Benha University
Benha Faculty of Engineering
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Exam (External)
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Mechanical Engineering Department
Time: 3 Hours.
B. Sc. Course Exam
Subject: Internal Combustion Engines
Code No.: M528

Model Answer

Question ① 《**10 marks**》

A- Define: Airless injection, Detonation, Indicated Power, volumetric efficiency, Specific fuel consumption, Mean Effective Pressure, Distillation, Pour point, Cetane Number, and Indicator diagram. (5)

Airless injection. A general term describing all methods of injecting fuel without the use of compressed air.

Detonation. A violent uncontrolled burning of a fuel in the combustion chamber.

Indicated power. The engine power developed in the combustion cylinder, as calculated from an indicator diagram.

Volumetric efficiency. Ratio of the volume discharged from a pump to the piston displacement of the pump. In diesel engines a term often used instead of the correct term *charge efficiency.*

Specific fuel consumption. The fuel consumption per hour divided by the brake horsepower developed, expressed in kg/kW.hr.

Mean effective pressure. The mean or average pressure which, acting on lie piston, would do the same work

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

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.

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

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

B- What are main forms of uncontrolled combustion phenomenon in SI engine? (5)

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.

Question ② (15 marks)

A- What are the SI and CI engine knock rating parameters? (10)

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_{l8}; 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 and thus there is a greater chance for engine knocking.

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:

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

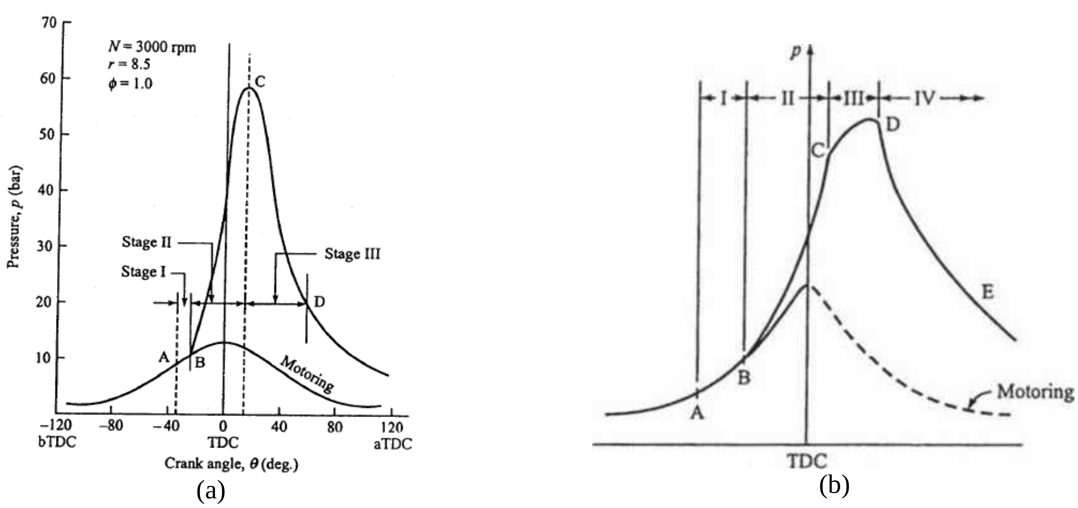
Generally, Diesel index gives a slightly higher values than CN.

B- State the combustion phases and the corresponding events in both SI and CI engines. (5)

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.



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

Question ③ (15 marks)

A- What are the main methods of engine supercharging? (5)

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.

B- What are the main changes required in the elementary carburetor to provide the required mixture according to engine load and engine speed (5)

The changes required in the elementary carburetor to provide the required mixture according to engine load and engine speed as follows:

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).

C- What are the main devices used to measure engine indicated power? (5)

The indicated power can be measured mechanically or electronically to receive the pressure/volume history of the engine cylinder:

1. **Mechanical Indicator:** The simplest form of engine mechanical indicator is the Dobbie Mclnncs indicator like that shown. It consists of a small cylinder fitted with a piston, the underside of which is placed in communication with the cylinder. The upper side of the indicator piston is kept in-communication with the atmosphere. A helical spring on the top of the piston has one end attached to the piston and the other to the cover of the indicator cylinder through which passes the piston rod which carries a pencil at its upper end attached to a drum, and the rotation of the drum is linked to the piston displacement by a cord wrapped around the drum. This pencil traces out the indicator (P-V) diagram on the paper. When the paper is unwrapped from the drum the area of the diagram can be found, and this corresponds to the indicated work per cylinder per cycle. The area can be determined by ‘counting squares’, by cutting the diagram out and weighing it, or by using a planimeter. The planimeter is a mechanical device that computes the area of the diagram by tracing the perimeter. The indicator diagram has a positive loop (the area between the compression and expansion curves) and a negative loop (the area between the suction and exhaust curves). The positive loop gives the gross work done by the piston during the cycle and the negative loop represents the pumping loss due to admission of fresh charge and removal of exhaust gases. To convert area to work a calibration constant is needed; alternatively imep can be found more directly. The diagram area is divided by its length to give a mean height. When this height is multiplied by the indicator spring constant (bar/mm) the imep can be found directly:

$$\text{imep} = \mathrm{K}h_d = K\frac{A_d}{l_d}$$

where A_d - the net diagram area, l_d - the diagram length, and h_d - the mean height of the diagram.
The indicated power than computed from:

$$iP = \frac{imep * SAN}{60 * i}$$

where S – Stroke, A – cyclinder corss section area, N – revolution speed, i – factor =1 for twp stroke or =2 for four stroke engine. Because of the inertia effects in moving parts - friction, backlash and finite stiffness - mechanical indicators can be used only at low speeds lower than 600 RPM. Also, this simple type of mechanical indicator is not sensitive enough to record the 'pumping losses’ during the induction and exhaust strokes.

2. Electronic indicator: The disadvantages of mechanical indicator can be removed when the sensitive with high frequency electronic measurement system is used to determine the indicated power of the engine; Figure 8. In this system the successive recording of the crank angle and the in-cylinder pressure can be obtained online and readout using computer software and/or digital storage oscilloscope. The

crank angle is measured by using a rotary encoder that may be sensitive to record each angle or even divide the angle into fractions. In this case a sensitive proximity installed above a geared disc is used to discretize the gear teeth. To differentiate the top dead center (TDC) location, another sensor may be needed or the corresponding tooth may be changed to give a different signal at TDC. The in-cylinder pressure is collected via a piezoelectric pressure sensor coupled with charge amplifier. The output signal of the piezo sensor is charge that will be converted into analog multiplied voltage signal by use of signal conditioning system. The voltage signal can be readout by data acquisition system or oscilloscope. To convert the time base readings of crank angle to a piston displacement base it is usual to assume constant angular velocity throughout each revolution, and the piston displacement (x) is given by:

$$x = r(1-\cos\theta) + \left[l - \sqrt{l^2 - r^2\sin^2\theta}\right]$$

where l – connecting rod length, r – crank radius (half stroke), θ – crank angle from TDC

Question ④ (20 marks)

A- The diesel engine has a compression ratio of 18:1 and operates on an air-standard Dual cycle. At 2400 rpm, the fuel injection starts tat 20° BTDC, combustion starts at 7° BTDC and lasts for 42° of engine rotation. The ratio of connecting rod length to crank offset is R = 3.8. Calculate ignition delay and the cycle cutoff ratio. (5)

1) Combustion starts at 7° bTDC and fuel injection starts at 20° bTDC (from Example Problem 5-4). Ignition delay in degrees of engine rotation:

$$\underline{\text{ID} = 13° \text{ of engine rotation}}$$

Ignition delay in seconds:

$$\text{ID} = (13°)/[(2400/60 \text{ rev/sec})(360°/\text{rev})] \underline{= 0.0009 \text{ sec}}$$

2) Combustion stops at 35° aTDC. Equation (2-14) is used to find cutoff ratio:

$$\beta = V/V_{\text{TDC}} = V/V_c = 1 + \tfrac{1}{2}(r_c - 1)[R + 1 - \cos\theta - \sqrt{R^2 - \sin^2\theta}]$$
$$= 1 + \tfrac{1}{2}(18-1)[3.8 + 1 - \cos(35°) - \sqrt{(3.8)^2 - \sin^2(35)°}]$$
$$\underline{= 2.91}$$

B- In a test of a gas-turbine combustor, saturated-liquid methane at 115 K is to be burned A four-stroke four-cylinder compression ignition engine has a cylinder diameter 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: (10)

a) Indicated mean effective pressure,
b) Excess air factor, and
c) 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:

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

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

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 $\dot{m}_f = 0.000567 + 0.0000577\text{N}_e$ 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

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

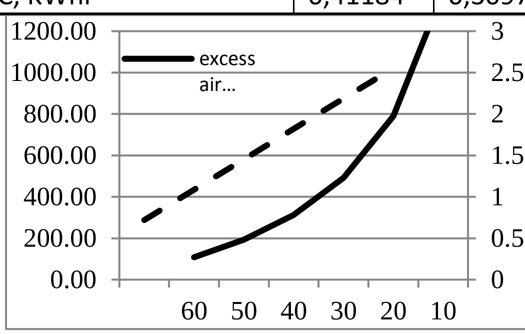
For general assumption the theoretical AF =16

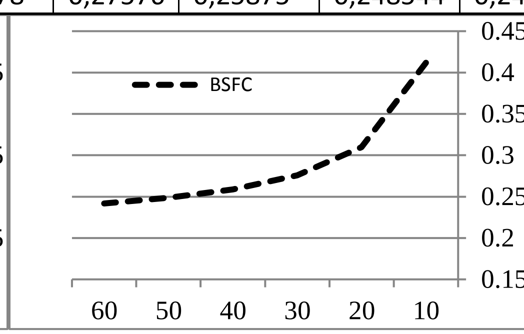
2- Excess air $\alpha = \frac{AF_{act}}{AF_{Th}} - 1 = \frac{0.0819/\dot{m}_f}{16} - 1 = \frac{0.0819/(0.000567 + 0.0000577\text{N}_e)}{16} - 1$

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





Good Luck,

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