ACE Engineering Academy
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ESE – 2019 MAINS
OFFLINE TEST SERIES

MECHANICAL ENGINEERING

TEST – 5
SOLUTIONS

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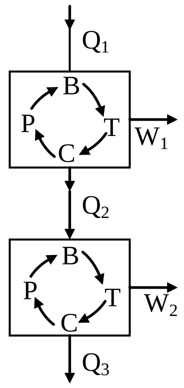
01(a).

Sol: If two Rankine cycles with two different working fluids are coupled in series, the heat lost by one is absorbed by the other, as in the mercury-steam binary cycle. Let η_1 and η_2 be the efficiencies of the topping and bottom cycles, respectively, and η be the overall efficiency of the combined cycle.



$$\eta_1 = 1 - \frac{Q_2}{Q_1} \quad \text{and} \quad \eta_2 = 1 - \frac{Q_3}{Q_2}$$

$Q_2 = (1-\eta_1)Q_1$ and $Q_3 = Q_2(1-\eta_2)$

$$\text{Now, } \eta = 1 - \frac{Q_3}{Q_1} = 1 - \frac{Q_2(1-\eta_2)}{Q_1}$$

$$= 1 - \frac{Q_1(1-\eta_1)(1-\eta_2)}{Q_1}$$

$\therefore \quad \eta = 1 - (1-\eta_1)(1-\eta_2)$ ------------ (1)

or $1-\eta = (1-\eta_1)(1-\eta_2)$

For n cycles coupled in series, the overall efficiency would be given by

$$1 - \eta = (1-\eta_1)(1-\eta_2) \ldots\ldots (1-\eta_n)$$

or $\quad 1-\eta = \prod_{i=1}^{n}(1-\eta_i)$

$\therefore$ Total loss = product of losses in all the cycles in series

For two-cycles coupled in series, from equation (i)

$$\eta = 1 - (1 - \eta_1 - \eta_2 + \eta_1\eta_2)$$

$\therefore \quad \eta = \eta_1 + \eta_2 - \eta_1\eta_1$

01(b).

Sol:

(i) ***Specific humidity:***

Specific humidity or humidity ratio (ω) is defined as the ratio of mass of water vapour (m_v) to the mass of dry air(m_a) in the mixture.

$$\omega = \frac{m_v}{m_a}$$

$$(\omega_s) = 0.622 \frac{P_s}{P_{atm} - P_s}$$

$$\therefore \ \omega = \frac{\frac{P_v v}{R_v T}}{\frac{P_a v}{R_a T}} = \frac{M_v}{M_a} \times \frac{P_v}{P_a} = \frac{18}{29} \times \frac{P_v}{P_a}$$

$$\omega = 0.622 \times \frac{P_v}{P_a}$$

$$\omega = 0.622 \times \frac{P_v}{(P_{atm} - P_v)} \ \frac{\text{kg of vapour}}{\text{kg of dry air}}$$

Saturated specific humidity

Where

P_v = water vapour pressure

P_a = dry air pressure

P_{atm} = atmospheric pressure

$P_{atm} = P_v + P_a$

(ii) **Relative Humidity:**

Relative Humidity (ϕ) is defined as the ratio of the partial pressure of the water vapour in the mixture to the saturation pressure (P_s) of water at the mixture temperature.

$$\phi = \frac{P_v}{P_s} = \frac{v_s}{v_v} = \frac{m_v}{m_{sat}}$$

$$\phi = \frac{P_v}{P_s} = \frac{\omega P_a}{0.622 P_s}$$

It can be shown that

$$\omega = 0.622\,\phi\left(\frac{P_s}{P_a}\right)$$

$$\phi = \left(\frac{\omega P_a}{0.622\ P_s}\right)$$

From above equations

$$\phi = \frac{\mu}{\left[1-(1-\mu).\left(\frac{P_s}{P_{atm}}\right)\right]}$$

Where

v_s = specific volume of saturated air

v_v =specific volume of water vapour

P_s = saturated pressure

μ = degree of saturation.

Note:

- ϕ = 0% $\Rightarrow$ no water vapour present in air
- ϕ=100% $\Rightarrow$ maximum amount of water vapour present in air

(iii) Wet bulb temperature:

But $T_{atm} > T_{sat}$

Hence, water vapour present in air is in superheated state.

- When unsaturated air is blown over the wet wick of a thermometer the water in the wick evaporates and air comes out as saturated air in this process cooling effect produces and the temperature drops to a new value and this temperature is called as the wet bulb temperature(WBT).
- Wet bulb depression(W.B.D)
  (W.B.D) = DBT – WBT
- WBD is maximum when ϕ = 0
  WBD = 0 when ϕ = 100%
  i.e. DBT = WBT

01(c).

Sol: Hot starting:

If the engine is started immediately after a hot shut down, the amount of fuel vapors entering the intake manifold will be high and the mixture formed in the combustion chamber will be too rich to ignite. It creates the problem of hot starting. To avoid it on hot days volatility temperature of first portion of fuel should be high. The requirement is opposite to that in cold starting. We use less volatile gasoline in summer than winter. In summer, than winter problem of hot starting can be avoided by proper design and proper placement of the fuel system. The fuel system should be placed away from hot engine parts.

Carburettor Icing:

Carburettor icing is formed due to vaporisation of gasoline into the air containing water vapor. The result is rapid drop in temperature of air fuel mixture and that of carburettor parts. Under some conditions ice is formed on throttle blade. Under idling conditions the ice slides down the throttle blade and restricts the passage preventing the flow of mixture past the throttle, thereby causing engine to run slower and stall Carburettor icing can be reduced by using less volatile fuels. Anti-icing additives like isopropyl alcohol or methyl alcohol and surface active materials which coat the metal surface with a film, thus minimising the tendency of ice to adhere to the surface.

Short and long trip economy:
In short trip driving the warm up period is quite significant. For efficient operation and greater economy, it requires fuel having relatively more volatility in the mid range section of distillation. In long trip driving the warm up period is insignificant compared to total driving. A gasoline having higher density will give more kilometres per litre in warm up engines.

Evaporation loss: Vaporisation and loss of lighter fractions of gasoline from the fuel tank and carburettor occur at all times. The evaporation loss depends on the vapor pressure of the fuel at storage temperature. It decreases fuel economy and antiknock quality of fuel since lighter fractions have higher antiknock properties. The evaporation loss is limited to 10% ASTM distillation temperature. The front end volatility temperature should be higher to reduce evaporation in order to reduce evaporation loss and vapor lock.

01(d)(i).

Sol: The ideal regenerative cycle is not practicable because:

(a) reversible heat transfer cannot be realize in finite time,

(b) heat exchanger in the turbine is mechanically impracticable, and

(c) the moisture content of the steam in the turbine is high, which leads to excessive erosion of turbine blades.

01(d)(ii).

Sol: Given:

m_a = 27 kg/sec,

CV = 43000 kJ/kg, m_f = 0.8 kg,

Thrust, F = 9 kN, c_i = 500 m/s,

$\dot{m}_f = \frac{0.18}{3600} \times 9000$ m = 0.45 kg/s

Air-fuel ratio = $\frac{27}{0.45}$ = 60 : 1

Thrust power, $P_T = F \times c_i$

$= 9 \times 500 = 4500$ kW

Heat input, Q = 0.45 × 43000 = 19350 kW

$\eta = \frac{P_T}{Q} = \frac{4500}{19350} \times 100$ = 23.26%

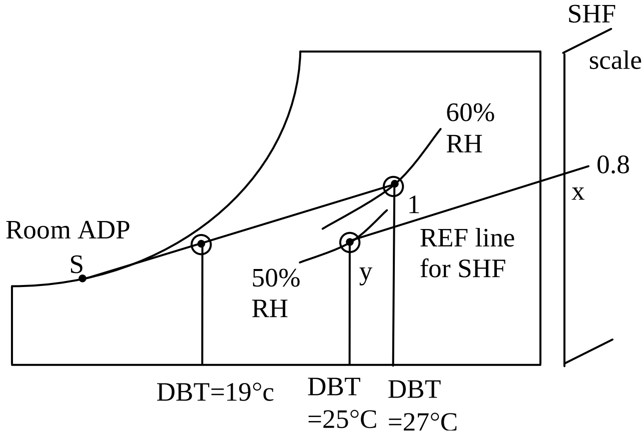
01(e).

Sol: Sensible heat load = SHL = 100,000 kJ/hr

Latent heat load = LHL = 60,000 kJ/hr

Sensible heat factor $= \frac{SHL}{SHL + LHL}$

$= \frac{100000}{100000 + 60000} = 0.8$



Mark DBT = 27°C

Point (1) and ϕ = 60%

On psychometric chart

From chart h_1 = 63 kJ/kg

$w_1 = 14 \times 10^{-3}$ kg vap/kg da

$v_1 = 0.87$ m^3/kg

Corresponding to SHF draw a line through (1) parallel to SHF reference line marked on chart. It intersects RH 100% line at room (ADP) point consider DBT = 19°C which is less than room DBT by 8°C

At 19°C erect a vertical which intersects the room (ADP) line drawn at (2) and (2) is the supply state of air to the room.

From chart,

h_2 = 49 kJ/kg da

$w_2 = 12 \times 10^{-3}$ m^3/kg. d.a

$v_2 = 0.843$ m^3/kg.d.a

Total heat load = SHL + LHL

= 160000 kJ/hr

$= \frac{160000}{3600}$ kW = 44.44 kW

$$\dot{m}_a(\text{kg/sec})(h_1 - h_2)\frac{\text{kJ}}{\text{kg}} = \text{THL(kW)}$$

$$\frac{\dot{V}_a}{60} \times \frac{1}{v^2}(h_1 - h_2)\frac{\text{kJ}}{\text{kg}} = \text{THL (kW)}$$

$\dot{V}_a$ = volume flow rate of air in cmm

$$\frac{\dot{V}_a}{60} \times \frac{1}{0.843}(63 - 49) = 44.44$$

$$\dot{V}_a = \frac{44.44 \times 60 \times 0.843}{14} = 160.57\,\text{cmm}$$

02(a).

Sol: Blade speed is given by

$$V_b = \frac{\pi DN}{60} = \frac{\pi \times 1 \times 3000}{60} = 157\,\text{m/sec}$$

Heat drop in the fixed blades is given by

$$\Delta h_{fixed} = \frac{V_i^2 - \phi V_0^2}{2 \times 1000 \times \eta}$$

Where,

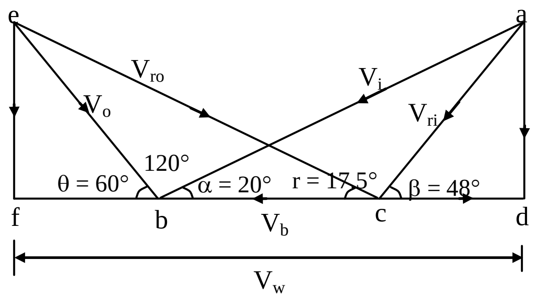
V_0 is the velocity at entry to fixed blades

V_i is the velocity at exit from fixed blades

η is the expansion efficiency

ϕ is the carry over co-efficient

It may be noted that velocity diagram is drawn for moving blades and therefore V_0 shown there is not the V_0 used in the equation. V_0 shown in the diagram is the absolute velocity of steam leaving the moving blades and going to second stage, if any. Thus this V_0 need not be equal to 100 m/sec. The values of V_0 = 100 and V_i = 250 are fixed blades before the moving blades.



The values are

V_{ro} = 202 m/sec, V_i = 250 m/sec,

V_o = 72 m/sec, V_{ri} = 116 m/sec

V_b = 157 m/sec, V_w = 270 m/sec

Therefore,

$$\Delta h_{fixed} = \frac{(250)^2 - 0.9 \times (100)^2}{2 \times 1000 \times 0.9}$$

Also, $\Delta h_{moving} = \frac{V_{ro}^2 - \phi V_{ri}^2}{2 \times 1000 \times \eta}$

Where,

V_{ri} is the relative velocity at entry to moving blades.

V_{ro} is the relative velocity at exit from moving blades.

The values of V_{ri} is measured from the velocity triangle bca drawn to scale.

V_{ri} = 116 m/sec

Therefore, $\Delta h_{moving} = \frac{V_{ro}^2 - 0.9 \times (116)^2}{2 \times 1000 \times 0.9}$

Also, degree of reaction is defined as,

$$R = \frac{\Delta h_{moving}}{\Delta h_{moving} + \Delta h_{fixed}}$$

$$\frac{1}{R} = \frac{\Delta_{moving} + \Delta h_{fixed}}{\Delta h_{moving}} = 1 + \frac{\Delta h_{fixed}}{\Delta h_{moving}}$$

Therefore,

$$\frac{1}{0.35} - 1 = \frac{\dfrac{(250)^2 - 0.9 \times (100)^2}{2 \times 1000 \times 0.9}}{\dfrac{V_{ro}^2 - 0.9 \times (116)^2}{2 \times 1000 \times 0.9}}$$

$$2.86 - 1 = \frac{(250)^2 - 0.9 \times (100)^2}{V_{ro}^2 - 0.9 \times (116)^2}$$

$$1.86 = \frac{6.25 \times 10^4 - 0.9 \times 10^4}{V_{ro}^2 - 1.213 \times 10^4}$$

$$V_{ro}^2 = \frac{5.35 \times 10^4}{1.86} + 1.213 \times 10^4$$

$= (2.875 + 1.213) \times 10^4 = 4.088 \times 10^4$

$V_{ro} = 202$ m/sec

Power developed $= \dfrac{m V_w V_b}{1000}$

$= \dfrac{1 \times 270 \times 157}{1000}$ = 42.39 kW/kg/sec

$$\eta_{stage} = \frac{V_w V_b}{1000(\Delta h_{moving} + \Delta h_{fixed})}$$

$$\Delta_{fixed} = \frac{V_i^2 - \phi V_o^2}{2 \times 1000 \times \eta}$$

$$= \frac{(250)^2 \times 0.9 \times (100)^2}{2 \times 1000 \times 0.9}$$

$= \dfrac{5.35 \times 10^4}{2 \times 1000 \times 0.9}$ = 29.7 kJ/kg

$$\frac{\Delta h_{fixed}}{\Delta h_{moving}} = 1.86$$

Therefore,

$$\Delta h_{moving} = \frac{\Delta h_{fixed}}{1.86} = \frac{29.7}{1.86} = 16\,kJ/kg$$

$(\Delta_{moving} + \Delta h_{fixed}) = 29.7 + 16 = 45.7$ kJ/kg

$\eta_{stage} = \dfrac{270 \times 157}{1000 \times 45.7}$ = 0.927 or 92.7 %

02(b).

Sol:

(i) ***Vortex tube system of refrigeration:***

- Vortex tube is a simple straight piece of tube into which compressed air flows tangentially and is so throttled that the central core of the air stream can be separated from the peripheral flow.

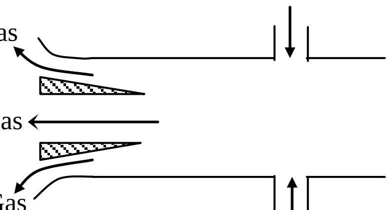


Fig: vortex tube refrigeration

- The central core of the air is separated either by uni-flow or counter flow method. The central core of the air stream is cold as compared to the hot gases at the periphery.

Production of low temperatures :

The various principles and processes involved in the production of low temperatures are as follows:

i) Throttling expansion of a liquid with flashing

ii) Reversible adiabatic expansion of a gas

iii) Irreversible adiabatic expansion (throttling) of a real gas

iv) Thermoelectric cooling

Adiabatic demagnetization

(ii) ***Thermostatic expansion valve:***

- It is the most widely used expansion valve as it is adaptable to any type of refrigeration system. It has very feeler Bulbhigh efficiency as well.

- Though its name is thermostatic yet it is not actuated by the change in temperature of the evaporator. It is actuated by the superheat of the refrigerant leaving the evaporator.
- Its working is based on maintaining a constant degree of sufficient superheat at the evaporator outlet. The evaporators remain filled with the refrigerant under all conditions of load.
- The principle of the thermostatic expansion valve is shown in fig. It consists of pressure bellows/diaphragm, a needle and the seat, a feeler bulb and the adjustable spring.

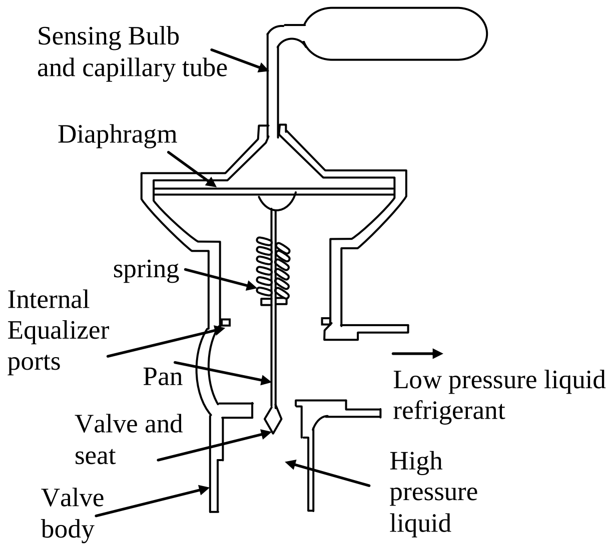


Fig: Thermostatic expansion valve

- The feeler bulb is fixed on the suction line at the outlet of the evaporator to sense the temperature changes of the refrigerant.
- The pressure of the feeler bulb liquid acts on one side of the bellows/diaphragm as it is connected to it by state of equilibrium because of the two opposing pressures.
- The valve setting gets disturbed, when the change in the degree of superheat is encountered, thereby it moves in the direction depending on which side the pressure is higher.
- Normally thermostatic expansion valves are adjusted for a 4.5 to 5.5^0C superheat

02(c).

Sol: Single cylinder 4 stroke

Volume of fuel consumed

$$V_f = 10\times10^{-6}\ m^3$$

Time taken, $t = 20.4$ sec

Density of fuel, $\rho_f = 700\ kg/m^3$

Mass flow rate of fuel = $\dot{m}_f = \dfrac{\rho_f V_f}{t}$

$$= \frac{10\times10^{-6}\times200}{20.4} = 0.343\times10^{-3}\ kg/sec$$

Volume of air consumed = $V_a = 0.1\ m^3$

Time = t = 16.3 sec

Mass flow rate of air = $\dot{m}_a = \dfrac{V_a\rho}{t}$

$$= \frac{(0.1)(1.175)}{16.3} = 7.21\times10^{-3}\ kg/sec$$

$$AFR = \frac{\dot{m}_a}{\dot{m}_f} = \frac{7.21\times10^{-3}}{0.343\times10^{-3}} = 21$$

$$BP = \frac{WN}{K}$$

W = 17 kgs, N = 3000 rpm

K = 5000, CV = 43,000 kJ/kg

$$BP = \frac{17\times3000}{5000} = 4.2\ kW$$

$$bsfc = \frac{\dot{m}_f\times3800}{BP} = \frac{0.343\times3600}{4.2}$$

$$= 294\ gm/kWhr$$

$$Br.\ Th.\eta = \frac{BP\times3600}{\dot{m}_f\times C_v}$$

$$= \frac{4.2\times3600}{0.294\times43000}\times100 = 28.48\ \%$$

03(a).

Sol: Altitude = 6000 m

AFR = 15:1

T = 300 K

P = 101.3 kPa

$t = t_s - 0.0065\ h$

$t_s = 27°C$

h = 6000 m

$t = 27° - 0.0065 \times 6000$

$= 27 - 39 = -12°C$

$= 273 - 12 = 261\ K$

$$h = 19220 \log_{10}\left(\frac{1.013}{P}\right)$$

h = 6000 m

$$6000 = 19220 \log_{10}\left(\frac{1.013}{P}\right)$$

$$\log_{10}\left(\frac{1.013}{P}\right) = \frac{6000}{19220} = 0.31217$$

$$= \frac{1.013}{P} = 2.05196$$

$$P = \frac{1.013}{2.05196} = 0.4937\ \text{bar} = 49.37\ \text{kPa}$$

Density of seal level = $\rho_0 = \frac{\rho_0}{RT_0}$

$$= \frac{101.3}{0.287 \times 300} = 1.176\ \text{kg/m}^3$$

Density of altitude = $\rho_{A\ell} = \frac{P}{RT}$

$$= \frac{49.37}{0.287 \times 261} = 0.659\ \text{kg/m}^3$$

$$\frac{(AFR)_{Altitude}}{(AFR)_{Sea\ level}} = \sqrt{\frac{\rho_{AL}}{\rho_0}}$$

$$\frac{(AFR)_{Altutude}}{15} = \sqrt{\frac{0.659}{1.176}}$$

$(AFR)_{Altitude} = 11.23$

03(b).

Sol: Combustion equation is

$$C_xH_y + a(O_2 + 3.76N_2) \rightarrow 8CO_2 + 0.9CO + 8.8O_2 + 82.3N_2 + bH_2O$$

Carbon Balance

$x = 8+0.9 = 8.9$

Nitrogen Balance

$3.76\times2\times a = 82.3\times2$

$$a = \frac{82.3}{3.76} = 21.89$$

Oxygen balance

$2a = 8\times2+0.9\times1+8.8\times2+b$

$2\times21.89 = 16+0.9+17.6+b$

$b = 9.28$

Hydrogen balance

$y = 2b = 2\times9.28 = 18.56$

Composition of fuel is

$C_xH_y \rightarrow C_{8.9}H_{18.56}$

$$\text{Air fuel ratio} = \frac{\text{mass of air}}{\text{Mass of fuel}}$$

$$= \frac{21.89[2\times16+3.76\times28]}{12\times8.9+1\times18.56}$$

$$= \frac{3005.06}{106.8+18.56}$$

$$= \frac{3005.06}{125.36} = 23.972$$

Stoichiometric Ratio

$$C_{8.9}H_{18.56}+a(O_2+3.76N_2)\rightarrow bCO_2+CH_2O+dN_2$$

Carbon Balance

$b = 8.9$

Hydrogen Balance

$2c = 18.56$

$c = 9.28$

Oxygen balance

$2a = 2b+c$

$$a = b+\frac{c}{2} = 8.9+\frac{9.28}{2} = 13.54$$

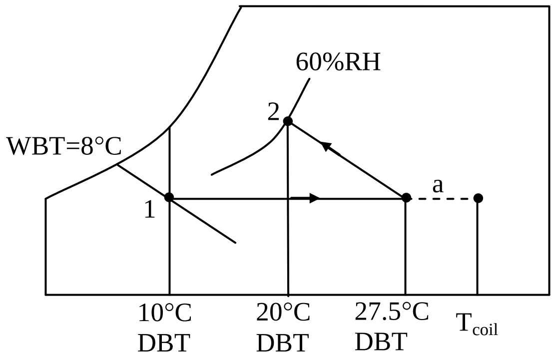
Nitrogen balance

2d = 7.52a

d = 3.76a = 3.76×13.54 = 50.91

03(c).

Sol:



Mark point (1) on psychometric chart

DBT = 10°C WBT = 8°C

From chart

h_1 =25 kJ/kg da

w_1 = 0.0058 kg vap/kg da

ν_1 = 0.81 m^3/kgda

Mark point (2) on psychometric chart

DBT = 20°C

RH = 60%

From chart

h_2 = 42 kJ/kgda

w_2 = 0.0088 kg vap/kg da

Draw a constant enthalpy line through (2).

From (1) drop a horizontal which intersects constant enthalpy line through (2) at (a)

h_a = h_2 = 42 kJ/kg da

w_a = w_1 = 0.0058 kg vap/kg da

T_a = 27.5°C

Amount of Air circulation

= 0.3 m^3/min/person

Number of persons = 50

Total amount of air circulation = 50 × 0.3

$$\dot{V} = 15\,m^3/min$$

Specific volume at inlet ν_1 = 0.81 m^3/kgda

Mass flow rate

$$= \dot{m} = \frac{\dot{V}}{\nu_1} = \frac{15}{0.81} = 18.52\,kg/min$$

Capacity of heating coil

$$= \dot{m}_a\left(\frac{kg}{sec}\right)(h_2 - h_1)\left(\frac{kJ}{kgda}\right)$$

$$= \frac{18.52}{60}[42 - 25] = 5.25\,kW$$

BPF of heating coil = 0.32

$$BPF = \frac{T_{coil} - T_a}{T_{coil} - T_1}$$

$$0.32 = \frac{T_{coil} - 27.5}{T_{coil} - 10}$$

0.32 T_{coil} – 3.2 = T_{coil} – 27.5

0.68 T_{coil} = 27.5 – 3.2 = 24.3

$$T_{coil} = \frac{24.3}{0.68} = 35.70 \approx 36°C$$

Capacity of humidifier

$$= \dot{m}_a(kg/sec)(w_2 - w_a)\frac{kgvap}{kgda}$$

$$= \frac{18.52}{60}[0.0088 - 0.0058]$$

= 9.26×10^{-4} kg/sec = 3.336 kg/hr

03(d)(i).

Sol: h = 35 mm of water

$$T_g = 280^o = 553\,K$$

$$T_a = 15^oC = 288\,K$$

$$m_a = 20\,kg/kg\,fuel$$

$$h = 353H\left[\frac{1}{T_a} - \frac{(m_a+1)}{m_a}\frac{1}{T_g}\right] mm\ of\ H_2O$$

$$35 = 353H\left[\frac{1}{288} - \left(\frac{20+1}{20}\right)\frac{1}{553}\right]$$

H = 63 meters.

03d(ii).

Sol:

- The maximum temperature of steam that can be used is fixed from metallurgical considertations, i.e. the materials used for the manufacture of the components which are subjected to the high pressure high temperature steam like the superheaters, valves, pipelines, inlet stages of the turbine and so on.
- The moisture content of steam in the later stages of the turbine is high, the entrained water particles along with the vapour coming out of the nozzles with high velocity strike the blades and erode their edges, as a result of which the life of the blades decreases.
- From the consideration of the erosion of blades in the later stages of a turbine, the maximum moisture content at the turbine exhaust is not allowed to exceed 12% of the quality of steam to fall below 88%.
- Therefore, with the maximum steam temperature at the turbine inlet, the minimum temperature of heat rejection and the minimum quality of steam at the turbine exhaust being fixed by the materials used, the ambient conditions, turbine inlet also gets fixed.

04(a).

Sol: From Willan's law

$$\dot{m}_s = a + bL$$

$\dot{m}_s$ = steam consumption rate in kg/hr

L = load in kW

a, b are constants

Steam apply at no load = 25000 kg/hr

$$25000 = a + b \times 0$$

a = 25,000 kg/hr

At full load

L = 100,000 kW

$$\dot{m}_s = 500{,}000\,\text{kg/hr}$$

$$\dot{m}_s = a + bL$$

500,000 = 25,000+b×100,000

b = 4.75 kg/kW hr

$$\dot{m}_s = a + bL$$

(a) steam rate $\dfrac{\dot{m}_s}{L} = \dfrac{a}{L} + b$

At 1/4 th load L = 25,000 kW

$$\text{Steamrate} = \frac{25{,}000}{25{,}000} + 4.75 = 575\,\text{kg/kWhr}$$

At 1/2 load L = 50,000 kW

$$\text{Steamrate} = \frac{25{,}000}{50{,}000} + 4.75 = 5.25\,\text{kg/kWhr}$$

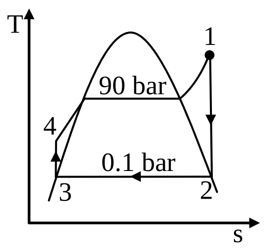
At 3/4 load L = 75,000 kW

$$\text{Steamrate} = \frac{25{,}000}{75{,}000} + 4.75 = 5.08\,\text{kg/kWhr}$$

At full load L = 100,000 kW

$$\text{Steamrate} = \frac{25{,}000}{100{,}000} + 4.75 = 5\,\text{kg/kWhr}$$

(b)



$$h_1 = 3511\frac{\text{kJ}}{\text{kg}}$$

$$s_1 = 6.8142\frac{\text{kJ}}{\text{kgK}}$$

$s_1 = s_2 < s_g$ at 0.1 bar. So wet state after expansion

$$s_1 = s_{f2} + x_2 (s_{fg})_2$$

$$s_{f2} = 0.6493; (s_{fg})_2 = 7.5009$$

$$6.8142 = 0.6493 + x_2 \times 7.5009$$

$$x_2 = \frac{6.8142 - 0.6493}{7.5009} = 0.822$$

$$h_2 = h_{f2} + x_2 (h_{fg})_2$$

$$= 191.83 + 0.822 \times 2392.8 = 2158.44 \text{ kJ/kg}$$

Neglecting pump work $h_3 = h_y$

$$h_3 = h_4 = 191.83 \text{ kJ/kg}$$

$$\eta_{rankine} = \frac{h_1 - h_2}{h_1 - h_4}$$

$$= \frac{3511 - 2158.44}{3511 - 191.83} = 0.407 \text{ or } 40.7\%$$

(c) $\eta_{actual} = \eta_{boiler} \times \eta_{cycle} \times \eta_{gen}$

$\eta_{boiler} = 100\%$ given

$$= \eta_{boiler} \times \frac{h_1 - h_2}{h_1 - h_4} \times \frac{\text{generator output}}{\dot{m}_s (h_1 - h_2)}$$

$$= 1 \times \frac{100{,}000}{500000 \times (3511 - 191.83)}$$

$$= 0.217 \text{ or } 21.7\%$$

(d) $\eta_{turbogenerator} = \dfrac{\text{generator output}}{\text{work output of turbine}}$

$$\frac{\text{generator output}}{\dot{m}_s (\text{kg/hr})(h_1 - h_2)\frac{\text{kJ}}{\text{kg}}} = \frac{100{,}000 \times 3600}{500{,}00 \times 1352.56}$$

$$= 0.532 \text{ or } 53.2\%$$

04(b).

Sol: Speed , N = 2000 rpm

Compression ratio , $r_k = 10$

Cylinder diameter, d = 0.1m

Stroke length , ℓ = 0.12 m

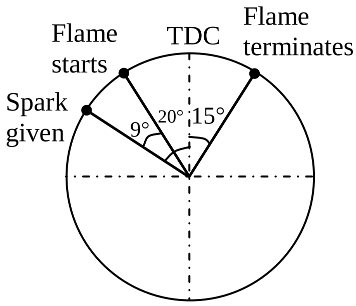
Connecting rod length , L = 20 cm = 0.2 m

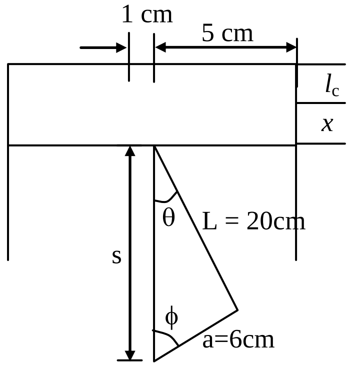
Spark plug offset, e = 0.01 m

Spark plug fired at 20° bTDC

Crank Rotation for flame propagation = 9°

Flame mode termination at 15° aTDC





Crank radius, $a = \dfrac{L}{2} = \dfrac{0.12}{2} = 0.06 \text{ m} = 6 \text{ cm}$

$$\frac{V_s}{V_c} = r_k - 1; \quad V_C = \frac{V_s}{r_k - 1}$$

$$V_c = \frac{\frac{\pi}{4} d^2 \ell}{r_k - 1}$$

$$\frac{\pi}{4} d^2 \ell_c = \frac{\frac{\pi}{4} d^2 \ell}{r_k - 1}$$

$$\ell_c = \frac{\ell}{r_k - 1} = \frac{12}{10 - 1}$$

(i) Piston displacement from TDC

$$x = L + a - a\cos\theta - \sqrt{L^2 - a^2 \sin^2\theta}$$

$$x = 20 + 6 - 6\cos 15^\circ - \sqrt{20^2 - 6^2 \sin^2 15^\circ}$$

$$= 26 - 6\cos 15^\circ - \sqrt{400 - 36 \sin^2 15^\circ}$$

$$= 6.0604 - 6\cos 15^\circ$$

$$= 0.2648 \text{ cm}$$

(ii) Flame travel distance $= \sqrt{(\ell_c + x)^2 + \left(\dfrac{d}{2} + e\right)^2}$

$$= \sqrt{(1.33 + 0.2648)^2 + (5 + 1)^2}$$

$$= 6.2083$$

Crank angle for which flame travels

$= 15° + 11° = 26°$

Time of flame travel, $t = \dfrac{\theta}{360} \times \dfrac{60}{N}$

$= \dfrac{26^o}{360^o} \times \dfrac{60}{2000}$

$= 2.1666 \times 10^{-3}$ sec

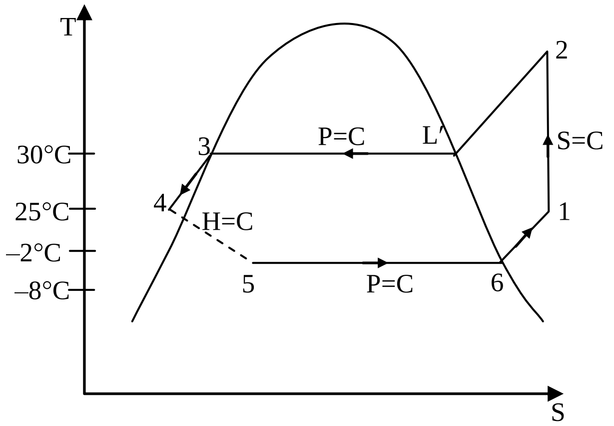
(iii) Effective flame speed = $\dfrac{\text{f t d}}{\text{time}}$

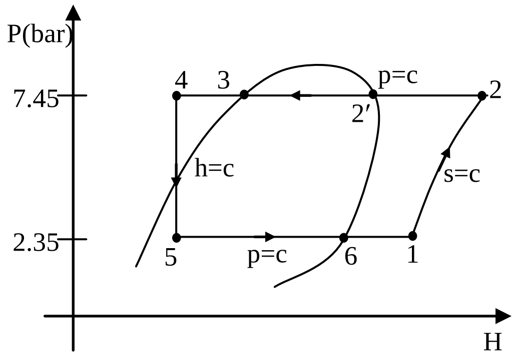
$= \dfrac{62083 \times 10^{-2}}{2.1666 \times 10^{-3}}$

$= 28.65$ m/sec

04(c).

Sol:





Net refrigerating effect = NRE

$= 20 \times 3.517 = 70.34$ kW

$h_6 = (h_g)_{-8°C} = 184.67$ kJ/kg

$h_2' = (h_g)_{30°C} = 199.62$ kJ/kg

$(h_3) = (h_f)_{30°C} = 64.59$ kJ/kg

$S_6 = (S_g)_{-8°C} = 0.7007$ kJ/kgK

$S_2' = (S_g)_{30°C} = 0.6853$ kJ/kg.K

$\nu_6 = 0.08$ m^3/kg ;

$\nu_2' = 0.02$ m^3/kg

$(C_{pg})_{EP} = 0.61$ kJ/kg.K

$(C_{p\ell})_{cp} = 1.235$ kJ/kg.K

$(C_{pg})_{cp} = 0.733$ kJ/kg.K

$S_1 = S_6 + (c_{p_g})_{EP}\, \ell n \dfrac{T_1}{T_6}$

$= 0.7007 + 0.61\ \ell n \dfrac{271}{265}$

$= 0.7144$ kJ/kg.K

$h_1 = h_6 + (c_{p_g})_{EP}(T_1 - T_6)$

$= 184.67 + 0.61\,(271 - 265)$

$= 188.33$ kJ/kg

$h_4 = h_3 - (c_{p_\ell})_{cp}(T_3 - T_4)$

$= 64.59 - 1.235\,(5)$

$h_4 = 58.42$ kJ/kg $= h_5$

$S_1 = S_2 = S_2^1 + (c_{p_g})_{cp}\, \ell n \dfrac{T_2}{T_2^1}$

$0.7144 = 0.6853 + 0.733\ \ell n \dfrac{T_2}{303}$

$\ell n \dfrac{T_2}{303} = \dfrac{0.7144 - 0.6853}{0.733} = 0.0397$

$T_2 = 303 \times e^{0.0397} = 315.3$ K

$h_2 = h_2^1 + (c_{p_g})_{cp}(T_2 - T_2^1)$

$= 199.62 + 0.733\,(315.3 - 303)$

$= 208.64$ kJ/kg

$\dfrac{\nu_6}{T_6} = \dfrac{\nu_1}{T_1}$

$\nu_1 = \dfrac{T_1}{T_6} \times \nu_6 = \dfrac{271}{265} \times 0.08 = 0.0818\, \dfrac{m^3}{kg}$

Specific volume at entrance to compressor = $\nu_1 = 0.0818$ m^3/kg

$$\dot{m}_r(\text{kg/sec})(h_1 - h_5)\frac{\text{kJ}}{\text{kg}} = \text{NRE}(\text{kW})$$

$$\dot{m}_r(188.33 - 58.42) = 70.34$$

Mass flow rate of refrigerant

$$\dot{m}_r = \frac{70.34}{129.91} = 0.5415\,\text{kg/sec} = 32.49 \text{ kg/min}$$

Work of compression (kW)

$$= \dot{m}_r(\text{kg/sec})(h_2 - h_1)\frac{\text{kJ}}{\text{kg}}$$

$$= 0.5415\ (208.64–188.33) = 11 \text{ kW}$$

$$\text{COP} = \frac{\text{NRE (kW)}}{w_C(\text{kW})} = \frac{70.34}{11} = 6.39$$

Clearance ratio = C = 0.02

$$\eta_{vol} = 1 + C - C\left(\frac{P_2}{P_1}\right)^{1/n}$$

$$= 1 + C - C\left(\frac{\nu_1}{\nu_2}\right)$$

$$\frac{\nu_2}{T_2} = \frac{\nu_2^1}{T_2^1}$$

$$\nu_2 = \frac{T_2}{T_2^1} \times \nu_2^1 = \frac{315.3}{303} \times 0.02 = 0.0208 \text{ m}^3\text{/kg}$$

$$\eta_{vol} = 1 + 0.02 - 0.02\left(\frac{0.0818}{0.0208}\right) = 0.9413$$

Number of cylinders = n = 2

Single acting,

Diameter = D

Length = 1.5 D

Speed = N = 1000 rpm

Volume flow rate (m^3/sec)

$$\dot{V} = \dot{m}_r\left(\frac{\text{kg}}{\text{sec}}\right) \times \nu_1\left(\frac{\text{m}^3}{\text{kg}}\right)$$

$$= 0.5415 \times 0.0818 = 0.0443$$

$$\frac{\pi}{4}D^2 L n \times \frac{N}{60} \times \eta_{vol} = \dot{V}\ (\text{m}^3/\text{sec})$$

$$\frac{\pi}{4}D^2 \times 1.5D \times 2 \times \frac{1000}{60} \times 0.9413 = 0.0443$$

$$D^3 = \frac{0.0443 \times 60 \times 4}{\pi \times 1.5 \times 2 \times 1000 \times 0.9413} = 2.99762 \times 10^{-4}$$

D = 0.067 m

Diameter = 6.7 cm

Length = 1.5 D = 1.5 × 6.7 = 10.05 cm

05(a).

Sol: Air injection system:

In air injection system, fuel is forced into the cylinder by means of compressed air. This system causes a bulky multi stage air compressor, this causes an increase in engine weight and reduces the brake power output. Good mixing of fuel with air with resultant higher mean effective pressure and ability to use the fuel of high viscosity are the advantages.

Solid injection:

In solid injection system the liquid fuel is injected into the combustion chamber without the aid of compressed air. Hence, it is called airless mechanical injection or solid injection system.

Depending upon the arrangement of the fuel pumps and injectors, solid injection system may be classified as:

(i) common rail system,

(ii) unit injector system,

(iii) individual pump and injector system.

Of these, the latter two are mostly used.

05(b).

Sol: The desirable characteristics of the working fluid in a vapour power cycle to obtain the best thermal efficiency are given below:

- The fluid should have a high critical temperature so that the saturation pressure at the maximum permissible temperature (metallurgical limit) is relatively low. It should have a large enthalpy of vaporization at that pressure.
- The saturation pressure at the temperature of heat rejection should be above the atmospheric pressure so as to avoid the necessity of maintaining vacuum in the condenser.
- The specific heat of liquid should be small so that little heat transfer is required to raise the liquid to the boiling point.
- The saturated vapour line of T-s diagram should be steep, very close to the turbine expansion process so that excessive moisture does not appear during expansion.
- The freezing point of the fluid should be below the room temperature, so that it does not get solidified while flowing through the pipelines.
- The fluid should be chemically stable and should not contaminate the materials of construction at any temperature.
- The fluid should be non-toxic, non-corrosive, not excessively viscous, and low in cost.

05(c).

Sol: T_{sat} = 30°C ; P_{sat} = 4.2431 kPa

ϕ_1 = Relative humidity = 0.75 or 75%

Apparatus dew point temperature

T_{ADP} = 12°C

Bypass factor of cooling coil = BPF = 0.15

Mass flow rate of air = $\dot{m}_a = 10\,\text{kg/sec}$

T_{sat} = 12°C ; P_{sat} = 1.4017 kPa

P_{atm} = 101.325 kPa

Enthalpy of condensate at 12°C = 50.24 kJ/kg.K

$$\phi_1 = \frac{P_{V_1}}{P_{sat}}$$

$$0.75 = \frac{P_{v_1}}{4.2431}$$

$$P_{v_1} = 3.182\,\text{kPa}$$

$$\omega_1 = 0.622\frac{P_{v_1}}{P_{atm} - P_{v_2}}$$

$$= 0.622 \times \frac{3.182}{101.325 - 3.182} = 0.622 \times \frac{3.182}{98.143}$$

$$\omega_1 = 0.0201\frac{\text{kg vap}}{\text{kg da}}$$

$$\omega_{ADP} = 0.622\frac{(p_{sat})_{ADP}}{p_{atm} - (P_{sat})_{ADP}}$$

At DPT RH = 100%

$$\therefore p_v = (p_{sat})_{ADP}$$

$$\omega_{ADP} = 0.622 \times \frac{1.4017}{101.325 - 1.4017}$$

$$= 0.622 \times \frac{1.407}{99.918} = 0.00876\frac{\text{kg vap}}{\text{kg da}}$$

$$BPF = \frac{\omega_2 - \omega_{ADP}}{\omega_1 - \omega_{ADP}}$$

$$0.15 = \frac{\omega_2 - 0.00876}{0.0201 - 0.00876}$$

$$\omega_2 = 0.00876 + 0.15\,(0.0201 - 0.00876)$$

$$= 0.01046\frac{\text{kg vap}}{\text{kg da}}$$

$$BPF = \frac{T_2 - T_{ADP}}{T_1 - T_{ADP}}$$

$$0.15 = \frac{T_2 - 12}{30 - 12}$$

T_2 = 12 + 0.15 (30–12) = 14.7°C

05(d).

Sol:

1. **Idling and low load:** (From no load to about 20% of rated power). The no load running mode of the engine is called idling. At idling and part load operation backflow occurs since the exhaust pressure is higher than the intake pressure. This increases the amount of residual gases. During the suction process the residual gases expand, there by reducing the fresh mixture inhaled.

   This requires the air fuel ratios used for idling and low loads about 20% of full load should be rich for smooth engine operation (F/A ratio 0.08:1 or A/F ratio 12.5:1).

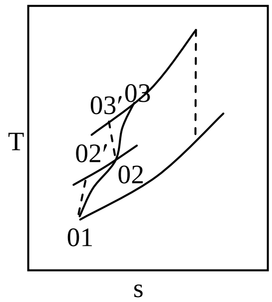
2. **Maximum power range:** (From about 75% to 100% rated power) It requires A/F about 14:1 besides providing maximum power rich mixture also prevents overheating of exhaust valve at high load to inhibit detonation at high loads, there is greater heat transfer to engine parts.

3. **Starting and warm up:** Starting from cold the speed as well as engine temperature are low hence much of heating ends supplied by the carburetor do not vaporize and remain in liquid form. Therefore during starting very rich mixture is required (A/F ratio 3:1)

4. **Acceleration:** due to opening the throttle the engine speed increases, however the main purpose of opening the throttle is to provide an increase in torque and whether or not an increase in speed follows depends on nature of load.

05(e).

Sol:



$$T_{02'} = T_{01}\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} = 273\times\left(\frac{1.05}{0.7}\right)^{0.286}$$

$$= 306.6\text{ K}$$

$$T_{01} = T_{01} + \frac{T_{02'} - T_{01}}{\eta_C}$$

$$= 273 + \frac{306.6-273}{0.83} = 313.5\text{K}$$

$$W_C = 1.005\times(313.5-273) = 40.7\,\text{kJ/kg}$$

With additional compressor

$$T_{03'} = 313.5\times\left(\frac{1.05}{0.7}\right)^{0.286} = 352\,\text{K}$$

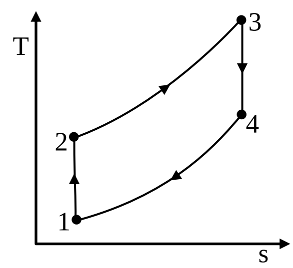
$$T_{03} = T_{02} + \frac{T_{03'} - T_{02}}{\eta_C}$$

$$= 313.5 + \frac{352.0-313.5}{0.83} = 359.96\,\text{K}$$

$$T_{03'} = \frac{344.3-273}{359.9-273}\times 100 = 82\%$$

06(a).

Sol:



P_1 = 1 bar,

T_1 = 273+27 = 300K,

T_3 = 273+820 = 1093 K

r_p = Pressure ratio $= \frac{P_2}{P_1} = \frac{P_3}{P_4}$

Simple cycle: 1–2

$Q = 0$; $s = c$

$$(T_2) = T_1 (r_p)^{\frac{\gamma-1}{\gamma}}$$

$T_2 = 600 \times 6^{0.4/1.4} = 500.5$ K

$$T_4 = \frac{T_3}{(r_p)^{\frac{\gamma-1}{\gamma}}} = \frac{1093}{(6)^{\frac{1.4-1}{1.4}}} = 655\,\text{K}$$

$$\text{Turbine work} = W_T = C_p(T_3 - T_4)$$
$$= 1.005[1093 - 655]$$
$$= 440.19\,\text{kJ/kg}$$

$$\text{Compressor work} = W_C = C_p(T_2 - T_1)$$
$$= 1.005(500.5 - 300)$$
$$= 201.5\,\text{kJ/kg}$$

$$W_{net} = W_T - W_C$$
$$= 440.19 – 201.5 = 238.69\ \text{kJ/kg}$$

$$\eta_{Th} = 1 - \left(\frac{1}{r_p}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= 1 - \left(\frac{1}{6}\right)^{\frac{1.4-1}{1.4}} = 0.4006 = 40.06\%$$

Maximum or optimum work condition for optimum work condition

$$T_2 = T_4 = \sqrt{T_1 T_3} = \sqrt{300 \times 1093} = 572.62\text{K}$$

$$W_{opt} = C_p\left[\sqrt{T_3} - \sqrt{T_1}\right]^2$$
$$= 1.005\left[\sqrt{1093} - \sqrt{300}\right]^2$$
$$= 1.005[33.06 - 17.32]^2$$
$$= 248.98\ \text{kJ/kg}$$

$$(\eta_{th})_{op} = 1 - \sqrt{\frac{T_1}{T_3}} = 1 - \sqrt{\frac{300}{1093}} = 0.4791 \text{ or } 47.61\%$$

Changes:

$$\%\ \text{changes in work} = \frac{W_{opt} - W_{net}}{W_{net}} \times 100$$
$$= \left(\frac{248.98 - 238.69}{238.69}\right) \times 100 = 4.31$$

$$\%\ \text{change in efficiency} = \frac{(\eta_{th})_{opt} - \eta_{th}}{\eta_{th}} \times 100$$
$$= \frac{47.61 - 40.06}{40.06} \times 100 = 18.85$$

06(b).

Sol:

(i) Supercharging of IC engines

The method of increasing the inlet air density, called supercharging, is usually employed to increase the power output of the engine. This is done by supplying air at a pressure higher than the pressure at which the engine naturally aspirates air from the atmosphere by using a pressure boosting device called a supercharger.

(ii) Morse Test:

The Morse test consists of obtaining indicated power of the engine without any elaborate equipment. The test consists of making inoperative, in turn, each cylinder of the engine and noting the reduction in brake power developed.

This test is applicable only to multi cylinder engines.

In this test the engine is first run at the required speed by adjusting the throttle in SI engine or the pump rack in CI engine and the output is measured. The throttle rack is locked in this position.

Then, one cylinder is cut out by short circuiting the spark plug in the SI engine or by disconnecting the injector in the CI engine. Under this condition all the other cylinders will motor the cut out cylinder and the speed and output drop. The engine speed is brought to its original value by reducing the load.

If there are k cylinders, then

$$ip_1 + ip_2 + ip_3 + ip_4 + \ldots + ip_k = \sum_{1}^{k} bp_k + fp_k \quad \text{----------(1)}$$

where, ip, bp and fp are respectively indicated, brake and frictional power and the suffix k stands for the cylinder number.

If the first cylinder is cut-off, it will not produce any power but it will have friction, then

$$ip_2 + ip_3 + ip_4 + \ldots + ip_k = \sum_{2}^{k} bp_k + fp_k \quad \text{--------------.(2)}$$

Subtracting Eqn (2) from Eqn (1)

$$ip_1 = \sum_{1}^{k} bp_k - \sum_{2}^{k} bp_k$$

Similarly we can find the indicated power of the cylinders, viz., $ip_2, ip_3, ip_4, \ldots, ip_k$

The total indicated power developed by the engine, ip_k, is given by

$$ip_k = \sum_{1}^{k} ip_k \quad \text{----------------------(3)}$$

When all the k cylinders are working, it is possible to find the brake power, bp_k, of the engine.

The frictional power of the engine is given by

$$fp_k = ip_k - bp_k \quad \text{------------------(4)}$$

(iii) Simple carburetor:

The process of preparation of combustible mixture by mixing the proper amount of fuel with air before admission to the engine cylinder is called carburation. A device which atomises the fuel and mixes it with air and prepares the charge for Otto cycle engine is called carburetor.

Simple Carburetor:

The carburetors are used on most SI engines for preparation of combustible air-fuel mixture charge. The figure shows a simple carburetor, which provides an air-fuel mixture for normal range at a single speed. It consists of a float chamber, fuel discharge nozzle and metering orifice, a venturi, a throttle valve and a chock valve.

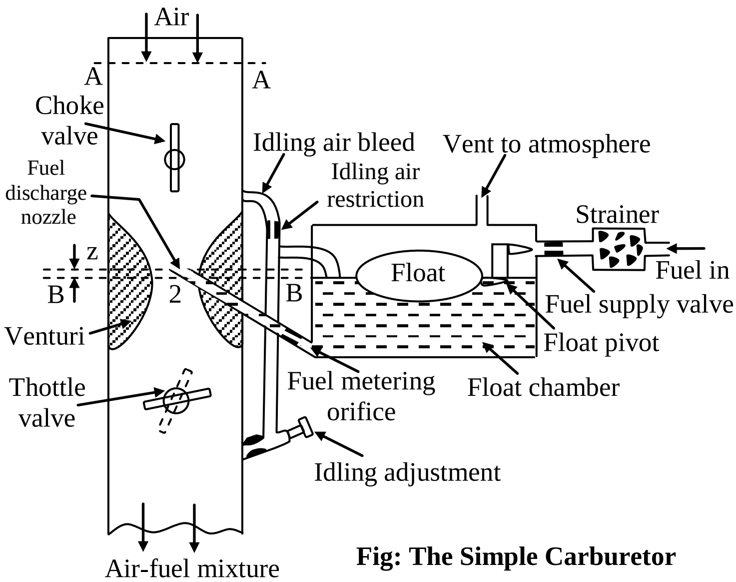


Fig: The Simple Carburetor

How it works: The basic carburetor as shown in fig. is built around a hollow tube called a throat. The downward motion of the piston creates a partial vacuum inside the cylinder that draws air into the carburetor's throat and past a nozzle that sprays fuel. The mixture of air and fuel produced inside the carburetor is delivered to the cylinder for combustion.

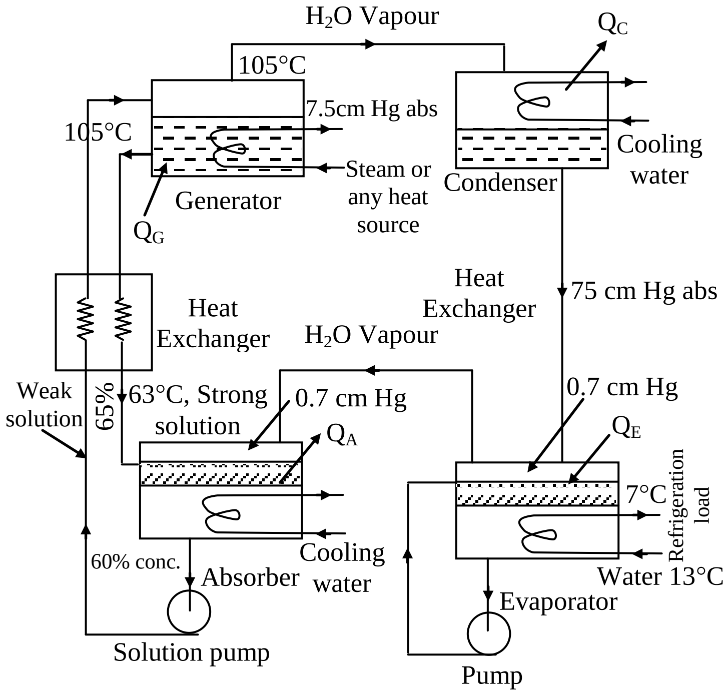
06(c)(i).

Sol:

Refrigeration Cycles :

There is another absorption refrigeration system, namely, lithium bromide-water vapour absorption (figure). Here the refrigerant is water and the absorbent is the solution of lithium bromide salt in water. Since water cannot be cooled below 0^0C, it can be used as a refrigerant in air conditioning units. Lithium bromide solution has a strong affinity for water vapour because of its very low vapour pressure. It absorbs water vapour as fast as it is released in the evaporator.

While the vapour compression refrigeration system requires the expenditure of 'high-grade energy in the form of shaft work to drive the compressor with the concomitant disadvantage of vibration and noise, the absorption refrigeration system requires only 'low-grade' energy in the form of heat to drive it, and it is relatively silent in operation and subject to little wear. Although the COP = Q_E/Q_G is low, the absorption units are usually built when waste heat is available, and they are built in relatively bigger sizes. One current application of the absorption system that may grow in importance is the utilization of solar energy for the generator heat source of a refrigerator for food presentation and perhaps for comfort cooling.



06(c) (ii).

Sol: $T_1 = 273 + 90 = 363$ K

$T_2 = 273 + 40 = 313$ K

$T_3 = 273 + 5 = 278$ K

$$COP = \eta_E \times (COP)_R$$

$$= \frac{T_1 - T_2}{T_1} \times \frac{T_3}{T_2 - T_3}$$

$$= \frac{363 - 313}{363} \times \frac{278}{313 - 278}$$

$$= \frac{50}{363} \times \frac{278}{45} = 0.85$$

07(a).

Sol: Initial state of air:

$T_1 = 17°C$

$\phi_1 = 60$ %

At DBT 17°C

$P_{sat} = 1.96$ kPa

$P_{atm} = 101.325$ kPa

$$\phi_1 = \frac{P_{v1}}{P_{sat}}$$

$0.6 = \frac{P_{v1}}{1.96}$;

$P_{v_1} = 1.176\,\text{kPa}$

$$\omega_1 = 0.622\frac{P_{v1}}{P_{atm} - P_{v1}}$$

$$= 0.622 \times \frac{1.176}{101.325 - 1.176}$$

$$= 0.622 \times \frac{1.176}{100.149} = 0.0073\frac{\text{kg vap}}{\text{kg d.a.}}$$

$P_{air} = P_{atm} - P_{v1} = 101.325 - 1.176$

$= 100.149$ kPa

$$P_{air} = \frac{P_{air}}{R_a T} = \frac{100.149}{0.287 \times 290} = 1.203\ \text{kg/m}^3$$

$$\nu_{air} = \frac{1}{\rho_{air}} = 0.8313\,\text{m}^3/\text{kg}$$

Volume flow rate of air = $\dot{v} = 0.5\text{m}^3/\text{sec}$

Mass flow rate of air = $\dot{m} = \frac{\dot{v}}{\nu_{air}} = \frac{0.5}{0.8313}$

$= 0.6014$ kg/sec

$T_{ADP} = 6°C$

$P_{sat} = 0.9433$ kPa $= P_v$

At RH 100%

$$\omega_{ADP} = 0.622\frac{P_{sat}}{P_{atm} - P_{sat}}$$

$$= 0.622 \times \frac{0.9433}{101.325 - 0.9433}$$

$$= 0.622 \times \frac{0.9433}{100.3817} = 0.00585\frac{\text{kg vap}}{\text{kg d.a.}}$$

Exit condition of air:

$T_2 = 9°C$

$P_{sat} = 1.1568$ kPa

$\phi_2 = 90$ %

$$\phi_2 = \frac{P_{v2}}{P_{sat}}$$

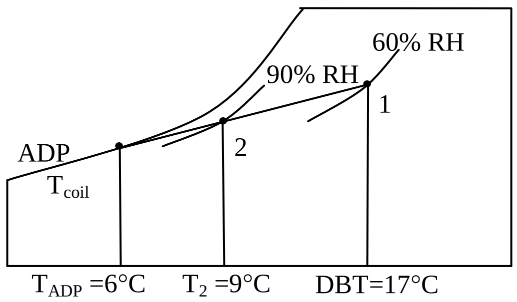
$$0.9 = \frac{P_{v2}}{1.1568}$$

$P_{v2} = 1.04$ kPa

$$\omega_2 = 0.622\frac{P_{v2}}{P_{atm} - P_{v2}}$$

$$= 0.622 \times \frac{1.04}{101.325 - 1.04}$$

$$= 6.45 \times 10^{-3}\frac{\text{kg vap}}{\text{kg d.a.}}$$



Rate of water condensation

$= \dot{m}_a(\text{kg/sec})(\omega_1 - \omega_2)$

$= 0.6014(0.0073 - 0.00645)$

$= 5.119 \times 10^{-4}$ kg/sec

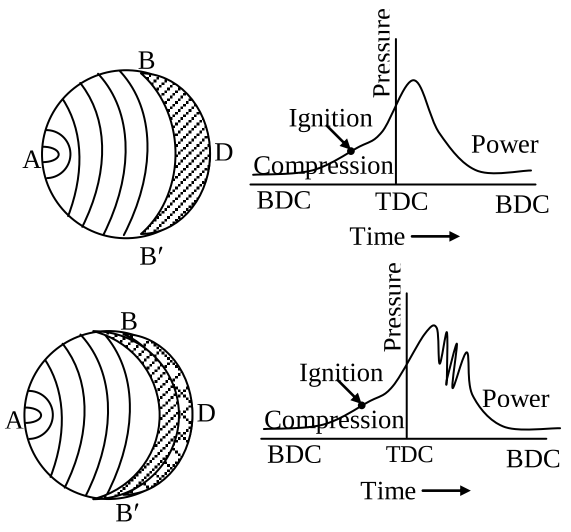
$= 1.8403$ kg/hr

$$\text{BPF} = \frac{T_2 - T_{adp}}{T_1 - T_{adp}} = \frac{9-6}{17-6} = \frac{3}{11} = 0.273$$

07(b).

Sol: The phenomenon of knock in SI engines:

If the temperature of the unburnt mixture exceeds the self – ignition temperature of the fuel and remains at or above this temperature during the period of preflame reactions(ignition lag), spontaneous ignition or auto-ignition occurs at various pin – point locations. This phenomenon is called knocking. The process of auto-ignition leads towards engine knock.



- In the normal combustion the flame travels across the combustion chamber from A towards D. The advancing flame front compresses the end charge BB′D farthest from the spark plug, thus raising its temperature.
- The temperature is also increased due to heat transfer from the hot advancing flame – front. If the temperature of the end charge had not reached its self – ignition temperature, the charge would not autoignite and the flame will advance further and consume the charge BB′D.
- This is the normal combustion process.
- If the end charge BB′D reaches its autoignition temperature and remains for some length of time equal to the time of preflame reactions the charge will autoignite, leading to knocking combustion.
- When flame has reached the position BB′, the charge ahead of it has reached critical autoignition temperature. During the preflame reaction period if the flame front could move from BB′ to only CC’ then the charge ahead of CC′ would autoignite.
- Because of the autoignition, another flame front starts travelling in the opposite direction to the main flame front. When the two flame fronts collide, a severe pressure pulse is generated.
- The gas in the chamber is subjected to compression and rarefaction along the pressure pulse until pressure equilibrium is restored. Gas vibration frequency in automobile engines is of the order of 5000 cps.

Knock Limited Parameters:-

1. **Knock Limited Compression ratio:** The knock limited compression ratio is obtained by increasing the compression ratio on a variable compression ratio engine until incipient knocking is observed.
2. **Knock Limited Inlet pressure**: The inlet pressure can be increased by opening the throttle or increasing supercharger delivery pressure until incipient knock is observed. An increase in knock limited inlet pressure indicates a reduction in the knocking tendency.
3. **Knock Limited Indicated mean effective pressure:** The indicated mean effective pressure measured at incipient knock is usually abbreviated as Klimep.

- This parameter and the corresponding fuel consumption are obviously of great practical interest.
- This number is defined as the ratio of Klimep with the fuel in question to Klimep with iso – octane when the inlet pressure is kept constant.

- This performance number is related to octane number and one of the advantages of this is that it can be applied to fuels whose knocking characteristics are superior to that of iso – octane, i.e., it extends the octane scale beyond 100.
- Relative Performance Number(rpn) which is defined as
- $$\text{rpn} = \frac{\text{Actual performance number}}{\text{Performance number corresponding to the imep of 100}}$$

4. Effect of engine variables on knock :-

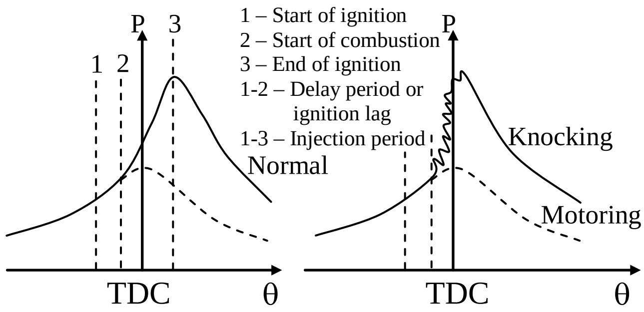
Density Factors: Reduce the temperature of the unburned charge should reduce the possibility of knocking by reducing the temperature of the end charge for autoignition. Any factor which reduces the density of the charge tends to reduce knocking by providing lower energy release.

Compression Ratio: Increase in compression ratio increases the pressure and temperature of the gases at the end of the compression stroke.

This decreases the ignition lag of the end gas and thereby increasing the tendency for knocking.

The phenomenon of knock in CI engines:

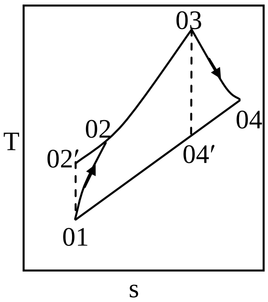
- A CI engine begins to give out black exhaust (due to incomplete combustion) when the AFR is close to the Stoichiometric ratio, because, in certain parts of the combustion chamber the AFR is too rich for complete lag, and during this period there is no pressure rise in the cylinder due to combustion.(Fig)



- Fuel injection in CI engines takes place over a certain time interval and as the initial droplets go through the delay period, additional droplets continue to enter the cylinder.
- If the ignition delay of the fuel is short, the first droplets will begin to burn soon after injection and a very small amount of fuel will accumulate in the combustion chamber.
- This will give normal combustion with smooth pressure variation in the cylinder.
- But if the ignition delay of the fuel is long, a considerable part of the fuel injected will accumulate and when ignition begins it will suddenly burn, causing rapid pressure rise and pressure fluctuations.
- This will result in vibrations and audible knocks in a phenomenon similar to SI engine detonation occur near the end of combustion, in CI engines knocks (called combustion or diesel knocks) due to detonation occur at the beginning of combustion.
- From the above we see that ignition delay of a fuel plays an important part in smooth combustion in a CI engine.
- The ignition delay of a CI engine fuel may be reduced by adding certain additives to the fuel.

07(c).

Sol:



The T-s diagram is as shown in the figure above.

$$T_{02'} = T_{01}\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} = 303\times 6^{0.2857} = 505.5\,K$$

$$T_{02} = T_{01} + \frac{T_{02'}-T_{01}}{\eta_C} = 303 + \frac{505.5-303}{0.87} = 535.8\,K$$

Neglecting the addition of fuel in the combustion chamber, we have $\dot{m}_f + \dot{m}_a \approx \dot{m}_a$.

$$\dot{m}_f = \frac{\dot{m}_a C_{pg}(T_{03}-T_{02})}{CV}$$

$$= \frac{80}{60}\times\frac{1.147\times(973-535.8)}{43100}$$

= 0.0155 kg/s

$$\frac{\dot{m}_a}{\dot{m}_f} = \frac{80}{60}\times\frac{1}{0.0155} = 86$$

A/F = 86

$$T_{04'} = T_{03}\left(\frac{p_{04}}{p_{03}}\right)^{\frac{\gamma-1}{\gamma}} = 973\times\left(\frac{1}{6}\right)^{0.2481}$$

= 623.8 K

$$T_{04} = T_{03} - \eta_T(T_{03}-T_{04'})$$

= 973 – 0.85 × (973 – 623.8) = 676.2 K

The net power of installation, W_N (neglect fuel)

$W_N = \dot{m}_a C_{pg}(T_{03}-T_{04}) - \dot{m}_a C_{pa}(T_{02}-T_{01})$

= 1.3333 × 1.147 × (973–676.2)–1.3333 × 1.005 × (535.8 – 303)

= 142 kW

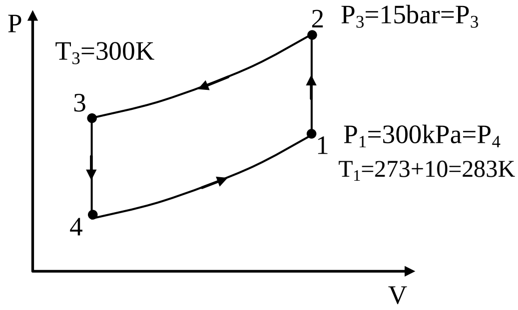
The overall thermal efficiency,

$$\eta_{th} = \frac{\text{Net power}}{\text{Heat input}} = \frac{142}{0.0155\times 43100}\times 100$$

= 21.25%

07(d).

Sol:



$P_2 = P_3 = 15$ bar; $P_1 = P_4 = 3$ bar

$T_1 = 283$ K; $T_3 = 300$ K

$$\text{Pressure ratio} = \frac{P_2}{P_1} = \frac{P_3}{P_4} = \frac{15}{3} = 5 = r_p$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 283(5)^{\frac{1.4-1}{1.4}} = 448\text{ K}$$

$$T_4 = \frac{T_3}{(r_p)^{\frac{\gamma-1}{\gamma}}} = \frac{300}{(5)^{\frac{1.4-1}{1.4}}} = 190\,K$$

Net refrigerating effect = NRE = 3.517 kW

$$\dot{m}_a(\text{kg/sec})c_{p_a}(\text{kJ/kg.K})(T_1-T_4) = \text{NRE(kW)}$$

$$\dot{m}_a\times 1.005\times(283-190) = 3.517$$

$$\dot{m}_a = \frac{3.517}{1.005\times 93} = 0.03763\,\text{kg/sec}$$

Compressor work (kW) = W_C

$= \dot{m}_a(\text{kg/sec})c_{p_a}(\text{kJ/kg.K})(T_2-T_1)$

= 0.03763 × 1.005 ×(448–283)

= 6.24 kW

Expander work (kW) = W_E

$= \dot{m}_a(\text{kg/sec})\times c_{p_a}(\text{kJ/kg.K})\times(T_3-T_4)$

$= 0.03763 \times 1.005 \times (300–190)$

$= 4.16$ kW

Net work $= W_{net} = W_C - W_E$

$= 6.24 - 4.16 = 2.08$ kW

$$\text{COP} = \frac{\text{NRE}(\text{kW})}{W_{net}(\text{kW})} = \frac{3.517}{2.08} = 1.69$$

Compressor inlet volume flow rate,

$P_1 = 300$ kPa

$\dot{m} = 0.0373$ kg/sec

$T_1 = 283$ K

$R = 0.287$ kJ/kg.K

$$\dot{V}_1 = \frac{\dot{m}RT_1}{P_1} = \frac{0.03763 \times 0.287 \times 283}{300}$$

$= 0.01019\ m^3/sec = 36.68\ m^3/hr$

Volume flow rate at turbine exit,

$P_4 = 300$ kPa

$\dot{m} = 0.03763$ kg/sec

$T_4 = 190$ K

$R = 0.287$ kJ/kg.K

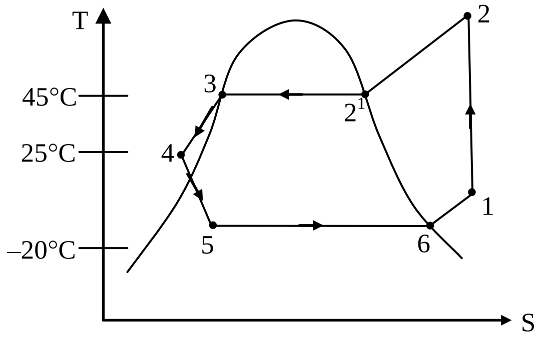
$$\dot{V}_4 = \frac{\dot{m}RT_4}{P_4} = \frac{0.03763 \times 0.287 \times 190}{300}$$

$= 6.84 \times 10^{-3}\ m^3/sec = 24.63\ m^3/hr$

08(a).

Sol: Net refrigerating effect = 10 tons

$= 10 \times 3.517 = 35.17$ kW



$(h_6) = (h_g)_{-20°C} = 178.7$ kJ/kg

$(h_3) = (h_f)_{45°C} = 79.7$ kJ/kg

c_{p_f} at condenser pressure = 1.02 kJ/kg.K

c_{p_g} at condenser pressure = 0.755 kJ/kg.K

c_{p_g} at evaporator pressure = 0.61 kJ/kg.K

$h_4 = h_3 - c_{p_f}(T_3 - T_4)$

$= 79.7 - 1.02(45–25)$

$= 59.3$ kJ/kg $= h_5$

Heat lost in sub-cooler = Heat gain in evaporator

$c_{p_f}(T_3 - T_4) = c_{p_g}(T_1 - T_6)$

$$T_1 = T_6 + \frac{c_{pf}}{c_{pg}}(T_3 - T_4)$$

$$= -20 + \frac{1.02}{0.61}(45 - 25)$$

$= 13.44°C$

$S_6 = (S_g)_{-20°C} = 0.7088$ kJ/kg.K

$$S_1 = S_6 + c_{p_g} \ell n \frac{T_2}{T_6}$$

$$= 0.7088 + 0.61 \ell n \frac{273 + 13.44}{273 - 20}.$$

$= 0.7845$ kJ/kg.K

$h_1 = h_6 + c_{pg}(T_1 - T_6)$

$= 178.7 + 0.61\,[13.44–(–20)]$

$= 199.09$ kJ/kg

$S_1 = S_2 = 0.7845$ kJ/kg.K

$$S_1 = S_2' + c_{p_g} \ell n \frac{T_2}{T_2'}$$

$$0.7845 = 0.6812 + 0.255 \ell n \frac{T_2}{318}$$

$$\ell n \frac{T_2}{318} = \frac{S_1 - S_2'}{c_{p_g}}$$

$$= \frac{0.7845 - 0.6812}{0.755} = 0.137$$

$T_2 = 318e^{0.137} = 364.69$ K

$h_2 = h_2' + c_{p_g}(T_2 - T_2')$

= 204.9 + 0.755 (364.69 – 318)

= 240.16 kJ/kg

$\dot{m}_r(kg/sec)(h_6 - h_5)$ kJ/kg=NRE (in kW)

$\dot{m}_r(kg/sec)(178.7 - 59.3) = 35.17$

$\dot{m}_r = \frac{35.17}{178.7 - 59.3} = \frac{35.17}{12.116}$ = 0.295 kg/sec

W_c(kW) = $\dot{m}_r(kg/sec)(h_2 - h_1)kJ/kg$

= 0.295(240.16–199.09) = 12.116 kW

COP = $\frac{NRE(kW)}{W_c(kW)} = \frac{35.17}{12.116}$=2.9

Condenser load in (kW)

= NRE(kW) + W_c(kW)

= 35.17 + 12.116

Condenser heat rejection = 47.286 kW

08(b).

Sol: 4 stroke engine

Bore diameter = D = 0.3 m

Stroke length = L = 0.46 m

Speed = N = 200 rpm

Carbon =C = 87%

Hydrogen = H = 13%

Composition of exhaust gases

CO_2 = 7% = C_1

O_2 = 10.5% = O_2

N_2 = 82.5% = N_2

Mass of air per kg fuel = m_a

$= \frac{CN_2}{33(C_1)} = \frac{87 \times 82.5}{33 \times 7} = 31.07\,kg/kg\,fuel$

Theoretical mass of air required

$= \frac{1}{0.23}\left[\frac{8}{3}C + 8\left(H_2 - \frac{O_2}{8}\right) + S\right]$

$= \frac{1}{0.23}\left[\frac{8}{3} \times 0.87 + 8(0.13 - 0) + 0\right]$

(Oxygen and sulphur in fuel is zero)

= 14.61 kg Air/kg fuel

Mass flow rate of fuel = $\dot{m}_f = 6.75\,kg/hr$

Air fuel ratio = $\frac{m_a}{m_f} = 31.07$

Actual mass flow rate of air = $\dot{m}_f \times AFR$

= 6.75 × 31.07

= 209.73 kg/hr

P_a = 100 kPa

T = 17°C = 273 + 17 = 290 K

R_a = 0.287 kJ/kg.K

Density of Air

$\rho_a = \frac{P_a}{R_a T} = \frac{100}{0.287 \times 290} = 1.201\,kg/m^3$

Theoretical air flow rate

$= \dot{m}_{TL} = V_s \times \rho_a \times \frac{N}{120}$

$= \frac{\pi}{4}D^2 L \times \rho_a \times \frac{N}{120}$

$= \frac{\pi}{4} \times 0.3^2 \times 0.46 \times 1.201 \times \frac{200}{120}$

= 0.06505 kg/sec

= 0.06505 × 3600 kg/hr

= 234.18 kg/hr

$\eta_{volumetric} = \frac{\dot{m}_a}{\dot{m}_{Th}} = \frac{209.73}{234.18} \times 100 = 89.56\%$

08(c).

Sol: 4 stroke of cylinders = 1

Cylinder diameter = D = 0.15 m

Stroke length = L = 0.25 m

Area of Indicator diagram = 450 mm^2

Length of Indicator diagram = L_d = 50 mm

Indicator spring constant

$= \frac{9.81 \times 10^4\,N/mm^2}{1.2} = 81.75\,kPa/mm$

Engine speed = N = 40 rpm

Brake Torque = 225 Nm

Fuel consumption = 3 kg/hr

Calorific value of fuel = 44,200 kJ/kg

Cooling water flow rate = $\dot{m}_w = 4\,kg/min$

Cooling water temperature rise

$= (\Delta T)_w = 42°C$

Specific heat of cooling water

$= c_{p_w} = 4.2\,kJ/kg.K$

Indicator spring rating pressure = 1.2 mm

$= 9.81\ N/cm^2$

$= 9.81 \times 10^4\ N/m^2$

= 98.1 kPa

Indicated mean effective pressure

$$= \frac{\text{Area of indicator diagram}}{\text{Length of indicator diagram}} \times \text{spring constant}$$

$$P_{mi} = \frac{450\,mm^2}{50\,mm} \times 81.75\,kPa/mm$$

= 735.75 kPa

$$\text{Indicated power (kW)} = \frac{P_{mi}\,L\,A\,N\,n}{120}$$

$$= \frac{735.75 \times 0.25 \times \frac{\pi}{4}(0.15)^2 \times 40 \times 1}{120}$$

= 1.083 kW

$$\text{Brake power (kW)} = \frac{2\pi NT}{60{,}000}$$

$$= \frac{\pi \times 40 \times 22.5}{30000} = 0.0942\,kW$$

Friction power = IP – BP

= 1.083 – 0.0942 = 0.988 kW

(i) Mechanical efficiency

$$\eta_m = \frac{BP}{IP} = \frac{0.0942}{1.083} = 8.69\ \%$$

$$\text{Heat liberated by fuel} = \dot{m}_f\left(\frac{kg}{sec}\right) \times cv\left(\frac{kJ}{kg}\right)$$

$$= \frac{3}{3600} \times 44{,}200 = 36.83\,kW$$

(ii) Brake thermal efficiency $= \dfrac{BP \times 3600}{m_f \times CV} \times 100$

$$= \frac{0.0942}{36.83} \times 100 = 0.255\ \%$$

(iii) Brake specific fuel consumption

$$= \frac{\dot{m}_f(kg/hr)}{BP\,(kW)} = \frac{3}{0.0942} = 31.84\ kg/kWhr$$

Heat converted to brake power (kW)

= 0.0942 kW

Heat converted to friction power (kW)

= 0.988 kW

Heat carried away by cooling water

$$\dot{\theta}_w = \dot{m}_w\left(\frac{kg}{sec}\right) \times c_{p_w}\left(\frac{kJ}{kg.K}\right) \times (dT)_w$$

$$= \frac{4}{60} \times 4.187 \times (42) = 11.73\,kW$$

Heat carried away by exhaust gases and radiation

$$= \{\text{Heat liberated by fuel}\} - \{BP + FP + \dot{\theta}_w\}$$

= 36.83 – 0.0942 – 0.988 – 11.73

= 24.01 kW

Description	kW	Description	kW	%
Heat supply by fuel (x)	36.83	1. BP	0.0942	0.255
		2. FP	0.988	2.68
		3. Heat carried away by exhaust and radiation	24.01	65.19
		4. Heat carried away by cooling water	11.73	31.85
Total	**36.83**	**Total**	**36.83**	**100%**