Model Answer

1) *A- Define the following terms: API gravity, Brake mean effective pressure, Air injection, Cetane number, Distillation, Idling, Indicator diagram, Pour point.*

API gravity. An arbitrary scale adopted by the American Petroleum Institute to designate the specific gravity of mineral oils. Diesel fuels range from 18 to 41 API.

Brake mean effective pressure. The mean effective pressure, corresponding to the brake horsepower developed. Abbreviated bmep.

Air injection. The system of injecting fuel into the combustion chamber of a diesel engine by means of a blast of highly compressed air.

Cetane number. A percentage indicating the ignition quality of diesel fuels.

Distillation. Separation of the more volatile parts of a liquid from those less volatile by vaporization and subsequent condensation.

Idling. Engine running without a load at the lowest speed possible.

Indicator diagram. A diagram obtained by means of an indicator; it shows the change of pressure in the engine cylinder.

Pour point. The lowest temperature at which fuel oil will just flow under test conditions. It is an indication as to how suitable a fuel is for cold-weather operation.

B- State the main differences between modern and conventional engines

Differences between old and modern engines:

1- Modern Engines are More Efficient
2- Modern Engines are More Powerful.
3- Modern Engines are Smaller
4- Modern Engines Work Smarter.
5- Modern Engines Have Partners.

C- What are main forms of uncontrolled combustion phenomenon in SI engine?

Forms of uncontrolled combustion in SIE include:
In the following subsection the uncontrolled combustion phenomenon are collected; including combustion due to auto-ignition by a hot spots or flame initiation by a hot surface (called *surface ignition*). The latter may by the spark plug insulator or electrode, the exhaust valve head, the carbon deposits on the combustion chamber surfaces, etc. Surface ignition occurring before the spark is called *pre-ignition* while that after sparking is called *post-ignition.*
Pre-ignition: is an uncontrolled inflammation of the combustible mixture in an engine by a hot surface before the spark occurs. This is the most severe form of uncontrolled combustion. It is equivalent to advancing the ignition, but since the hot spot surface is larger than the spark, the combustion rate would be faster than that of the normal combustion, creating very high cylinder pressures and temperatures and thus resulting in excessive negative compression work and increased heat loss to the walls.
Run-on Surface Ignition: Run-on mode is the engine idling condition at which the fuel is supplied throughout idling jet while engine continue to fire this charge. In this case, mixture firing might be a spontaneous ignition due to a hot surface in the cylinder. The spontaneous ignition is affected by (a) elevated temperature of inlet mixture, (b) poor cooling of the combustion chamber surface, (c) duration of the valve overlap, and (d) a high compression ratio. At idling speed, the combustion chamber surface is not properly

cooled due to poor coolant circulation.

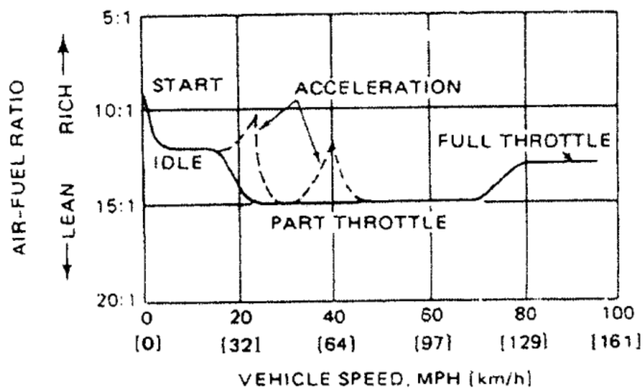
Run-away Surface Ignition: In severe cases, a surface ignition in one cycle can heat the surface such that the surface temperature still high in successive cycles leading to a series of earlier pre-ignitions; this mode is called the run-away surface ignition. That results in considerable damage to pistons and other engine parts.

Wild Ping: Wild ping is one or several irregular, but very sharp, combustion knocks caused by early surface ignition from deposit particles after the inlet valve is closed.

Rumble: is the name assigned to intermittent roughness caused by combustion chamber deposits which create secondary flame fronts.

2) *A- What are the main SI engine mixture requirements?*

In spark ignition engine a carburetor must provide a proper quantity of fuel within air according to the engine load and speed as shown in figure below. It can be concluded that the fuel air mixture is varied from slightly lean to rich mixture to be able to ignite the supplied mixture.



In diesel engine the engine power is controlled by the fuel injection system so that the air-fuel mixture is generally very lean (from equivalence ratio of 0.65 to 0.4). In this case the combustion mainly by diffusion mechanisms that dilutes with air.

B- Explain different devices used to measure engine brake effective power?

The brake effective power can be measured using dynamometers such as:

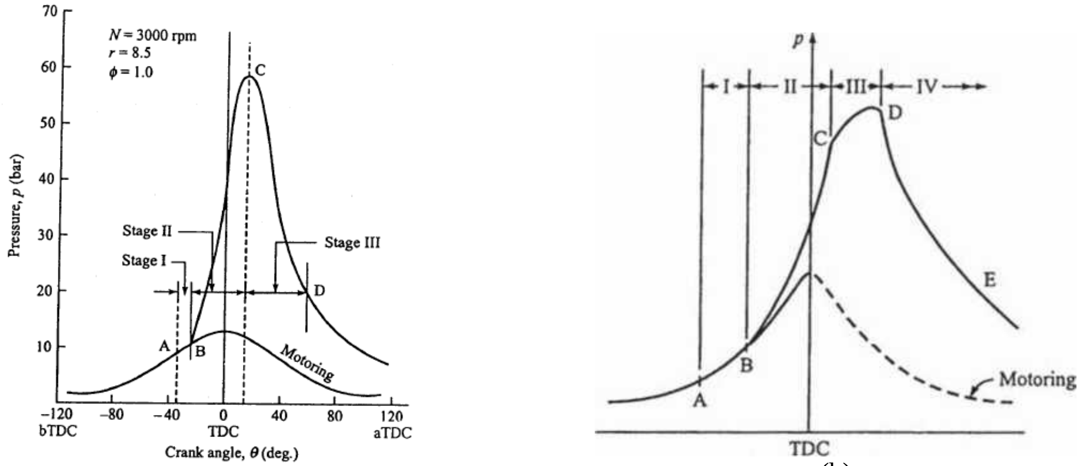
1- Electric dynamometers (both DC and AC generators to converts the mechanical power into electric power that can be measured easily),
2- Friction dynamometers to measure the torque needed to mechanically brake engine
3- Hydraulic dynamometers (water or oil dynamometers), and other dynamometers

C- State the combustion phases and the corresponding events in both SI and CI engines.

As shown in Figure below, the pressure-crank angle (P-θ) diagram indicating three combustion stages in a spark-ignition engine; including early flame development, flame propagation, and late stage of burning. The corresponding pressure is compared with that of the motoring case; obtained when the engine is not firing.
The combustion process which continually takes place in an operating diesel engine is basically represented by the pressure vs. crank angle diagram shown in Figure 3-3-b. At point A, the injector starts to inject fuel into the combustion chamber. A finite time elapses, during process AB, before the fuel droplets reach the ignition temperature, although when this happens most of the fuel injected during the first stage ignites spontaneously, causing an abrupt pressure rise at B and also during the process BC (second stage). The rest of the fuel injected during stage 3 is burnt as soon as it enters the combustion chamber. The CI engine combustion can be subdivided into three phases; Phase 1: Initial Premixed Combustion (Rapid or uncontrolled combustion), Phase 2: Main Combustion (Mixing-controlled combustion, and Phase 3: Post-combustion (Late combustion or afterburning. Thus the flame travel pattern includes four events:

1- Spark initiation
2- Early flame development

3- Flame propagation
4- Flame termination.



(a) (b)

The in-cylinder pressure-crank angle diagram for (a) for SIE, and (b) for CIE

3) A- State the main indicators describing the ignition quality or in-equality of diesel and gasoline fuels.

Fuel knock rating determines whether or not a fuel will knock in a given engine under the given operating conditions.

(a) Knock Rating of SI Engine Fuels

For fuels suitable for SI engines, the fuel resistance to knock in SI engines is determined by the fuel's octane number. The higher the octane number (ON), the higher the fuel resistance to knock and the higher the attainable compression ratio without knocking. The octane number depends on the engine design and the operating conditions during the test. As a standard, The ON is based on two hydrocarbons which define the ends of the scale; isooctane (C_8H_{18}; 2-2-4-trimethyl pentane) of a very good antiknock fuel is assigned a rating of 100 ON and normal heptane (n-C_7H_{16}) of very poor antiknock qualities is assigned a rating of zero ON. A gasoline of 90 ON has the same tendency to detonate in the test engine as that of a mixture containing 90% isooctane and 10% normal heptane by volume. There are two common procedures for ON determination - research method (Testing code: ASTM D - 2699) and motor method (Testing Code: ASTM D - 2700) and the received number is called **Research ON** (RON) and **Motor ON** (MON), respectively. In the motor method, the engine operating conditions are more severe (table 2-1) and thus there is a greater chance for engine knocking.

Table 2-1: Engine operating conditions for research and motor methods

Variable	*Research method*	*Motor method*
Inlet temperature	52 °C	149°C
Inlet pressure	Atmospheric	Atmospheric
Humidity (kg/kg dry air)	0.0036 - 0.0072	0.0036 - 0.0072
Coolant temperature	100°C(212°F)	100°C (2!2°F)
Engine speed	600 ipm	900 rpm
Spark advance	13°BTDC (constant)	19^0-26° BTDC (varies with compression ratio)
Air/fuel ratio	Adjusted for maximum knock	Adjusted for maximum knock

Since the motor method is more severe operating conditions than the research method, the MON is always lower than the RON, and the difference between these numbers is called the *fuel sensitivity*, i.e. $Fuel\ Sensitivity = RON - MON$. It is a measure of the extent to which a gasoline is downgraded under severe conditions. The higher sensitivity indicates poor performance under severe conditions. Generally, paraffins are the least sensitive, while olefins, naphthenes and aromatics are more sensitive. Therefore, the straight-run gasolines containing a high percentage of saturated hydrocarbons have low sensitivity, while the

cracked gasolines containing a large percentage of unsaturated hydrocarbons have high sensitivity.
Automobile engines run on the road under variable speed, load and weather conditions. The spark-timing also changes with speed. On the other hand, the CFR engines used to determine ON run at constant speed, full throttle, and fixed spark-timing. Therefore, the road octane number requirement differs from both RON and MON. The **road ratings** of fuels usually lie between the two number according to the following expression:

$$Road\ ON = aRON + bMON + c$$

where *a, b, c* are experimentally determined constants. For most gasolines used in automobiles, $a \approx b \approx 0.5$ and $c \approx 0$ give good agreement with practical results. In this case the mean value of RON and MON is known as antiknock index.
Fuels superior to isooctane in antiknock qualities have ON greater than 100. In this case, a mixture of isooctane and tetraethyl lead is used to determine their ON according to ASTM standards defined as follows:

$$ON = 100 + \left[28.28\,TEL/\left\{1 + 0.736\,TEL + \sqrt{1 + 1.472\,TEL - 0.035216\,TEL^2}\right\}\right]$$

where, TEL is in milliliters of TEL per US gallon in isooctane.
Performance number (PN) is another measure of antiknock effectiveness, it is related to ON and tetraethyl lead in isooctane; as the amount of tetraethyl lead in isooctane increases, the antiknock quality also increases non-linearly. The performance number is the ratio of the knock-limited indicated mean effective pressure (klimep) of the test fuel to the knock-limited indicated mean effective pressure (klimep) of isooctane, according to the following relation:

$$PN = \frac{klimep\ of\ test\ fuel}{klimep\ of\ isooctane}$$

Isooctane is assigned to have 100 PN. A fuel rated at 120 PN can produce approximately 1.2 times the power (without knock) that it can develop with a 100 PN fuel (without knock). ON and PN are related by the following relation:

$$ON = 100 + \frac{PN - 100}{3}$$

This relation attempts to extend the octane scale beyond 100. A fuel having 100 ON will have 100 PN, but a fuel having 120 ON will have 160 PN.
The knock rating of a fuel can also be expressed in terms of the **highest useful compression ratio (HUCR)** obtained by carrying out the test on a variable compression ratio engine under specified operating conditions, when the spark timing and mixture strength have been adjusted to give the maximum efficiency. The compression ratio is raised under specified conditions until knocking conditions are reached

(b) Knock Rating of Cl Engine Fuels

The methods for determining the ignition quality of CI engine fuels are (a) the cetane number and (b) the diesel index.
The cetane number determines the ignition quality of a diesel fuel; the higher the value of cetane number, the lower the ignition delay period and the lower the knocking tendency. The cetane number scale is defined by blending cetane (n-hexadecane $C_{16}H_{34}$) assigned with cetane number of 100 and isocetane (Hepta-methylnonane, HMN) having low ignition quality that is assigned with a cetane number of 15. In the original procedure α-methylnapthalene ($C_{11}H_{10}$) with a cetane number of zero represented the bottom of the scale, but HMN, as more stable compound is replaced it. Generally, the fuel Cetane Number (CN) is given by:

$$CN = Percent\ of\ n - cetane + 0.15 \cdot Percent\ of\ HMN$$

The ASTM method for rating CN of any fuel is determined by performing the test on a CFR engine. It is a single cylinder, variable compression ratio engine. The operating conditions are: engine speed 900 rpm, intake air temperature 65.6°C (150°F); coolant temperature 100°C; injection timing 13°bTDC; injection pressure 10.3 MPa. The compression ratio of the engine is varied until the combustion starts at TDC, i.e. an ignition delay period of 13° is obtained. This compression ratio is now kept fixed and the test engine with the specified operating conditions is run with the different blends of the reference fuels until combustion starts once again at TDC. Knowing the percentage of both the reference fuels in the blend, the cetane number is calculated as in the above relation. The recommended CN are 25 to 35 for low speed engines, 35 to 45 for medium speed engines and 45 to 60 for high speed engines. The ignition quality of a diesel fuel has a nonlinear relationship with the cetane number, but it is not a serious problem, since Cl engines burn fuels in a narrow range of cetane scale. For fuels of low CN as cracked fuels and fuels of other than paraffinic base, certain additives are needed to raise the fuel CN to the desirable values. These additives reduce the self-ignition temperature of the fuel, such as Isopropyl nitrate, ethyl nitrate, amyl nitrate, ethyl nitrite, butyl peroxide and methyl acetate. The leaded compounds used to enhance the ON, are not suitable additives for diesel fuels as they increase the ignition delay and hence the knocking tendency increases.

The diesel index depends on the fact that aromatic hydrocarbons mix completely with aniline at comparatively low temperatures, whereas the paraffins require considerably higher temperatures before becoming completely miscible. First, the 'aniline point' is determined. It is the lowest temperature at which equal volumes of the fuel and aniline become just miscible. The aniline point is determined by heating a mixture consisting of equal volumes of the test sample and freshly distilled, water free aniline, C_6H_7N, until a clear solution is obtained. Then, while the solution is cooling, the temperature at which turbidity appears is noted. The diesel index is computed from the following expression:

$$Diesel\ Index = \frac{Aniline\ point\ (°\mathrm{F}) \cdot API\ gr.}{100} = \left(Aniline\ (°\mathrm{C}) \cdot \frac{9}{5} + 32\right) \cdot \frac{API\ gr.}{100}$$

Generally, Diesel index gives a slightly higher values than CN.

B- Compare between combustion knocking of CI and SI engines.

The knocking in the SI engine and in Cl engine have essentially the same basic cause, i.e. auto-ignition followed by a rapid pressure rise, but they differ in the following points:

1- In the SI engine, knocking occurs due to auto-ignition of the end-gas spots, i.e. it occurs usually at the end of combustion, while in the Cl engine knocking occurs at the start of combustion (Figure 5-1).
2- The knocking intensity in SI engines (disturbances in the pressure rise) is higher than that of CI engines. This may be owing to the effect of charge state; in SI engine it is a homogeneous charge so knocking will produce a very high rate of pressure rise and high peak pressure, but in Cl engine a heterogeneous so as the combustion is initiated a normal smooth pressure rise continues until the end of fuel injection, even CI engines possess higher peak pressure due to the use of higher compression ratios.
3- As the charge within Cl engine during compression stroke is only air, there is no question of pre-ignition occurrence, which may be the major reason for SI engine knocking.
4- In the SI engine, the sound level due to knocking can be detected by human ear as the engine is usually running smooth, while in Cl engine there is no clear distinction between knocking and the starting of normal combustion.

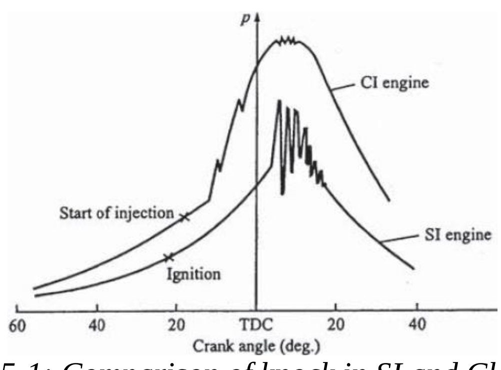


Figure 5-1: Comparison of knock in SI and Cl engines

The factors tend to prevent knock in SI engines are the same factors that promote knock in Cl engines. For example, to reduce the tendency for SI engine knocking, the auto-ignition of the last part of the charge should not occur and so a long delay period and a high self-ignition temperature are required. While for CI engines, the achievement of early flame due to lower delay period and lower self-ignition temperature would prevent engine knocking. It may also be noted that a good SI engine fuel is a bad Cl engine fuel and vice-versa.

C- What are the main methods of engine supercharging?

The following are the three basic methods are used for engine supercharging:

1- Mechanical system, where a blower or a compressor driven by the engine is used to provide the compressed air to the engine. It is the most common method of supercharging. Superchargers can spin at speeds as high as 50,000 to 65,000 rotations per minute (RPM).
2- Turbocharger: In this method, a turbocharger - a compressor and a turbine mounted on a single shaft - is used to increase the inlet air density. The energy of the hot exhaust gases is recovered by the turbine and the turbine output work is used to derive the air compressor.
3- Pressure wave supercharging: also known as a wave rotor is a type of supercharger technology that couples the pressure waves produced by an internal combustion engine exhaust gas pulses to compress the intake air. In this system cconsiderable part of the blow-down energy is converted into exhaust pulses as soon as the exhaust valve opens. Towards the end of exhaust the pressure in the exhaust pipe drops below the scavenging and large air pressure making scavenging quite easy.

4) A- Compare between combustion knocking of CI and SI engines.

The sudden combustion of fuel air mixture before flame envelop will be more possible in case of premixed gaseous mixture than for fuel spray. So that, the diesel knock is less important than gasoline detonation since in diesel there is a heterogeneous mixture so at high temperature and pressure the flame propagation will support fuel evaporation. In spark ignition engine the occurrence of detonation depends on many parameters as fuel octane number, initial conditions of the premixed mixture, ignition timing, and the compression ratio.

Diesel combustion starts when the injected fuel is vaporized and mixed with oxygen at the auto-ignition temperature. So that if the injected part is large enough a very sharp pressure rise will be obtained (this region is controlled by the chemical kinetics of premixed mechanism). While the rate of reaction of the other part of inject fuel is controlled by fuel-air mixing rate. Any change in the injection characteristics or timing may lead to high change in the combustion behavior; this change may not a predictable, so in this case the combustion is described as un-controllable process.

B- What are the main types of engine tests?

1- Measurement of brake power: The brake power is the power output of the engine. Measurement of the brake power involves the determination of the torque and the angular speed of the engine output shaft. The torque measuring device is called dynamometer. The engine is connected to a dynamometer which can be loaded in such a way that the torque exerted by the engine can be measured. It is capable of providing an adjustable and measurable torque. The dynamometer may be of the absorption or the transmission type. The absorption dynamometer converts the work done by the engine on the dynamometer into heat. In the transmission type of dynamometer the torque transmitted by the driving shaft is measured directly and consequently it does not itself absorb any of the engine’s work output. The majority of engine test-work utilizes an absorption type dynamometer. The important types of absorption dynamometers are described below.

2- Measurement of indicated power: The power that should be developed due to conversion of chemical energy of fuel into thermal energy during the in-cylinder gaseous expansion can be determined from the indicator diagram. The indicator is an instrument which produces a graphic record of the pressure inside the engine cylinder for every position of the piston as it reciprocates. This can be determined mechanically or electronically to receive the pressure/volume history of the engine cylinder.

3- Measurement of fuel consumption: The fuel consumption may be based on gravimetric or volumetric base; the first includes successive measure of mass reduction during time, while the other include the volume flow rate during time. The rate of fuel consumption can be measured in different ways by the following methods; Measure the time to consume known volume of specific fuel, Orifice type flow meter, Turbine flow meter, Pressure drop measurements, and Rotameter.

4- Measurement of air flow rate: Due to the opening and closing of inlet valves during engine charging, there will be high disturbances in the air flow rate within the inlet manifold. Thus, it will be difficult to measure the air flow rate accurately as there will be different pulsating pressure waves. These disturbances will be maximum in the case of a single cylinder four-stroke engine running on full throttle at low speed, and will be minimum for greater number of cylinders and higher engine speed, where air flow becomes more steady. To prevent these pulsation a large air box is installed at the inlet of engine. This air box has a capacity of more than 100 times that of the engine capacity.

5- **Measurement of engine speed:** The engine rotational speed can be measured by speedometer that may be mechanical tachometer, electrical revolution counters (proximity), optical, or frequency tachometers. The mechanical tachometer is a simple device which may be integral with the dynamometer. Hand-held revolution counters are also used which may be held in frictional contact with a rotating shaft to indicate the number of revolutions made. The time is separately recorded. The method is suitable for very slow rotational speeds. For high rotational speeds, an electronic transducer monitors the rotating shaft and produces electrical impulses at a frequency proportional to shaft speed. These impulses are fed to a gated electronic counter which sums the number of impulses over a fixed period of time. The rpm is digitally displayed by the counter. A light sensitive device such as a photo-transistor may also be used for speed measurement. One-half of the periphery of the shaft is pointed black to destroy its reflecting properties. Thus for every shaft revolution, the transistor will receive one burst of reflected light. The transistor will function as a high frequency switch, allowing current to flow only during its reception of reflected light. The electrical pulses so produced are counted and the shaft speed is displayed.

6- **Measurement of spark timing:** For spark-timing measurement, flash may be initiated by an external electrical impulse by winding the stroboscope triggering lead, once or twice round the spark plug high-tension lead. Each time the plug fires, a small current is simultaneously induced in the coil of the trigger lead, so that the plug spark and the light flash from the stroboscope occur at the same instant. Stroboscope light is directed on to a semi-circular scale, graduated in degrees. The mid-point of the scale is marked zero and it indicates the TDC position. The apparently stationary pointer marked on the crankshaft pulley moving over the scale will indicate the ignition point in relation to TDC.

7- **Measurement of temperature and cooling air flow rate:** The temperature within the air box, at exhaust, for cooling water inlet and outlet and the oil can be measured by the thermocouples. Also cooling water flow rate can be determined from the gravity or the volumetric base

C- What are the main changes required in the elementary carburetor to provide the required mixture according to engine load and engine speed

The changes required in the elementary carburetor to provide the required mixture according to engine load and engine speed as follows (See Figure 4-1):

1. **Choke**: A *choke* must be added to enrich the mixture during cold starting and warm-up to ensure. During the cold engine starting, the air flow rate is low producing low manifold vacuum, which draws less fuel from the jet causing the too lean fuel to ignite. By shutting off the most of air supplied to the main jets, an excess fuel will flow to balance the low vacuum and this will produce rich mixture necessary for cold starting. As the engine is started, the choke valve will be opened automatically as the engine warms up.
2. **Compensating tube:** The main metering system must be compensated to provide a constant lean or stoichiometric mixture over 20 to 80% of the air flow range. The compensating well is vented to atmosphere and gets its fuel supply from the float chamber through a restricted orifice. The main jet delivers a richer mixture with increase in air flow. The compensating jet would give a mixture that is too lean and which becomes still leaner with the increases in engine speed and load due to the fact that more air is drawn in from the well when it is emptied and a constant amount of fuel is discharged.
3. **Idling port:** must be added to meter the fuel flow at idle and light loads to provide a rich mixture, since at these conditions the elementary carburetor fails to satisfy the mixture requirements. This system comes

into action during starting, idling and low-speed operations. The idling system is a small fuel line from the float chamber to a point nearer to the engine side of the throttle. This line usually contains a fixed idling fuel orifice to meter the fuel. An idle mixture adjusting screw controls the richness of the mixture at idling; when adjusted the mixture produces a smooth engine idling speed.

4. **Acceleration port:** An *accelerator port* must be included to give additional fuel only when the throttle is suddenly opened thus a rapid response to supply a mixture within required mixture strength is ensured. The acceleration system consists from a cylinder coupled with the float chamber and plunger linked to the throttle shaft. Due to sudden change in the throttle position, the plunger moves downwards (upwards) forcing (pulling) the fuel to leave (enter) the cylinder.
5. **Full load port:** An *enrichment system* must be included into the simple carburetor to supply the engine with a rich mixture at engine full load so the maximum power is obtained.
6. ***Altitude compensation*** is necessary to adjust the fuel flow, which makes the mixture rich when air density is lowered.
7. Increase in the magnitude of the pressure drop available for controlling the fuel flow is provided by introducing *boost venturis* (Venturis in series) or *Multiple-barrel carburetors* (Venturis in parallel).

5) A four-stroke four-cylinder compression ignition engine has a cylinder diameter of 99 mm, and stroke of 95 mm. An experiment is run on the engine at speed of 2800 rpm, and the following reading are obtained:

a. Volumetric efficiency is 0.85,
b. brake constant is 1500
c. the following data at various load

Brake load (Kg)	*5.35*	*10.7*	*16.1*	*21.4*	*29.2*
Rate of fuel consumption (kg/min)	*0.068*	*0.102*	*0.137*	*0.1717*	*0.222*

Draw the relation between the effective power of the engine and:

1- Indicated mean effective pressure,
2- Excess air factor, and
3- Brake specific fuel consumption (kg/kW.hr).

For 4-cycliners CIE the brake power $N_e = WN/(\text{Brake Constant})\,\text{KW}$ receiving the following table:

Brake load (Kg)	5,35	10,7	16,1	21,4	29,2
Engine power (KW)	9,99	19,97	30,05	39,95	54,51
Rate of fuel consumption (kg/min)	0,068	0,102	0,137	0,1717	0,222

For air density of 1.2 kg/m^3 the theoretical amount of air to fill engine cylinders will be:

$$\dot{m}_{Air,Th} = \frac{\pi}{4} d_c^2 S \frac{zN}{i60} \rho_a = \frac{\pi}{4}(0.099)^2 0.095 \frac{4*2800}{2*60} 1.2 = 0.0819 \quad \text{kg/S}$$

For engine volumetric efficiency of 85% where: $\eta_{Vol} = \dfrac{\dot{m}_{Air}}{\dot{m}_{Air,Th}}$ then $\dot{m}_{Air} = 0.0696$ kg/S

By curve fitting data to linear relation; according to Willan's line, then:
m_f=A+B*N_e kg/min at two distinct points, it can be found that B=0.00346 and A=0.0334 to get fuel mass flow rate per min.

or $\dot{m}_f = 0.000567 + 0.0000577 N_e$ kg/S thus at m_f=0 then N_e=N_f=9.8 KW

the relation between required variables into effective power are:

1- The indicated mean effective pressure imep=N_i/(Rate of Swept Volume) or

$$imep = (N_e + 9.8)\frac{4i60}{\pi d_c^2 SzN} = (N_e + 9.8)\frac{4*2*60}{\pi(0.099)^2 0.095*4*2800} = (N_e + 9.8)*14.651 \quad \text{KPa}$$

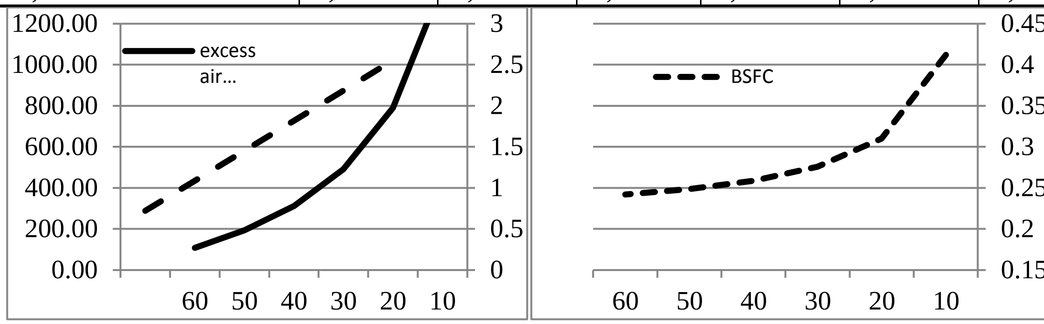
For general assumption the theoretical AF =16

2- Excess air $\alpha = \frac{AF_{act}}{AF_{Th}} - 1 = \frac{0.0819/\dot{m}_f}{16} - 1 = \frac{0.0819/(0.000567 + 0.0000577N_e)}{16} - 1$

3- specific fuel consumption $BSFC = 3600\frac{\dot{m}_f}{N_e} = 3600\frac{0.000567 + 0.0000577N_e}{N_e}$ KWhr

Applying these equations for power from 10 to 60 KW to get the following table:

Engine power (KW)	10	20	30	40	50	60
imep, KPa	287,89	434,40	580,91	727,42	873,93	1020,44
excess air factor	3,474432	1,974288	1,22748	0,780435	0,482836	0,270477
BSFC, KWhr	0,41184	0,30978	0,27576	0,25875	0,248544	0,24174



6) *A four stroke-four cylinders spark ignition engine has a cylinder diameter of 89 mm and piston stroke of 91mm. An experiment is done at speed of 3500 rpm, and the following reading are taken:*
   a. *Brake load when all cylinders are firing is 26 kg*
   b. *Brake load when only three cylinders are firing is 18.2 kg*
   c. *Brake constant is 2000*
   d. *Pressure drop across the air box orifice of diameter 5 cm is 10 cm H_2O, with CD = 0.61*
   e. *Chemical formula of the used fuel is C_8H_{18}*
   f. *Fuel density is 0.74 gm/cm^3*
   g. *Fuel consumption is 0.82 L during 3 min*
   h. *Rate of cooling water is 81 L in 60 sec*
   i. *Temperature rise of cooling water across the engine is 8 ^oC*
   j. *Temperature of the exhaust gases is 670 ^oC*
   k. *Temperature and pressure of the ambient air 300 K and 1 bar*

   Calculate: 1) Heat balance of the engine (kW), 2) Thermal efficiency of the engine, 3) Mechanical efficiency of the engine, 4) Volumetric efficiency of the engine, and 5) The excess air factor.

For 4-cycliners W_4=26 kg and that for 3-cycliners W_3=18.2 kg

Then brake power $N_{e,4} = WN/(\text{Brake Constant}) = 26*3500/2000 = 45.5$ KW

and the corresponding power for only 3-cyliners

$N_{e,3} = WN/(\text{Brake Constant}) = 18.2*3500/2000 = 31.85$ KW

According to Morse test, the indicated power from the shut-off cylinder will be:

$N_{i,1} = N_{e,4} - N_{e,3} = 13.65$ KW

Assuming the engine cylinders are the same, then the engine indicated power will be

$N_i = zN_{i,1} = 4*13.65 = 54.6$ KW

The engine mechanical efficiency is defined by: $\eta_{mech} = \frac{N_{e,4}}{N_i} = 83.3\%$

To perform heat balance it is necessary to calculate the mass flow rate of cooling water and air.
For air (air density is calculated from conditions to be 1.18 kg/m^3):

$$\dot{m}_{Air} = C_d A\sqrt{2\Delta P\rho_a} = 0.61*\frac{\pi}{4}(0.05)^2\sqrt{2*(0.1*9.81*1000)*1.18} = 0.0576 \text{ kg/S}$$

Theoretical amount of air will be $AF_{Th} = \frac{(8+18/4)32}{(8*12+18)0.233} = 15.059$ kg per kg of fuel

The actual fuel flow rate $\dot{m}_{Fuel} = \frac{Volume}{Time}\rho_{Fuel} = \frac{0.82/1000}{3*60}740 = 3.37*10^{-3}$ kg/S

Thus actual Air-To-Fuel ratio will be $AF_{act} = \frac{\dot{m}_{Air}}{\dot{m}_{Fuel}} = 17.08$ kg per kg of fuel

So excess air factor will be $\alpha = \frac{AF_{act}}{AF_{Th}} - 1 = 0.134$ or air used is 113.4 % of theoretical.

The theoretical amount of air to fill engine cylinders will be:

$$\dot{m}_{Air,Th} = \frac{\pi}{4}d_c^2 S\frac{zN}{i60}\rho_a = \frac{\pi}{4}(0.089)^2 0.091\frac{4*3500}{2*60}1.18 = 0.0779 \text{ kg/S}$$

Thus engine volumetric efficiency will be: $\eta_{Vol} = \frac{\dot{m}_{Air}}{\dot{m}_{Air,Th}} \approx 74\%$

Total energy supplied to engine: $Q_{Fuel} = \dot{m}_{Fuel}HV = 3.37*42 = 141.54$ KW

Thus engine thermal efficiency will be: $\eta_{Therm} = \frac{N_e}{Q_{Fuel}} = 32.15\%$

The cooling water flow rate $\dot{m}_{CW} = \frac{Volume}{Time}\rho_W = \frac{81/1000}{60}1000 = 1.35$ kg/S

Thus heat for cooling will be $Q_{CW} = \dot{m}_{CW}C_{PW}\Delta T_W = 1.35*4.2*8 = 3.37*42 = 45.36$ KW

The fraction of heat for cooling will be $\eta_{CW} = \frac{Q_{CW}}{Q_{Fuel}} = 32.04\%$

For energy in the exhaust $Q_{Ex} = \dot{m}_{Air}C_{Ex}\Delta T_{Ex} = 0.0576*1.13*(670-27) = 41.85$ KW

The fraction of heat into Exhaust $\eta_{Ex} = \frac{Q_{Ex}}{Q_{Fuel}} = 29.57\%$

The un-account heat loss will be: $Q_{Uacc} = Q_{Fuel} - (N_e + Q_{CW} + Q_{Ex}) = 8.83$ KW
or $\eta_{Uacc} = 100 - (\eta_{Ther} + \eta_{CW} + \eta_X) = 6.24\%$.

Best wishes for all, Dr. Ali M.A. Attia