ESO201A

Lecture#38

(Class Lecture)

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By

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Gas Power Cycles

Two types of gas power cycles will be discussed. They are Otto and Diesel cycles. The processes in the cycle take place in a piston-cylinder device and hence, the analysis is based on closed system. In both engines, energy is provided by burning a fuel within the system boundaries. That is, they are internal combustion engines or in short, I.C. engines. Because of this combustion process, the composition of the working fluid changes from air and fuel to combustion products during the course of the cycle. However, considering that air is predominantly nitrogen that undergoes hardly any chemical reactions in the combustion chamber, the working fluid closely resembles air at all times.

It may be noted that actual cycles are open cycles whereas the ideal cycles are closed cycles.

Air-standard Assumptions

The actual gas power cycles are rather complex. To reduce the analysis to a manageable level, the following approximations are made. These are known as the air-standard assumptions.

1. The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.

2. All the processes that make up the cycle are internally reversible.

3. The combustion process is replaced by a heat-addition process from an external source.

4. The exhaust process is replaced by a heat-rejection process that restores the fluid to its initial state.

5. Air has constant specific heats whose values are taken at 25°C. With this the air-standard assumptions are called cold-air-standard assumptions.

A cycle for which the air-standard assumptions are applicable is frequently referred to as an air-standard cycle.

The air-standard assumptions previously stated provide considerable simplifications in the analysis without significantly deviating from the actual cycles. This simplified model enables us to study qualitatively the influence of various parameters on the performance of the actual engines.

An Overview of Reciprocating Engines

The basic components of a reciprocating engine are shown in Fig. 1. The piston reciprocates in the cylinder between two fixed positions called the top dead centre (TDC) which is the position of the piston when it forms the smallest volume in the cylinder and the bottom dead centre (BDC), the position of the piston when it forms the largest volume in the cylinder.

The distance between the TDC and the BDC is the largest distance that the piston can travel in one direction, and it is called the stroke of the engine. The diameter of the piston is called the bore. The air (in the case of compression-ignition engine) and air-fuel mixture (in the case of spark-ignition engine) is drawn into the cylinder through the intake valve, and the combustion products are expelled from the cylinder through the exhaust valve.

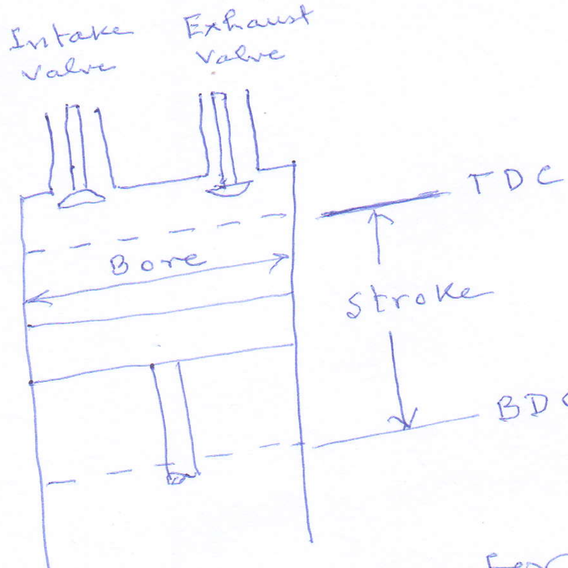
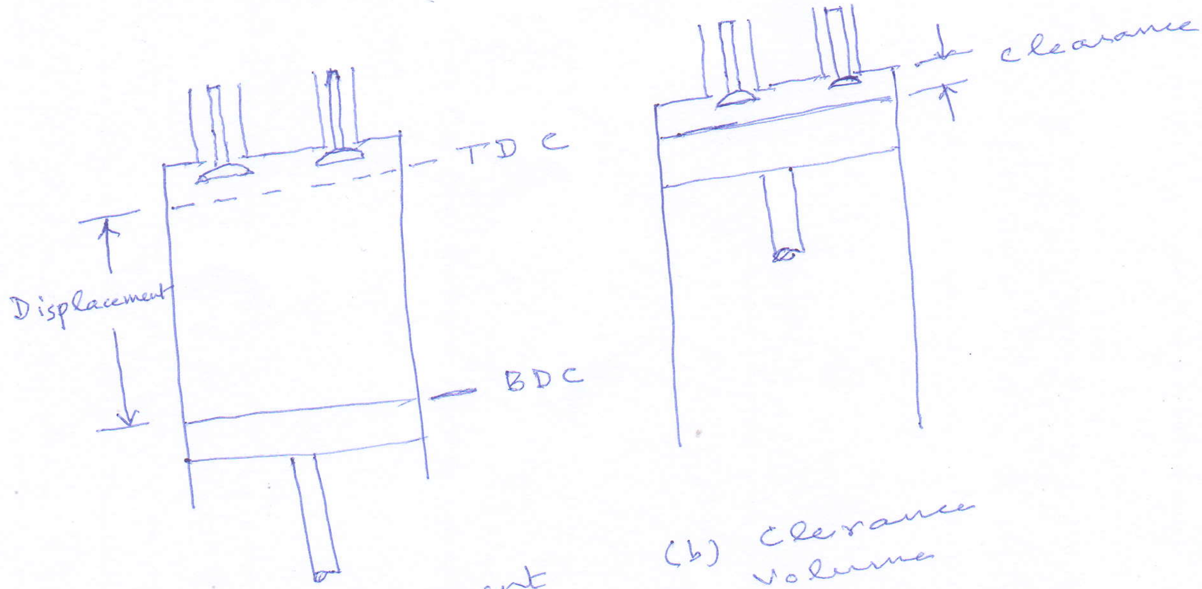


Fig. 1 Nomenclature for reciprocating engines

The minimum volume formed in the cylinder when the piston is at TDC is called the <u>clearance volume</u> (Fig. 2). The volume displaced by the piston as it moves between TDC and BDC is called the <u>displacement volume</u>. The ratio of the maximum volume formed in the cylinder to the minimum (clearance) volume is called the <u>compression ratio</u> r of the engine (eq. (1)).

$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}} \qquad (1)$$



(a) Displacement volume

(b) Clearance volume

Fig. 2 Displacement and clearance volumes of a reciprocating engine

Another term frequently used in conjunction with reciprocating engines is the mean effective pressure (MEP). It is an equivalent pressure that, if it acted on the piston during the entire power stroke, would produce the same amount of net work produced in an actual cycle (Fig. 3).

$$W_{net} = MEP \times \text{Piston area} \times \text{stroke}$$

$$= MEP \times \text{Displacement volume}$$

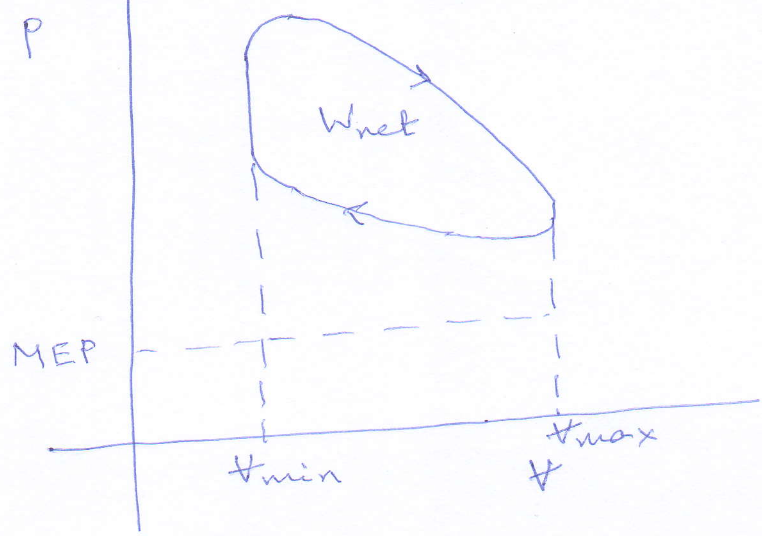
or

$$MEP = \frac{W_{net}}{V_{max} - V_{min}}$$

$$= \frac{w_{net}}{v_{max} - v_{min}} \qquad (2)$$

The mean effective pressure is a measure of net work per cycle and is used as a parameter to compare the performances of reciprocating engine of equal size (same bore and stroke).

$W_{net} = MEP\,(V_{max} - V_{min})$



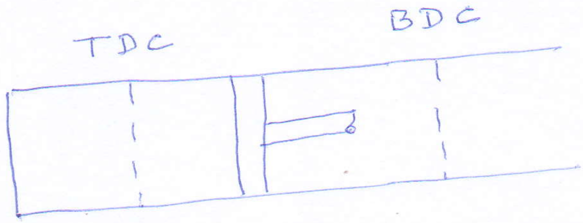


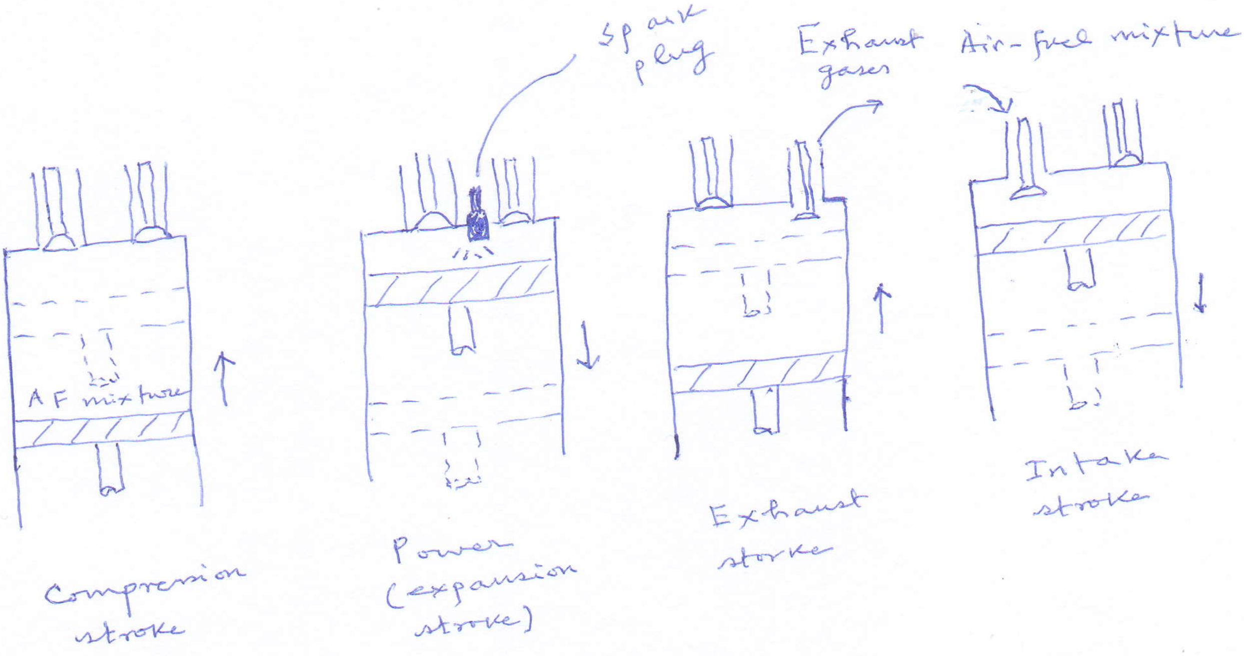
Fig. 3 The net work output of a cycle is equivalent to the product of the mean effective pressure and the displacement volume

Reciprocating engines are classified as spark-ignition (SI) engines or compression-ignition (CI) engines, depending on how the combustion process in the cylinder is initiated. In SI engines, the combustion of the air-fuel mixture is initiated by a spark plug. In CI engines, the fuel is sprayed into the compressed air and as a result the air-fuel mixture is self-ignited since the temperature of the mixture rises above its self-ignition temperature.

Next we discuss Otto and Diesel cycles, which are the ideal cycles for the SI and CI reciprocating engines, respectively.

Otto Cycle: The Ideal Cycle for Spark-Ignition Engines

The Otto cycle is the ideal cycle for spark-ignition reciprocating engines. It was developed by Nicolaus A. Otto in 1876 in Germany. A schematic of each stroke as well as P-v diagram for an actual four-stroke SI engine is given in Fig. 4



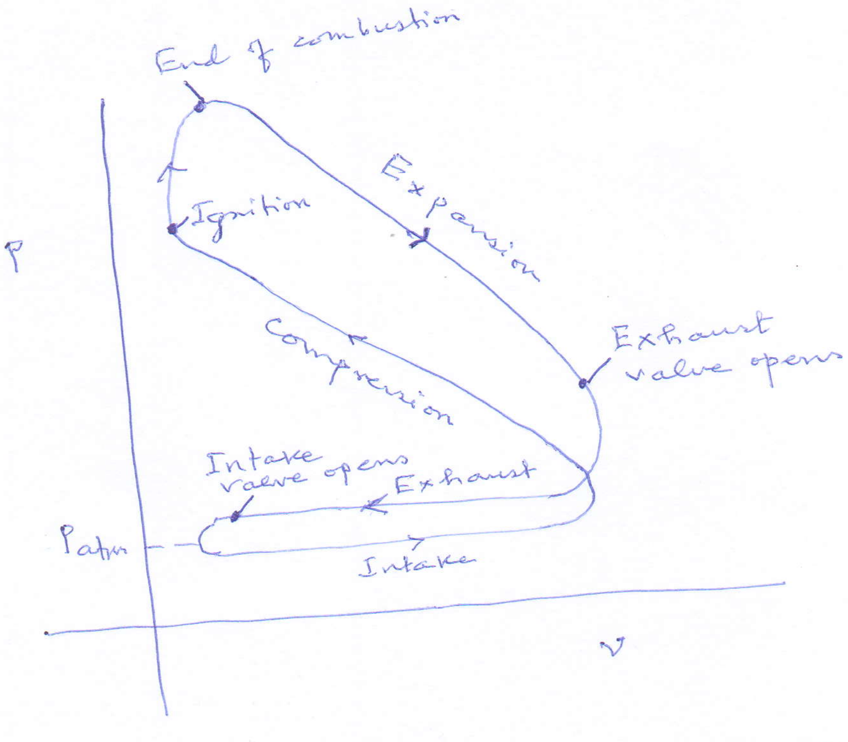


Fig. 4 Actual 4-stroke SI engine and the P-v diagram

Initially, both the intake and exhaust valves are closed, and the piston is at its lowest position (BDC). During the compression stroke, the piston moves upward, compressing the air-fuel mixture. Shortly before the piston reaches its highest position (TDC), the spark plug fires and the mixture ignites, increasing the pressure and temperature of the system. The high-pressure gases force the piston down, which in turn forces the crankshaft to rotate, producing a useful work output during the expansion or power stroke. At the end of this stroke, the piston is at its lowest position and the cylinder is filled with combustion products. Now the piston moves upward one more time, purging the exhaust gases through the exhaust valve (the exhaust stroke), and down a second time, drawing in fresh air-fuel mixture through the intake valve (the intake stroke). Notice that the pressure in the cylinder is slightly above the atmospheric value during the exhaust stroke and slightly below during the intake stroke.

The thermodynamic analysis of the actual four-stroke cycle is not a simple task. However, the analysis can be simplified significantly if the air-standard assumptions are utilized. The resulting cycle, which closely resembles the actual operating conditions, is the ideal Otto cycle. It consists of four internally reversible processes:

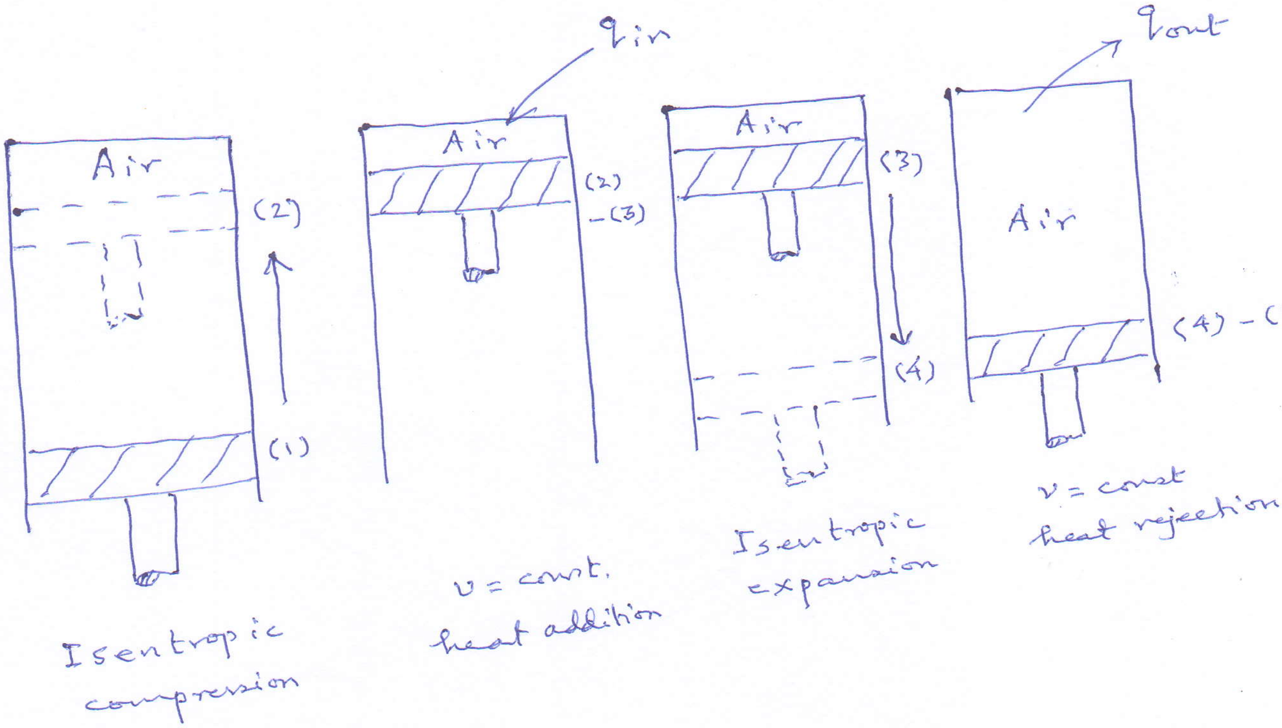
1 - 2 Isentropic compression

2 - 3 Constant-volume heat addition

3 - 4 Isentropic expansion

4 - 1 Constant - volume heat rejection

The execution of the Otto cycle in a piston-cylinder device together with a P - v diagram is illustrated in Fig. 6. The T - s diagram of the Otto cycle is given in Fig. 7.



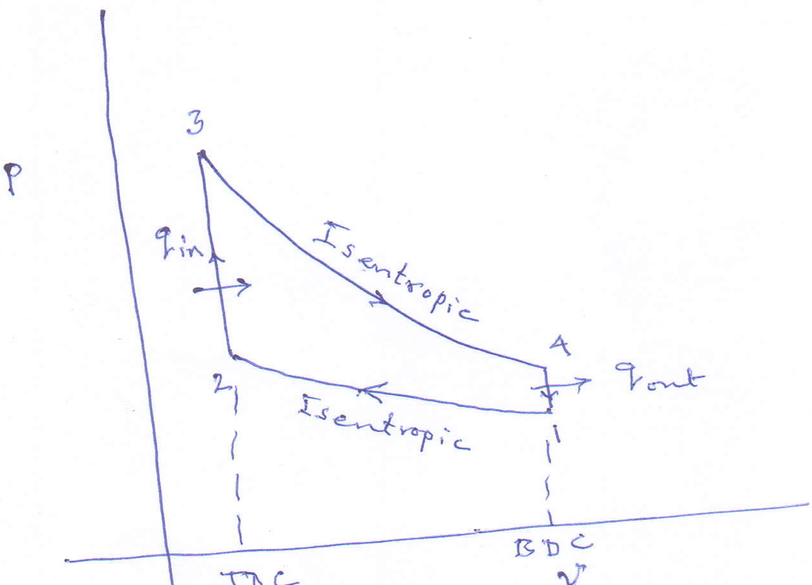


Fig. 6 Ideal 4-stroke SI engine and the p-v diagram

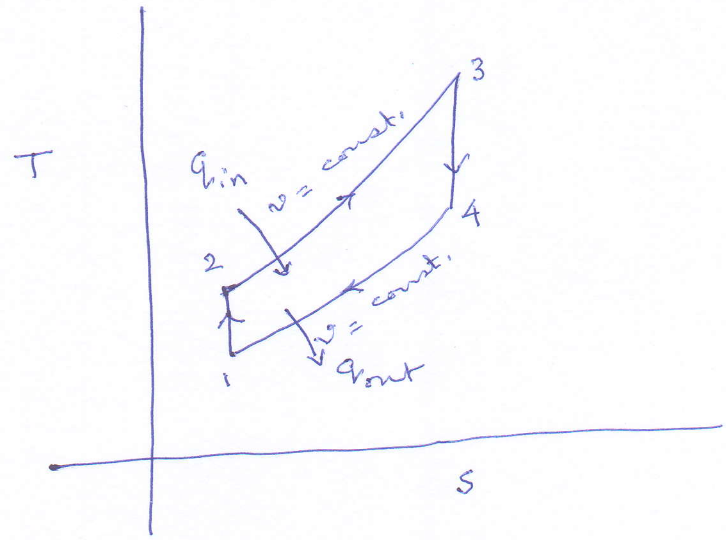


Fig. 7 T-s diagram of the ideal Otto cycle

The Otto cycle is executed in a closed system, and disregarding the changes in kinetic and potential energies, the energy balance for any of the processes is expressed, on a unit-mass basis, as

$$(q_{in} - q_{out}) + (w_{out} - w_{in}) = \Delta u \qquad (3)$$

No work is involved during the two heat transfer processes since both take place at constant volume.

Therefore, heat transfer to and from the working fluid can be expressed as

$$q_{in} = u_3 - u_2 = c_v (T_3 - T_2) \qquad (4)$$

and

$$q_{out} = u_4 - u_1 = c_v (T_4 - T_1) \qquad (5)$$

Then the thermal efficiency of the ideal Otto cycle under the cold air standard assumptions becomes

$$\eta_{th,\,Otto} = \frac{w_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

$$= 1 - \frac{(T_4 - T_1)\,c_v}{(T_3 - T_2)\,c_v}$$

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

$$= 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} \qquad (5a)$$

Processes 1-2 and 3-4 are isentropic, and $v_2 = v_3$ and $v_4 = v_1$. Thus,

Since $P_1 v_1 = R T_1$

$$P_2 v_2 = R T_2$$

$$P_1 v_1^K = P_2 v_2^K$$

$$\Rightarrow \frac{P_1}{P_2} = \left(\frac{v_2}{v_1}\right)^K$$

$$\frac{P_1}{P_2}\frac{v_1}{v_2} = \frac{T_1}{T_2}$$

$$\Rightarrow \left(\frac{v_2}{v_1}\right)^K \left(\frac{v_1}{v_2}\right) = \frac{T_1}{T_2}$$

$$\Rightarrow \frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{K-1} = \left(\frac{v_3}{v_4}\right)^{K-1} \quad (6)$$

Since

$$P_3 v_3 = R T_3$$

$$P_4 v_4 = R T_4$$

$$P_3 v_3^K = P_4 v_4^K \quad (7)$$

$$\Rightarrow \frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{K-1}$$

From eqs. (6) and (7),

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{K-1} = \left(\frac{v_3}{v_4}\right)^{K-1} = \frac{T_4}{T_3} \quad (8)$$

$$\Rightarrow \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Now,

$$\eta_{th} = 1 - \frac{T_1}{T_2}\left(\frac{T_4}{T_1} - 1\right) \Big/ \left(\frac{T_3}{T_2} - 1\right)$$

$$= 1 - \left(\frac{v_3}{v_4}\right)^{K-1} \left(\frac{T_3}{T_2} - 1\right) \Big/ \left(\frac{T_3}{T_2} - 1\right)$$

$$= 1 - \left(\frac{v_3}{v_4}\right)^{K-1} = 1 - \left(\frac{v_2}{v_1}\right)^{K-1} \quad (9)$$

$$= 1 - \frac{1}{(v_1/v_2)^{K-1}} = \boxed{1 - \frac{1}{r^{K-1}}}$$

where $$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2} \qquad (10)$$

is the compression ratio and k is the specific heat ratio c_p/c_v. $k = 1.4$ for air at $T = 298$ K.

Equation (9) reveals that for a fixed k, thermal efficiency increases with r. Figure 8 shows the plot of η_{th} vs. r ($k = 1.4$).

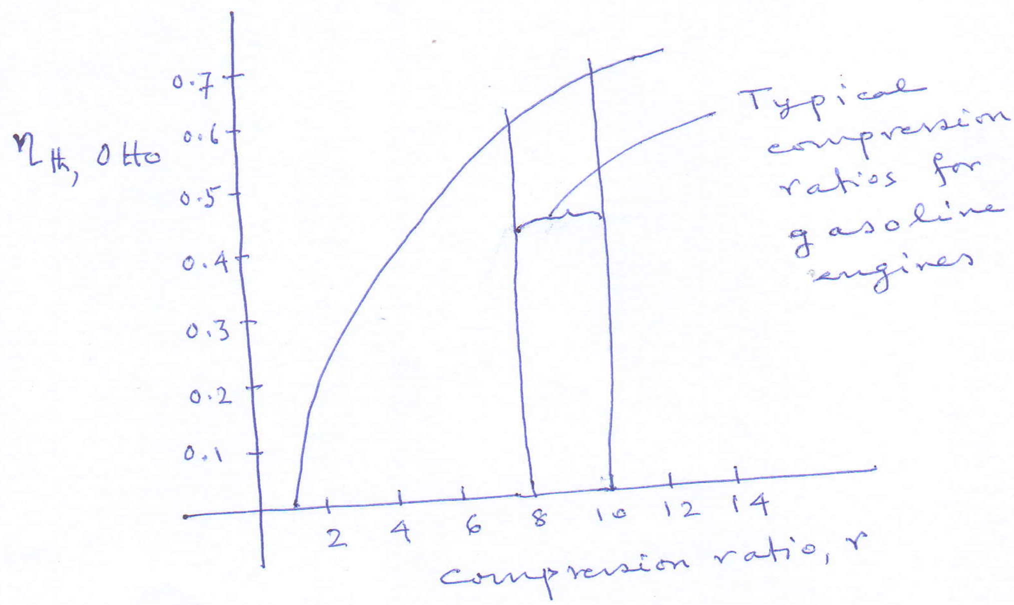


Fig. 8 Thermal efficiency of the ideal Otto cycle as a function of compression ratio ($k = 1.4$).

For a given compression ratio, the thermal efficiency of an actual spark-ignition engine is less than that of an ideal Otto cycle because of the irreversibilities, such as friction, and other factors such as incomplete combustion.

We can observe from Fig. 8 that the thermal efficiency curve is rather steep at low compression ratio but flattens out starting with around $r = 8$. Also, at high values of r, the temperature of the air-fuel mixture rises above the autoignition temperature of the fuel (the temperature at which the fuel ignites without the help of a spark) during the combustion process, causing an early and rapid burn of a fuel at some point or points ahead of flame front, followed by almost instantaneous inflammation of the end gas. This premature ignition of the fuel, called autoignition, produces an audible noise, which is called engine knock. Autoignition in spark-ignition engines cannot be tolerated because it hurts performance and cause engine damage. The requirement that autoignition not be allowed places an upper limit on the compression ratios that can be used in spark-ignition I.C. engines.

The second parameter affecting the thermal efficiency of an ideal Otto cycle is the specific heat ratio K. For a given compression ratio, an ideal Otto cycle using a monatomic gas (such as argon or helium, K = 1.667) as the working fluid will have the highest thermal efficiency. The specific heat ratio, K, and thus the thermal efficiency of the ideal Otto cycle, decreases as the molecules of the working fluid get larger (Fig. 9). At room temperature (T = 298 K) it is 1.4 for air, 1.3 for CO_2 and 1.2 for ethane. The working fluid in actual engines contains large molecules such as carbon dioxide, and the specific heat ratio decreases with temperature, which is one of the reasons that the actual cycles have lower thermal efficiencies than the ideal Otto cycle. The thermal efficiencies of actual spark-ignition engines range from about 25 to 30%.

Fig. 9 The thermal efficiency of the Otto cycle increases with the specific ratio k of the working fluid.

Diesel Cycle: The Ideal cycle for Compression-Ignition Engines

The Diesel cycle is the ideal cycle for CI reciprocating engines. In the CI engine, first developed by Rudolph Diesel in the 1890s, in Germany, the air is compressed to a temperature that is above the autoignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. See Fig. 10.

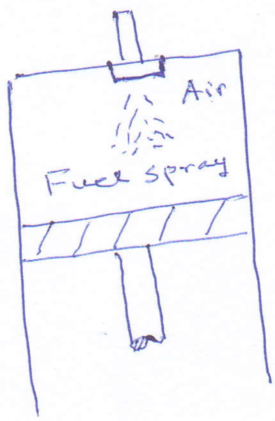
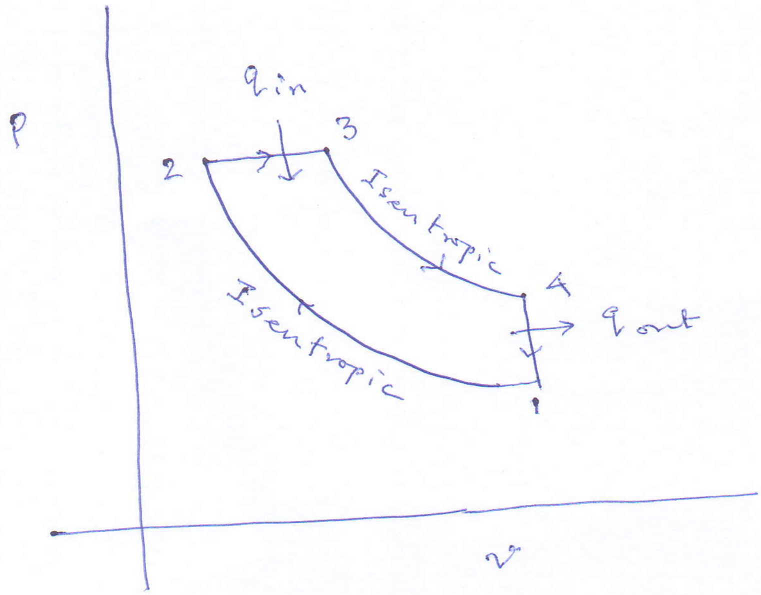


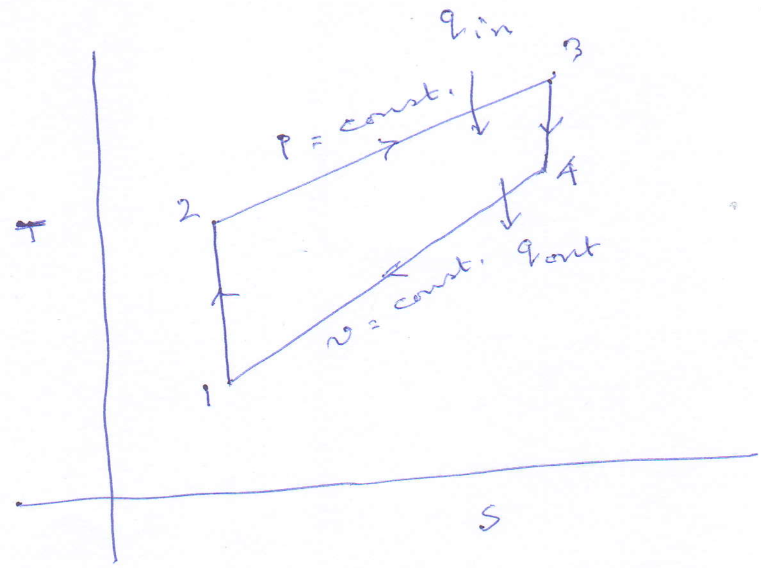
Fig. 10 The Diesel engine

In gasoline engines, a mixture of air and fuel is compressed during the compression stroke, and the compression ratios are limited by the onset of autoignition or engine knock. In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of autoignition. Therefore, diesel engines can be designed to operate at much higher compression ratios, typically between 12 to 24.

The fuel injection process in diesel engines starts when the piston approaches TDC and continues during the first part of the power stroke. Therefore, the combustion process in these engines takes place over a longer interval. Because of this longer duration, the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat addition process. The rest three processes are exactly same as those of Otto cycle. The P-v and T-s diagrams for the ideal Diesel cycle are shown in Figs. 11(a) and 11(b), respectively.



(a) P – v diagram



(b) T – S diagram

Fig. 11 P – v and T – S diagrams for the ideal Diesel cycle

Noting that the Diesel cycle is executed in a piston-cylinder device which forms a closed system, the amount of heat transferred to the working fluid at constant pressure and rejected from it at constant volume can be expressed as

$$q_{in} - w_{b,out} = u_3 - u_2$$

$$\Rightarrow q_{in} = w_{b,out} + (u_3 - u_2)$$

$$= P_2(v_3 - v_2) + (u_3 - u_2)$$

$$= (P_2 v_3 + u_3) - (P_2 v_2 + u_2)$$

$$= (P_3 v_3 + u_3) - (P_2 v_2 + u_2)$$

$$= h_3 - h_2 \qquad (11)$$

$$= c_p (T_3 - T_2)$$

and

$$-q_{out} = u_1 - u_4$$

$$\Rightarrow q_{out} = u_4 - u_1 \qquad (12)$$

$$= c_v (T_4 - T_1)$$

Then the thermal efficiency of the ideal Diesel cycle under the cold-air-standard assumption becomes

$$\eta_{th,\,Diesel} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

$$= 1 - c_v(T_4 - T_1)/c_p(T_3 - T_2)$$

$$= 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} \qquad (13)$$

Considering the process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\Rightarrow \quad T_2 = T_1 \left(\frac{v_1}{v_2}\right)^{k-1} \qquad (14)$$

$$= T_1 r^{k-1}$$

Considering the constant-pressure process 2-3, we have

$$P_2 v_2 = R T_2$$

$$P_3 v_3 = R T_3$$

Since $P_2 = P_3$,

$$\frac{v_2}{T_2} = \frac{v_3}{T_3} \qquad (15)$$

$$\Rightarrow \quad \frac{T_3}{T_2} = \frac{v_3}{v_2} = r_c$$

r_c is called the cut-off ratio, which is the ratio of the cylinder volumes after and before the combustion process.

From eq. (15),

$$T_3 = T_2\, r_c$$

Using eq. (14),

$$T_3 = T_1\, r^{k-1}\, r_c \qquad (16)$$

considering 3-4,

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

$$\Rightarrow T_4 = T_3\left(\frac{v_3}{v_2}\cdot\frac{v_2}{v_4}\right)^{k-1}$$

$$= T_3\left(\frac{v_3}{v_2}\Big/\frac{v_4}{v_2}\right)^{k-1}$$

$$= T_3\left(\frac{r_c}{r}\right)^{k-1} \qquad (17)$$

Hence, using eq. (16),

$$T_4 = T_1\, r^{k-1}\, r_c\left(\frac{r_c}{r}\right)^{k-1} \qquad (18)$$

$$= T_1\, r_c^{\,k}$$

Finally, from eq. (13),

$$\eta_{th,\,Diesel} = 1 - \frac{T_4 - T_1}{K(T_3 - T_2)}$$

$$\Rightarrow \eta_{th,\,Diesel} = 1 - \frac{1}{K}\left[\frac{T_1 r_c^{K} - T_1}{T_1 r^{K-1} r_c - r^{K-1} T_1}\right]$$

$$= 1 - \frac{1}{K}\left[\frac{T_1 (r_c^{K} - 1)}{T_1 (r^{K-1})(r_c - 1)}\right]$$

$$= 1 - \frac{1}{K}\left[\frac{r_c^{K} - 1}{(r^{K-1})(r_c - 1)}\right]$$

$$= 1 - \frac{1}{r^{K-1}}\left[\frac{r_c^{K} - 1}{K(r_c - 1)}\right] \quad (19)$$

It may be noted that the efficiency of the Diesel cycle is different from that of the Otto cycle only in the bracketed factor. This factor is always greater than unity. Hence, for given compression ratio, the Otto cycle is more efficient. In practice

The operating compression ratios of diesel engines are much higher compared to spark-ignition engines working on Otto cycle. The normal range of compression ratio for diesel engine is 12 to 24 whereas for spark-ignition engines it is 8 to 10. Due to the higher compression ratios used in diesel engines the efficiency of a diesel engine is more than that of the gasoline engine. Thermal efficiencies of large diesel engines range from about 35 to 40 percent.

Figure 12 shows the thermal efficiency of the ideal Diesel cycle as a function of compression and cut-off ratios ($k = 1.4$). As the cut-off ratio decreases, the efficiency of the Diesel cycle increases. For the limiting case of $r_c = 1$, the quantity in the brackets becomes unity, and the efficiencies of the Otto and Diesel cycles become identical.

$$\left[\ \eta_{th,\,Diesel} = 1 - \frac{1}{r^{k-1}} \lim_{r_c \to 1} \left[\frac{r_c^{k} - 1}{k(r_c - 1)}\right]\right.$$

(Using L'Hospital's rule)

$$= 1 - \frac{1}{r^{k-1}} \lim_{r_c \to 1} \frac{k r_c^{k-1}}{k(1)}$$

$$= 1 - \frac{1}{r^{k-1}} \cdot (1)^{k-1} = 1 - \frac{1}{r^{k-1}} = \eta_{th,\,Otto} \ \Big]$$

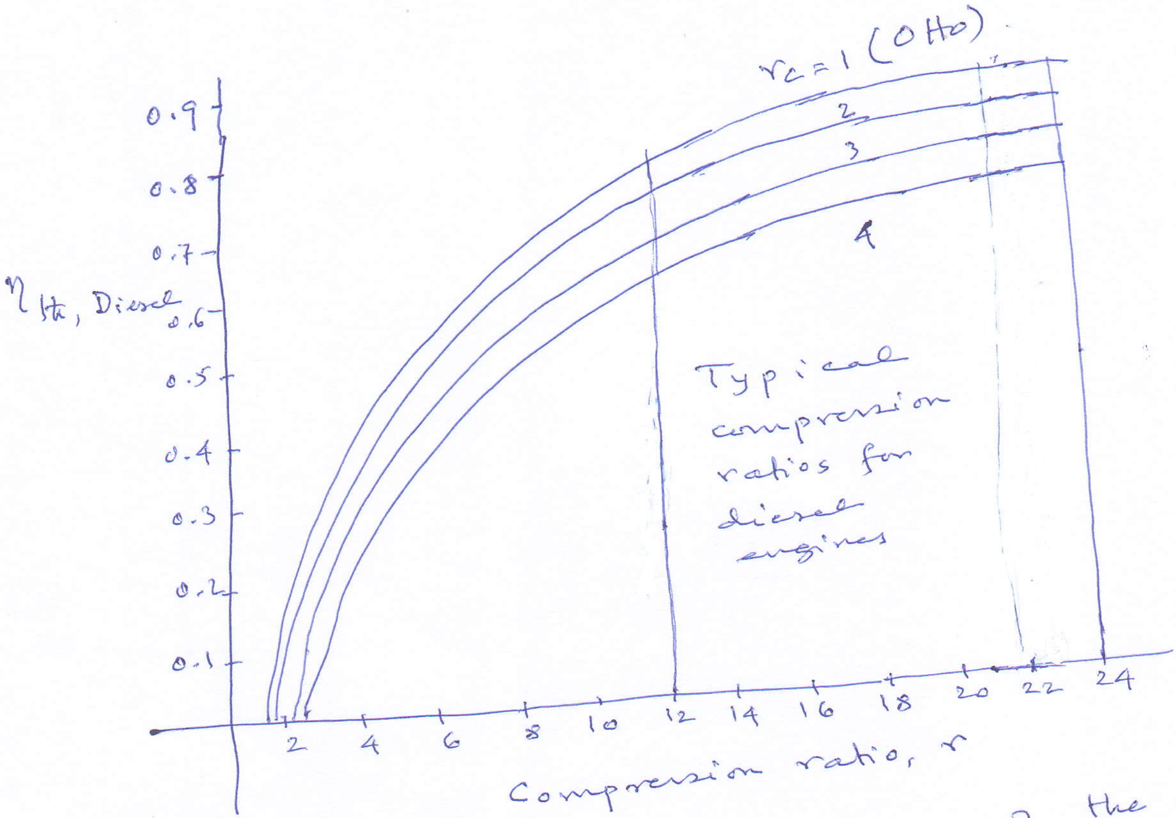


Fig. 12 Thermal efficiency of the ideal Diesel cycle as a function of compression and cut-off ratios ($k = 1.4$)

Dual cycle

In modern high-speed CI engines, fuel is injected into the combustion chamber much sooner compared to the early diesel engines. Fuel starts to ignite late in the compression stroke, and consequently part of the combustion occurs almost at constant volume. Fuel injection continues until the piston reaches the top dead centre, and combustion of the fuel keeps the pressure high well into the expansion stroke. Thus, the entire combustion process can better be modelled as the combination of constant-volume and constant-pressure processes. The ideal cycle based on this concept is called the dual cycle and P-v and T-s diagrams for it are given in Figs. 13(a) and 13(b), respectively. Note that both the Otto and the Diesel cycles can be obtained as special cases of the dual cycle. Dual cycle is a more realistic model than diesel cycle for representing modern, high-speed compression ignition engines. The efficiency of a dual cycle lies between that of the Otto cycle and the diesel cycle having the same compression ratio.

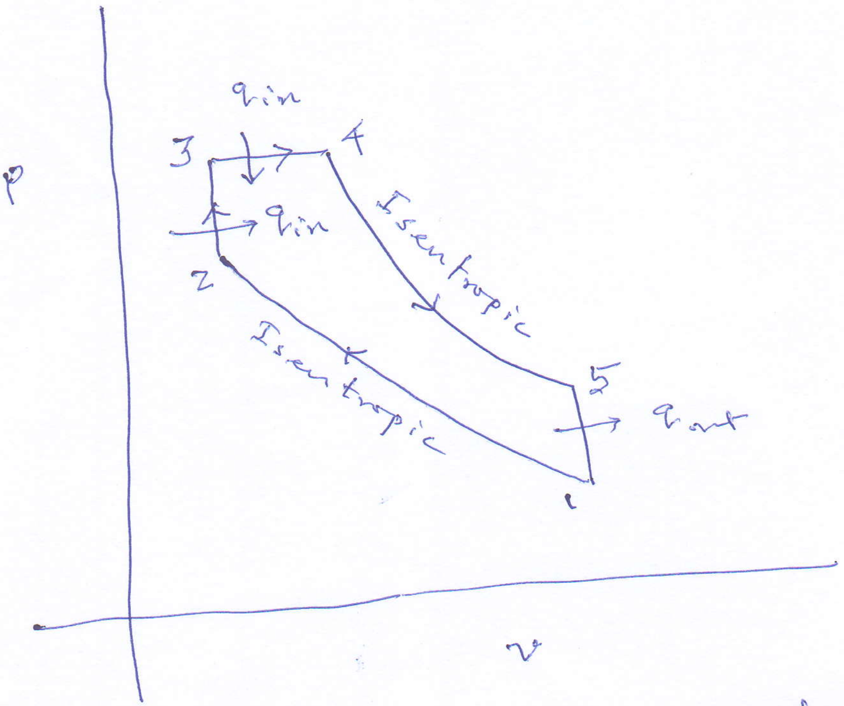


Fig. 13(a) P- v diagram of an ideal dual cycle

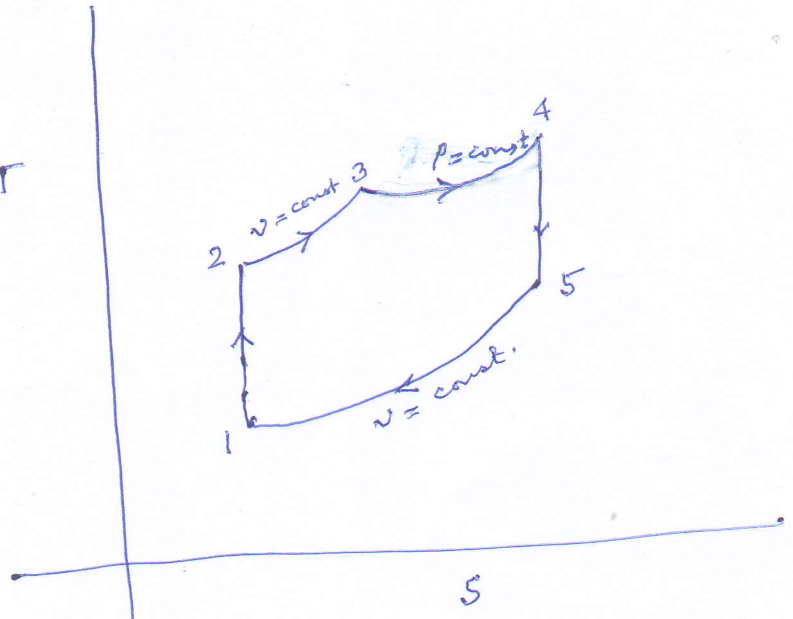


Fig. 13 (b) T- s diagram of an ideal dual cycle