6 THEORY

6.1 Introduction.

The Internal Combustion (IC) engine is still today the predominant prime mover for a host of applications such as vehicle propulsion, marine propulsion, generator sets etc. Two principal classes of IC engines exist. The first, where the combustion is initiated by a spark is termed 'Spark Ignition' (SI) and embodies Petrol and Gas engines. The second, where combustion is initiated spontaneously by virtue of the rise in temperature during the compression process is termed 'Compression Ignition' (CI), and embodies Diesel engines. The following sections describe the theory associated with IC engines and standard experiments designed to demonstrate these principals.

6.2 Criteria for engine performance.

The overall efficiency of a power plant is usually expressed as

$$\eta = \frac{W}{Q_{net,p}}$$

Q = 44 000 KJ/Kg (1)
lower heating value (H_2O in vapor form at engine exit)

Where W is the net power output per unit mass of fuel and $Q_{net,p}$ is the net calorific value of the fuel at constant pressure. The net output of an IC engine is called the *brake power* (b.p.), and for this reason the overall efficiency is termed the *brake thermal efficiency* , η_b. If the brake power is expressed in kW, the fuel flow m_f in kg/s, and $Q_{net,p}$ in kJ/kg, then

$$\eta_b = \frac{b.p.}{(m_f \times Q_{net,p})} \qquad (2)$$

The *specific fuel consumption* (s.f.c) is frequently used as an alternative criterion of performance and is defined as the rate of fuel consumption per kW of brake power and is expressed as follows

$$s.f.c = \frac{m_f}{b.p.} \qquad (3)$$

As can be seen, for any given fuel the s.f.c. is inversely proportional to η_b. Most common hydrocarbon fuels have very similar calorific values, and the s.f.c. can therefore be used when comparing efficiency of engines using different fuels.

In IC engines, an appreciable part of the losses is due to mechanical friction, and it is informative if the thermodynamic and friction losses are separated. The overall efficiency is therefore analysed as the product of the *indicated thermal efficiency* η_i and the *mechanical efficiency* η_m, as follows

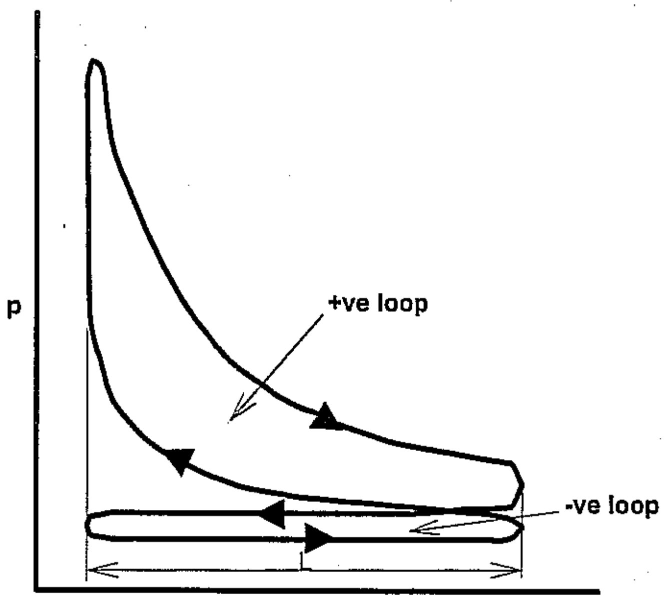
$$\eta_b = \left(\frac{i.p.}{m \times Q_{net,p}}\right) \bullet \left(\frac{b.p}{i.p}\right) = \eta_i \times \eta_m \qquad (4)$$

The *indicated power* (i.p.) is the actual rate of work done by the working fluid on the piston, and the difference between the i.p. and the b.p. is the power absorbed by mechanical friction (piston, bearings etc.). As the name implies, i.p. can be determined from the *indicator diagram* which for a four stroke engine has the form as shown in the figure below The negative work of the small anticlockwise loop is termed the *pumping power*, and is the result of viscous friction in the induction and exhaust strokes. The difference between a four stroke IC engine and other reciprocating machines therefore, is that the area of the small anticlockwise loop must be subtracted from the area of the main diagram to determine *Indicated mean effective pressure* (i.m.e.p.), which would then be the height of a rectangle of the same area as this resultant area and with the same length as the p-v diagram (L). The i.p. can then be expressed as follows.

$$\text{i.p.} = 100 \times p_m \times S \times A \times C \times nc \qquad \text{kW} \qquad (5)$$

where S is the stroke in m; A is the piston area in m^2, nc the number of cylinders, and p_m the i.m.e.p. in bar. C is the number of machine cycles per second and for a four stroke engine, this is equal to half the engine speed (rev/s) whilst for a two stroke engine it equals the engine speed.

Engine Indicator diagram



Whereas i.m.e.p. is considered a useful guide to engine performance, in practice it is usual to use the concept of *brake mean effective pressure* (b.m.e.p.) as an indicator. The b.m.e.p., denoted by p_{mb}, is defined in a similar way to p_m above.

$$\text{b.p.} = 100 \times p_{mb} \times S \times A \times C \times nc \qquad \text{kW} \qquad \text{(i)} \qquad (6)$$

where p_{mb} is again in bar. The b.m.e.p. may be regarded as that part of the i.m.e.p. which is imagined to contribute to brake power, the remainder being used to overcome friction. An increase in b.m.e.p. therefore implies that the cylinder volume (S x A), can be smaller for a given output.

In the Automotive 1 engine test set the parameters monitored relating to the engine brake power are the engine speed and the electrical output of the Alternator (Volts and Amps) driven by the engine. This output can be converted to *Brake Power* (b.p.) as follows.

$$\text{b.p.} = \frac{V \times I \times 100}{\mu} \qquad \text{Watts} \qquad (7)$$

where V is the Alternator output volts, I is the Alternator output Current and μ is the Alternator efficiency. μ varies with speed and output current and values can be obtained from figure 2. The brake power can then be expressed in terms of Torque and engine speed as follows.

$$\text{b.p.} = \frac{2 \times \Pi \times N \times T}{1000} \quad \text{kW} \qquad \text{(ii)}$$

(8)

where the engine speed N is in rev/sec. It can be seen from equations (i) & (ii) that Torque and b.m.e.p. are directly proportional and can be related by the following expression.

$$T = \frac{10^5}{4 \times \Pi} \text{ x } p_{mb} \text{ x S x A x N x nc Nm}$$ for a four stroke engine, and

(9)

and $$T = \frac{10^5}{2 \times \Pi} \text{ x } p_{mb} \text{ x S x A x N x nc Nm}$$ for a two stroke engine.

6.3 Comparisons between SI and CI Engines.

6.3.1 SI Engines.

The air standard cycle employed in SI engines is the Otto cycle and analysis shows that the cycle efficiency improves along with the brake thermal efficiency, with increase in compression ratio. There is an upper limit to which the compression ratio can be increased however, which is related to the fact that the fluid compressed in the cylinder is a mixture of fuel vapour and air. The temperature of the mixture increases during compression and too high a compression ratio could result in *pre-ignition* before spark-ignition. Even if the compression ratio is not high enough to cause pre-ignition, *detonation* may result. Combustion begins near the spark plug a short time after the formation of the spark and sets of a relatively slow flameless reaction occupying a delay period until the flame front develops. This delay period although short (0.002 seconds)can translate in high speed engines to appreciable crank movement and as a result the point of ignition is usually well in advance of top dead centre. As the flame front spreads in a uniform manner across the combustion space, it compresses the unburnt mixture before it. The temperature of the unburnt portion is raised therefore by both the radiation from the flame as well as by the compression and so if the original compression and temperature were to high self-ignition will occur. The resulting violent pressure rise and associated shock wave as the flame front accelerates rapidly are allied to the detonation affect. Modern engines and fuels tend to permit compression ratio's of upto 10:1

Having fixed the compression ratio the next factor governing performance, is the rise in pressure obtainable during combustion, as an increase in peak pressure will result in an increase in b.m.e.p. and consequently b.p.. The peak pressure is fixed by the amount of fuel that can be burnt, and hence the amount of oxygen available. Theoretically it will be a maximum if a stoichiometric mixture is used i.e. just sufficient oxygen to burn all the fuel. Stoichiometric air to fuel ratio's (AFR) by weight are almost identical for all liquid fuels, i.e. approximately 14.5:1. In practice, maximum brake thermal efficiency is achieved at a slightly weak mixture, giving complete combustion of the fuel, and maximum power with a slightly rich mixture where complete combustion of the oxygen occurs.

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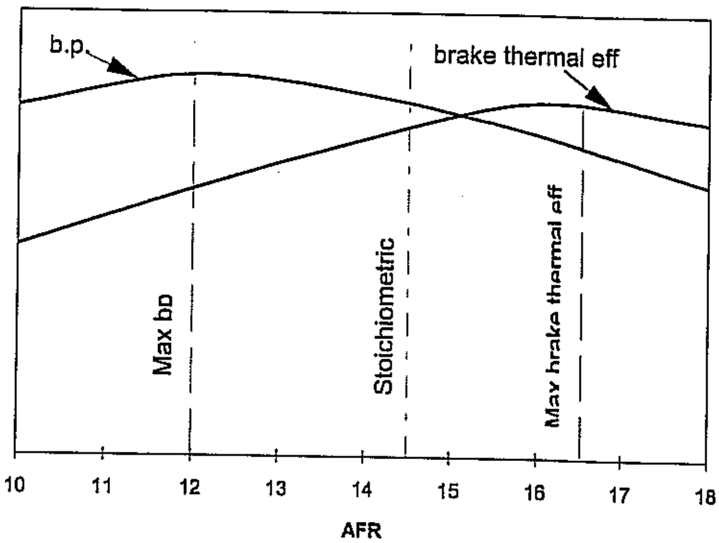
Effect of mixture strength on performance



As can be seen from the graph one way to regulate the power output would be to change the mixture AFR, however reliable ignition by spark is only achievable over a narrow band of AFR values. Instead approximately stoichiometric ratio's are maintained at all loads with regulation being achieved by varying the mass of the mixture induced in each working cycle. This method is referred to as *Throttle* or *Quantity governing.* Throttling results in sub-atmospheric pressures in the cylinder during the induction stroke, thus increasing the pumping loss. Part load efficiency of SI engines is consequently poor.

6.3.2 CI Engines.

Since the fuel is introduced into the cylinder of a CI engine only when combustion is required, i.e. towards the end of the compression stroke, pre-ignition cannot occur. Moreover, since the fuel is injected at a controlled rate, the simultaneous combustion of the whole mixture cannot happen, thus the problem of detonation associated with SI engines does not arise. It follows that compression ratio's in CI engines can be much higher, and in fact there is a lower limit of about 12:1 below which compression ignition of common diesel fuels is not possible.

In a CI engine most of the combustion takes place at relatively constant pressure, such that the maximum pressure reached in the cylinder is largely determined by the compression ratio employed. The upper limit of compression ratio is therefore fixed by the strength of the cylinder, con rod, bearings etc., whose stresses are determined by the peak pressure forces. Usually compromises between high efficiency and low weight/cost result in maximum limit on compression ratio's in the order of 20:1 A comparison of air-standard efficiencies of the Diesel and Otto cycles suggests that the CI engine will be more efficient, a fact borne out in practice.

The factor limiting i.m.e.p. (and consequently b.m.e.p.) of a CI engine, is the change in volume which occurs during the combustion at constant pressure. This is a maximum when all the fuel that can be burnt in the oxygen available is injected into the cylinder, suggesting again, stoichiometric AFR. In practice, because the fuel is injected as fine liquid droplets, mixing has to take place at the point of injection to ensure full combustion. Despite careful attention to inlet port, fuel injector, and combustion chamber design, it is not possible to burn all the fuel completely if stoichiometric ratio's are used, and the minimum AFR corresponding to full load usually lies between 18:1 and 25:1. Consequently the engine must be larger than would be the case if it were capable of burning stoichiometric ratio's. This results in CI engines being generally larger and heavier than SI engines although the former has a higher thermal efficiency.

It is found that very weak mixtures can be ignited and burnt in a CI engine so that it is possible to govern power output by varying fuel supply. Although this results in a increase in indicated thermal efficiency at part load, the fall in mechanical efficiency more than outweighs this effect, and the brake thermal efficiency always falls off. Nevertheless the reduction in efficiency with decrease in load is not so marked as in SI engines. Governing the power output by varying the mixture is usually referred to as *Quality governing.*

6.4 Engine performance characteristic.

Up to now engine performance has been considered in terms of work done per complete cycle, however the actual power output will depend on the rotational speed. It would seem that, other factors being equal, the power output of an engine can be raised by increasing its speed upto its mechanical limit. In practice it is found that the maximum indicated work per cycle varies considerably with speed, and beyond a certain speed the i.p. will fall with further increase in speed. The reduction is chiefly due to decrease in the mass of the charge induced in each cycle. Theoretically a naturally aspirated engine should induce a mass equivalent to its swept volume at ambient pressure and temperature. As a result of fluid friction and charge expansion, significantly less is taken in at high engine speeds when gas velocities are high and the manifold is hot.

This "breathing" capacity of an engine is expressed in terms of *volumetric efficiency* defined as

$$\eta_v = \frac{V_c}{V_s} \qquad (10)$$

where V_c is the volume of induced charge per induction stroke , and V_s is the engine swept volume. The volume of the induced charge is the combination of the volume of air and fuel induced, and at typical AFR values, due to the relative densities of the two fluids the fuel volume flow is very small in comparison to the air. When comparing engine performance, it is often standard practice to correct power readings to a reference pressure and temperature, thus compensating for differing ambient conditions and there effect on mass charge. There are various standards in existence for this corrective procedure, some merely accounting for variations in pressure and temperature, whilst others include corrections for humidity and consequently moisture content in the air. A typical correction factor, α, for SI engines, to multiplied by the observed power for correction purposes, is given as

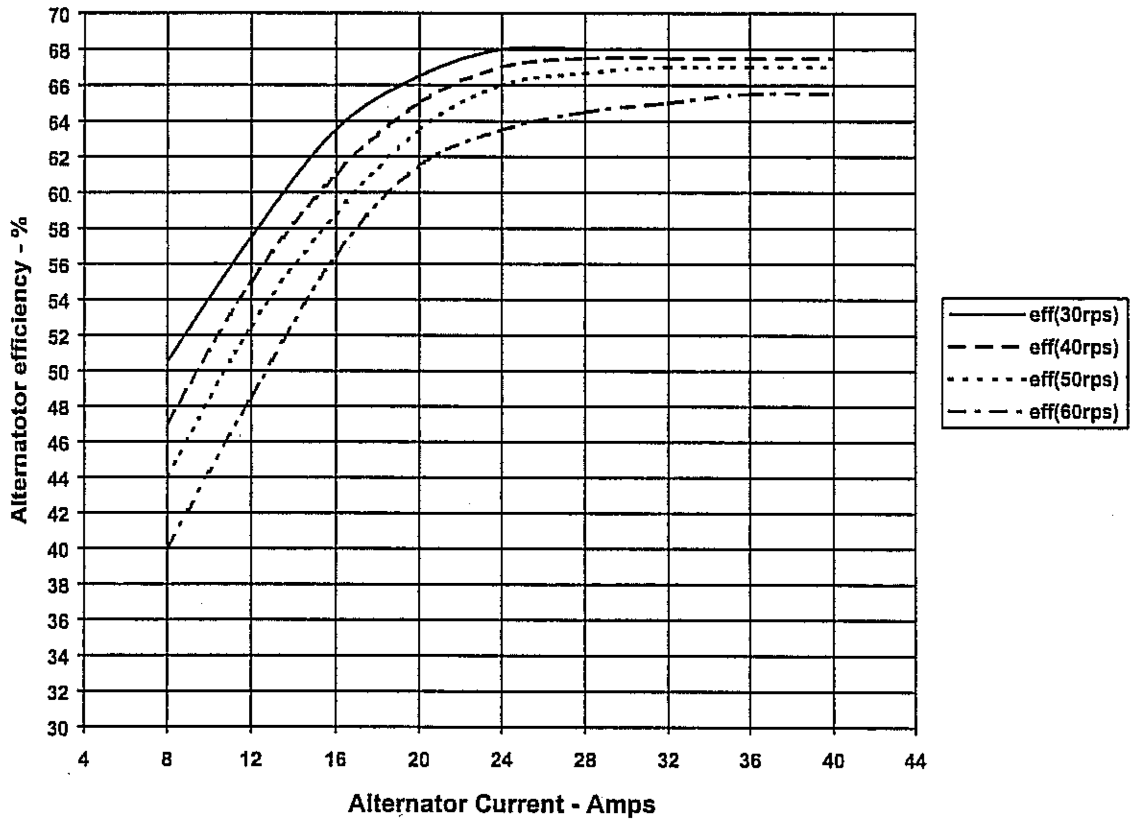
$$\alpha_s = \left(\frac{p_r - \varnothing_r p_{sr}}{p_y - \varnothing_y p_{sy}}\right)^{1.2}\left(\frac{T_y}{T_r}\right)^{0.6} \quad \text{(taken from the British standard BS 5514)} \qquad (11)$$

where p is absolute barometric pressure in kPa, p_S is the saturated water vapour pressure in kPa at the applicable temperature (at $\varnothing = 100$), $\varnothing$ is the relative humidity, and T is the absolute air temperature in deg K, Subscript r corresponds to the value under standard reference conditions, and these are given in the standard as p = 100kPa, T = 298 K, and $\varnothing = 0.3$. Subscript y corresponds to the values under test ambient conditions.

The indicated work per cycle also falls at low speeds. This is due partly to increased leakage of charge past valves and piston for which more time is available at low speeds. It is also due to reduced turbulence and increased cooling time available, thus making combustion less efficient. A typical power curve against speed therefore is as shown below.

It is evident that the power developed in an IC engine, is dependant on the mass of charge induced, and it would seem that only the ambient conditions prevalent may effect this. A much greater increase in power can be obtained by compressing the charge prior to induction with some type of *Supercharger*. A Roots blower or centrifugal compressor is normally used for this purpose, the former gear driven from the crankshaft, whilst the latter obtaining its power from turbine driven from the exhaust gases, and is termed a *Turbocharger*. Although the net increase in power obtained from supercharging or turbocharging can be quite considerable, it has little effect on brake thermal efficiency, as the fuel must be increased in proportion to the air charge to maintain the required AFR.

FIGURE 2 ALTERNATOR EFFICIENCY CURVES



12 APPENDIX 2 - AIR FLOW MEASUREMENT BY ORIFICE PLATES

12.1 Introduction.

Both the inlet and discharge air flows are measured by square edged orifice plates with corner tappings designed in accordance with section 7 of British Standard BS 1042: Part1: 1992. The working equations for the orifice plates are:-

$$\text{Volume Flow Rate, } \dot{Q} = C\,\varepsilon\,E\frac{\pi}{4}d^2\left(2\frac{\Delta p}{\rho}\right)^{0.5}$$

where :

$\dot{Q}$	=	Volume flow rate, m^3 / sec
d	=	Orifice diameter, m
ρ	=	Air density at upstream conditions, kg / m^3
Δp	=	Pressure difference across orifice, Pa
C	=	Coefficient of discharge
ε	=	Expansibility factor.
E	=	Velocity of approach factor.

The constants C, ε, and E are expressed as follows

$$\text{Coefficient of Discharge, } C = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^{8} + 0.0029\beta^{2.5}\left(\frac{10^6}{\mathrm{Re}_D}\right)^{0.75}$$

$$\text{Area Ratio, } \beta = \left(\frac{d}{D}\right)^2 \qquad \text{where D= Upstream diameter, m}$$

$$\mathrm{Re}_D = \text{Reynolds number referred to diameter D} = \frac{4\dot{Q}}{\Pi \nu d}$$

where ν = Kinematic viscosity, m^2/sec

$$\text{Expansibility factor, } \varepsilon = 1 - \left(0.41 + 0.35\beta^4\right)\frac{\Delta p}{\kappa\, p_1}$$

p_1 = Absolute upstream static pressure, Pa

κ = Isentropic exponent

$$\text{Velocity of Approach Factor, } E = \frac{1}{\left(1 - \beta^2\right)^{0.5}}$$

It can be seen that volumetric flow rate $\dot{Q}$ and Renolds number Re are dependant on one another and to satisfy the equation therefore re-iteration is necessary.

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12.2 Application To Inlet Air Flow Orifice And Surge Tank.

The orifice diameter fitted to the inlet surge tank used to measure the inlet air flow is sized to suit the engine, (the diameter is stamped on the plate), whilst the air is drawn from a free space which is effectively of infinite diameter, thus:

$D = \infty$ so that the area ratio,	$\beta = 0$
therefore,	$C = 0{,}5959$
And, by calculation the velocity of approach factor,	$E = 1{,}0$

Substituting into

$$\text{Volume Flow Rate, } \dot{Q} = C\,\varepsilon\,E\frac{\pi}{4}d^2\left(2\frac{\Delta p}{\rho}\right)^{0.5}$$

$\varepsilon = 1$

$d = 13\,mm$

yields :

$$\dot{Q} = 0.66188\ \varepsilon\, d^2\left(\frac{\Delta p}{\rho}\right)^{0.5}$$

and assuming a perfect gas the air density $\rho = P / R' T$ where R can be assumed to be 287.1 J/kg K

Note: The orifice diameter is stamped on the orifice plate on top of the surge chamber