Benha University	**Department: Mechanical Eng.**
Benha Faculty of Engineering	**Program: Mechanical Power**
Date: 30/5/2017	**Time: 3 hrs.**
Semester: External	**Subject: Internal Combustion Engines**
Examiner: Ali M.A. Attia	**Code: M1332**
Total Points: 90	**No. of Pages: 9**

Model Answer

Question ① 《*30 marks*》

A- Define: Fire point, Cloud point, Squish, Knocking, Lift-off length, Pre-ignition, Distillation, Airless injection, Cetane number, Indicator diagram, Pour point (10)

- ✓ Fire point: The fire point is the temperature at which enough vapors will rise to produce a continuous flame above the liquid fuel.
- ✓ cloud point is the temperature below which the wax content of the petroleum oil separates out in the form of a solid.
- ✓ Squish is radial inward motion of the gas mixture
- ✓ Knocking is the auto-ignition followed by a rapid pressure rise
- ✓ lift-off length is the axial distance between the injection nozzle and the diffusion flame.
- ✓ Pre-ignition is an uncontrolled inflammation of the combustible mixture in an engine by a hot surface before the spark occurs.
- ✓ Distillation is the separation of the more volatile parts of a liquid from those less volatile by vaporization and subsequent condensation,
- ✓ Airless injection is the fuel injection using very high pressure without the aid of the compressed air
- ✓ Cetane number is 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,
- ✓ Pour point is 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- What are the main parameters affecting engine volumetric efficiency? Hence state methods to measure/determine engine friction power (10)

Variables affecting volumetric efficiency:

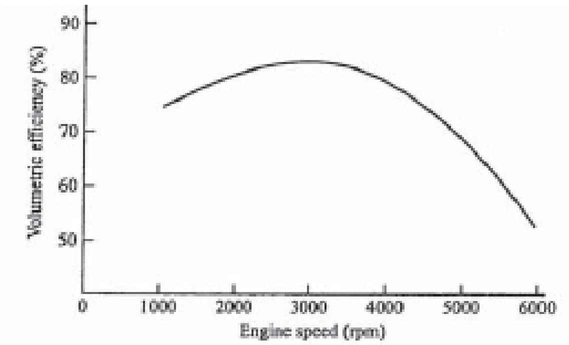
1- Engine speed: The variation of volumetric efficiency with engine speed is shown in Figure below. It can be noticed that, the maximum volumetric efficiency is obtain at a certain engine speed, then decreases at higher and lower speeds.

2- Fuel: For SI engine, the volumetric efficiency depends on the fuel type and how and when it is added, the fuel/air ratio, the fraction of fuel vaporized in the intake system, and the fuel heat of vaporization.

3- Heat transfer in the intake system: The temperature of the intake system is generally higher than the surrounding air temperature and will therefore heat the incoming air.

4- Viscous drag and flow restrictions: As the air passes through many paths (as air filter, carburetor, throttle plate, intake manifold, and intake valve), the pressure drop either due to viscous drag and friction in these passages will produce a higher reduction of air pressure entering the cylinder than that of the surrounding atmospheric air, and so the volumetric efficiency is reduced.

5- Exhaust gas reseals and exhaust gas recirculation: During the exhaust stroke, a fraction of burned gases does not go out of the cylinder but they are trapped in the clearance space..

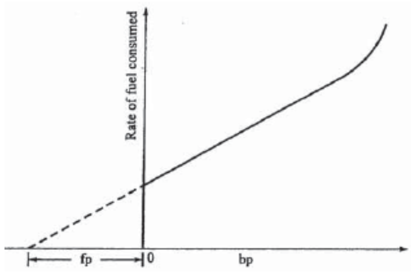


Dependence of volumetric efficiency on engine speed

Methods to measure/determine 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:** This method of determining the friction power and so indicated power is used for unthrottled compression ignition engine at specific engine speed no matter number of cylinders. 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. From this, the indicated power and mechanical efficiency can be evaluated. 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. The motoring is done to crank the engine at the same speed at which it was operating previously. The test is conducted as rapidly as possible. The torque is measured under firing and under motoring conditions from which the bP and fP are evaluated. Thus the iP and mechanical efficiency can be determined. The friction power determined by this method is reasonably good, but not accurate. Although the coolant temperature will change little during changeover, the piston and cylinder wall temperatures will change markedly, since the engine is prevented from firing during the test. The temperature of the working parts within the engine is low, and it is the temperature of the working parts which affects the viscous drag and hence the friction power. Also in absence of the exhaust blow-down, the pumping losses are not representative. The motoring method is suitable for assessing the relative contribution to the friction power of the many moving parts within an engine. Components such as piston rings, valve gear, the camshaft and all accessories can be removed in turn and the motor torque measured.

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

C- State methods of (i) engine supercharging, and (ii) fuel injection in engine cylinder (10)

Methods of 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 are powered mechanically by belt- or chain-drive from the engine's crankshaft. Most are driven by an accessory belt, which wraps around a pulley that is connected to a drive gear. The drive gear, in turn, rotates the compressor gear. To pressurize the air, a supercharger must spin rapidly – more rapidly than the engine itself. Making the drive gear larger than the compressor gear causes the compressor to spin faster. 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.

Methods of fuel injection include:

1. Air-injection system: Air-injection applies to systems injecting air along with the liquid fuel. It requires an air compressor for supplying air at 70 bar or even a higher pressure. A camshaft driven fuel pump meters and discharges a definite quantity of fuel into the injection valve. The following are the advantages of the air-injection system:
   - Very good atomization and distribution of the fuel, resulting in comparatively high mean effective pressures.
   - High viscous fuels, which are less expensive than those used by the engine having the solid

injection system, can be utilized without any problem.

- The fuel pump is required to develop less pressure than that required by the engine having the solid injection system.

The following are the disadvantages of the air-injection system.

- Complexity of the engine due to a multistage air compressor.
- A separate mechanical linkage is required to operate the fuel valve at the proper time.
- The fuel in the combustion chamber ignites very near to the injection nozzle, which may result in overheating and burning of the fuel valve and the valve seat.
- The fuel valve seating requires regular attention to guard against any leakage.

2. Airless-or solid-injection system: In this system the fuel is injected at a very high pressure into the combustion chamber without the aid of the compressed air. The main parts of this system are the fuel pump and the injector. Depending upon the location of the fuel pumps, the grouping, the method of actuating the pumps and the methods used to meter the fuel, the solid injection systems can be classified as follows:
   i. Individual pump system or the divided fuel-feed device. In this system, each cylinder is provided with one pump and one injector.
   ii. Unit injector system or the undivided fuel-feed device. The high-pressure pipe line connecting the individual pump and the associated injector can be avoided by the design of a unit injector. In this system the pump and the injector nozzle are combined in one housing.
   iii. Distributor system. The individual pump system, described earlier, requires a separate metering and compression pump for each cylinder, which increases the cost of the system.
   iv. Common rail system: in this system the pump serves to deliver fuel under high pressure into a common rail, called the header, with the pressure held constant by a pressure regulating valve.

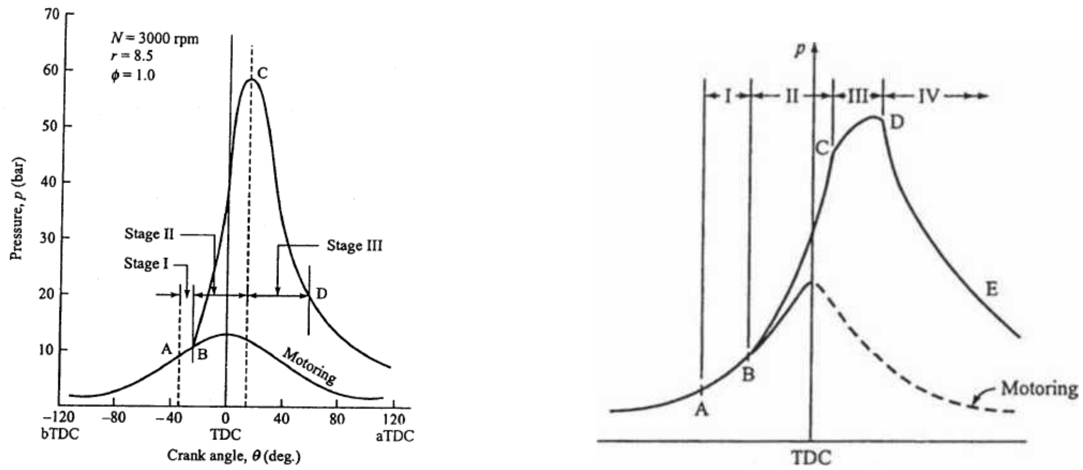
Question ② (30 marks)

A- What are the main combustion phases in both Spark ignition engines and compression ignition engines? Hence compare between Spark ignition engines and compression ignition engines. (10)

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 below. 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

Compare between SI and CI engines

SI engine	***Cl engine***
•It works on Otto cycle.	•It works on Diesel or Dual cycle.
•A fuel having higher self-ignition temperature is desirable, such as gasoline	•A fuel having lower self-ignition temperature is desirable such as diesel oil.
•Air and fuel mixture in gaseous form is inducted through the carburettor in the cylinder during the suction stroke.	•Only air is introduced into the cylinder during the suction stroke then fuel is injected at high pressure.
•The throttle valve of the carburettor controls the quantity of the charge, thus the charge quality remains almost fixed during normal running at variable speed and load; a quantity governed engine.	•The amount of air inducted is fixed but the amount of fuel injected is varied by regulating the quantity of fuel in the pump. The air-fuel ratio is varied at varying load. So, it is a quality governed engine.
•Spark is required to burn the fuel. For this, an ignition system with spark plugs is required. Because of this it is called a spark-ignition (SI) engine.	•Combustion of fuel takes place on its own with-out any external ignition system as the compressed air temperature exceeds fuel self-ignition temperature.
•A compression ratio of 6 to 10.5 is employed. The upper limit is fixed by the anti-knock quality of fuel. Engine tends to knock at higher compression ratios.	•A compression ratio of 14 to 22 is employed. The upper limit of compression ratio is limited by the rapidly increasing weight of the engine.
•Part load efficiency is poor, since even at part load the air/fuel ratio is not much varied. In order to improve the part load efficiency of the SI engine.	•Part load efficiency is good. As the load decreases, the fuel supply to the engine can also be reduced and lean mixture to the engine is then supplied.
•The cost of the petrol is higher than that of the diesel oil.	•The cost of diesel oil is less than that of petrol that has higher specific gravity.
•Noise and vibration are less because of less engine weight.	•Noise and vibrations are more because of heavier engine components due to higher compression ratio.
•The main pollutants are carbon monoxide (CO), oxides of nitrogen (NO_X) and hydrocarbons (HC)	•Apart from CO, NO_X and HC, soot or smoke particles are also emitted to the atmosphere.

B- What are the main engine emissions for Spark ignition engines and compression ignition engines? Hence provide main strategies to control engine emissions. (10)

For the same load and engine conditions, diesel engines emit 100 times more sooty particles than gasoline engines. Diesel engines also produce nearly 20 percent of the total nitrogen oxides (NOx) in outdoor air and 26 percent of the total NOx from on-road sources. Diesel buses and trucks are important contributors to smog (ground-level ozone) and fine toxic soot, two pollutants that have recently come under increased scrutiny because of their important public health impacts.

Thus, diesel engines produce higher amounts of NOx and PM (Diesel particulate matter (DPM) is largely comprised of elemental carbon particles, organic particles (unburned hydrocarbons), and sulfates) than some other prime movers and thus require control strategies; they are also notable for their low emissions of other pollutants – such as carbon monoxide (CO) and hydrocarbons (HC). Also, diesel powered engines produce significantly less carbon dioxide (CO2) per kilowatt-hour than those powered by gasoline, natural gas, or LPG.

The strategies applied to reduce the emissions

1- Engine Modifications

The new direction is to use the microprocessor to control the factors affecting the formation of different pollutants. The use of the microprocessor is joined with different experimental and theoretical data according to the exhaust gas temperature, oxygen concentration in the exhaust, and the engine operating conditions. The engine modifications include the improvements of combustion chamber designs to maximize air-fuel mixing and minimize local hot spots in addition to other improvements related to fresh air inlet and fuel injections.

2- Fuel Modifications

There are various strategies to improve fuel characteristics in purpose of emissions reductions including use additives to the current used fuel (composed of precious and non-precious metals (platinum, cerium) that can lead to NOx reduction from 5% to 10% and PM reduction from 10% to 40%), use of low sulfur diesel fuels (that leads to PM reduction ≈ 10% without any effect on NOx), use water-fuel emulsion (that leads to reduction of NOx by 15% and PM around 50%), and finally use of alternative fuels such as natural gas, Biodiesel, Propane and Ethanol.

3- After-treatment Strategies

Achievement of the regulated levels of NOx and PM may not be attained by the engine design improvements, so that, to reach lower levels of NOx and PM, various after-treatments of exhaust gasses will be necessary. Controlling NOx emissions from a diesel engine is inherently difficult because diesel engines are designed to run lean. In the oxygen-rich environment of diesel exhaust, it is difficult to chemically reduce NOx to molecular nitrogen. While a number of emissions control techniques show experimental promise, the following strategies have already achieved a practical level of commercialization in a variety of applications. By the use of diesel particulate filters catalysts, harmful particulates from cars have been reduced by over 95% and those from heavy-duty engines by around 90%. The challenge now is to target those emissions which remain critical for air quality and health - particulates and NOx - and to ensure that the increasing number of vehicles on the roads does not offset the gains already made. Most after-treatment strategies require the uses of ultra-low-sulfur diesel fuel (15 ppm) to prevent contamination of the various catalysts.

C- The spark plug is fired at 18° BTDC in an engine running at 1800 rpm. It takes 8° C.A. to start the combustion process while the flame is terminated at 12° ATDC. If engine bore diameter is 10 cm and the spark plug is offset 1 cm from the centerline of the cylinder. Calculate the effective flame front speed during flame propagation. If this engine will be operated at 3600 rpm calculate how much ignition timing must be advanced to maintain flame termination occurring at 12°ATDC. For this engine, consider that the improved in flame speed is proportional to engine speed with a constant factor of 0.7. (10)

Rotational angle during flame propagation is from 10°BTDC to 12° ATDC, which equals 22° C.A.
Time of flame propagation:

$$t = \frac{22}{360 * 1800/60} = 2.04\, ms$$

Maximum flame travel distance:
D_{max} = bore/2 + offset = (0.010/2) + (0.01) = 0.06 m
Thus the average flame speed will be:

$$s_f = \frac{D_{max}}{t} = \frac{0.06}{0.00204} = 29.5\ m/s$$

Since $\frac{s_f}{N} \propto 0.85$ then $s_{f2} = 0.7 * N2 * \frac{s_{f1}}{N1} = 0.7 * 3600 * \frac{29.5}{1800} = 41.3\ m/s$
Thus the time consumed by the flame to reach the end oc cylinder during combustion will be:

$$t_2 = \frac{D_{max}}{s_{f2}} = \frac{0.06}{41.3} = 1.4528\ ms$$

This time will be consumed during angle rotation obtained as follows:

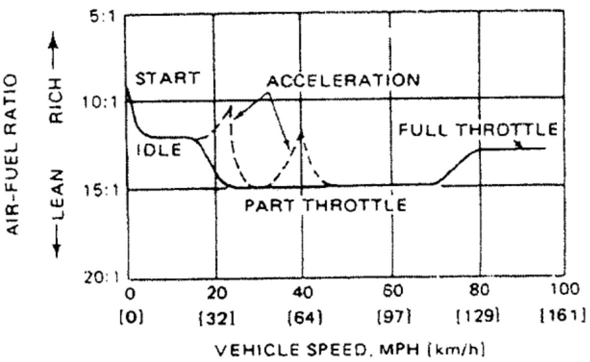
$$\triangle\, \theta = t_2 * \frac{N_2}{60} * 360 = 0.0014528 * 3600 * \frac{360}{60} = 31.4\,°\, C.A.$$

To keep moment of flame termination, considering ignition lag is constant, then the combustion shall start at $\theta = 19.4\,°\, BTDC$ or ignition will occur at 27.4 ^{0} C.A. BTDC, thus the spark is advanced by 9.4 ^{0} C.A.

<u>Question ③ (30 marks)</u>

A- State mixture requirements for SIE operation and state the main changes required in elementary carburetor to provide these requirements (10)

The maximum indicated thermal efficiency occurs at a fuel/air ratio of slightly leaner than the chemically correct fuel/air ratio as the excess air ensures complete combustion of the fuel. However, very lean mixture is not favorable as flame speed is reduced and so burning time loss s increased. The minimum time loss occurs at the slight rich mixture where flame speed is maximum. The mixture requirements for idling, part load, and high power ranges are shown in Figure below. The fuel system must vary the air-fuel ratio as engine load and operating conditions change. To start a cold engine, a rich mixture is needed. After the engine warms up, it can run on a lean mixture. When starting cold, the mixture is rich (about 9:1). During idle, the mixture leans out about 12:1; the engine is said to idle when it is operating at no external load with the throttle almost closed. At medium speed, it leans out further to about 15 : 1 (stoichiometric condition). If the driver "steps on the gas" to accelerate, the throttle valve opens and the engine suddenly begins taking in additional air. Extra fuel must be delivered to temporarily rich the mixture (shown in Figure below) or the engine will stall. The mixture is also enriched at full or wide-open throttle.



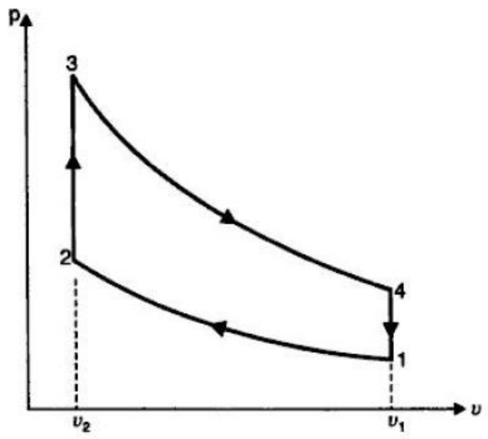
The modern carburetor is mainly the simple carburetor with addition to the following elements:

1. **Choke**: A *choke* must be added to enrich the mixture during cold starting and warm-up to ensure. 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, thus

gradually supplying the leaner mixture.

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.
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.
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.

B- For engine working on Otto cycle between given lower and upper temperature limits, show that for maximum work to be done per Kg, the ratio of compression is given by: $\boldsymbol{r} = (\frac{T_3}{T_1})^{\mathbf{1.25}}$. Also, prove that the intermediate temperatures for this condition are: $\mathbf{T_2} = \mathbf{T_4} = \sqrt{\mathbf{T_1 T_3}}$. (5)



Applying 1st law of thermodynamics for Otto cycle, then the cycle work done can be evaluated to be:

$$w = q_{add} - q_{rej} = c_v(T_3 - T_2) - c_v(T_4 - T_1)$$

For isentropic compression: $\frac{T_2}{T_1} = r^{\gamma-1}$ or $T_2 = T_1.r^{\gamma-1}$

Similarly for isentropic expansion: $T_3 = T_4.r^{\gamma-1}$ or $T_4 = T_3/r^{\gamma-1}$

Thus:

$$w = c_v(T_3 - T_1.r^{\gamma-1}) - c_v(T_3/r^{\gamma-1} - T_1)$$
$$= c_v(T_3 - T_1.r^{\gamma-1} - T_3/r^{\gamma-1} + T_1)$$

From this relation, work done is function of T_3, T_1 and r. thus for fixed temperature limits, at maximum work, $\frac{dw}{dr} = 0$ or

$$\frac{dw}{dr} = -(\gamma-1)T_1.r^{\gamma-2} - (1-\gamma)T_3.r^{-\gamma} = 0$$

Or $T_1.r^{\gamma-2} = T_3.r^{-\gamma}$ or $\frac{T_3}{T_1} = r^{2(\gamma-1)}$ then $r = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}}$ for air γ=1.4

then for maximum work $r = \left(\frac{T_3}{T_1}\right)^{1.25}$

at this condition,

$$T_2 = T_1 . r^{\gamma-1} = T_1 . \left(\frac{T_3}{T_1}\right)^{\frac{\gamma-1}{2(\gamma-1)}} = T_1 . \left(\frac{T_3}{T_1}\right)^{\frac{1}{2}} = \sqrt{T_1 . T_3}$$

Also,

$$T_4 = T_3 \div . r^{\gamma-1} = T_3 \div \left(\frac{T_3}{T_1}\right)^{\frac{\gamma-1}{2(\gamma-1)}} = T_3 \div \left(\frac{T_3}{T_1}\right)^{\frac{1}{2}} = \sqrt{T_1 . T_3}$$

Or

$$T_2 = T_4 = \sqrt{T_1 . T_3}$$

C- A single-cylinder 4-stroke gas engine of 20 cm bore and 38 cm stroke was tested with the following results: (15)
Barometric pressure of 720 mm Hg, atmospheric temperature of 17^0C, gas consumption of 0.165 m3/min at pressure of 8.8 mm water over atmospheric pressure, gas heating value is 18 MJ/ m3 at NPT, gas density is 0.61 kg/m^3 at NPT, hydrogen contents in gas is 13% on mass basis, air of 1.75 kg/min has Cp=1.05 kJ/kg.K and the exhaust gas temperature is 400^0C and steam Cp=2.1 kJ/kg.K. the received indicator diagram has two loops; positive loop with MEP of 560 kPa, and negative MEP of 50 kPa, engine speed of 250 rpm, brake torque of 400 N.m, cooling water rate is 5 kg/min with rise temperature of 40^0C.

Calculate: (a) the percentage of negative pumping power relative the positive indicated power, (b) the mechanical efficiency, (c) the engine heat balance relative to the supplied energy.

The percent of negative to positive indicated power is $\frac{50}{560} = 8.9\%$

The net indicated power is the positive indicated power minus the pumping power. Or

$$\mathrm{Ni} = \mathrm{IMEP}.\frac{\pi}{4}.\mathrm{d}^2.S.z.\frac{N}{i} = (560 - 50) * \frac{3.14}{4} * 0.2^2 * 0.38 * \frac{250}{60 * 2} = 12.7\ kW$$

Engine brake power: $Ne = T.2\pi.N = 400 * 2 * 3.14 * 250/60 = 10.5\ kW$

The engine mechanical efficiency: $\eta_{mech} = \frac{10.5}{12.7} = 82.7\%$

NPT: 760 mm Hg, 273 K, given gas conditions: 720+8.8/13.6=720.65 mm Hg and 190 K.

Thus gas flow at NPT is:

$$V_{NPT} = \frac{VP}{T}\frac{T_{NPT}}{P_{NPT}} = \frac{0.165 * 720.65}{290}\frac{273}{760} = 0.147\ m^3/min$$

Heat balance:

1- Supplied energy: $Qf = \dot{V}_f.HV = 0.147 * \frac{18000}{60} = 44.1\ kW$
2- Effective power: Ne=10.5 kW or 23.8% of supplied energy
3- Heat loss due to water cooling: $Qc = \dot{m}_w.cp\Delta T = \frac{5}{60} * 4.186 * 40 = 13.95\ kW$ or 31.6% of supplied energy
4- Heat loss in exhaust: total mass flow rate; $\dot{m}_g = \dot{m}_f + \dot{m}_a = 0.165 * 0.61 + 1.75 = 1.85\frac{kg}{min} = 0.0308\ kg/s$ from this mass, the water steam flow rate is; $\dot{m}_{steam} = 0.1/60 * 9 * 0.13 = 0.002\ kg/s$, thus the dray exhaust will be 0.0288 kg/s then $Qex = \dot{m}_{dr}.cp\Delta T + \dot{m}_{steam}.hfg = 0.0288 * 1.05 * 383 + 0.002 * (2250 + 2.1 * 383 - 17 * 4.186) = 17.5\ kW$ or 39.7% of supplied energy.
5- The unaccount of heat loss due to radiation and convection: $Qu = Qf - (Ne + Qc + Qex) = 44.1 - (10.5 + 13.95 + 17.5) = 2.15\ kW$ or 4.9% of the supplied energy.

Best Wishes for all,
Ali M.A. Attia

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B- What are the main parameters affecting engine volumetric efficiency? Hence state methods to measure/determine engine friction power (10)

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B- For engine working on Otto cycle between given lower and upper temperature limits, show that for maximum work to be done per Kg, the ratio of compression is given by: $\boldsymbol{r} = (\frac{T_3}{T_1})^{1.25}$. Also, prove that the intermediate temperatures for this condition are: $\mathbf{T_2 = T_4 = \sqrt{T_1 T_3}}$. (5)

C- A single-cylinder 4-stroke gas engine of 20 cm bore and 38 cm stroke was tested with the following results: (15)

Barometric pressure of 720 mm Hg, atmospheric temperature of 17^{0}C, gas consumption of 0.165 m3/min at pressure of 8.8 mm water over atmospheric pressure, gas heating value is 18 MJ/ m3 at NPT, gas density is 0.61 kg/m^3 at NPT, hydrogen contents in gas is 13% on mass basis, air of 1.75 kg/min has Cp=1.05 kJ/kg.K and the exhaust gas temperature is 400^{0}C and steam Cp=2.1 kJ/kg.K. the received indicator diagram has two loops; positive loop with MEP of 560 kPa, and negative MEP of 50 kPa, engine speed of 250 rpm, brake torque of 400 N.m, cooling water rate is 5 kg/min with rise temperature of 40^{0}C.

Calculate: (a) the percentage of negative pumping power relative the positive indicated power, (b) the mechanical efficiency, (c) the engine heat balance relative to the supplied energy.

Best Wishes for all,
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