Benha University
Benha Faculty of Engineering
Mechanical Engineering Department
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Internal Combustion Engine (M528)
First semester – Supplementary Exam
Time: 3 hr.
Date: 1/1/2014

Model Answer

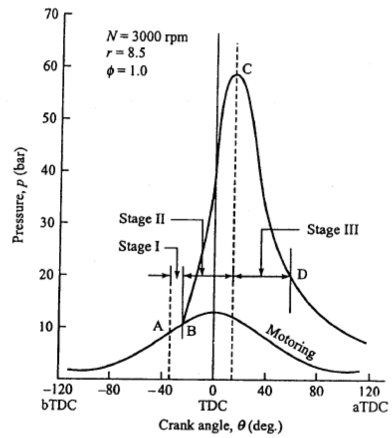
1) A- What are the combustion phases for spark ignition engine? *(5 P)*

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.

1- Early flame development stage - stage (AB): This stage is called *ignition lag* or *early flame development phase.* It corresponds to the time for the growth and development of a self-propagating nucleus of the flame.

2- Flame propagation stage - stage (BC): This stage is called the *main stage,* during which the propagation of the flame spreads practically at a constant speed. This stage starts as a first measur-able pressure rise against the motoring curve is observed up to a moment where the maximum pressure is attained.

3- Late burning stage - stage (CD): This stage is called *late burning, afterburning, or flame termination.* Theoretically, the maximum pressure occurs as the combustion is completed, but actually in engine cylinder a part of the heat is still liberated after this point.



B- Stat the components of fuel injection system? *(5 P)*

components required in a fuel-injection system include:

1. *Pumping elements:* Pumping elements pump will supply fuel under high pressure from the fuel tank to the cylinder through pipe lines and injectors.
2. *Metering elements:* Metering elements measure and supply the fuel at the rate required by the engine speed and load.
3. *Fuel injection pump:* to rise the fuel pressure up to a value that meets the good atomization requirements.
4. *Metering controls:* Metering controls adjust the rate of the metering elements for changes in engine speed and load.
5. *Distributing elements - injector:* Distributing elements distribute the metered fuel equally among the cylinders due to proper atomization.
6. *Cam or electronic control for Timing controls:* Timing controls adjust the start and the stop of injection.

2) A- Derive the relation for air to fuel ratio for simple carburetor neglecting the compressible effect of air. *(5 P)*

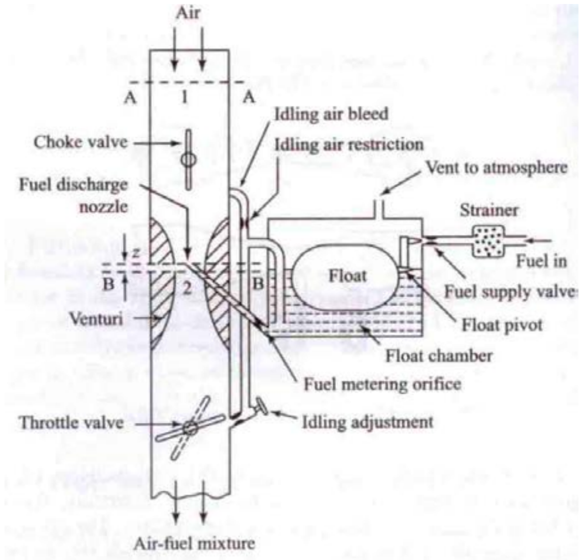


Figure: A schematic representation for simple carburetor

Considering the above Figure, now applying the steady-flow energy equation between the sections AA and BB and considering unit mass of air flow, then:

$$q - w \quad = \quad (h_2 - h_1) + \frac{1}{2}\left(C_2^2 - C_1^2\right) \qquad (1)$$

Here, q and w are the heat and work transfers from the entrance to the throat and h and C stand for enthalpy and velocity respectively. If we assume reversible adiabatic conditions, and there is no work transfer, q=0, w=0, and if approach velocity C1≈0 we get

$$C_2 \quad = \quad \sqrt{2(h_1 - h_2)} \qquad (2)$$

For air as a perfect gas, thus $h = c_p T$ then:

$$C_2 = \sqrt{2c_p(T_1 - T_2)} \qquad (3)$$

If we assume that the distance from the inlet to the venture throat is short, we can consider it to be isentropic in the ideal case, *then* $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$ or:

$$T_1 - T_2 \quad = \quad T_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right] \qquad (4)$$

Substituting for $T_1 - T_2$ from Eq. 5 in Eq. 3, we get

$$C_2 = \sqrt{2c_p T_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \qquad (5)$$

From continuity equation, then theoretical air mass flow rate is given by:

$$\dot{m}_a = \rho_1 A_1 C_1 = \rho_2 A_2 C_2 \qquad (6)$$

where A_1 and A_2 are the cross-sectional areas at the air inlet (point 1) and venturi throat (point 2). To calculate the mass flow rate of air at the throat, we have assumed the flow to be isentropic till the throat so the equation relating p and v (or ρ) can be used.

$$p_1 v_1^\gamma = p_2 v_2^\gamma \quad \text{or} \quad \frac{p_1}{\rho_1^\gamma} = \frac{p_2}{\rho_2^\gamma}, \text{ thus } \rho_2 = \rho_1\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \qquad (7)$$

then theoretical air mass flow rate will be:

$$\dot{m}_a = \rho_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} A_2 \sqrt{2c_p T_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \qquad (8)$$

From ideal gas law, we have $\rho_1 = \frac{p_1}{RT_1}$, thus:

$$\dot{m}_a = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} \frac{p_1}{RT_1} A_2 \sqrt{2c_p T_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

and rearranging the above equation we have

$$\dot{m}_a = \frac{A_2 p_1}{R\sqrt{T_1}} \sqrt{2c_p \left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right]} \qquad (9)$$

Since the fluid flowing in the intake is air, we can put in the approximate values of R = 287 J/kgK, c_p = 1005 J/kgK at 300K, and γ = 1.4 at normal temperature (≈ 300 K).

$$\dot{m}_a = 0.1562 \frac{A_2 p_1}{\sqrt{T_1}} \sqrt{\left(\frac{p_2}{p_1}\right)^{1.43} - \left(\frac{p_2}{p_1}\right)^{1.71}} = 0.1562 \frac{A_2 p_1}{\sqrt{T_1}} \phi \qquad (10)$$

where $\phi = \sqrt{\left(\frac{p_2}{p_1}\right)^{1.43} - \left(\frac{p_2}{p_1}\right)^{1.71}}$

Here, pressure p is in N/m^2, area A is in m^2, and temperature T is in K. If we take the ambient temperature T_1 = 300K and ambient pressure $p_1 = 10^5$ N/m^2, then theoretical air mass flow rate will be:

$$\dot{m}'_a = 901.8 A_2 \phi \qquad (11)$$

Due to the behavior of Vena-contracts during flow throughout orifice, the estimated theoretical mass flow rate of air can be converted into an actual value by multiplying $\dot{m}_a$ by the discharge coefficient $C_{d,a}$. Thus

$$\dot{m}_{Air} = 0.1562 C_{d,a} \frac{A_2 p_1}{\sqrt{T_1}} \phi \qquad (12)$$

Note that, the value of the through discharge coefficient is obtained experimentally by using know air mass flow are and estimated mass flow rate, thus $C_{d,a} = \frac{\dot{m}_{Air}}{\dot{m}_a}$. Anyway, the values of discharge coefficient and throat cross area are constant for a given venturi, thus:

$$\dot{m}_{Air} \propto \frac{p_1}{\sqrt{T_1}} \phi \qquad (13)$$

Now applying energy and continuity equation for fuel. Since the fuel is a liquid before mixing with the air, it can be considered as incompressible flow. Applying Bernoulli's equation between the atmospheric conditions prevailing at the top of the fuel surface in the float bowl, which corresponds to point 1 and the point where the fuel will flow out, at the venturi, which corresponds to point 2. Fuel flow will take place because of the drop in pressure at point 1 due to the venturi effect. Thus

$$\frac{p_1}{\rho_f} - \frac{p_2}{\rho_f} = \frac{C_f^2}{2} + gz \qquad (14)$$

where ρ_f is the density of the fuel in kg/m^3, C_f is the velocity of the fuel at the exit of the fuel nozzle (fuel jet), and z is the depth of the jet exit below the level of fuel in the float bowl. This quantity must always be above zero otherwise fuel will flow out of the jet at all times. The value of z is usually of the order of 10 mm. From Eq. 14 we can obtain an expression for the fuel velocity at the jet exit as:

$$C_f = \sqrt{2\left[\frac{p_1 - p_2}{\rho_f} - gz\right]} \qquad (15)$$

From the continuity equation, the theoretical fuel mass flow rate is determined by:

$$\dot{m}_f = \rho_f A_f C_f = A_f \sqrt{2\rho_f\left(p_1 - p_2 - \rho_f gz\right)} \qquad (16)$$

where A_f is the exit area of the fuel jet in m^2. For fuel nozzle of discharge coefficient $C_{d,f}$ given by:

$$C_{d,f} = \frac{\dot{m}_{fuel}}{\dot{m}_f}\text{, then } \dot{m}_{fuel} = C_{d,f} A_f \sqrt{2\rho_f\left(p_1 - p_2 - \rho_f gz\right)} \qquad (17)$$

Since

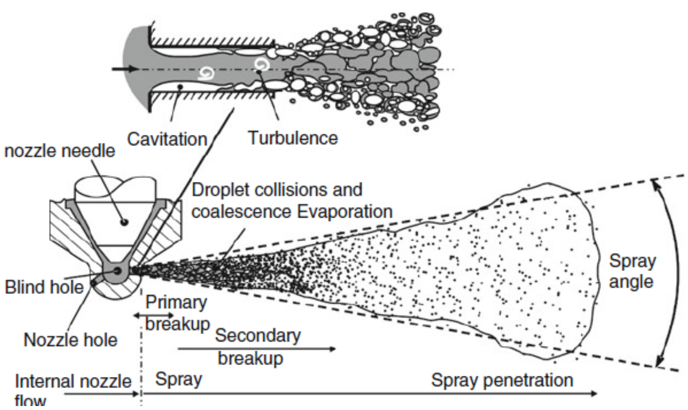
$$\frac{A}{F} = \frac{\dot{m}_{Air}}{\dot{m}_{fuel}} = 0.1562 \frac{C_{d,a}}{C_{d,f}} \frac{A_2}{A_f} \frac{p_1 \phi}{\sqrt{2\rho_f T_1\left(p_1 - p_2 - \rho_f gz\right)}} \qquad (18)$$

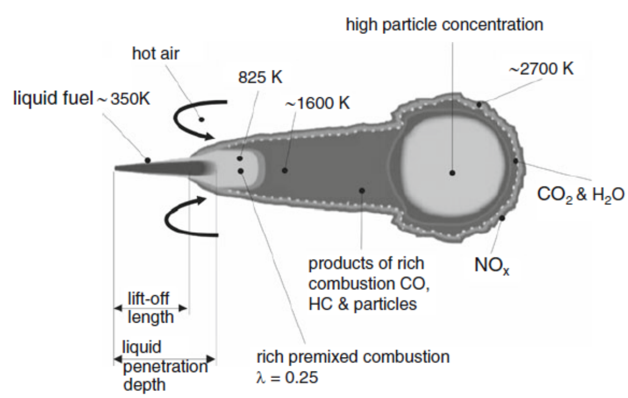
For T_1 = 300K and $p_1 = 10^5$ N/m^2 then:

$$\frac{A}{F} = 901.8 \frac{C_{d,a}}{C_{d,f}} \frac{A_2}{A_f} \frac{\phi}{\sqrt{2\rho_f\left(p_1 - p_2 - \rho_f gz\right)}}$$

B- Describe with a sketch the structure of the fuel spray within diesel engine cylinder with emphasize on the conditions for emission formation. *(5 P)*

Spray is divided into dense spray (near the fuel nozzle where fuel is mainly broken and fuel droplets are formed) and the secondary spray (where additional break up of existing droplets into smaller droplets due to aerodynamic forces occurs – fuel vapor region). After that there will be a region of air entrainment at the outer surface and at the spray central products of rich combustion are produced (mainly soot). At the end of the fuel spray there will be combustion of fully vaporized and mixed fuel with air. In this region most of formed soot is oxidized at high temperature and NO_x also formed.

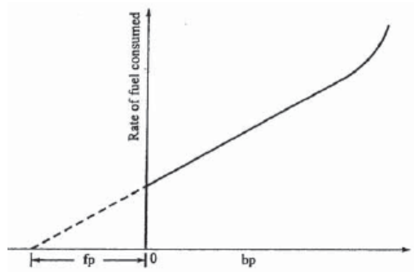




3) A- What are the experimental methods used to determine the engine friction power? *(5 P)*

If the indicated power and brake power are known there is not problem in determining the engine friction power, as $fP = iP - bP$. But this is not usually available, in this case the determination of engine friction power can be performed experimentally from the data of brake power depending on the engine type using a specific procedure. These methods will be described as follows:

1- **Willan's line method:** In this method a relation between fuel consumption and engine load is plotted at a given engine speed. A straight line law exists between the rate at which fuel is consumed and the engine load or brake power; commonly called Willan's line as shown in Figure below. By extrapolation, the fuel flow rate to give zero brake power. Since the fuel flow rate at zero necessary to overcome friction, and consequently, the amount of negative brake power at zero rate of fuel consumption represents the friction power. This method is not suitable for use with petrol engines as the engine load is controlled by the throttle opening thus the aerodynamic losses are changed and so the engine friction power is not fixed even the engine speed is maintained constant..



Representation of Willan's line

2- **Morse Test:** in this test method, the engine is running at specific speed and the brake power is measured. Then the work from the first cylinder is subtracted either by removing fuel injector (for diesel engine) or by removing the ignition supply (for petrol engine). According to this event, the engine speed is changed, to maintain this speed, the supplied fuel is changed, at this moment, the brake power is measured; which is produced by all engine cylinder except the first. Thus, the reduction of the brake power is due to the subtraction of the indicated power produced by the first cylinder that consumes the same power as a friction to maintain the engine speed or $iP_1 = bP_K - bP_{K-1}$. This sequence can be repeated for all engine cylinders to determine the indicated power of individual cylinders. For K cylinders, then the indicated power will be:

$$iP = bP - fP$$

$$\sum_{i=1}^{K} iP_k = \sum_{i=1}^{K} bP_k + \sum_{i=1}^{K} fP_k = bP_{For\ whole\ engine} - fP_{for\ whole\ engine}$$

Sometimes the engine cylinder can be assumed to be identical, in this case $iP = K * iP_1$. Thus this test method is not suitable for single cylinder engine.

3- **Motoring Test:** In the motoring test the engine is first run at a given speed and load conditions for sufficient time so that the temperature of the engine components, lubricating oil and cooling water reaches a steady state. A swinging field type electric dynamometer is used to absorb the power during this period. The ignition is then switched off and by suitable electric switching devices the dynamometer is converted to run as a motor at the same speed at which it was operated previously. The torque is measured under firing and under motoring conditions from which the bP and fP are evaluated.

4- **By measuring both indicated and brake powers.**

B- What are the major deficiencies of simple carburetor? *(5 P)*

The major **deficiencies of the simple (elementary) carburetor** include:

1. At low loads, the mixture becomes leaner; the engine requires the mixture to be enriched at low loads. The mixture is richest at idle.
2. At intermediate loads, the equivalence ratio increases slightly as the air flow rate increases; the engine requires an almost constant equivalence ratio.
3. As the air flow approaches the maximum (Wide-Open Throttle value WOT) value, the equivalence ratio remains essentially constant; the engine requires an equivalence ratio of about 1.1 at maximum engine power.
4. The elementary carburetor cannot compensate for transient phenomena in the intake manifold. It also cannot provide a rich mixture during engine starting and warm-up.

5. It cannot adjust to changes in ambient air density due to changes in altitude.

4) *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:*

(15 P)

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:

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

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

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

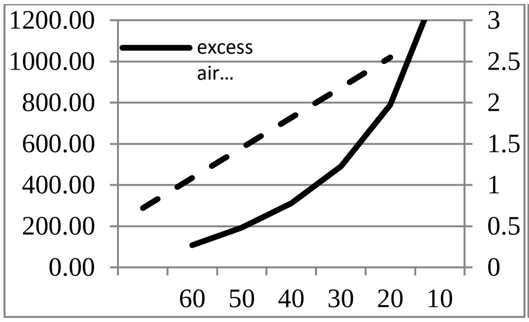
For general assumption the theoretical AF =16

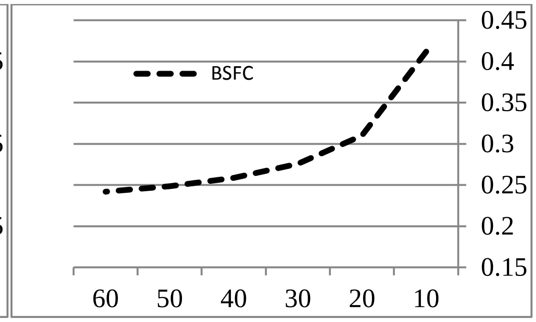
2- Excess air $\alpha = \frac{AF_{act}}{AF_{Th}} - 1 = \frac{0.0819/\dot{m}_f}{16} - 1 = \frac{0.0819/(0.000567 + 0.0000577\text{N}_e)}{16} - 1$

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





5) *The power output of a six-cylinder four-stroke engine is absorbed by a hydraulic dynamometer for which the law is* $\frac{WN}{20000}$ *kW, where the brake load W is in newton and the speed N is in rpm. The air consumption is measured by an air-box with a sharp-edged orifice system. The following observations are made:* *(15 P)*

Orifice diameter = 30 mm	*Engine speed = 2500 rpm*
Coefficient of discharge = 0.6	*C/H ratio by mass = 83/17*
Pressure drop across the orifice=14 cm of Hg	*Ambient pressure = 1 bar*
Bore = 100 mm	*Time for 100 cc of fuel consumption = 18 s*
Stroke =110 mm	*Ambient temperature = 27°C*
Brake load = 540 N	*Fuel density = 780 kg/m*^{3}

Calculate the volumetric efficiency, the bmep, the bp, the torque, the bsfc and the percentage of excess air.

Air density; $\rho_a = \frac{P}{RT} = \frac{101.325}{0.287*300} = 1.17\ \text{kg/m}^3$

Air mass flow rate:

$$\dot{m}_{air} = C_d A_o \sqrt{2gh\rho_f\rho_a} = 0.6 * \frac{\pi}{4} * 0.03^2 \sqrt{2 * 9.81 * 0.14 * 13600 * 1.17} = 0.0887\ \text{kg/S}$$

The theoretical mass of air to fill engine's cylinders:

$$\dot{m}_{th} = \rho_{air}.\frac{N}{i}.Z.Vs = 1.17 * \frac{2500}{2*60} * 6 * \frac{\pi}{4} * 0.1^2 * 0.11 = 0.126\ \text{kg/S}$$

Thus engine volumetric efficiency $\eta_{vol} = \frac{m_{act}}{m_{th}} = \frac{0.0887}{0.126} = 0.702$

Engine brake power: $bP = \frac{WN}{20000} = \frac{540*2500}{20000} = 67.5\ \text{kW}$

Thus the brake mean effective pressure: $bmeP = \frac{bP*60}{LAN/iZ} = \frac{67.5*60}{0.11*\frac{\pi}{4}*0.1^2*\frac{2500}{2}*6} = 625\ \text{kPa}$

The engine torque is computed from engine brake power: $bP = T\omega$ or $T = \frac{bP}{\omega} = \frac{67.5*1000}{2*\pi*\frac{2500}{60}} = 258\ \text{N.m}$

Fuel mass flow rate: $\dot{m}_f = \frac{V_f\rho_f}{Time} = \frac{100*0.78*3600}{18*1000} = 15.6\ \text{kg/hr}$

The BSFc is: $BSFC = \frac{\dot{m}_f}{bP} = \frac{15.6}{67.5} = 0.231 \quad \left[\frac{kg}{kW.hr}\right]$

The theoretical air/fuel ratio is: $AF_{Th} = \frac{1000}{233} * \left(0.83 * \frac{32}{12} + 0.17 * 8\right) = 15.34\ \frac{\text{kg of air}}{\text{kg of fuel}}$

While the actual AF ratio is: $AF = \frac{\dot{m}_{air}}{\dot{m}_f} = \frac{0.0887*3600}{15.6} = 20.47$

Thus the excess air factor will be: $x = \frac{AF - AF_{Th}}{AF_{Th}} = 0.33$

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