Benha University
Benha Faculty of Engineering
Mechanical Engineering Department
Examiners: Dr. Ali Mahmoud Ali Attia

Internal Combustion Engine (M528)
Second Semester – Final Exam
Date: 8/06/2014
Model Answer.

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1) *A- What is the effect of supercharging on SI engine performance? State the cases where it is proper to supercharging SI engine* *(5 P)*

The purpose of supercharging is to increase the power output without much deterioration of the engine efficiency or fuel economy. This will lead to overheating of the cylinder walls which may get damaged. The high inlet pressure also increases the peak pressure, so higher mechanical stresses. Moreover, for SI engines there will be a greater tendency for engine knocking. This is why the supercharging is not recommended for SI engines. To obtain greater power output by supercharging without the loss of efficiency and fuel economy, the knocking tendency of the fuel is suppressed by lengthening the ignition delay; either by use of new fuels or by the addition of a small quantity of tetraethyl lead. The injection of water in the intake manifold of the engine cylinder also reduces the knocking tendency. Thus SI engines can be safely supercharged in three cases:

1- Use of advanced combustion method as direct gasoline and stratifying combustion
2- Use of fuels having higher octane number as oxygenated and alcohol fuels
3- At high altitude where the initial temperature is reduced.

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.

C- State 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.

2) A- Derive the relation for air to fuel ratio for simple carburetor neglecting the compressible effect of air. *(8 P)*

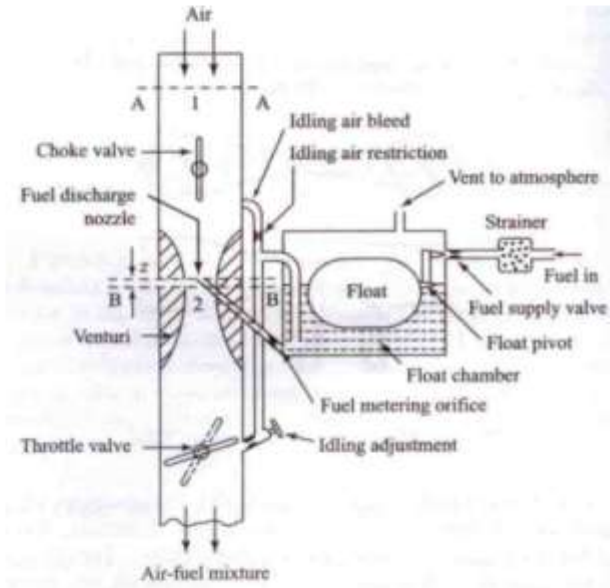


Figure: A schematic representation for simple carburetor

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:

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

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

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

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:

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

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 can be estimated by:

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

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: $C_{d,f} = \frac{\dot{m}_{fuel}}{\dot{m}_f}$, 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:

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

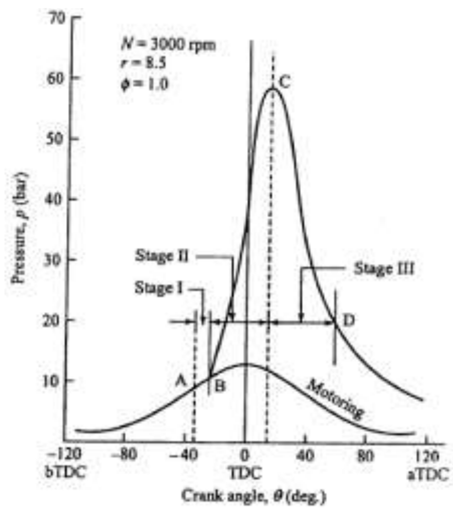
B- *Show in the indicated diagram the different events during the SI combustion. In this case, for SI engine at 1800 rpm the spark is started at 18º C.A. BTDC and the ignition lag about 8 º C.A. and the combustion process continued up to 12 º C.A. ATD. If the engine cylinder has diameter of 84 mm and the spark plug is offset 8 mm from the center. Assuming constant mixture properties and the end of combustion conditions what will be the angle at which spark should be started if the engine speed becomes 3000 rpm .* *(5 P)*

The combustion process in SI engines contains four events:

1- Spark initiation
2- Early flame development
3- Flame propagation
4- Flame termination.

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 measurable 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.



For spherical flame propagation from spark plug to the cylinder wall, the maximum flame travelling distance will be =0.084/2+0.008 = 0.05 m

The duration of the combustion process = (18-8)+12 = 22 º C.A. at 1800 rpm.

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Or the time of combustion will be: $t_c = \frac{22}{360 * \frac{1800}{60}} = 2.04\ mS$

In this case the average flame will be: $S_f = \frac{Distance}{time} = \frac{0.05}{0.00204} = 24.55\ m/S$

Due to the increase of the turbulence due to the increase of engine speed, the flame speed is proportionally increased according to: $\frac{S_{f,2}}{S_{f,1}} = A\left(\frac{N_2}{N_1}\right)$ where A is a proprtiona factor from 0.8 to 0.9. Let A=0.85

Then $S_{f,2} = 0.85 \cdot S_{f,1}\left(\frac{N_2}{N_1}\right) = 0.85 * 24.55 * \frac{3000}{1800} = 34.77\ m/S$

For constant distance traveled by the flame, the combustion duration will be:

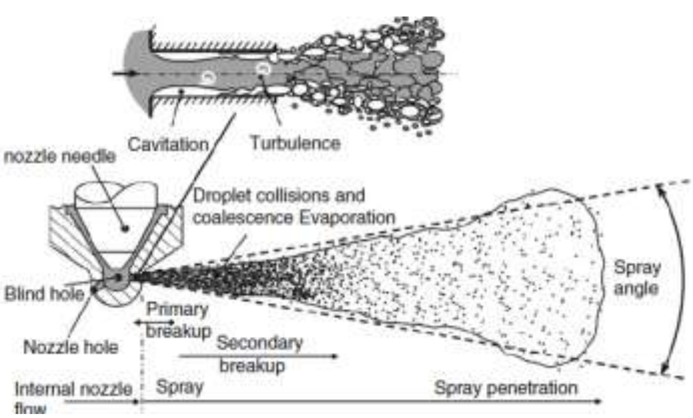
$$t_c = \frac{Distance}{Flame\ speed} = \frac{0.05}{34.77} = 1.44\ mS$$

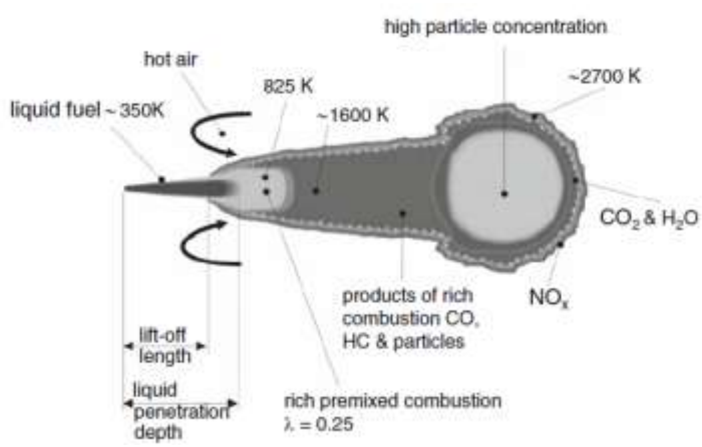
Thus $\theta_{comb} = t_c \cdot 360 \cdot \frac{3000}{60} = 25.9\,°\ C.A.$

For flame termination at θ = 12 º C.A. ATDC, then the combustion start at $\theta = 25.9 - 12 = 13.9\,°\ C.A.BTDC$, for ignition lag of 8 º C.A., then the spark will start $\theta = 13.9 + 8 = 21.9\,°\ C.A.BTDC$ thus the spark is advanced by $21.9 - 18 = 3.9\,°\ C.A.$

3) *A- 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.

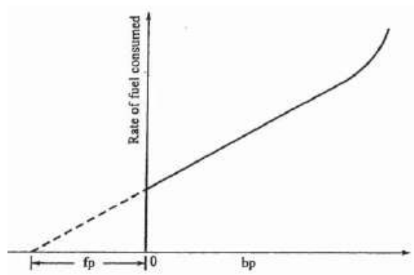




B- What are the experimental methods used to determine the engine friction power?

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.

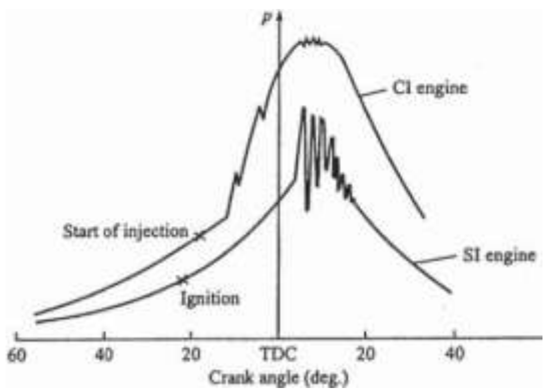
3- **Motoring Test:** In the motoring test the engine is first run at a given speed and load conditions for sufficient time so that the temperature of the engine components, lubricating oil and cooling water reaches a steady state. A swinging field type electric dynamometer is used to absorb the power 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.**

C- Differentiate between the engine knocking in SI and CI engines? With emphasize on the parameters that would reduce engine knocking in both engine types? (5 P)

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 in the first part, e. at the start of combustion (see figure below).
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.
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.



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; Table below presents a comparative statement of the various factors to be varied in order to reduce knock in SI and Cl engines.

Factors tending to reduce knock in SI and Cl engines

Factor	SI engines	Cl engines
Self-ignition temperature	High	Low
Time lag or delay period of fuel	Long	Short
Compression ratio	Low	High
Inlet temperature	Low	High
Inlet pressure	Low	High
Combustion chamber wall	Low	High
Engine speed, rpm	High	Low
Cylinder size	Small	Large

4) *A- A four-stroke four-cylinder compression ignition engine has a cylinder diameter of 77 mm, and stroke of 80 mm. An experiment is run at atmospheric conditions of 1 bar and 300 K on the engine at speed of 1500 rpm, and the following reading are obtained:* *(7 P)*

   a. *Diameter of the orifice of the air box = 2.65 cm*
   b. *Pressure drop across the orifice = 10 cm H_2O*
   c. *Coefficient of discharge across the air orifice = 0.65*
   d. *The rate of fuel mass flow rate - $\dot{m}_f$ (kg/s) to the engine changes with the engine brake power - bP (kW) according to the following relation: $\dot{m}_f = 0.0011 + 5.94 * 10^{-5} bP$,*

Draw the relation between the engine brake power and the following:

1- *Thermal efficiency,*
2- *Mechanical efficiency, and*
3- *Excess air factor (Take the theoretical A/F = 15.5).*

Solution:

For air (air density is calculated from conditions to be 1.16 kg/m^3):

$$\dot{m}_{Air} = C_d A\sqrt{2\Delta P\rho_a} = 0.65 * \frac{\pi}{4}(0.0265)^2\sqrt{2*(0.1*9.81*1000)*1.16} = 0.0171 \quad \text{kg/S}$$

1- The engine thermal efficiency will be: $\eta_{th} = \frac{bP}{\dot{m}_f \cdot LCV} = \frac{bP}{LCV(0.0011+5.94*10^{-5}bP)}$

The engine mechanical efficiency will be: $\eta_m = \frac{bP}{iP} = \frac{bP}{fP+bP}$

From Willan's line, use given relation at fuel flow rate of zero to get the friction power, then: $fP = \frac{0.00111}{5.94 \cdot 10^{-5}} = 18.69\ kW$

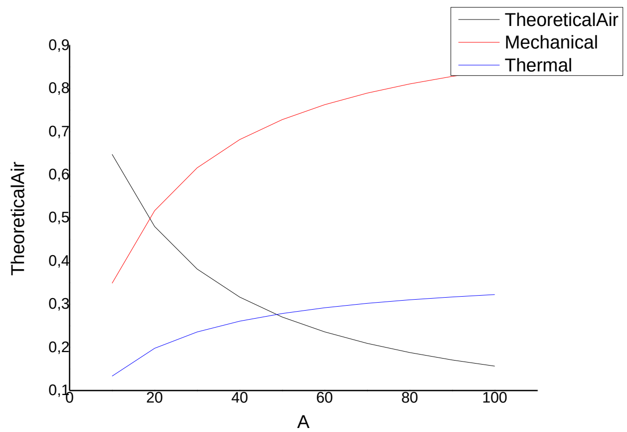
2- Then mechanical efficiency will be: $\eta_m = \frac{bP}{18.69+bP}$

For diesel engine at specific speed, the air mass flow rate is constant, and the excess air factor will depends only on fuel mass flow rate and hence brake power.

Theoretical air factor $\alpha = \frac{AF_{act}}{AF_{Th}} - 1 = \frac{0.0171/\dot{m}_f - 15.5}{15.5}$ or,

3- Theoretical air factor will be: $\alpha = \frac{0.0171}{15.5(0.0011+5.94*10^{-5}bP)} - 1$

Use the three equation to draw the required relations as shown below:



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B- 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: *(8 P)*

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

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

Solution:

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

Air mass flow rate:

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

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

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

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

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

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

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

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

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: $x = \frac{AF - AF_{Th}}{AF_{Th}} = 0.33$