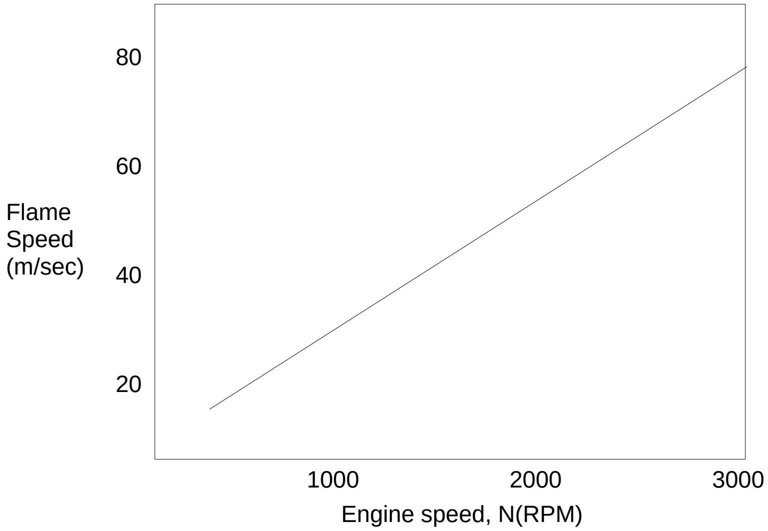

Figure 7.4: Flame speed vs Engine speed

by igniting a homogeneous mixture using a spark plug. The main difference arises when the flame is initiated and the flame travels at a certain direction, dictated by the flame propagation, whereas combustion in CI engines there is no flame propagation with a direction. Combustion in a CI engine is a nonsteady process where a nonhomogeneous mixture is controlled through fuel injection. The mixture is nonhomogeneous since air is the only substance being compressed until late in the compression stroke. Injection of the fuel occurs at about 15° bTDC and ends at about 5° aTDC. Following are the steps that the fuel goes through, after injection, in order to cause the proper combustion.

1.*atomization*: the fuel droplets break into smaller droplets.

2.*vaporization*: the small droplets of fuel vaporize in the chamber due to high temperatures. About "90% of the fuel injected into the cylinder has been vaporized within 0.001 second after injection."(Pulkrabek, p.252)

3.*mixing*: after vaporization of the fuel, the fuel mixes with the air to form a combustible air-fuel mixture.

4.*self-ignition*: self-ignition usually starts around "8° bTDC, 6-8°, after the start of injection." (Pulkrabek, p.252) At this point some of the mixture will ignite. These small reactions are caused by high temperature within the chamber. They are exothermic and further raise the temperature of the combustion chamber.

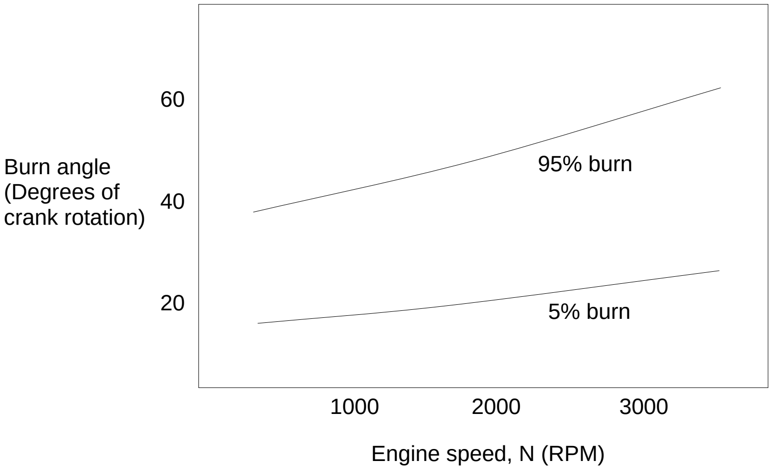

Figure 7.5: Burn angle vs Engine speed

5.*combustion*: combustion finally takes place after the self ignition of the air-fuel mixture, throughout the combustion chamber. At the time of combustion, around 70-95% of the fuel in the combustion chamber is in the vapor state. At this time many flame fronts develop at different places throughtout the combustion chamber, with the aid of the self-ignited mixture. When all of the combustible air-fuel mixture has been used the temperature and pressure rise. The increase in temperature and pressure further increase self-ignition points and combustion increases. Throughout this process liquid fuel is still being injected into the combustion chamber. The amount of fuel that is injected dictates the rate of combustion since the fuel has to be atomized, vaporized, mixed and finally combusted. Figure 7.6 shows the relationship between cylinder pressure and crank angle.

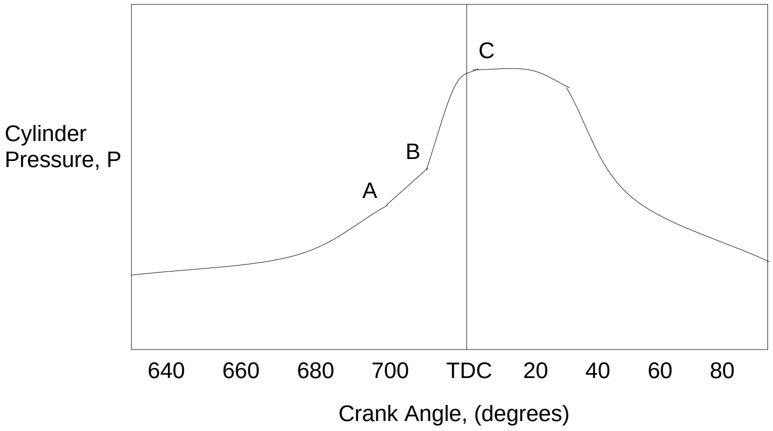

Figure 7.6: Cylinder pressure vs Crank angle

Chapter 8

Heat Transfer In IC Engines

Heat transfer in IC engines is a very serious problem since you need high temperatures to combust the fuel but you also need to keep the temperature at a controllable level in order to operate the engine safely. Once the temparature in the engine has reached intolerable values the engine block and components may suffer damage. Therefore it is essential to have a heat removal process which will maintain the engine at a safe operating condition. A water jacket or air through finns are two ways that reduce the temperature in the engine.

8.1 Engine Temperatures

There are components in an engine that suffer more from high temperatures than others. Figure 8.1 shows the temperature distribution for an IC engine.

The components with the highest temperatures are the spark plug, the piston face, and exhaust valve and port. The problem wirth these components are that they are not only the hottest components but they are also very difficult to cool. They are very difficult to cool because their location in the engine does not permit them for proper heat transfer. The design of the engine is such that there is not much space for a water jet to cool of the components. The spark plug is located in the middle and fastened to the chamber. There is really no water flow that will aid in the reduction of the temperature of the hot spot. The exhaust valve and port carry the steady state flow of hot exhaust gases. The exhaust gases heat the valve through convective heat transfer. Because of the geometry created by the valve mechanism it is very difficult to have a water jet or even a finned surface to provide the necessary cooling. The third part of the chamber where heat is a problem is the piston face. The geometry of the piston face gives very little opportunity for cooling. Figure 8.2 shows the relationship between temperature and time. At about 60 seconds a steady temperature is reached.

8.2 Heat Transfer In Intake System

The heat transfer involved in the intake system occurs when air or an air-fuel mixture comes into the manifold. The intake manifold is hotter than the air-fuel mixture because of its proximity to the engine components or the design of the manifold. The intake manifold can be designed to heat the air-fuel mixture, so that the mixture can start to vaporize once it has entered the combustion chamber. One way of heating the manifold is to put it in close proximity with other hot components. The manifold will heat through convective heat transfer. Electricity and hot coolant flow are other ways in which the manifold can also be heated. After the manifold is heated then the air-fuel mixture is heated through convective heat transfer. Equation 8.1 shows the heat transfer problem associated with the air-fuel mixture and the manifold walls.

$$\dot{Q} = hA(T_{wall} - T_{gas}) \tag{8.1}$$

where T=temperature

h=convective heat transfer coefficient

A=inside surface area of intake manifold

The advantage of heating the intake manifold is that the fuel vaporizes sooner. By making the fuel vaporize at an earlier time, there is more time available for the fuel and the air to mix, therefore providing a homogeneous mixture that is ready for combustion in the combustion chamber.

8.3 Heat Transfer In Combustion Chamber

The heat transfer involved in the combustion chamber involves all three modes of heat transfer. Conduction, convection, and radiation all play very important roles when defining the heat transfer characteristics of the combustion process in an IC engine. During the intake stroke the air-fuel mixture is cooler than the cylinder walls and heat transfer to the mixture occurs. This process further vaporizes the rest of the fuel until all the fuel has been vaporized. At this same time evaporative cooling takes place, which lowers the heat in the compressive stroke. In evaporative cooling the substance, in this case the mixture, is evaporated because of an applied heat source. In order to vaporize the fuel energy has to be used in the form of heat from the hot cylinder walls. Since energy was used, in the form of heat, then there is less energy in the cylinder walls, and hence the cylinder walls are cooled. After the compression stroke and during combustion there is heat transfer to the sorroundings from the hot gas through the cylinder walls. The peak gas temperatures of combustion are on the order of 3000 K and that is why the cylinder walls of the chamber overheat. The only way in which energy can be transferred away from the combustion chamber is through convection and conduction. This form of heat transfer will keep the walls of the cylinder from melting. Figure 8.3 shows the two two types of cylinder walls used for heat transfer purposes. The figure shows an air and liquid cooled engine. The air cooled engine uses fins whereas the liquid cooled engine uses a coolant. The heat transfer per unit surface area is:

$$\dot{q} = \frac{\dot{Q}}{A} = \frac{T_g - T_c}{\frac{1}{h_g} + \frac{\Delta x}{k} + \frac{1}{h_c}} \tag{8.2}$$

In equation 8.2 T_g=gas temperarure in the combustion chamber
T_c=coolant temaperarure
h_g=convection heat transfer coefficient on the gas side
h_c=convection heat transfer coefficient on the coolant side
Δx=thickness of the combustion chamber wall
k=thermal conductivity of the cylinder wall

Since the process of combustion in an IC engine is a cyclic process some of the coefficients and temperatures will no be constant during the cycle. The gas temperature in the combustion chamber will not be constant over the cycle. As a matter of fact, the gas temperature in the combustion chamber will be greater than the cylinder walls' temperature during the expansion stroke and lower during most of the compression stroke. The coolant temperature will be fairly constant and any changes will occur over a longer period of time. The convective heat transfer coefficient on the gas side will vary greatly within a cycle because of changes in gas motion, swirl, turbulance, and velocity. The convection heat transfer coefficient on the coolant side will be fairly constant throughout the cycle. The thermal conductivity of the cylinder wall will also be fairly constant because the cylinder wall temperature is maintained at an overall constant temperature.

Heat transfer to and from the cylinder walls occur throughout the four strokes of the combustion process. During the intake stroke the cylinder walls are hotter than the fuel, which causes vaporation of the fuel. Heat transfer from the cylinder walls to the fuel is through convection. During the compression stroke the gases become warmer than the cylinder walls and during the expansion stroke the greatest amount of heat transfer from the gases to the cylinder wall occurs. In the exhaust stroke the gases have cooled down so there is not a significant amount of heat transfer.

Since the process is cyclic the amount of heat transfer at a particular point in the combustion process should renain fairly constant. Also the heat transfer at any particualr point can be either positive or negative, which means thatthe heat is travelling either away or to the cylinder walls. Figure 8.4 demonstrates the cycle-to-cycle variation in heat transfer for a particular point. Even though heat transfer is fairly constant there are still small variations.

The heat transfer from cycle to cycle varies for different points in an engine. Figure 8.5 shows the heat transfer at three different locations in the combustion chamber for a single cylinder during one cycle. There is significant variation in the heat transfer for these points.

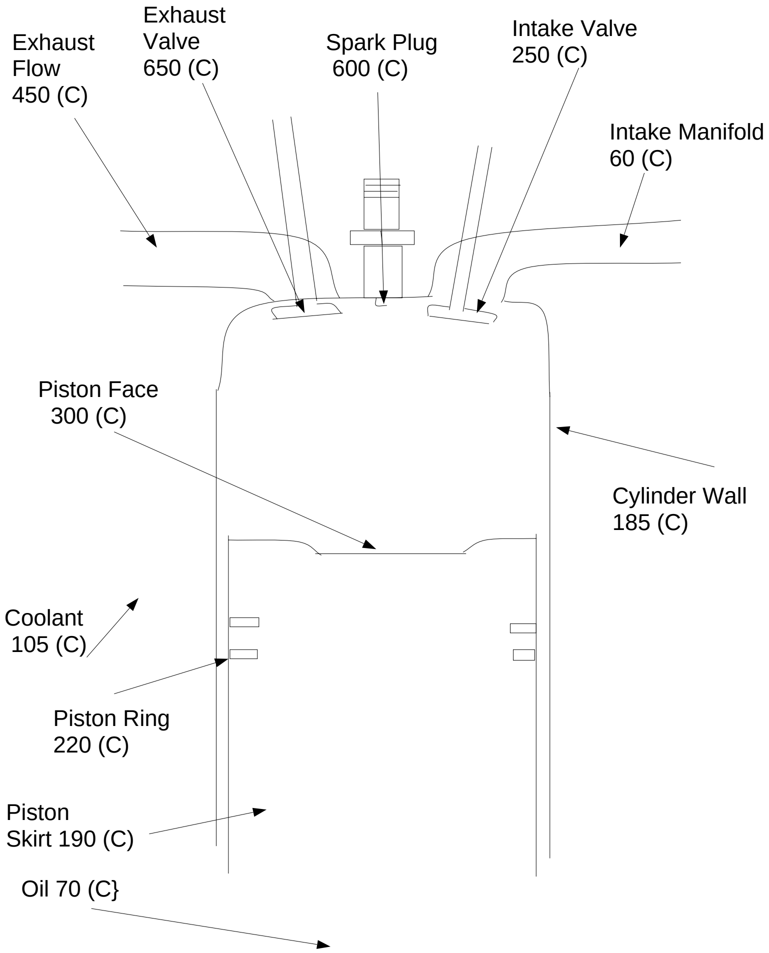

Figure 8.1: Temperatures of engine components

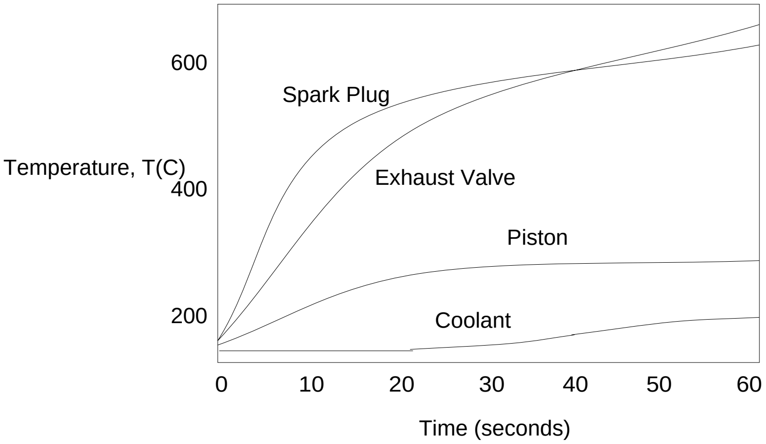

Figure 8.2: Temperarure vs. time

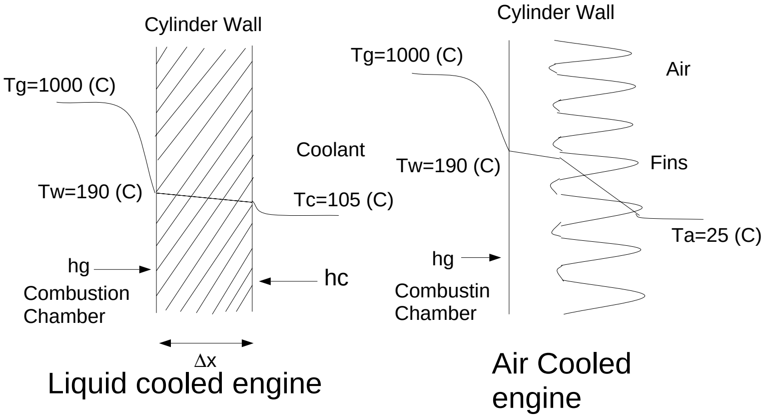

Figure 8.3: Liquid and air cooled engines

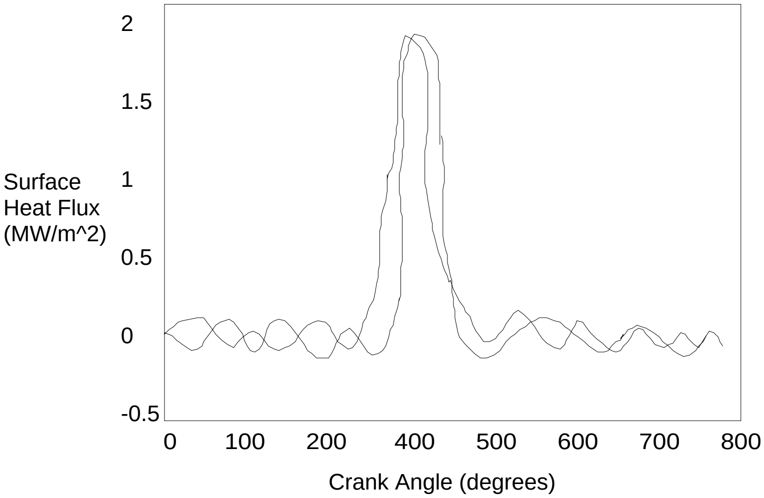

Figure 8.4: Heat flux vs crank angle

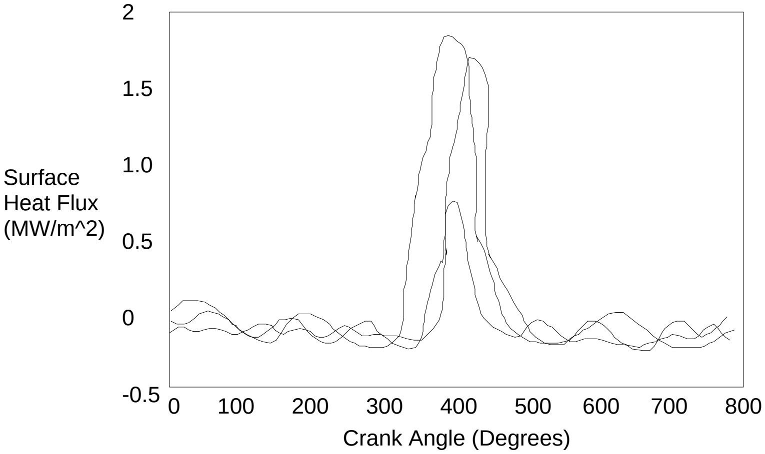

Figure 8.5: Heat flux vs crank angle

Chapter 9

Turbocharging

9.1 Introduction

In order to increase the power of an engine there has to be an increase in pressure, and hence force exerted on the piston, during the power stroke. The amount of work that the power stroke delivers is basically determined by the air-fuel mixture in the combustion chamber. The combustion that occurs during the end of the comprssion stroke and throughout the power stroke is determined by how much air is mixed with the fuel. When air is compressed its density increases but volume decreases. Hence compressing air at the beginning of an engine cycle increases the power output by increasing the amount of air that is mixed with fuel. Since the total volume of occupied space within the cylinder is decreased when compressing air, more air can be used to combust with the fuel. This i sexactly what superchargers and turbochargers do. They increase th epressure to increase the density of the air to make the engine increase in power. The most common supercharger is a mechanical supercharger. Figure 9.1 shows this supercharger and its arrangement with the engines. The compressor is driven using power fromthe engine. In a turbocharger, shown in Figure 9.3, a combination of a compressor and turbine are used. Although this requires the use of another shaft, the engine power is not used to provide the work needed to run the compressor. The exhaust gases go into the turbine, which uses the energy content in the hot gases to run the shaft that runs the compressor.

9.2 Superchargers

The most common supercharger is a mechanical supercharger. Figure 9.1 shows this supercharger and its arrangement with the engine. In this configuration the compressor receives atmospheric air. This air is then compressed so that the density increases. After the air has been compressed the engine takes this compressed air and mixes it with the fuel in the combustion chamber. The imiting factor in the maximum power an engine can deliver is limited by “the amount of fuel that can be burned efficiently inside the engine cylinder.” (Heywood, p.248) This in turn is limited by the “the amount of air that is introduced into each cylinder each cycle.” (Heywood, p.248) So by introducing a greater amount of air into the cylinder the opportunity for combustion increases, therefore incresing the power of the engine. After the engine has burned the mixture it is released through the exhaust manifold into the atmosphere.

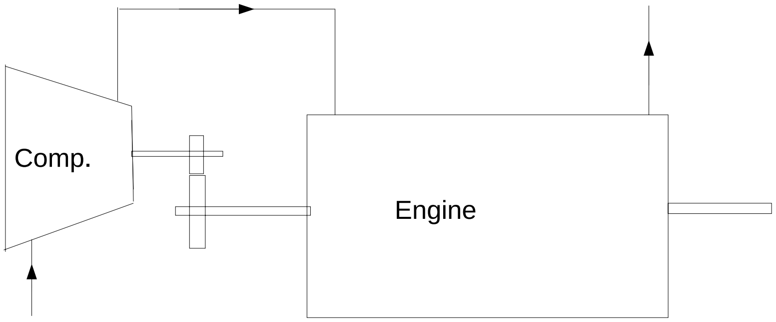

Figure 9.1: Supercharger

9.2.1 Compressors

The compressor used in the supercharger uses energy to produce work. Where the compressor gets this energy is essential in determining total engine efficiency. Because the power output of the engine increases when using a supercharger some of this power is used to drive the compressor. The amount of work that is produced by the piston is therefore divided into running the compressor and running the engine. In a turbocharger the exhaust gas is used to drive the compressor, therefore all the work from the cylinder is used in running the engine. The work input into the compressor can be computed from the first law of thermodynamics. Equation 13.3, represents the first law of thermodynamics when applied to a control volume around the turbomachinery component.

$$\dot{Q} - \dot{W} = \dot{m}[(h + \frac{C^2}{2} + gz)_{out} - (h + \frac{C^2}{2} + gz)_{in}] \tag{9.1}$$

With the use of equation 13.3 the work-transfer equation can be equated for a compressor. For compressors the quantity of work involved is much greater than the heat losses to the surroundings or frictional losses, therefore the term $\dot{Q}$, in equation 13.3 is equal to 0. Also the work related to the kinetic and potential energy of the compressor is also 0 as compared to to the work involved. Equation 9.2 describes the relationship between the work rate input and the enthalpy change in the system.

$$-\dot{W} = \dot{m}(h_{0,out} - h_{0,in}) \tag{9.2}$$

In equation 9.2 the enthalpy terms are defined in terms of stagnation enthlpies. Stagnation enthalpy is definesd as the static plus the dynamic enthalpy. The dynamic enthalpy is related to the velocity associated with the fluid. The stagnation enthalpy is defined through equation 9.3.

$$h_0 = h + \frac{C^2}{2} \tag{9.3}$$

An *h*-*s* diagram is used to directly relate the enthalpy to work. Figure 9.2 provides a direct relation between the enthalpy and entropy. The area under the curve represents the work. Taking the difference in the enthalpy also provides a relation describing the efficiency of the compressor. The efficiency is defined as the theoretical work required divided by the actual work the compressor uses. In Figure 9.2 the p_{02} and p_{01} term represent the stagantion pressures at the oulet and inlet. The stagantion pressure is defined using equation 9.4.

$$p_0 = p(\frac{T_0}{T})^{\frac{\gamma}{\gamma-1}} \tag{9.4}$$

where T_0 is equal to the stagantion temperature, defined as:

$$T_0 = T + \frac{C^2}{2c_p} \tag{9.5}$$

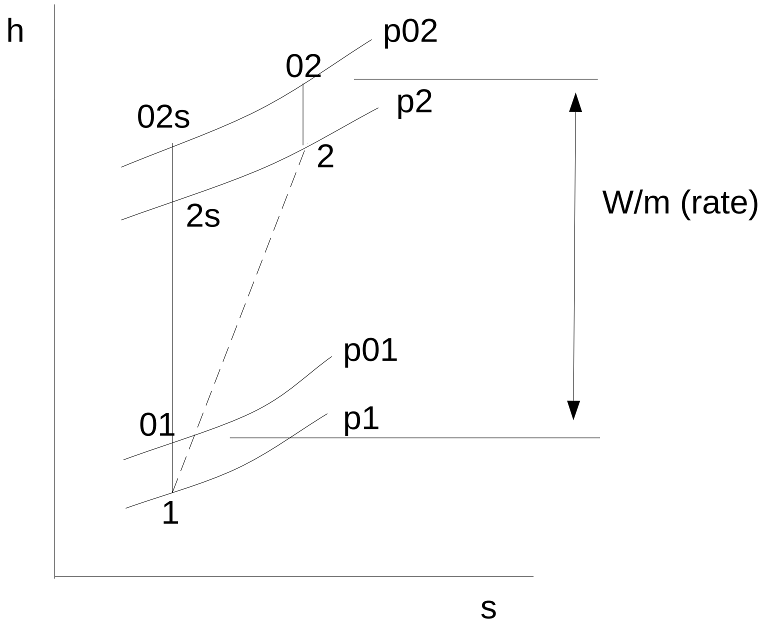

Figure 9.2: h-s Diagram for Compressor

The compressor isentropic efficiency corresponding to the *h*-*s* diagram is defined by equation 9.15.

$$\eta = \frac{h_{02s} - h_{01}}{h_{02} - h_{01}} \tag{9.6}$$

Because it is difficult to measure the enthalpy it is sometimes convenient to model the gas as an ideal gas. The efficiency for a model of an ideal gas corresponding to:

$$\eta = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}} \tag{9.7}$$

and since the process 01 to 02s is isentropic,

$$T_{02s} = T_{01}(\frac{p_{02}}{p_{01}})^{\frac{\gamma-1}{\gamma}} \tag{9.8}$$

the efficiency is now expressed in terms of temperatures and pressures. The efficiency, 9.9, is now:

$$\eta_c = \frac{(\frac{p_{02}}{p_{01}})^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{02}}{T_{01}} - 1} \tag{9.9}$$

The work-transfer rate or power required to drive the compressor is now obtained in equation 9.10.

$$-\dot{W} = \frac{\dot{m}_i c_p T_{01}}{\eta_c}[(\frac{p_{02}}{p_{01}})^{\frac{\gamma-1}{\gamma}} - 1] \tag{9.10}$$

9.3 Turbochargers

In a turbocharger, shown in Figure 9.3, a combination of a compressor and turbine are used. Although this requires the use of another shaft, the engine power is not used to provide the work needed to run the compressor. After the exhaust gases leave the exhaust manifold through the process of supercharging as described above the exhaust gases go into a turbine. The turbine uses the energy content in the hot gases to run the shaft that runs the compressor. The turbine basically expands the gas mixture, which is at high temperature, and transfers some of the energy into useful work. There are basically two ways of turbocharging engines. They are *constant-pressure turbocharging* and *pulse turbocharging.* In constant-pressure turbocharging the exhaust manifold is designed such that the pressure pulses sent by the reciprocating engine die out, in order to have steady flow reach the turbine.

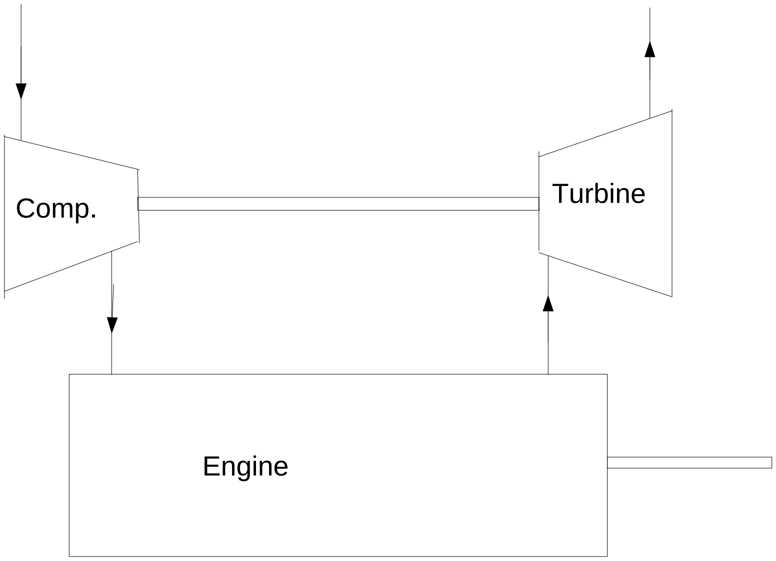

Figure 9.3: Turbocharger

The volume of the exhaust manifold will have been increased to accomodate constant-pressure turbocharging. The disadvatage of constant-pressure turbocharging is that by letting the pressure pulses die out in order to have steady flow the energy inherent in having a fluid at high velocity will die out since the velocity will have decreased significantly. In pulse turbocharging the kinetic energy associated with blowdown of the exhaust manifold is used by the turbine. Small cross-sectional pipes connect from the exhaust port to the turbine in order to maintain this kinetic energy. The problem with pulse turbocharging is that the turbine does not receive a steady flow. To compensate for this defficiency the pipes connecting to the turbine are grouped such "that the exhaust pulses are sequential and have minimum overlap." (Heywood, p.263) This provides a steady flow to the turbine.

9.3.1 Turbines

The turbine in a turbocharger acts as an aid in the total efficiency of the purpose of turbocharging. By having the exhaust gases go into the turbine energy is being reused to drive the compressor. The reason the turbine can do work is because the temperature of the gases are hot enough to have high energy content. In a turbine the gases do work on the blades of the turbine, but the end effect is the turbine producing work on a shaft. Since the turbine does work on a shaft to make the compressor run it is said that the turbine does positive work. The work term in terms of enthalpy for a turbine is shown in equation 9.11.

$$\dot{W} = \dot{m}_e(h_{03} - h_{04}) \tag{9.11}$$

The enthalpy terms in equation 9.11 are once again staganation enthalpy. The *h*-*s* diagram that describes the process of a turbine is shown in Figure 9.4. From this diagram the efficiency of the turbine can be solved for using the enthalpy terms. The efficiency for a turbine can be ecaluated by dividing the actual power output by the theoretical work output. For a turbine the turbine efficiency is shown in equaiton 9.12

$$\eta_T = \frac{h_{03} - h_{04}}{h_{03} - h_{04s}} \tag{9.12}$$

Since exhaust gas of an engine is very difficult to analyze, it is helpful to analyze the gas as an ideal gas with constant specific heats. Eqaution 9.13 represents the efficiency when an ideal gas is used. From the same analysis

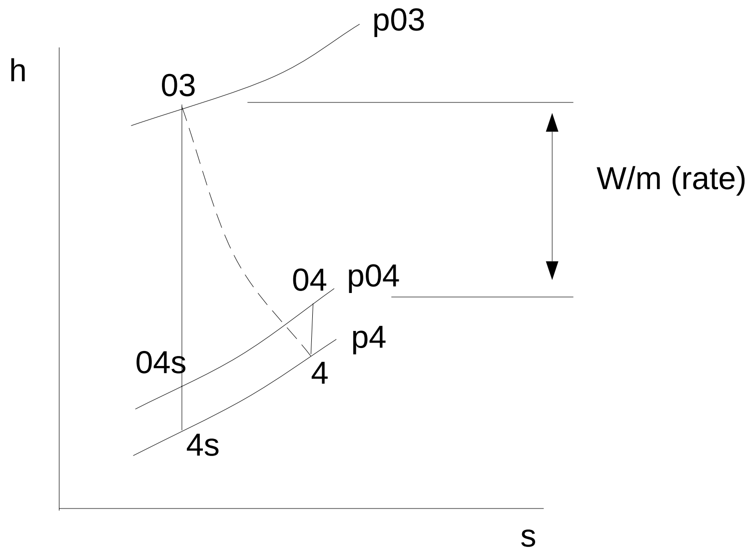

Figure 9.4: h-s Diagram for Turbine

used earlier the efficiency is:

$$\eta_T = \frac{T_{03} - T_{04}}{T_{03} - T_{04s}} = \frac{1 - (\frac{T_{04}}{T_{03}})}{1 - (\frac{p_{04}}{p_{03}})^{\frac{\gamma-1}{\gamma}}} \tag{9.13}$$

and since the process 03 to 04s is isentropic,

$$T_{04s} = T_{03}(\frac{p_{04}}{p_{03}})^{\frac{\gamma-1}{\gamma}} \tag{9.14}$$

The work rate, or power, is:

$$\dot{W} = \dot{m}_e(h_{03} - h_{04}) = \dot{m}_e c_{p,e} \eta_T T_{03}[1 - (\frac{p_{04}}{p_{03}})^{\frac{\gamma_e-1}{\gamma_e}}] \tag{9.15}$$

Chapter 10

Friction and Lubrication

10.1 Friction

Friction in IC engines is a major problem because it deteriorates the cylinder and other components in the engine. Effective lubrication is needed in order to maintain the engine at safe operating conditions. Oil is a good lubricant since it provides necessary friction lubrication.

Friction presents problems to the engine by reducing the power output and hence its efficiency. Friction is classified as a loss in form of power in equation 10.1,

$$\dot{W}_f = (\dot{W}_i)_{net} - \dot{W}_b \tag{10.1}$$

where $(\dot{W}_i)_{net} = (\dot{W}_i)_{gross}$-$(\dot{W}_i)_{pump}$ and
subscipt f=friction
i=indicated
b=brake

It is a loss since it takes away from the whole power of the engine. The i, in the above equations indicates the power from the combustion chamber. The b, indicates brake power which corresponds to the power from the crankshaft. Gross power is the power from the compression and expansion stroke. Pump power is the power from the exhaust and intake strokes.

Since there are many different engines with different speeds the best way to measure friction loss is by measuring the mean effective pressure. The friction loss is most often reffered to as a friction mean effective pressure (fmep). The relationship between work and pressure, and power and pressure is shown in eqaution, 10.2, and equation ,10.3, respectively.

$$w = (mep)V_d \tag{10.2}$$

$$\dot{W} = (mep)V_d(\frac{N}{n}) \tag{10.3}$$

In the above eqautions V_d=displacement volume

N=engine speed

n=number of revolutions per cycle

The frictional mean effective pressure is then,

$$fmep = \frac{\dot{W}_f}{[V_d(\frac{N}{n})]} \tag{10.4}$$

The frictional forces in a piston are more or less the same for the intake, compression, and exhaust stroke. During the compression stroke the pressure and forces are greater so the frictional forces increase. The piston contributes as much as 50% of the total friction to the engine. The piston rings also contribute around 20% of the total friction to the engine. At high speeds the friction involved with the engine increases. At speeds greater than $15\frac{m}{s}$ the danger of having structural failure increases.

10.2 Forces on Piston

The components that contribute most to the total friction loss in an engine are the valves, pistons and piston rings. Other components such as the water pump, oil pump, and alternater do not cause so much frictional losses. Figure 10.1 shows various components that contribute to frictional loss in an engine.

The forces on a piston are shown in Figure 10.2. The various forces are reactions to other forces during motion of the piston. For example the thrust force is a reaction to the force of the connecting rod when the piston is in motion. The friction force is a reaction to the sliding piston during motion. The thrust force is not a constant force but changes with many conditions. Since the force applied to the piston by the connecting rod is not constant and changes in direction, the thrust force also changes in magnitude and direction. The thrust force also changes with acceleration, pressure, and friction force, all of which vary during the engine cycle. A force analysis can

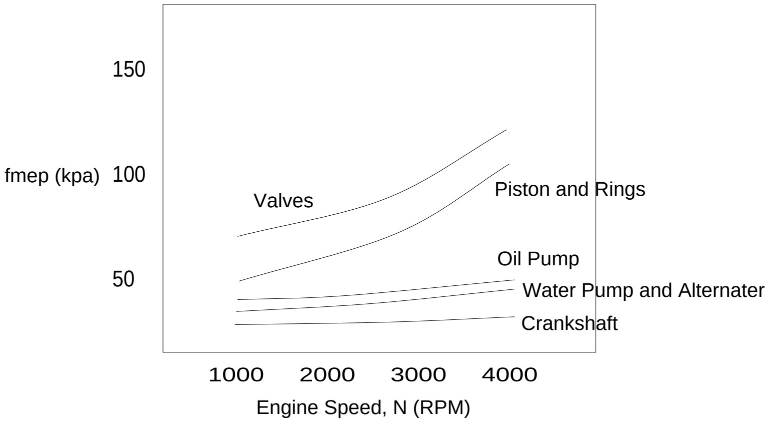

Figure 10.1: fmep vs Engine Speed

be done on the piston to determine the thrust force exerted on the piston. Assuming positive in the indicated x-dir and positive in the indicated y-dir, and summing the forces in the x-dir give equation 10.5.

$$\Sigma F_x = m(\frac{dU_p}{dt}) = -F_r cos\phi + P(\frac{\pi}{4})B^2 + F_f \tag{10.5}$$

where ϕ=angle between the connecting rod and ceneterline of piston
m=mass fo the piston
$\frac{dU_p}{dt}$=acceleration of the piston
F_r=force of the connecting rod
P=pressure in the combustion chamber
B=bore
F_f=friction between the piston and cylinder walls

The equation that satisfies the forces in the y-dir is shown in eqaution 10.6.

$$\Sigma F_y = 0 = F_r cos\phi - F_t \tag{10.6}$$

Combining equations 10.5 and 10.6 gives an equation that can be used to solve for the thrust force, F_t.

$$F_t = [-m(\frac{dU_p}{dt}) + P(\frac{\pi}{4})B^2 + F_f]tan\phi \tag{10.7}$$

To reduce friction in engines modern engines use a different design for the piston. The piston skirt and weight are reduced. By reducing the skirt

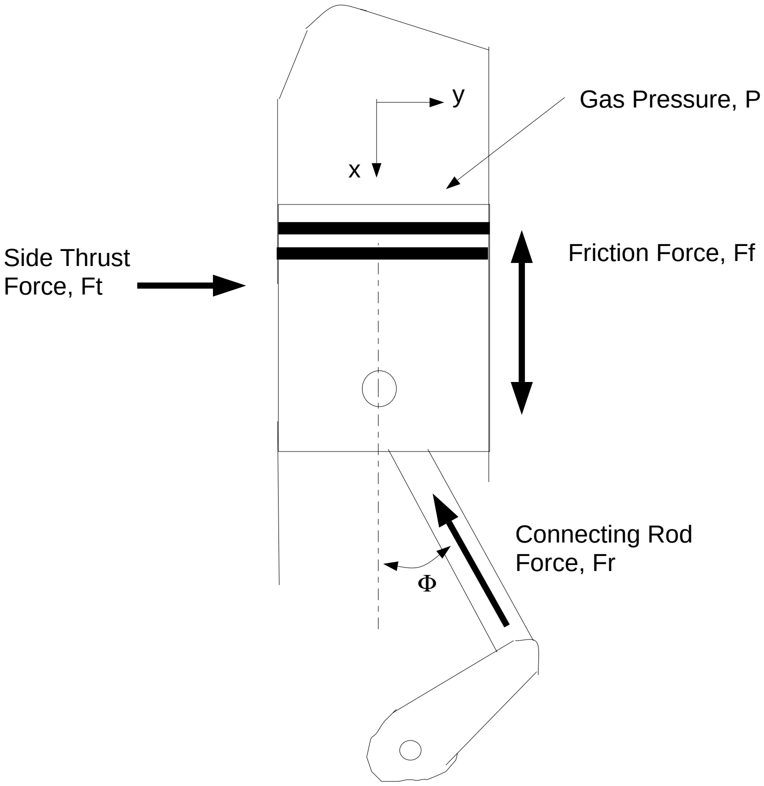

Figure 10.2: Forces on Piston

of the engine, the rubbing surface are of the piston is reduced and therefore the friction is reduced. But as a consequence of reducing the piston skirt the radial movement of the piston increases. In order to accomodate this movement the tolerances are reduced. When the weight of the piston is reduced the inertia is reduced and hence the accelaration is reduced. Therefore by reducing the weight and skirt of the piston the friction is decreased. Figure 10.3 shows how oil film thickness varies with the position of the piston. The film thickness is small both at TDC and BDC, but when the piston reaches high speeds the thickness increase. For the four strokes that the engine goes through the film thickness is a minimum at TDC and BDC, or where the crank angle is 180, 360, 540, and 720.

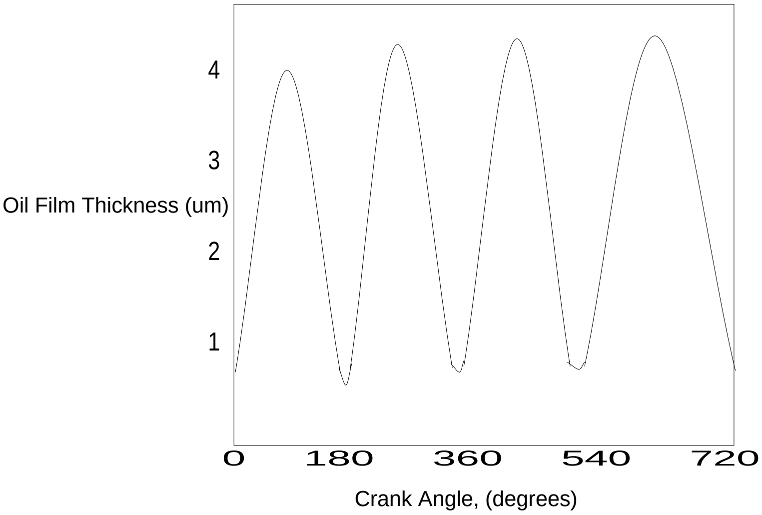

Figure 10.3: Oil Film Thickness vs Crank Angle

10.3 Lubrication

The lubrication process for reducing the friction in an engine is essential for making the engine run well. There are three oil distribution systems. The first involves a splash system where oil is splashed into other componensts of the engine by the rotating crankshaft. The second system involves using an oil pump to distribute the oil into engine components. The third lubrication process is a combination of the two above.

In the splash system, the crankcase is used as the oil sump (reservoir). As the crankshaft rotates oil is splashed to the oter parts of the engine. Many small four-stroke cycle engines use (lawn mowers, golf carts, etc.) this type of lubrication process.

The second type of lubrication process is the use of an oil pump. Oil is circulated throughout the engine through passages built into parts that are to be lubricated. In a typical engine there are passages built into the connecting rods, valve stems, push rods, rocker arms, valve seats, engine block and many other moving components. Most automobiles use a dual system, where pressurized oil flow from the oil pump and splash from the crankshaft are used to distribute the lubricating oil. Aircraft engines use pressurized oil systems with the oil reservoir located seperate from the crankcase. Oil pumps are eiter electric or can be driven directly off the engine.

Chapter 11

Lubrication

11.1 Introduction

When there is contact between two surfaces friction develops because of the relative movement between the two surfaces. Friction between metal surfaces causes wear. To decrease friction, the two mating surfaces have to be separated by a lubricant. A lubricant is a thin fluid film that separates to surfaces so as so reduce the friction between them. By reducing the friction between surfaces wear is reduced, hence increasing the life of the machine. If a proper lubricant is used, wear of parts will be minimized. This chapter talks about thin fluid film lubrication. The two types of fluid film lubrication are hydrodynamic and hydrostatic lubrication.

11.2 Hydrodynamic Lubrication

In hydrodynamic lubrication pressure is self induced by the relative motion of the walls. This type of lubrication is the most useful in terms of its applications. There are two design variables that are considered in the analysis. The pressure that is self induced by the relative motion of the walls is one design that is strived for. By having an understanding of the pressures generated the maximum load applied on the fluid film can then be determined. The second design to strive for is the film thickness. If you have a load applied, then you know the force applied on one of the walls. If you know the force that is applied on an area, then the pressure needed to keep the surfaces from contact is known. The fluid film can then be calculated from knowing the pressure. From fluid mechanics Newton's law of viscosity is expressed in equation 11.1,

$$\tau = \mu \frac{dv}{dy} \tag{11.1}$$

where τ equals the shear stress and μ is the dynamic viscosity. In equation 11.1 $\frac{dv}{dy}$ is the velocity of the fluid divided by the clearance of the film. Before any analysis can be done on the fluid film some assumptions are done. The assumptions associated with the theory of hydrodynamic lubrication are:

- Laminar-flow conditions prevail and fluid is said to be Newtonian.
- Inertia forces are small compared to viscous shear forces and may be neglected.
- Fluid is incompressible, therefore the volume flowing past any section in unit time will be constant.
- Pressure in the film is a function of x only. The pressure in the *y-dir* is a constant.
- Velocity in the film is a function of both x and y.
- Viscosity of fluid as it passes through bearing remains constant.

Figure 11.1 shows a converging fluid wedge. The dimensions are not proportional to the real situation. The distance h_o is actually much smaller than the distance shown. Assumming an infinite width of film in the *y-dir* a force analysis can be done on a differential fluid element. A force equilibrium diagram on a unit volume of hydrodynamic film is shown in Figure 11.2. By summing the forces in the horizontal direction an equation can be obtained to solve for the pressure distribution. Equation 11.2 describes the force equilibrium of a fluid element when the inertial forces are neglected.

$$(p + \frac{dp}{dx}dx)bdy + \tau bdx - (\tau + \frac{\partial \tau}{\partial y}dy)bdx - pbdy = 0 \tag{11.2}$$

After simplifying equation 11.2, equation 11.3 is obtained.

$$\frac{dp}{dx} = \frac{\partial \tau}{\partial y} \tag{11.3}$$

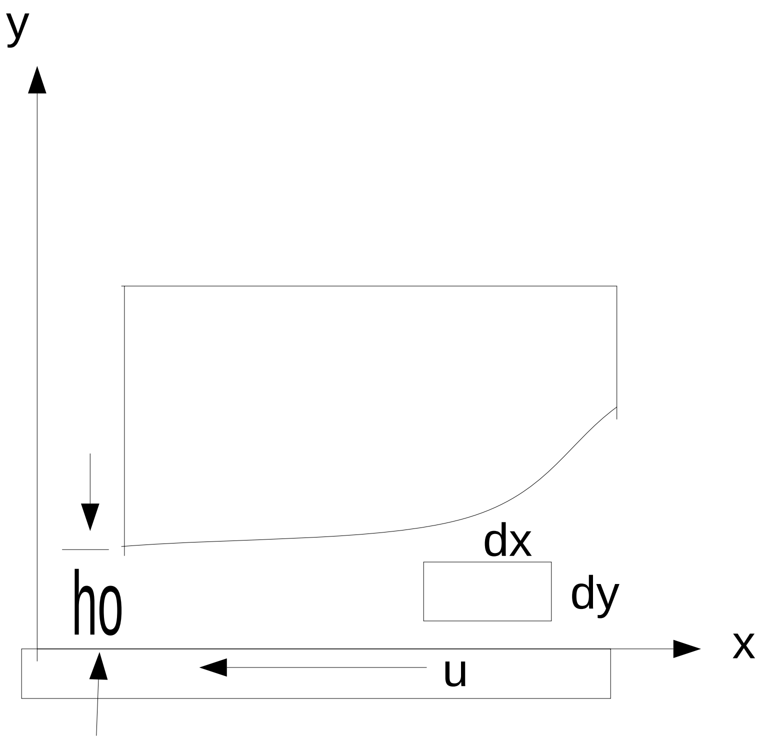

Figure 11.1: Converging fluid wedge

An expression for $\frac{\partial \tau}{\partial y}$ is needed in order to solve the pressure term. Since v is both a function of both x and y, the partial derivative of equation 11.1 is taken and substituted into equation 11.3. Equation 11.4 then gives an expression for the pressure in terms of the viscosity and velocity of the fluid and not shear stress.

$$\frac{dp}{dx} = \mu \frac{\partial^2 v}{\partial y^2} \tag{11.4}$$

In order to solve for pressure term the velocity needs to be integrated twice. When integrating and solving for the constants of integration equation 11.5 is obtained. This equation is expressed in terms of a pressure term, the velocity of the plate, u, the clearance, h, viscosity and variable y.

$$v = \frac{1}{2\mu} \frac{dp}{dx}(y^2 - hy) + u(\frac{y}{h} - 1) \tag{11.5}$$

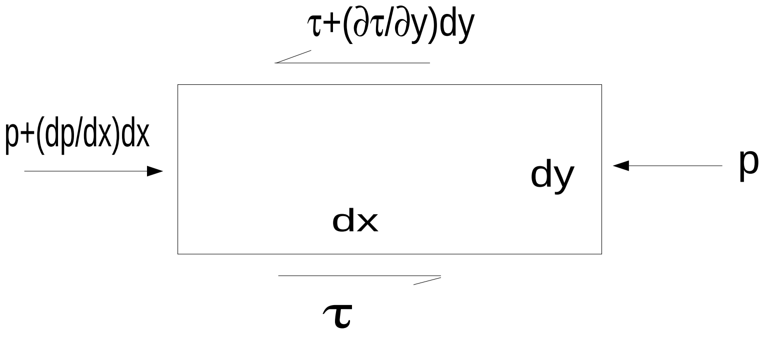

Figure 11.2: Force equilibrium on element of hydrodynamic film

If equation 11.5 were used to solve for the pressure term by taking its derivative and plugging it into equation 11.4, everything will divide out and nothing would be solved for. An alternate method to solve for the pressure is used using flows. The flow passing through any section is given by equation 11.6,

$$Q = \int_0^h vbdy \tag{11.6}$$

To solve for $\frac{dp}{dx}$ two equations for the flow rate are required. Once the two equations are obtained they are set equal to each other and an expression for the pressure is given in terms of the plate velocity, viscosity of fluid, and film thickness. The first of these equations is obtained by plugging equation 11.5 into 11.6. An expression for the flow in terms of pressure, plate velocity, and film thickness is given by equation 11.7.

$$Q = b(-\frac{1}{12\mu}\frac{dp}{dx}h^3 - \frac{uh}{2}) \tag{11.7}$$

The second expression for the flow needs to have either a constraint or a condition so that the resulting equation can be different from the one just derived and therefore be equated. Since the pressure varies along the path the fluid film takes there is a maximum pressure, where $\frac{dp}{dx} = 0$. At this instant the film thickness h, will be denoted as h^*. By substituting $\frac{dp}{dx} = 0$ and h^* into equation 11.7 the flow at maximum pressure is given by equation 11.8.

$$Q = -\frac{u}{2}bh^* \tag{11.8}$$

Since the flow is constant equations 11.7 and 11.8 are set equal to each other and an expression for the build up of pressure is obtained. Equation **??** represents the build up of pressure for many geometries that involve fluid film lubrication.

11.3 Hydrostatic Lubrication

The second type of fluid film lubrication is hydrostatic lubrication. In this type of lubrication there is very little relative motion between the surfaces. In this case it may be desirable to introduce a thin fluid film lubricant from an external high pressure source through a cut or groove. Since the lubricant is being pressurized into a clearance the pressure is not self induced but rather externally induced. This is known as externally pressurized lubrication. In this type of lubrication the bearing surfaces are kept separated at all times even if the surfaces are stationary. Relative sliding motion does not occur.

The rigidity or stiffness of the lubricant film is an important variable of the design. Bearing have been designed to support loads of about 25 to 30 million pounds per inch. This amount of rigidity means that the fluid film in the clearance space of the bearing has the same rigidity as a cube of steel with a side of 10 inches. Thus lubricants can provide the necessary rigidity and can provide a bearing film that is stiffer than the actual metal structure that contains it.

When dealing with hydrostatic lubrication the friction between the two surfaces is small. The coefficient of friction is on the order of 0.000005. Mathematical analysis of hydrostatic fluid film lubrication is essential to determine a relationship that correlates the design variables. The design variables are film thickness and viscosity. Other important constraints are set for these design variables such as the flow rate, the pressure required for the flow rate, and the load applied on the fluid film. Figure 11.3 represents a thrust bearing with a recess. The expression for the flow of an incompressible fluid is expressed in equation 11.9.

$$Q = \frac{\Delta P b h^3}{12 \mu l} \tag{11.9}$$

Equation 11.10 describes the flow for the thrust bearing shown in Figure 11.3. This equation is derived when equation 11.9 is applied to the thrust

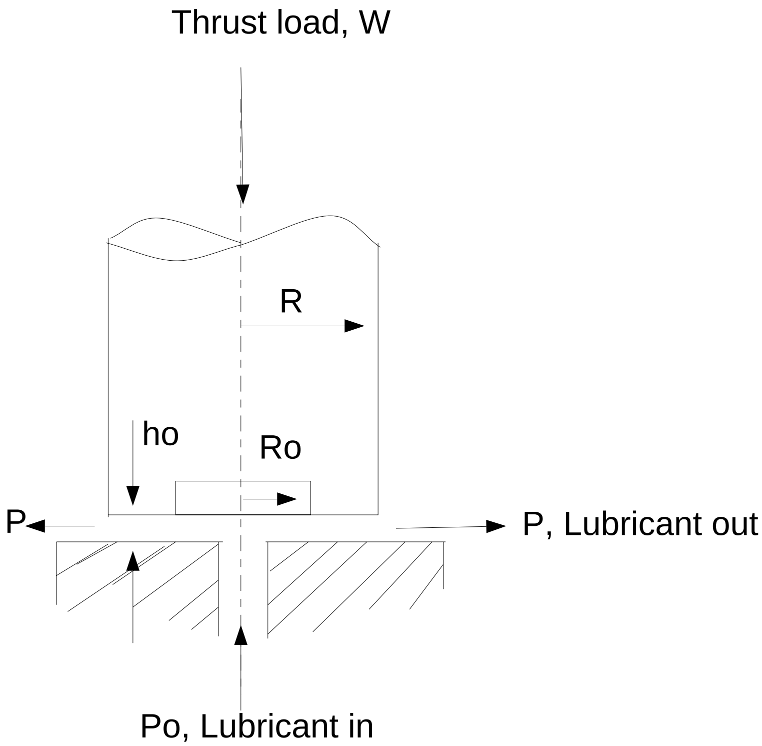

Figure 11.3: Thrust bearing

bearing.

$$Q = -\frac{dP 2\pi r h_o^3}{12\mu dr} \tag{11.10}$$

The negative sign in front of the pressure term is used because of the pressure drop. By solving for the presure term in equation 11.10, integrating, and solving for the constants of integration an overall expression for pressure is obtained. Equation 11.11 then describes the pressure for the thrust bearing at any radius, r.

$$p = \frac{6\mu Q}{\pi h_o^3} \ln \frac{R}{r} \tag{11.11}$$

By setting $r = R_o$ the required inlet pressure can be obtained. From the above equation the pressure necessary to avoid contact between the surfaces of the bearing is derived and expressed in equation 11.13.

$$P_o = \frac{6\mu Q}{\pi h_o^3} \ln \frac{R}{R_o} \tag{11.12}$$

The required “flow needed to maintain a predetermined film thickness” is derived from equation 11.13. By solving for Q the flow is given by,

$$Q = \frac{P_o \pi h_o^3}{6\mu \ln(R/R_o)} \tag{11.13}$$

The load carrying capacity can be obtained by an integration. The total load carrying capacity will be the sum of the forces exerted on the area of the recess by inlet pressure, P_o, and by the variable pressure p acting on the rest of the area of the bearing. The load carrying capacity is then:

$$W = P_o(\pi R_o^2) + \int_{R_o}^{R} p(2\pi r dr) \tag{11.14}$$

By integrating and solving for the constants of integration equation 11.15 expresses the load carrying capacity in terms of the inlet pressure, recess radius, and shaft radius.

$$W = \frac{P_o \pi}{2} \frac{R^2 - R_o^2}{\ln(R/R_o)} \tag{11.15}$$

Chapter 12

Adiabatic Engine

12.1 Introduction

An adiabatic process is one in which there is no heat added or removed from an isolated system. Heat is not transferred into or out of the system. The amount of work done by the process is therefore equal to the total change in energy. In an internal combustion engine the engine is the system. There is work done on the system and by the system. There is also heat transfer from the engine to the environment, through the coolant system. A system where the adiabatic process is employed to a certain extent is the adiabatic engine. In theory the adiabatic engine has no heat loss. The change in energy for the system, which is the diesel engine, is due to work done by the engine and work done on the engine. Some advantages of the adiabatic engine are described below.

- The removal of cooling water along with the radiator, fan, and water pump have made the adiabatic engine more cost effective.

- The increase in temperature due to the insulating ceramic material has increased the fuel ecocnomy.

- Reductions in NO_x, unburned hydrocarbons, and carbon monoxide is also expected.

- The density of ceramics is lower than that of metals so the new engine is more lightweight therefore increasing fuel economy.

12.2 Adiabatic Diesel Engine

In practice it is impossible to have a 100% adiabatic engine. At best the engine can reach 50-60% of adiabatic with advanced ceramics. In many cases the adiabatic engine is called the low heat rejection engine (LHRE), which more accurately describes the technology available today. As described earlier in an adiabatic engine there is no heat added or rejected. Theoretically one would like to make use of the exhaust that is released by the engine. The use of a turbocharger idealizes the no heat rejected concept by taking the high temperature exhaust and transferring work to the engine.

The adiabatic diesel engine with waste heat utilization is a very rewarding concept since there is energy being extracte from the hot exhaust gases. The brake fuel consumption is reduced because of the following:

- Insulation of the combustion chamber, exhaust and intake ports, and the exhaust manifolds.
- Elimination of the cooling system and the associated parts.
- Waste exhaust heat utilization by turbocharging.

The advantages of using an adiabatic turbocharged diesel engine are:

- Reduced fuel consumption
- Reduced emissions and white smoke
- Multi-fuel capability
- Reduced noise level
- Improved reliability and reduced maintenance
- Longer life
- Smaller installed volume
- Lighter weight

12.2.1 Engine Operating Environment

Figure 12.1 shows a *p-v* diagram demonstrating the difference between a water cooled and an uncooled engine. In the adiabatic engine the pressure and temperature is greater than that of the cooled engine. With greater temperature the engine thermal efficiency of the engine increases. With greater pressure the brake, or mechanical efficiency, increses due to the greater amount of force exerted on the piston and hence on the crankshaft.

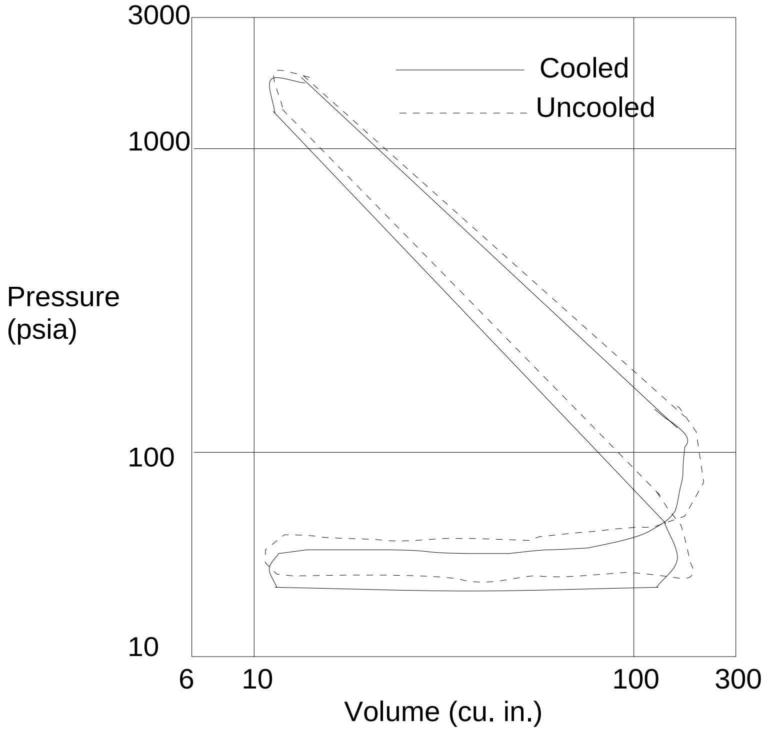

Figure 12.1: p-v diagram

Figure 12.2 shows the temperature of the engine block wall versus the crank angle of the engine. As the zirconia insulation thickness increases, the surface temperature increases. This is due to the fact that there is less heat transfer to the surroundings since the zirconia provides the insulation. The first curve is a curve of an iron wall. This provides a reference to measure the effectiveness of the ceramic insulation. With a thickness of 0.1 *in* of ceramic insulation temperatures can reach as high as 1250° F. When the temperature increases the gas mixture is able to combust faster than if it was at a lower temperature. The need for higher pressures is also reduced by increasing the temperature. With these such high temperatures the thermal efficiency increases.

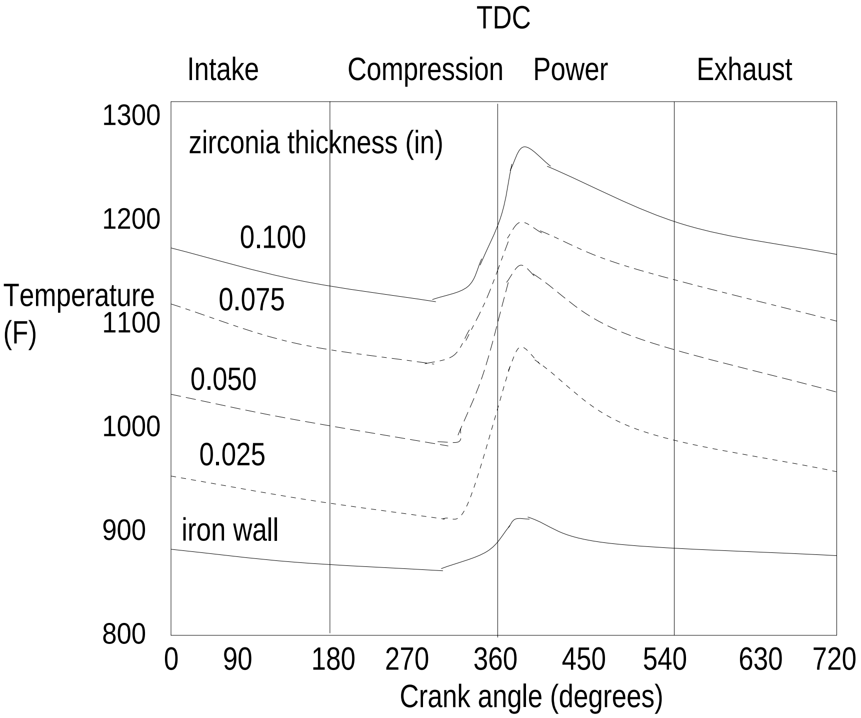

Figure 12.2: Temperature vs. Crank angle

12.2.2 Materials

Choosing the proper material for the adiabatic engine is not an easy task. With such high temperatures ceramics seem to offer the best desired properties. Super-alloy metallic design requires a variation of selected materials. Such materials are molybdenum, chromium, nickel, titanium, etc. These materials however do not provide the high resistance to temperature that ceramics provide. The availability of ceramic materials greatly influenced the selection of materials for the adiabatic engine. Figure 12.3 shows how the mechanical properties of alloys weaken as temperature increases. When the engine reaches such temperatures up to 1000°C the alloys used for the engine begin to weaken. Ceramics are best for the adiabatic engine since the strength is high at extremely overnecessary temperatures. The temperatures at which ceramics can be used for in the adiabatic engine range up to 1400°C. The design temperature for the engine is not even this high. Table 1 describes three basic materials considered for the design of the adiabatic engine. The materials are broken down into metals, and two types of ceramics. The first type of ceramic are the high performance ceramics. These ceramics have high temperature resistance but high thermal conductivities. The other type of ceramics are the glass ceramics. These have low thermal conductivities but they also posess low temperature resistance. Glass ceramics are considered for their insulation effectiveness. Some common high performance ceramics such as silicon nitride (Si_3N_4) and silicon carbide (SiC) lack insulation properties. Zirconia and glass ceramics lack high temperature strength. There is a need for a ceramic material that can have both high temperature strength and insulation properties. Until an almost perfect material can be designed for the adiabatic engine different materials will be tested and designed.

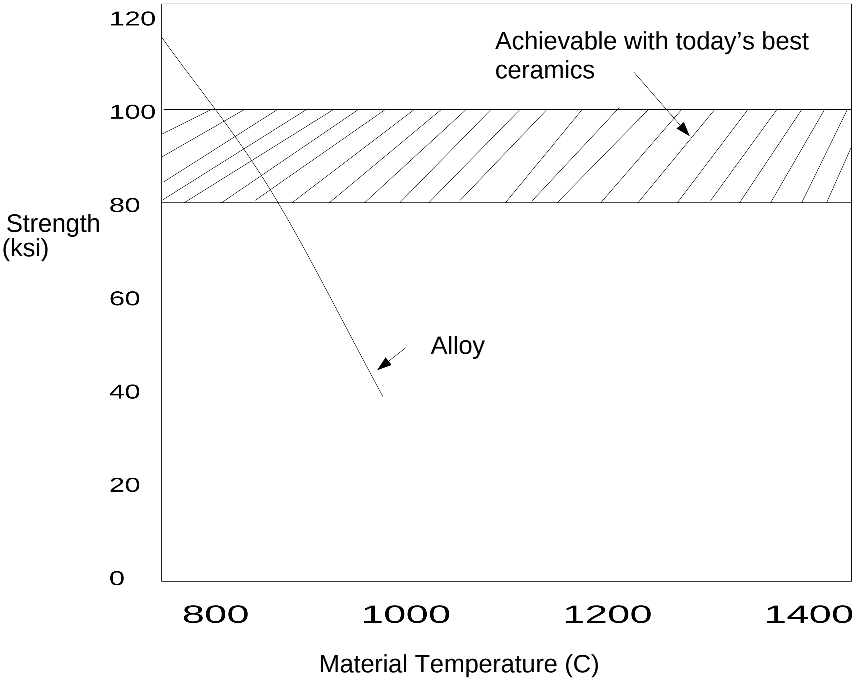

Figure 12.3: Strength vs Material Temperature

Material	Thermal Conductivity	Thermal Exp. Coefficient	Tensile Strength	MAX Design Temperature
Metallic	0.054	14.4	30	1000
High Perf. Ceramics	0.043	3.2	40	1350
Glass Ceramics	0.004	0.7	9	1000

When a ceramic material does become available, it will have to sustain long term durability. Aging properties of materials are highly critical. The "important long term properties of materials that should be determined in any of these promising materials are" shown below. (Evans, p.149)

- phase change
- high temperature creep
- oxidation and wear
- corrosion and deposits

12.2.3 Problems With the Adiabatic Engine

Some of the problems with the production of the adiabatic engine are as follow:

- high temperature tribology
- insulating ceramics
- low cost fabrication
- low cost finishing and machining
- quality control methods

Chapter 13

Chemical and Phase Equilibrium

13.1 Introduction

This chapter deals with chemical and phase equilibrium of pure substances and mixtures. The chemical equilibrium of a reaction in a single phase is considered. The discussion on hand deals with ideal gas mixtures. Phase equilibrium is also considered. Gibbs function and its uses is also discussed. The use of Gibbs function and chemical potential to solve for equilibrium constants in one and two phase equilibrium reactions is discussed.

13.2 Equilibrium Criteria

A system is in thermodynamic equilibrium when it is isolated from its sorroundings and there are no "observable macroscopically observable changes." (Moran, p.684) In order to have equilibrium the temperature needs to be constant throughout the system. If the system is not at a constant temperature then there will be a variance in temperature. When there is a temperature variance there is heat transfer within the system. So even if the system is isolated there can be heat transfer which will make the system not be in equilibrium. Another way for the system not to be in equilibrium is if it has unbalanced forces. So the system can be in thermal and mechanical equilibrium but there still might be the possibility that it is not in complete equilibrium. The process of a chemical reaction, a transfer of mass, or both might still make the system not in equilibrium. In this section criteria are used to decide whether or not a system is in equilibrium or not. These criteria are "developed using the conservation of energy principle and the second law of thermodynamics." (Moran, p.684)

13.3 Gibbs Function

When a system is said to be in equilibrium at constant temperature and pressure the Gibbs function has a value of zero. The Gibbs function is given in equation 13.1.

$$G = H - TS = U + pV - TS \tag{13.1}$$

By differentiating and solving for the Gibbs function differential form the following equation is obtained.

$$dG - Vdp + SdT = -(TdS - dU - pdV) \tag{13.2}$$

Using the energy balance in differential form, or the fist law of thermodynamics, an expression for the right side of equation 13.2 can be obtained. Using the first and second law of thermodynamics the following expression can be obtained:

$$TdS - dU - pdV \geq 0 \tag{13.3}$$

Substituting equation 13.3 into eqaution 13.2 gives the following expression for the differential form of the Gibb equation.

$$dG - Vdp + SdT \leq 0 \tag{13.4}$$

Any process taking place at a constant pressure and temparature will have a zero value for any change in temperature and pressure. Equation 13.4 defines the Gibbs function for a system at fixed temperature and pressure. The Gibbs function for a reversible process is given by equation 13.5.

$$dG]_{T,p} \leq 0 \tag{13.5}$$

The above expression expresses the equilibrium of a system as decresing with an irreversibl eprocess. The lower the Gibbs function the more in equilibrium the system is. Therefore when

$$dG]_{T,p} = 0 \tag{13.6}$$

the system is said to be at equilibrium.

13.4 Chemical Potential

The chemical potential is an expression formulated from Gibbs function. Any extensive property of a single phase, single component system is a function of two independent intensive properties and the size of the system. The two indepemdent intensive properties are pressure and temperature. The size of the system is defined by the number of moles. For a single phase multi-component system the Gibbs function is expressed through equation 13.7.

$$G = G(T, p, n_1, n_2, ..., n_j) \tag{13.7}$$

If each mole number is multiplied by alpha, α, equation 13.7 can be differentiated with respect to alpha holding temperature, pressure, and the mole number fixed. If a value of *one* is substituted for alpha then an expression for the Gibbs function in terms of a *chemical potential* is obtained.

$$G = \Sigma_{i=1}^{j} n_i (\frac{\partial G}{\partial n_i})_{T,p,n_l} \tag{13.8}$$

The partial derivatives in equation 13.21 are given the name chemical potential. The chemical potential is defined in equation 13.9.

$$\mu_i = (\frac{\partial G}{\partial n_i})_{T,p,n_l} \tag{13.9}$$

Through thermodynamic substitutions it can be shown that the chemical potential can be obtained for an ideal gas. The chemical potential for an ideal gas is shown in equation 13.10,

$$\mu_i = \bar{g}_i^o + \bar{R}T \ln \frac{y_i p}{p_{ref}} \tag{13.10}$$

where $\bar{g}_i^o$ is the Gibbs function of component i, evaluated at temperature T, pressure p, reference pressure of 1 atm, and mole fraction y_i. Therefore the relationship between Gibbs function and the chemical potential is expressed as follow:

$$G = n\mu \tag{13.11}$$

13.5 Chemical Equilibrium

13.5.1 Equation of Reaction Equilibrium

The purpose of determining the chemical equilibrium is to "establish the composition present at equilibrium for a specified temperature and pressure." (Moran, p.689) An important parameter for dtermining the equilibrium composition is the *equilibrium constant*. If we consider the following reaction between a gaseous mixture of hydrogen oxygen to produce water, an exppresion for the equation of equilibrium can b eobtained. The reaction under consideration is shown in equation 13.12

$$1H_2O + \frac{1}{2}O_2 \rightarrow 1H_2O \tag{13.12}$$

Using Gibbs equation an expression of the mixture between two states having the same temperature and pressure, but compositions that differ infinitesimally is equal to:

$$dG]_{T,p} = \mu_{H_2}dn_{H_2} + \mu_{O_2}dn_{O_2} + \mu_{H_2O}dn_{H_2O} \tag{13.13}$$

From equation 13.12 it can be seen that for every mole of H_2 and $\frac{1}{2}O_2$ there will be a mole of water. Therefore using the Gibbs equation and knowing that for equilibrium the Gibbs function needs to be zero the equation of reaction equilibrium for the above reaction is shown in equation 13.14.

$$1\mu_{H_2} + \frac{1}{2}\mu_{O_2} = 1\mu_{H_2O} \tag{13.14}$$

It is necessary to develop an equation of reaction equilibrium for a general case. If a chemical reaction is given by the following general equation,

$$\nu_A A + \nu_B B \rightarrow \nu_C C + \nu_D D \tag{13.15}$$

then the relationships bewteen the indiviual reactanst and products are:

$$\frac{-dn_A}{\nu_A} = \frac{-dn_B}{\nu_B} = \frac{dn_C}{\nu_C} = \frac{dn_D}{\nu_D} \tag{13.16}$$

Equation 13.16 is a definition of the ratios of a reaction. If there is an increase in the moles concentration of component D then there has to be a reduction in the mole concentration of component A or B. Using equation 13.11, the relationships between components of equation 13.16, and the equilibrium definition for Gibbs function the it equation of reaction equilibrium is:

$$\nu_A\mu_A + \nu_B\mu_B = \nu_C\mu_C + \nu_D\mu_D \tag{13.17}$$

13.6 Phase Equilibrium

13.6.1 Equilibrium Between Two Phases Of A Pure Substance

An expression for equilibrium of a two phase system can be obtained from Gibbs function, or equation 13.11. For a system in equilibrium each phase is at the same temperature and pressure and the Gibbs function for the system is:

$$G = n'\bar{g}'(T,p) + n''\bar{g}''(T,p) \tag{13.18}$$

The primes denote the phases one and two respectively. In order to have equilibrium both of the components of the above equation need to be in equilibrium. If either one or the other component increases in its amount present then the other needs to be compensated by a decrease in its amount to have equilibrium. By differentiating and keeping temperature and pressure constant an expression for the differential of G is obtaoined.

$$dG]_{T,p} = (\bar{g}' - \bar{g}'')dn' \tag{13.19}$$

Note that in equation 13.19 the subtitution *dn”=-dn’* has been substituted. At equilibrium the $dG]_{T,p}$=0, so

$$\bar{g}' = \bar{g}'' \tag{13.20}$$

Equation 13.19 is the counterpart of phase equilibrium to the Gibbs function in chemical equilibrium.

Through equation 13.20 the Clapeyron equation can be obtained. For two phases at equilibrium the variations in pressure are related to the variations in temperature by $p = p_{sat}(T)$. By differentiating equation 13.20 with respect to temeperature gives the following equation,

$$\frac{\partial\bar{g}'}{\partial T})_p + \frac{\partial\bar{g}'}{\partial p})_T\frac{dp_{sat}}{dT} = \frac{\partial\bar{g}''}{\partial T})_p + \frac{\partial\bar{g}''}{\partial p})_T\frac{dp_{sat}}{dT} \tag{13.21}$$

By substituting the following equations into equation 13.21 the Clapeyron equation is obtained. From the Maxwell relations

$$v = \frac{\partial\bar{g}}{\partial p})_T \tag{13.22}$$

and

$$-s = \frac{\partial \bar{g}}{\partial T})_p \tag{13.23}$$

The Clapeyron equation is then,

$$\frac{dp_{sat}}{dT} = \frac{1}{T}(\frac{\bar{h}'' - \bar{h}'}{\bar{v}'' - \bar{v}'}) \tag{13.24}$$

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