

# ESE – 2019 MAINS OFFLINE TEST SERIES

## **MECHANICAL ENGINEERING**

# TEST – 7 Solutions

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**01(a).** 

Sol: Taking the direction normal to the surface of the plate to be the x direction with x = 0 at the top surface, the mathematical formulation can be expressed as

$$\frac{d^2T}{dx^2} = 0$$

Integrating the differential equation twice with respect to x yields

$$\frac{dT}{dx} = C_1$$
$$T(x) = C_1 x + C_2$$

where  $C_1$  and  $C_2$  are arbitrary constants. Applying the boundary condition give

$$\mathbf{x} = \mathbf{0}: -\mathbf{k} \frac{\mathrm{dT}(\mathbf{0})}{\mathrm{dx}} = \dot{\mathbf{q}}_{o} = -\mathbf{k}\mathbf{C}_{1} \Longrightarrow \mathbf{C}_{1} = -\frac{\dot{\mathbf{q}}_{o}}{\mathbf{k}}$$

 $x = 0: \ T(0) = T_o = C_2$ 

Substituting  $C_1$  and  $C_2$  into the general solution, the variation of temperature is determined to be

$$T(x) = -\frac{\dot{q}_o}{k}x + T_o$$

At the top surface (x = 0), the net heat flux absorbed by the solar collector is

$$\begin{split} \dot{q}_{o} &= \alpha \dot{q}_{solar} - \epsilon \sigma \left( T_{o}^{4} - T_{sur}^{4} \right) - h \left( T_{o} - T_{\infty} \right) \\ \dot{q}_{o} &= (0.9)(500) - (0.9)(5.67 \times 10^{-8})[(35 + 273)^{4} \\ &- (0 + 273)^{4}] - (5)(35 - 25) \\ \dot{q}_{o} &= 224 \text{ W/m}^{2} \end{split}$$

**01(b).** 

Sol: 2 – Stroke engine :

BP = 368 kWFP = 73.6 kW 
$$\begin{split} \dot{m}_{f} &= 180 \text{ kg/hr} \\ AFR &= 20 \\ (CV)_{fuel} &= 42000 \frac{\text{kJ}}{\text{kg}} \\ IP &= BP + FP = 368 + 73.6 = 441.6 \text{ kW} \\ \eta_{mech} &= \frac{BP(\text{kW})}{IP(\text{kW})} = \frac{368}{441.6} \times 100 = 83.33 \end{split}$$

Air consumption =  $20 \times 180 = 3600$  kg/hr Indicated thermal efficiency

$$= \frac{IP(kW) \times 3600}{\dot{m}_{f}\left(\frac{kg}{hr}\right) \times CV\left(\frac{kJ}{kg}\right)}$$
$$= \frac{441.6 \times 3600}{180 \times 42000} = 0.2102 = 21.02\%$$

Brake thermal efficiency

$$= \frac{BP(kW) \times 3600}{\dot{m}_{f}\left(\frac{kg}{hr}\right) \times CV\left(\frac{kJ}{kg}\right)}$$
$$= \frac{368 \times 3600}{180 \times 42000} = 0.1752 = 17.52\%$$

**01(c).** 

## Sol: The desirable properties of an ideal refrigerant are divided into three categories.

- 1. Thermodynamic properties.
- 2. Safe working properties.
- 3. Physical properties.

#### 1. Thermodynamic properties:

- A good refrigerant should have **low boiling point** at atmospheric pressure.
- On the other hand, it is desirable that the **operating pressures should be positive**. i.e., above atmospheric pressure so that there is no leakage of air or moisture in the system.

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- It is desirable that the freezing point of the refrigerant should be much lower than the evaporator temperature so that the refrigerant does not freeze in the evaporator.
- *Latent heat.* Since the cooling in a refrigeration cycle is produced by the evaporation of liquid refrigerant, it is desirable that the refrigerants should have high latent heat.
- *Evaporator and condenser pressures*. The operating pressures in the evaporator and condenser should be positive but not very high as it would result in high capital as well as operating costs.

As discussed earlier, positive pressure in evaporator and condenser is desirable to avoid leakage of air or moisture in the system.

#### Coefficient of performance and power per ton.

- These two parameters are vital in selecting a refrigerant.
- *Critical temperature and pressure.* The critical temperature of a vapour is the temperature above which the vapour cannot be condensed irrespective of the pressure on the system.

#### 2. Safe working properties:

• *Flammability*: It is desirable that the refrigerants should be non-flammable and

non-explosive individually and when mixed with lubricating oil or air.

• *Toxicity:* Leakage of the refrigerant from the refrigeration system is inevitable. It is therefore desirable that it does not have any adverse effect on the human beings.

### Effect on refrigerated products:

• It is desirable that the refrigerants which are used in applications such as domestic refrigerator or cold storage plants where materials are stored for preservation do not adversely affect these products if by accident it comes in contact with these.

#### **Non-Corrosively:**

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The refrigerant should not affect the materials used in the refrigeration system. Keeping in mind the refrigerant to be used for the system, the materials are selected.

### **Chemical Stability:**

• The refrigerants should be chemically stable at the temperatures and pressures normally encountered in the system.

### 3. Physical Properties:

• *Leak-tendency*: The refrigerant may either leak out of the system or air may leak in the system. The most common causes of leakage are wearing out of joints or the materials used for the fabrication of the system. A dense refrigerant will have fewer tendencies to leak as compared to less density refrigerant. Moreover the detection of leaks should be easy to avoid loss of refrigerant.



- *Viscosity:* Viscosity of refrigerant both in liquid and vapour state plays an important role as it determines the heat transfer rate in evaporator and condenser. Lower the viscosity better is the heat transfer rate.
- *Thermal conductivity:* Thermal conductivity also determines the heat transfer rate in evaporator and condenser. Higher thermal conductivity of refrigerant is desirable.
- Specific heat: As the specific heat of refrigerant in liquid and vapour state is different in order to get maximum refrigerating effect. it is desirable that liquid refrigerant should have low specific heat to increase the sub-cooling of liquid and vapour refrigerant, high specific heat to decrease the super heating of liquid. The overall effect of this will be an increase in the refrigerating effect.

#### Freon-12 (R –12):

- $(CCl_2F_2)$  It is a quite widely used refrigerant
- non flammable,
- non-toxic and
- non-explosive.
- It is chemically stable and is difficult to break even under extreme operating conditions.
- If it is brought in contact with open flame or heater elements, it decomposes into highly toxic constituents.
- It has not only excellent safe properties but also condenses at moderate pressures under normal atmospheric conditions.

- Its boiling point at atmospheric pressure is 29.8°C, freezing point is –157.7°C,
- It is miscible with compressor lubricating oil under all operating conditions.
- As it condenses at moderate pressure and has low boiling point. It is suitable for all applications and can be used with different type of compressors.
- Its main disadvantages is low refrigerating effect which is however not so serious as compared to its other advantages.

#### **01(d).**

Sol: We have

Draught across the fuel bed = 25 mm of water =  $g \times 25 \text{ N/m}^2$ 

Draught to impart the velocity head

$$= \frac{V^{2}}{2g} = \frac{10 \times 10}{2g} = 5.1 \text{ metres}$$
$$= 5.1 \times 1.293 \times \text{gN/m}^{2}$$
$$= 5.1 \times 1.293 \times \text{g} = 6.6 \times \text{g N/m}^{2}$$

∴ Total draught developed by the force draught fan= g (25 + 6.6) N/m<sup>2</sup> = 310 N/m<sup>2</sup>
 Also volume of air handled by fan is given by

$$V = \frac{mRT}{n}$$

Where m is the mass of air handled

= 10,000×13 = 130000 kg/hr  
T = (20+273) = 293°K, p = 1×10<sup>5</sup> N/m<sup>2</sup>,  
R = 287 N-m°K  
∴ V = 
$$\frac{130,000 \times 287 \times 293}{1 \times 10^5}$$
  
=  $\frac{130,000 \times 287 \times 293}{60 \times 10^5}$ 

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:. Power = 
$$\frac{310 \times 1860}{60 \times 1000 \times 0.815} = 11.8 \text{ kW}$$
  
Actual power =  $\frac{\text{Power}}{\eta_{f}} = \frac{11.8}{0.815} = 14.48 \text{ kW}$ 

01(e)(i).

Sol:

- The availability of a non-flow system in a particular state is the maximum work that the system could conceivably deliver to something other than the surroundings as its state changes to the dead state, exchanging heat only with the environment. Here it should be noted that the maximum possible work would be obtained if the system proceeds reversible externally and internally from one state to another.
- The availability of a steady flow system is the maximum work that can be delivered by the system to something other than the surroundings as the state changes from that at entry to the control volume to the dead state at the exit boundary, mean while exchanging heat only with the surroundings.

#### 01(e)(ii).

**Sol:** Let at any instant pressure in the tank be P, volume of the tank be V and mass of air be "m"

PV = mRTVdp = dmRT .....(i)

[::V&Tare constants]

Now after time 'dt' let the mass in the tank become m-dm

Now, mass 'dm' having volume 'Qdt' comes out at pressure = P - dp and temperature = T, where Q = volume flow rate

$$(P-dp)Qdt = dmRT$$
  
 $pQdt = dmRT.....(ii)$   
From (i) & (ii)

From (i) & (ii)

$$-Vdp = PQdt$$

(-ve sign indicates that pressure is decreasing)

$$-\frac{dP}{P} = \frac{Qdt}{V}$$
$$-\frac{dP}{P} = 0.1 \times \frac{dt}{1}$$
$$-\int_{200}^{P} \frac{dP}{P} = 0.1\int_{0}^{10} dt$$

Solving, we get P = 73.58kPa



Sol:



- 1) Fresh air condition: 40°C DBT, 27°C WBT,  $h_1 = 85 \text{ kJ/kg}, \omega_1 = 17.5 \text{ g/kg d.a}$
- 2) Air condition after mixing chamber: On line, joining point (1) and (4), point (2) is located such that  $v_2 = 0.87 \text{ m}^3/\text{kg}$

: 
$$h_2 = 58 \text{ kJ}/\text{kg}, T_2 = 28.5^{\circ}\text{C}$$
 (DBT),

$$\omega_2=\!11.5\,g/kg\,d.a$$

$$\dot{m}_2 = \frac{\dot{V}_2}{v_2} = \frac{4500}{0.87} = 5172 \text{ kg} / \text{min}$$



$$\dot{m}_{1} = 0.2 \times 5172 = 1034 \text{ kg/min} (\text{freshair})$$

$$\dot{m}_{2} - \dot{m}_{1} = 4138 \text{ kg/min} (\text{recirculated air})$$
Room sensible heat gain =  $\dot{m}_{3}(h_{4'} - h_{3})$ 

$$= \frac{5172}{60}(49 - 37) = 1034.4 \text{ kW}$$
Room latent heat gain =  $\dot{m}_{3}(h_{4} - h_{4'})$ 

$$= \frac{5172}{60}(51 - 49)$$

$$= 172.4 \text{ kW}$$
Load on the cooling coil =  $\dot{m}_{2}(h_{2} - h_{3})$ 

$$= \frac{5172}{60} \times (58 - 37) = 1810.2 \text{ kW}$$
3). Air condition after cooling coil:  
13° C DBT, RH = 100%, h\_{3} = 37 \text{ kg/kJ}
4). Room air condition:  
25° C DBT, 50% RH, h\_{4} = 51 \text{ kJ/kg}
**02(b).**  
**Sol:** Absolute steam velocity = V<sub>1</sub> = 360 m/sec  
Nozzle angle =  $\alpha_{1} = 20^{\circ}$   
Diameter of rotor = D = 0.955 m  
Speed = N = 3000 rpm  
Axial thrust on blades = 0  
Blade speed  
 $U = \frac{\pi DN}{60} = \frac{\pi \times 0.955 \times 3000}{60} = 149.94 \text{ m/sec}$ 



 $\alpha_1 = 20^\circ$  $V_{w_1} = V_1 \cos \alpha_1 = 360 \cos 20 = 338.3 \,\text{m/sec}$  $V_{f_1} = V_1 \sin \alpha_1 = 360 \sin 20 = 123.13 \, \text{m/sec}$  $Bc = V_{w_1} - U = 338.3 - 149.94 = 188.36 \text{ m/sec}$  $\mathbf{V}_{r_1} = \sqrt{\left(\mathbf{BC}\right)^2 + \left(\mathbf{V}_{r_1}\right)^2}$  $=\sqrt{(188.36)^2+123.13^2}$  $=\sqrt{35479.49+15161}$ = 225.03 m/sec  $(KE)_{exit} = (KE)_{entrance} - (0.19)(KE)_{entrance}$  $(KE)_{entrance} = 0.81 (KE)_{entrance}$  $\frac{\mathrm{Vr}_2^2}{2} = \frac{\mathrm{KVr}_1^2}{2}$  $Vr_2 = \sqrt{K}V_m$  $= 0.9 \times 225.03 = 202.53$  m/sec As there is no axial thrust  $V_{f_1} = V_{f_2} = 123.13 \,\text{m/sec}$  $V_{f_2} = V_{r_2} \sin \beta_2$  $123.13 = 202.5 \sin \beta_2$  $\beta_2 = \sin^{-1}\left(\frac{123.13}{202.5}\right) = 37.45$  $V_{r_2} = \cos \beta_2 = 202.53 \cos 37.45$ = 160.79 m/sec $V_{r_2} \cos\beta_2 > U(V_{w_2} \text{is in opposite direction to } V_{w_1})$  $V_{w_2} = V_{r_2} \cos\beta_2 - U$ = 160.79 - 149.94 = 10.85 m/sec Mass flow rate of steam =  $\dot{m} = 1$  kg/sec power =  $\frac{\dot{m}.U}{1000} (V_{w_1} + V_{w_2})$  $=\frac{1\times149.94}{1000}(338.3+10.85)=52.35 \text{ kW}$ 



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#### **02(c).**

Sol: Given: Compression ratio, r = 12Cut-off ratio,  $\beta = 1.615$ Maximum pressure,  $p_1 = p_4 = 52.17$  bar  $p_1 = 1$  bar  $T_1 = 62 + 273 = 335$  K Indices of compression and expansion, n = 1.35

(a) Processes 1-2 and 4-5 are polytropic therefore, there will be best transfer during these processes as well

Heat transfer during the process 1-2

$$Q_{1-2} = \left(\frac{\gamma - n}{\gamma - 1}\right) \left(\frac{p_1 V_1 - p_2 V_2}{n - 1}\right)$$
$$= \left(\frac{\gamma - n}{\gamma - 1}\right) \left(mR\right) \left(\frac{T_1 - T_2}{n - 1}\right)$$

For unit mass,

$$Q_{1-2} = \left(\frac{1.4 - 135}{1.4 - 1}\right) \left(0.287 \left(\frac{335 - 799}{1.35 - 1}\right)\right)$$
$$= -47.56 \text{ kJ/kg}$$

Heat transfer during the process 2-3,

$$Q_{2-3} = c_p(T_3 - T_2) = 0.718(1456 - 799)$$
  
= 471.7 kJ/kg

Heat transfer during the process 3-4,

$$\begin{split} Q_{3\text{-}4} &= c_p(T_4 - T_3) = 1.005(2351 - 1456) \\ &= 899.5 \text{ kJ/kg} \end{split}$$

Heat transfer during the process 4-5,

$$Q_{4-5} = \left(\frac{\gamma - n}{\gamma - 1}\right) (K) \left(\frac{T_4 - T_5}{n - 1}\right)$$
$$= \left(\frac{1.4 - 1.35}{1.4 - 1}\right) (0.287) \left(\frac{2351 - 1165}{1.35 - 1}\right)$$
$$= 121.6 \text{ kJ/kg}$$

Heat transfer during the process 5-1,

$$Q_{5-1} = Q_{2-3} + Q_{3-4} + Q_{4-3}$$
  
= 471.7 + 899.5 + 121.6  
= 1492.8 kJ/kg  
and heat rejected Q<sub>2</sub> = Q<sub>1-2</sub> + Q<sub>5-1</sub>  
= 47.56 + 596  
= 643.56 kJ/kg  
Work done, W = Q<sub>1</sub> - Q<sub>2</sub>  
= 1492.8 - 643.56 = 849.2 kJ/kg  
Efficiency,  $\eta = \frac{W}{Q_1} = \frac{849.2}{1492.8} = 0.569 = 56.9\%$ 

(b) Swept volume,

$$V_1 = V_1 - V_2 = V_1 \left(1 - \frac{1}{r}\right) = \frac{mRT_1}{p_1} \left(\frac{r-1}{r}\right)$$

For unit mass,

$$V_{1} = \frac{RT_{1}}{p_{1}} \left(\frac{r-1}{r}\right) = \frac{287 \times 335}{1 \times 10^{5}} \times \frac{11}{12}$$
$$= 0.8813 \text{ m}^{3}/\text{kg}$$

Mean effective pressure,

$$p_{\rm m} = \frac{W}{V_1} = \frac{849.2 \times 10^3}{0.8813} = 9.635 \times 10^5 \,\text{N/m}^2$$

= 9.635 bar

**03(a).** 

Sol:

$$P = 0.2m^{3} = 0.4m^{3}$$

$$V_{2} = 0.2m^{3} = 0.4m^{3}$$

$$V_{3} = 0.4m^{3}$$

$$T = C$$
  
 $P_1 = 100 \text{ KPa}$   
 $V_1 = 0.4 \text{m}^3$   
 $T_1 = 273 + 27 = 300 \text{k}$ 



 $T_{1} = T_{2} = 300k$ Argon mono atomic gas  $\gamma = 5/3$   $m = \frac{P_{1}V_{1}}{RT_{1}} = \frac{100 \times 0.4}{0.20785 \times 300} = 0.6415 kg$   $R = \frac{\overline{R}}{Molecular Wt}$   $= \frac{8.314}{40} = 0.20785 kJ / kg.K$   $C_{v} = \frac{R}{\gamma - 1} = \frac{0.20785}{0.667} = 0.3116 \frac{kJ}{kgK}$   $P_{1}V_{1} = P_{2}V_{2};$   $P_{2} = P_{1} \times \frac{V_{1}}{V_{2}} = 100 \times \frac{0.4}{0.2} = 200 kPa$   ${}_{1}Q_{2} = {}_{1}W_{2} = P_{1}V, \ ln \frac{P_{1}}{P_{2}}$   $= P_{1}V_{1} ln \frac{V_{2}}{V_{1}} = 100 \times 0.4 ln \frac{0.2}{0.4}$ = -27.73 kJ

#### <u>2–3</u> p=c

 $\frac{V_2}{T_2} = \frac{V_3}{T_3}; T_3 = \frac{V_3}{V_2} \times T_2$ =  $\frac{0.6}{0.2} \times 300 = 900 \text{ K}$  ${}_2W_3 = P_2(V_3 - V_2)$ = 200(0.6-0.2) = 80 kJ ${}_2Q_{3-2}W_3 = {}_2U_3 = \text{m.C}_v(T_3-T_2)$  ${}_2Q_3-80 = 0.6415 \times 0.3116(900-300)$  ${}_2Q_3 = 199.94 \text{ kJ}$ Total heat transfer =  ${}_1Q_2 + {}_2Q_3$ = -27.73 + 199.94= 172.21 kJ **03(b).** 

**Sol:** We assume the inner surface of the pipe to be at 0°C at all times. The thermal resistances involved and the rate of heat transfer are

$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(1.2/1)}{2\pi(0.16)(0.5)}$$
$$= 0.3627^{\circ}C/W$$
$$R_{conv, o} = \frac{1}{h_o A} = \frac{1}{(40)[(\pi(0.024)(0.5))]}$$
$$= 0.6631^{\circ}C/W$$

$$R_{total} = R_{pipe} + R_{conv, o}$$

$$= 0.3627 + 0.6631 = 1.0258$$
°C/W

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{total}} = \frac{[0 - (-5)]^{\circ}C}{1.0258^{\circ}C/W} = 4.874 W$$

The total amount of heat lost by the water during a 14-h period that night is

$$Q = \dot{Q}\Delta t = (4.874)(14 \times 3600) = 245.7 \text{ kJ}$$

The amount of heat required to freeze the water in the pipe completely is

$$m=\rho V=\rho\pi r^2 L$$

$$= (1000)\pi(0.01)(0.5) = 0.157$$
 kg

 $Q = mh_{fg} = (0.157)(333.7) = 52.4 \text{ kJ}$ 

The water in the pipe will freeze completely that night since the amount heat loss is greater than the amount it takes to freeze the water completely (245.7 > 52.4)

**03(c).** 

1.

Sol:  $P_v = \phi P_{sat} = 0.5 \times 3.17 = 1.585 \text{ kPa}$  $0.622 P_v = 0.622 \times 1.585$ 

$$\omega = \frac{v}{P - P_v} = \frac{100 - 1.585}{100 - 1.585}$$
  
= 0.01 kg / kg of d.a



2. 
$$h = h_a + h_v = c_{pa}T + \omega(2500 + c_{pv}T)$$
  
 $= 1.005 \times 25 + 0.01 \times (2500 + 1.82 \times 25)$   
 $= 50.58 \text{ kJ/kg of d.a}$   
3.  $m_a = \frac{P_a V}{R_a T} = \frac{(100 - 1.585) \times (5^3)}{0.287 \times 298} = 143.8 \text{ kg}$   
4.  $m_v = \frac{P_v V}{R_v T} = \frac{1.585 \times 5^3}{\left(\frac{8.314}{18}\right) \times 298} \left[ \therefore R_v = \frac{R_u}{M} \right]$   
 $= 1.44 \text{ kg}$ 

**03(d).** 

**Sol:** Power input to the FD fan,  $P = \frac{V \times \Delta p}{\eta_{mech}}$ 

$$\dot{V} = \frac{P \times \eta_{mech}}{\rho_{w}gh} = \frac{5 \times 10^{3} \times 0.78}{1000 \times 9.81 \times 0.180} = 2.21 \text{m}^{3}/\text{s}$$

Density of air,

$$\rho_a = \frac{p}{RT} = \frac{1.02 \times 10^5}{287 \times (273 + 16)} = 1.23 \text{kg} / \text{m}^3$$

(i) Mass flow rate of air,

$$\dot{m}_a = \rho_a \times \dot{V} = 1.23 \times 2.21 = 2.72 \text{ kg/s}$$

Assuming complete combustion of fuel, the mass flow rate of the gases,

$$\dot{m}_{g} = \dot{m}_{a} + \dot{m}_{f} = 2.72 + \frac{500}{3600} = 2.855 \text{ kg/s}$$

(ii) Energy lost by the gas = energy gained by air in air pre heater

 $2.855 \times 1.01 \times \Delta T = 272 \times 1.005 \times 75$ 

- $\therefore \Delta T =$  temperature drop of the flue gases = 71°C
- $\therefore$  Temperature of the flue gases leaving the plant = 225-71 = 154°C
- (iii) Making an energy balance for the economiser

$$\dot{m}_{c}c_{w}(125-40) = \dot{m}_{c}c_{ps} \times (395-225)$$

$$\therefore m_{w} = \frac{2.855 \times 1.01 \times 170}{4.2 \times 85} = 1.375 \text{ kg/s}$$

**04(a).** 

Sol: <u>Piston cylinder boundary is taken to be our</u> <u>control volume boundary.</u>

From mass conservation,

$$\frac{\mathrm{d}\mathrm{m}_{\mathrm{i}}}{\mathrm{d}\mathrm{t}} - \frac{\mathrm{d}\mathrm{m}_{\mathrm{e}}}{\mathrm{d}\mathrm{t}} = \left(\frac{\mathrm{d}\mathrm{m}}{\mathrm{d}\mathrm{t}}\right)_{\mathrm{cv}} \Longrightarrow - \frac{\mathrm{d}\mathrm{m}_{\mathrm{e}}}{\mathrm{d}\mathrm{t}} = \left(\frac{\mathrm{d}\mathrm{m}}{\mathrm{d}\mathrm{t}}\right)_{\mathrm{cv}} \dots$$
......(i)

where,  $m_i = mass$  entering the control volume,  $m_e = mass$  leaving the control volume

From energy conservation,

$$\frac{d}{dt} [m_i h_i + Q] - \frac{d}{dt} [m_e h_e + W] = \left(\frac{dU}{dt}\right)_{cv}$$
$$\frac{dQ}{dt} - h_{avg} \frac{dm_e}{dt} - \frac{dW}{dt} = \left(\frac{dU}{dt}\right)_{cv}$$

[Although exit enthalpy is also a function of time (because temperature changes with time), we have taken exit enthalpy as average value (between temperature 80°C and 20°C i.e.,

$$=\frac{+ \text{ enthalpy of refrigerant at } 20^{\circ}\text{C}}{2}) \qquad \text{as}$$

given in the question.

Thus, 
$$\frac{d}{dt}(m_e h_e) = h_{avg} \frac{dm_e}{dt}$$
]

$$\frac{dQ}{dt} + h_{avg} \left(\frac{dm}{dt}\right)_{cv} - \left(\frac{dW}{dt}\right)_{cv} = \left(\frac{dU}{dt}\right)_{cv} using (i)$$

Integrating the above expression, we get  $Q + h_{avg} (m_2 - m_1) - W = m_2 u_2 - m_1 u_1 \quad [m_1$ = initial mass in c.v,  $m_2$  = find mass in c.v]

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 $h_{avg}$ 



Where, W = P(V<sub>f</sub> - V<sub>i</sub>)  
= 
$$500 \times (1 \times 0.042115 - 2 \times 0.032659)$$
  
=  $-11.6 \text{ kJ}$   
 $\Rightarrow Q + 290.23 \times (-1) + 11.6 = 1 \times 242.42$   
 $-2 \times 290.86$   
 $\Rightarrow Q = -60.67 \text{ kJ}$ 

#### **04(b).**

#### Sol:

(a) When the nozzle lip Z is neglected: Rate of mass flow of air,

$$\dot{m}_a = Cd_n A_2 \sqrt{2\rho_a \Delta p}$$

Rate of mass flow of fuel,  $\dot{m}_{f} = Cd_{f}A_{j}\sqrt{2\rho_{f}\Delta\rho}$ Airfuel ratio,  $\frac{\dot{m}_{a}}{\dot{m}_{f}} = \frac{Cd_{a}A_{2}}{Cd_{f}A_{1}}\sqrt{\frac{p_{a}}{p_{1}}}$   $= \frac{Cd_{a}d_{2}^{2}}{Cd_{f}d_{1}^{2}}\sqrt{\frac{p_{0}}{p_{1}}}$  $= \frac{0.82}{0.70} \times \left(\frac{22}{12}\right)^{2} \times \sqrt{\frac{12}{250}} = 15.75$ 

(b) When the nozzle lip z is considered:
 m
<sub>a</sub> will remain the same

but, 
$$\dot{m}_{f} = Cd_{f}A_{1}\sqrt{2p_{1}(\Delta p - xZp_{1})}$$
  
Now,  
 $gZp_{1} = 9.81 \times 0.004 \times 750 = 29.43 \text{ N/m}^{2}$ 

$$gzp_1 = 9.81 \times 0.004 \times 750 = 29.43$$
 N/III  
= 0.0002943 bar  
 $\Delta p - gZp_1 = 0.075 - 0.0002943 = 0.0747$  bar  
Air/fuel ratio,

$$\frac{\dot{m}_{a}}{m_{f}} = \frac{Cd_{a}}{Cd_{f}} \frac{A_{2}}{A_{1}} \sqrt{\frac{p_{a}\Delta p}{p_{f}(\Delta p - gZp_{1})}}$$
$$= \frac{0.82}{0.7} \left(\frac{22}{1.2}\right)^{2} \sqrt{\frac{1.2 \times 0.075}{750 \times 0.0747}} = 15.78$$

- (c) When the nozzle lip is provided the flow of fuel will start only when there is sufficient depression to overcome the nozzle lip effect. The depression is created by the critical velocity of air
- ... Minimum velocity of air or critical velocity at the throat

$$C_{1} = \sqrt{\frac{2gZp_{f}}{p_{a}}}$$
$$= \sqrt{\frac{2 \times 9.81 \times 0.04 \times 750}{1.2}} = 7 \text{ m/s}$$

**04(c).** 

Sol:

:10:



Work done in compressor =  $\eta_m \times$  work done in turbine

$$c_p (T_2 - T_1) = 0.92 \times c_p (T_3 - T_4)$$
  
 $450.94 - 288 = 0.92 \times (933 - T_4)$   
 $\Rightarrow T_4 = 755.9 \text{ K}$   
In HP turbine,

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$$\eta_{\rm T} = \frac{T_3 - T_4}{T_3 - T_{4'}}$$

$$0.84 = \frac{933 - 755.9}{933 - T_{4'}}$$

$$T_{4'} = 722.16 \,\text{K}$$

$$\frac{T_3}{T_{4'}} = \left(\frac{p_3}{p_{4'}}\right)^{\frac{\gamma - 1}{\gamma}}$$

$$p_{4'} = \frac{4}{\left(\frac{933}{722.16}\right)^{3.5}} = 1.63 \,\text{bar}$$
In LP turbine,

 $\frac{T_{5}}{T_{6'}} = \left(\frac{p_{5}}{p_{6'}}\right)^{\frac{\gamma-1}{\gamma}}$  $\frac{898}{\mathrm{T}_{6'}} = \left(\frac{1.63}{1}\right)^{0.286}$  $T_{6'} = 780.90 \text{ K}$  $\eta_{\rm T} = \frac{T_5 - T_6}{T_5 - T_{6'}}$  $0.8 = \frac{898 - T_6}{898 - 780.9}$  $T_6 = 804.3 \text{ K}$  $\in = \frac{\mathbf{T}_7 - \mathbf{T}_2}{\mathbf{T}_6 - \mathbf{T}_2}$ 

$$\begin{split} 0.75 &= \frac{T_7 - 450.94}{804.3 - 450.94} \\ T_7 &= 716 \text{ K} \\ Q &= c_p \left(T_3 - T_7 + T_5 - T_4\right) \\ &= 1.005 \times (933 - 716 + 898 - 755.9) \\ &= 360.9 \text{ kJ/kg} \\ W_C &= c_p \left(T_2 - T_1\right) \\ &= 1.005 \times (450.94 - 288) \\ &= 163.75 \text{ kJ/kg} \\ W_T &= c_p \left(T_3 - T_4\right) + c_p \left(T_5 - T_6\right) \\ &= 1.005 \times [(933 - 755.9) + (898 - 804.3)] \\ &= 272.15 \text{ kJ/kg} \\ \eta_{th} &= \frac{W_T - W_C}{Q} = \frac{272.15 - 163.75}{360.89} = 30\% \end{split}$$



#### **05(a).**

#### Sol: The boiling curve:

Interface Film boiling Nucleate boiling evaporation or free convection Ш IV VI Π V T Bubbles Bubbles rise Radiation Unstable condense in Stable Coming into to surface film superheated film Play liquid A R Qs (W/m<sup>2</sup>) Heat transferred by Superheated liquid rising to the liquid-State beginning Spheroidal vapour interface where evaporation takes place Boiling curve Excess temperature  $\Delta t_{ex} = t_P - t_{sat}$ 

The boiling curve reveals range of conditions associated with saturated pool boiling on a q verses  $\Delta T_{ex}$ 

#### **Pool boiling regimes:**

#### (i) Pure convection with liquid rising to surface for evaporation (I):

In this region free–convection currents are responsible for motion of the fluid near the surface. In this region the liquid near the heated surface is superheated slightly, and it subsequently evaporates when it rises to the surface.

#### (ii) Nucleate boiling with bubbles condensing in liquid (II):

In region II bubbles begin to form on the surface of the wire and are dissipated in the liquid after breaking away from the surface. This region indicates the beginning of nucleate boiling.

#### (iii)Nucleate boiling with bubbles rising to surface (Peak temperature)(III):

As the temperature excess is increased further, bubbles form more rapidly and rise to the surface of the liquid, where they are dissipated. Eventually, bubbles are formed so rapidly that they blanket the heating surface and prevent the inflow of fresh liquid from taking their place. At this point the bubbles coalesce and form a vapor film which covers the surface. The heat must be conducted through this film before it can reach the liquid and effect the boiling process.

The thermal resistance of this film causes a reduction in heat flux.

#### (iv) Partial nucleate boiling and unstable film boiling (IV):

This region represents a transition from nucleate boiling to film boiling and is unstable.



#### **(v)** Film boiling is stabilized (V):

The surface temperatures required to maintain stable film boiling are high.

#### (vi)Radiation becomes a dominant mechanism for heat transfer (VI):

In this region a significant portion of the heat lost by the surface may be the result of thermal radiation

05(b).

**Sol:** Speed = N = 2000 rpm

Diametric of bore = d = 9 cms Spark plug offset = e = 8 mm

= 0.8 cms



Crank angle through which flame travels = 20 + 15.85

= 26.5°

Time of liquid =  $t = \frac{\theta}{360} \times \frac{60}{N}$  $=\frac{26.5}{360}\times\frac{60}{2000}=2.208\times10^{-3}$ Flame travel distance =  $\frac{d}{2} + e = \frac{9}{2} + 0.8$ = 5.3 cm $\frac{\text{flame travel dis tance}}{\text{time}} = \frac{5.3 \times 10^{-2}}{2.208 \times 10^{-3}} = 24 \text{ m/s}$ Flame speed =



#### **05(c).**

#### Sol: Domestic Electrolux (Three-Fluid) System



 $P_e = Conderser \ pressure = NH_3 \ pressure$ 

 $P_e$  = Evaporator pressure = Partial pressure of H<sub>2</sub>gas+Partial pressure of NH<sub>3</sub>

Strong solution  $\rightarrow$  contains as much NH<sub>3</sub> as possible

Weak solution  $\rightarrow$  contains considerably less NH<sub>3</sub>

#### **Description:**

This system is also known as three fluid system ( $NH_3$ , water and  $H_2$ ).  $NH_3$  is Refrigerant and water is absorbent. A light gas ( $H_2$ ) compared to the refrigerant vapour is introduced into the system. This gas acts as a carrier fluid. The following step by step sequence of operation is applicable to this system:

- 1. As shown in the figure, the system consists of a generator. In this generator, heat transfer  $Q_G$  causes production of NH<sub>3</sub> vapour.
- 2. NH<sub>3</sub> vapour together with water vapour moves upto the rectifier. This is also known as separator.
- 3. The water vapour returns to the generator and then to the absorber. If this water vapour is not removed will enter the evaporator as water and the water will be freezed while passing through the expansion valve.
- 4. NH<sub>3</sub> vapour enters the condenser. This vapour is converted into a liquid form and this liquid NH<sub>3</sub> then enters to the evaporator.
- 5. In the evaporator a light gas (H<sub>2</sub>) is charged. The pressure of H<sub>2</sub> is adjusted in such a way that the partial pressure of NH<sub>3</sub> in the evaporator (as per Dalton's Law) be equal to the desired evaporator temperature. For example in order to have evaporator temperature of  $-18^{\circ}$ C, the



saturation pressure is 2 bar. For a system of having a total pressure 18 bar, the hydrogen in the evaporator must be charged with a pressure of 16 bar.

- 6. Liquid  $NH_4$ , evaporates in the evaporator absorbing heat  $Q_E$  from the surroundings.
- 7. Liquid  $NH_3$  with  $H_2$  gas enters the absorber in which cooled, weak solution enters from the top.
- 8. Water will absorb NH<sub>3</sub> liquid. This will generate heat. The absorber be cooled by circulating cooling water around the absorber. The absorption power of the absorber will be better at lower temperature.
- 9. The warm light H<sub>2</sub> gas has a tendency to move up and to enter the evaporator. In order to prevent H<sub>2</sub> gas entering the condenser a u-bend is provided at the exit of the condenser.
- 10. The strong solution passes through the heat exchanger. The strong solution is heated up by hot weak solution returning from the generator to the absorber.
- 11. This heating of the strong solution will reduce the heat requirement in the generator. This will also reduce the temperature of the weak solution. This reduction in temperature will help the absorption process.
- 12. The expansion value is eliminated from this system (see figure). This is because of the use of  $H_2$  as the third fluid. Since the sum of the partial pressure of  $H_2$  and  $NH_3$  in the evaporator is the same as that of  $NH_3$  vapour present in the condenser, by adjusting the pressure of  $H_2$ , in evaporator, the pressure of  $NH_3$  is reduced and based on the application the partial pressure of  $H_2$  can be adjusted and that will fix the partial pressure of  $NH_3$  in the evaporator. The circulation is effected by gravity and thus, there is no moving part in the system.

#### **05(d).**

**Sol:**  $T_1 = 22^{\circ}C$ ,  $T_2 = 83^{\circ}C$ 

$$P_1 = 100 \text{ kPa}, P_2 = 100 \text{ kPa}$$
  
 $V_1 = 3.7 \text{ m/s}, V_2 = 9.1 \text{ m/s}$ 

By applying S.F.E.E between entry and exit of hair dryer,

$$\dot{m}\left(h_{1}+\frac{V_{1}^{2}}{2000}\right)+\frac{dQ}{dt}=\dot{m}\left(h_{2}+\frac{V_{2}^{2}}{2000}\right)+\frac{dw}{dt}$$

where,

 $h_1 = 1.005 \times 295 =$  $h_2 = 1.005 \times 356 =$  $\frac{dQ}{dt} = 0$ 

$$\dot{m} = \rho_2 A_2 V_2 = \frac{P_2}{RT_2} A_2 \times V_2$$
$$= \frac{100 \times 18.7 \times 10^{-4} \times 9.1}{0.287 \times 356} = 0.017 \text{ kg/s}$$

(a) Thus we get  $\frac{dw}{dt} = 1.04 \text{ kW}$ 

(b) 
$$s_2 - s_1 = c_p \, \ln \left( \frac{T_2}{T_1} \right) - R \, \ln \left( \frac{P_2}{P_1} \right) = 1.005 \, \ln \left( \frac{356}{295} \right) = 0.189$$

Increase in available energy

$$(h_2 - h_1) + \left(\frac{V_2^2 - V_1^2}{2000}\right) - T_0(s_2 - s_1) = 1.005 \times (83 - 22) + \frac{9.1^2 - 3.7^2}{2000} - 295 \times 0.189$$
  
= 61.3 + 0.0345 - 55.76 = 5.574 kJ/kg

Minimum work required =  $0.017 \times 5.574 = 0.094$  kW

Second law efficiency =  $\frac{0.094}{1.04} = 0.0911 = 9.1 \%$ 

Sol:



Because  $T_1 = T_3$  and  $T_2 = T_4$  &  $r_p = r_{p1} \times r_{p2}$ 

$$W_{T} = C_{p} [T_{5} - T_{6}]$$
$$W_{net} = C_{p} T_{5} \left[ 1 - \frac{1}{(r_{p})^{\frac{\gamma-1}{\gamma}}} \right] - 2C_{p} T_{l} \left[ (r_{p})^{\frac{\gamma-1}{2\gamma}} - 1 \right]$$

Taking  $\frac{\gamma - 1}{\gamma} = x$ ;  $T_5 = T_{max}$ ,  $T_1 = T_{min}$ 



$$\begin{split} W_{net} &= C_{p} \Bigg[ T_{max} \Bigg( 1 - \Bigg( \frac{1}{r_{p}} \Bigg)^{x} \Bigg) \Bigg] - 2 C_{p} \times T_{min} \Bigg[ (r_{p})^{\frac{x}{2}} - 1 \Bigg] \\ &\therefore \ \frac{dW}{d(r_{p})} = C_{p} \Bigg[ T_{max} \Big( x(r_{p})^{-(x+1)} \Big) - 2 T_{min} \frac{x}{2} (r_{p})^{\frac{x}{2}-1} - 0 \Bigg] \\ T_{max} \Big( r_{p} \Big)^{-(x+1)} &= T_{min} \Big( r_{p} \Big)^{\frac{x}{2}-1} \Big) \\ &\left( \frac{T_{max}}{T_{min}} \Bigg) = (r_{p})^{\frac{x}{2}+x} = (r_{p})^{\frac{3x}{2}} \\ r_{p} &= \Bigg( \frac{T_{max}}{T_{min}} \Big)^{\frac{2}{3} \left( \frac{\gamma}{\gamma - 1} \right)} \\ &\therefore \ \left( r_{p} \right) = \Bigg( \frac{1200}{300} \Big)^{\frac{2}{3} \left( \frac{\gamma}{\gamma - 1} \right)} = 25.4 \end{split}$$

Sol:



:17:

The system may be modelled as a three-surface enclosure with one surface reradiating. The rate at which energy must be supplied to the heated surface may then be obtained from eq.

$$q_{1} = \frac{E_{b1} - E_{b2}}{\frac{1 - \varepsilon_{1}}{\varepsilon_{1}A_{1}} + \frac{1}{A_{1}F_{12} + \left[\left(\frac{1}{A_{1}F_{1R}}\right) + \left(\frac{1}{A_{2}F_{2R}}\right)\right]^{-1}} + \frac{1 - \varepsilon_{2}}{\varepsilon_{2}A_{2}}}$$

From symmetry,

$$F_{12} = F_{1R} = F_{2R} = 0.5$$

Also, 
$$A_1 = A_2 = W \times L$$

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Where L is the duct length.

Hence,

$$q_{1}' = \frac{q_{1}}{L} = \frac{5.67 \times 10^{-8} \text{ W/m}^{2} \text{ K}^{4} (1200^{4} - 500^{4}) \text{K}^{4}}{\frac{1 - 0.8}{0.8 \times 1 \text{ m}} + \frac{1}{1 \text{ m} \times 0.5 + (2 + 2)^{-1} \text{ m}} + \frac{1 - 0.4}{0.4 \times 1 \text{ m}}}$$
$$q_{1}' = 37 \text{ kW/m} = -q_{2}'$$

The temperature of the insulated surface may be obtained from the requirement that  $J_R = E_{bR}$ , where  $J_R$  may be obtained from equation. However, to use this expression  $J_1$  and  $J_2$  must be known. Applying the surface energy balance, to surfaces 1 and 2, if follows that

$$J_{1} = E_{b1} - \frac{1 - \varepsilon_{1}}{\varepsilon_{1}W} q'_{1} = 5.67 \times 10^{-8} \text{ W/m}^{2} \text{ K}^{4} (1200 \text{ K})^{4} - \frac{1 - 0.8}{0.8 \times 1 \text{ m}} \times 37000 \text{ W/m} = 108323 \text{ W/m}^{2}$$
$$J_{2} = E_{b2} - \frac{1 - \varepsilon_{2}}{\varepsilon_{2}W} q'_{2} = 5.67 \times 10^{-8} \text{ W/m}^{2} \text{ K}^{4} (500 \text{ K})^{4} - \frac{1 - 0.4}{0.4 \times 1 \text{ m}} \times (-37000 \text{ W/m}) = 59043 \text{ W/m}^{2}$$

From the energy balance for the reradiating surface,

$$\frac{108323 - J_{R}}{\frac{1}{W \times L \times 0.5}} - \frac{J_{R} - 59043}{\frac{1}{W \times L \times 0.5}} = 0$$

Hence,

$$J_{R} = 83683 \text{ W/m}^{2} = E_{bR} = \sigma T_{R}^{4}$$
$$T_{R} = \left(\frac{83683 \text{ W/m}^{2}}{5.67 \times 10^{-8} \text{ W/m}^{2} \text{ K}^{4}}\right)^{1/4} = 1102 \text{ K}$$

06(b).

Sol:



Single acting

Reciprocating compressor

Diameter of cylinder = d = 0.1 m



Stroke length = l = 0.12 m speed = N = 2800 Clearance ratio = C = 0.04 Temperature after sub cooling = 35°C h<sub>1</sub> = (hg)<sub>-20°C</sub> = 397.53 kJ/kg h<sub>3</sub> = (hf)<sub>44°C</sub> = 254.30 kJ/kg S<sub>1</sub> = (Sg)<sub>-20°C</sub> = 1.7841 kJ/kg.K S<sub>2</sub><sup>1</sup> = (Sg)<sub>44°C</sub> = 1.6943 kJ/kg.K h<sub>2</sub><sup>1</sup> = (hg)<sub>44°C</sub> = 416.43 kJ/kg v<sub>2</sub><sup>1</sup> = (vg)<sub>44°C</sub> = 0.0136 m<sup>3</sup>/kg h<sub>4</sub> = h<sub>3</sub> - C<sub>pl</sub> (T<sub>3</sub>-T<sub>4</sub>) = 254.3 - 1.3 (44-35) = 242.6 kJ/kg = 45

Specific heat of superheated vapour at 16.885 bar

$$\begin{split} \text{pressure} &= 0.95 \text{ kJ/kg.K} \\ \nu_1 &= (\nu_g)_{-20^\circ\text{C}} = 0.0925 \text{ m}^3/\text{kg} \\ \text{super heated vapour is ideal gas} \\ 1\text{-}2 \text{ S} - \text{C} \\ S_1 &= S_2 = 1.7841 \text{ kJ/kg.K} > 1.6943 \text{ kJ/kg.K} \\ \text{At condenser pressure} \\ \text{Hence in super heated state} \\ S_1 &= S_2^1 + c_{p_v} \ell n \frac{T_2}{T_2^1} \\ 1.7841 &= 1.6943 + 0.95 \ \ell n \frac{T_2}{273 + 44} \\ \ell n \frac{T_2}{317} &= \frac{1.7841 - 1.6943}{0.95} = 0.0945 \\ T_2 &= 317e^{0.0945} = 348.42 \text{ K} \end{split}$$

$$h_{2} = h_{2}^{1} + c_{p_{v}} (T_{2} - T_{2}^{1})$$
  
= 416.43 + 0.95 (348.42–317)  
= 446.28 kJ/kg



$$\frac{\mathbf{v}_2}{\mathbf{T}_2} = \frac{\mathbf{v}_2^1}{\mathbf{T}_2^1}$$
$$\mathbf{v}_2 = \mathbf{v}_2^1 \times \frac{\mathbf{T}_2}{\mathbf{T}_2^1}$$
$$= 0.0136 \times \frac{348.42}{317} = 0.01495 \text{ m}^3/\text{kg}$$

Volumetric efficiency of compressor

$$\eta_{\text{vol}} = 1 + c - c \left(\frac{P_{\text{com}}}{P_{\text{evap}}}\right)^{\frac{1}{4}}$$
$$= 1 + c - c \left(\frac{\nu_1}{\nu_2}\right)$$
$$= 1 + 0.04 - 0.04 \left(\frac{0.0928}{0.01495}\right) = 0.7917 \text{ or } 79.17 \%$$

Volume flow rate (m<sup>3</sup>/sec)

$$\dot{m}_{r} \times 0.0928 = \frac{\pi}{4} (0.1)^{2} (0.12) \times 1 \times \frac{2800}{60} \times 0.7917$$

$$\dot{m}_r = 0.375 \text{ kg/sec}$$

Mass flow rate of refrigerant

$$\dot{m}_r = 0.375 \, \text{kg/sec}$$

Cooling load (kW) =  $\dot{m}_r (kg/sec)(h_1 - h_4)kJ/kg = 0.375 (397.53 - 242.6) = 58.09 kW$ 

 $=\frac{58.09}{3.517}=16.52(T.R)$ 

Power needed to drive the refrigerator =  $\dot{m}_r (kg/sec)(h_2 - h_1)(kJ/kg)$ 

$$= 0.375 (446.28 - 397.53) = 18.28 \text{ kW}$$





#### 06(c)(i).

Sol: The sources of air in the condenser are due to the following :

- Leakage through packing glands and joints.
- Leakage through condenser accessories, such as, atmospheric relief valve, etc.,
- Air associated with exhaust steam may also liberate at low pressure.
- In the jet condenser, the dissolved air in the cooling water liberates at low pressures.

The effects of presence of air in a condenser are :

- The pressure in the condenser is increased, this reduces the work done by the engine or turbine.
- Partial pressure of steam and temperature are reduced. The steam tables tell us that at lower pressure, the latent heat of steam is more. In order to remove this greater quantity of heat, more cooling water has to be supplied and, thus under cooling of the condensate is likely to be more severe resulting in lower overall efficiency.
- The presence of air reduces the rate of condensation of steam since the abstraction of heat by the circulating cooling water is partly from the steam and partly from the air.
- The rate of heat transfer from the vapor is reduced due to poor thermal conductivity of air. Thus, the surface are of the tubes has to be increased for a given condenser duty.
- An air extraction pump is needed to remove air still some quantity of steam escapes with the air even after shielding to the air extraction section. This reduces the amount of condensate. Moreover, the condensate is undercooled, with the result that more heat has to be supplied to the feed water in the boiler.



#### 06(c)(ii).

#### Sol: Burning Time Losses:

The crankshaft normally rotates through 40° or more between the time the spark is produced and the time the charge is completely burned. The time in degree crank angle (°CA) depends upon the flame speed and the distance between the position of spark plug and the farthest side of combustion chamber. The flame travel distance can be reduced by locating the spark plug at the centre of the cylinder head or by using more than one spark plug. A hemispherical combustion chamber often uses two spark plugs mounted on the cylinder head on opposite sides to reduce the flame travel distance. The motion of the flame front depends upon how fast the heat is transferred from the flame front to the unburned mixture just ahead of the flame front. Heat is generated by the chemical reaction at the flame front.

As the crankshaft rotates, the piston moves and if the piston motion during combustion is taken into account the burning time losses are determined, which results in loss of work and efficiency.

However, the burning time lossses are quite large, if:

(a) The fuel/air ratio is made too lean, or too rich.

(b) The throttle is partially closed, reducing the suction pressure.

(c) The point of ignition is not property set.



Fig: Actual P-V diagram



#### **07(a).**

**Sol:** With reference to figure.

$$\therefore \quad h_2 = h_g = 2706.7 \text{ kJ/kg},$$

 $h_2 - h_6 = h_{fg} = 2201.9 \text{ kJ/kg}$ 

If m is the rate of stream extraction for process heating

$$\begin{split} m(h_2 - h_6) &= 1.163 \times 10^3 \\ m &= \frac{1.163 \times 10^3}{2201.9} = 0.528 \text{ kg/s} = 1901.4 \text{ kg/h} \\ s_1 &= 7.0901 = s_f + x_3 s_{fg} = 0.52 + x_3 \times 7.815 \\ x_3 &= 0.84 \\ h_3 &= 149.79 + 0.84 \times 2416 = 2180.59 \text{ kJ/kg} \\ (i) \quad \text{Total work output,} \\ W_T &= m_s(h_1 - h_2) + (m_s - m)(h_2 - h_3) \\ 5.6 \times 10^3 &= m_s \times 738.6 + m_s \times 526.11 - 277.8 \\ m_s &= \frac{5877.8}{1264.7} = 4.648 \text{ kg/s} = 16731 \text{ kg/h} = 16.73 \text{ t/h} \\ h_7 &= 504.7 + 1 \times 10^{-3}(10 - 2) \times 100 = 508.5 \text{ kJ/kg} \\ h_5 &= 149.79 + 1 \times 10^{-3} \times (40 - 0.06) \times 100 \\ &= 153.0 \text{ kJ/kg} \end{split}$$
(ii) 
$$Q_1 &= (m_s - m)(h_1 - h_5) + m(h_1 - h_7) = (4.648 - 0.528)(3445 - 153.8) + 0.528(3445.3 - 508.73) \\ &= 4.120 \times 3291.5 + 0.528 \times 2936.357 \\ &= 15113.25 \text{ kJ/s} = 15.113 \text{ MW} \end{split}$$

:23:



(iii) 
$$\eta_{\text{boiler}} = \frac{Q_1}{m_f \times C.V.} = \frac{15.111}{w_f \times 25} = 0.88$$
  
 $m_f = 0.687 \text{ kg/s} = 2473.2 \text{ kg/h} = 2.473 \text{ t/h}$   
(iv)  $Q_2 = (w_s - w)(h_3 - h_4)$   
 $= 4.12 \times 2030.8 = 8367 \text{ kW}$   
 $= 8.367 \text{ MW}$ 

(v) If 
$$m_c$$
 = water flow rate in the condenser,

Q<sub>2</sub> = w<sub>c</sub>c<sub>p</sub>(t<sub>2</sub> −t<sub>1</sub>)  
∴ w<sub>c</sub> = 
$$\frac{8367}{4.187 \times 6}$$
 = 333.05 kg/s = 0.333 m<sup>3</sup>/s



Sol:



:24:

Area of piston = A

Extra volume = 
$$\eta V_1 - V_0 = \frac{\eta 2 V_0}{\eta + 1} - V_0 = \frac{\eta V_0 - V_0}{\eta + 1} = \frac{V_0(\eta - 1)}{(\eta + 1)}$$

Extra length, 
$$x = \left(\frac{\eta - 1}{\eta + 1}\right) \frac{V_0}{A}$$



 $P_0V_0 = P_1(V_0 + Ax)$  (: Isothermal process)



 $\mathbf{P}_1 = \frac{\mathbf{P}_0 \mathbf{V}_0}{\mathbf{V}_0 + \mathbf{A}\mathbf{x}}$ Similarly,  $P_2 = \frac{P_0 V_0}{V_0 - Ax}$ Now,  $P_1A + F = P_2A \Longrightarrow F = (P_2 - P_1)A$  $F = \frac{2P_0V_0A^2x}{V_0^2 - A^2x^2}$  $dW = \int F dx$  $W = P_0 V_0 A^2 \int_{0}^{\left(\frac{\eta - 1}{\eta + 1}\right) \frac{V_0}{A}} \frac{2x}{V_0^2 - A^2 x^2} dx$  $V_0^2 - A^2 x^2 = z$  $-2xA^{2}dx = dz$  $2xdx = \frac{-dz}{\Delta^2}$  $W = P_0 V_0 A^2 \int \frac{-dz}{Z A^2} = -P_0 V_0 [\ell n z]$  $= -P_0 V_0 \ell n (V_0^2 - A^2 x^2)$  $= -P_0 V_0 \Big[ \ell n \Big( V_0^2 - A^2 x^2 \Big)_0^{\left(\frac{\eta - 1}{\eta + 1}\right) \frac{v_0}{A}} \Big]$  $= P_0 V_0 \ell n \left[ \frac{(\eta + 1)^2}{4\eta} \right]$ 

07(c).

Sol:



From the convection rate equation,

 $T_s = T_{\infty} + q / \overline{h}A$ 



where  $q = Nq_c = 2500$  W and  $A = L^2 = 0.04m^2$ . The convection coefficient is given by the turbulent flow correlation

 $\overline{h} = \overline{Nu}_{L} = 0.037 r E_{L}^{4/5} P r^{1/3}$ 

where

$$\operatorname{Re}_{L} = (u_{\infty}L/v) = \frac{(2 \times 0.2)}{0.96 \times 10^{-6}} = 4.17 \times 10^{5}$$

and hence  $\overline{h} = 0.037 (4.17 \times 10^5)^{4/5} (5.2)^{1/3} (0.62 / 0.2) = 6228 \text{ W/m}^2 \text{.K}$ 

The plate temperature is then

 $T_s = 17^{\circ}C + 2500/(6228)(0.20)^2 = 27^{\circ}C$ 

For an individual component, a rate equation involving the component's contact resistance can (b) be used to find its temperature,

$$q_{c} = (T_{c} - T_{s}) / R_{t,c} = (T_{c} - T_{s}) / (R_{t,c}'' / A_{c})$$
$$T_{c} = T_{s} + q_{c} R_{t,c}'' / A_{c} = 27^{\circ} C + 25 (2 \times 10^{-4}) / 10^{-4}$$
$$T_{c} = 77^{\circ} C$$

**07(d).** 

#### Sol: Advantages:

Cooling Liquid	Air Cooling
Compact design of engines with small frontal area.	Design is simpler as no water jackets are required.
Fuel consumption of high compression liquid	Fuel consumption is high.
cooled engine is low.	
Due to even cooling of cylinder and head due to	Less maintenance due to absence of cooling pipes
jacketing reducers cylinder head and valve heat	and radiator.
temperatures.	
Frontal location is not required as cooling system	No danger of coolant leakage
can be located any where	
The size of engine does not give any problems as	Engine is not subject to freezing problems.
design of cooling system is concerned	
	Less weight of cooling system improves power to
	weight ratio
	Engine is self contained and requires no
	components like radiator, header pump etc.
	Easy installation and cheaper to renew



#### **Disadvantages :**

Liquid cooled	Air cooled
Water circulation is dependent on additional	Used only for small and medium engines.
system like pump	
Power of pump is high effects power output of	Cooling is not uniform
engine	
Cooling system failure causes server damage	Atmospherics temperatures must be low for
to engine	better heat removal.
Cost in high	High working temperatures due to air cooling
Considerable amount is spent on maintenance	Produces aero-dynamic noise.
of cooling system.	
	Fuel consumption is higher.
	Lower compression ratios for engine
	If fan is used it absorbs 5% of power.

#### **08(a).**

Sol:



The cooling water outlet temperature may be obtained from the overall energy balance,

$$T_{c,o} = T_{c,i} + \frac{q}{m_c c_{p,c}} = 20^{\circ}C + \frac{2 \times 10^9 W}{3 \times 10^4 \text{ kg/s} \times 4179 \text{ J/kgK}} = 36.0^{\circ}C$$

The problem may be classified as one requiring a heat exchanger design calculation.

First, we determine the overall heat transfer for use in the NTU method.

$$\mathbf{U} = \frac{1}{\left(1/h_{i}\right) + \left(1/h_{o}\right)}$$

Where  $h_i$  may be estimated from an internal flow correlation. With

$$Re_{D} = \frac{4m}{\pi D\mu} = \frac{4 \times 1 \text{kg/s}}{\pi (0.025 \text{ m}) \times 855 \times 10^{-6} \text{ N.s/m}^{2}} = 59567$$

The flow is turbulent,

$$Nu_D = 0.023 \ Re_D^{4/5} \ Pr^{0.4} = 0.023 \ (59567)^{0.8} \ (5.83)^{0.4} = 308$$

Hence,

$$h_i = Nu_D \frac{k}{D} = 308 \times \frac{0.613 \text{ W}/\text{mK}}{0.025 \text{ m}} = 7543 \text{ W}/\text{m}^2\text{K}$$

$$U = \frac{1}{\left[\frac{1}{7543} + \frac{1}{11000}\right]m^2 K / W} = 4474 W/m^2 K$$

Using the design calculation methodology, we note that

 $C_h = C_{max} = \infty$ 

$$C_{max} = m_c c_{p,c} = 3 \times 10^4 \text{ kg/s} \times 4179 \text{ J/kgK} = 1.25 \times 10^8 \text{ W/K}$$

From which

$$\frac{C_{min}}{C_{max}} = C_r = 0$$

The maximum possible heat transfer rate is

$$q_{\text{max}} = C_{\text{in}} \left( T_{\text{h,i}} - T_{\text{c,i}} \right) = 1.25 \times 10^8 \text{ W/K} \times (50 - 20) \text{ K} = 3.76 \times 10^9 \text{ W}$$
$$\varepsilon = \frac{q}{q_{\text{max}}} = \frac{2 \times 10^9 \text{ W}}{3.76 \times 10^9 \text{ W}} = 0.532$$

We find NTU = 0.759

$$L = \frac{NTU \times C_{min}}{U(N2\pi D)} = \frac{0.759 \times 1.25 \times 10^8 \text{ W/K}}{4474 \text{ W/m}^2 \text{K}(30000 \times 2 \times \pi \times 0.025 \text{ m})} = 4.51 \text{ m}$$

**08(b)(i).** 

Sol: The arrangement of the system is shown in figure.



**ESE-2019 Mains Test Series** 

The efficiency of the engine is given by

$$\eta = \frac{T_1 - T_2}{T_1} = \frac{W}{Q_1} = \frac{W}{W + Q_2}$$
  
T<sub>1</sub> = ?,  
T<sub>2</sub> = 27 + 273 = 300 K;  
Q<sub>2</sub> = 1520  
Substituting the values in the c

Substituting the values in the above equation

$$0.4 = \frac{T_1 - 300}{T_1} = \frac{W}{W + 1520}$$
  

$$\therefore \quad 0.4 T_1 = T_1 - 300$$
  

$$0.6 T_1 = 300$$
  

$$\therefore \quad T_1 = 500 \text{ K.}$$
  

$$W = 0.4 W + 608$$
  

$$0.6 W = 608$$
  

$$W = 1013.3 \text{ kJ/min}$$
  

$$\therefore \text{ Power} = \frac{1013.3}{60} \text{ kW} = 16.9 \text{ kW}$$

#### **08(b)(ii).**

**Sol:** From property relation

Tds = dh - vdp

$$ds = \frac{dh}{T} - v\frac{dp}{T}$$

For two states at A and B the entropy change of the system.

$$\int_{S_{A}}^{S_{B}} ds = \int_{T_{A}}^{T_{B}} \frac{c_{p} dT}{T} - \int_{p_{A}}^{p_{B}} 0.287 \frac{dp}{p}$$

$$s_{B} - s_{A} = 1.005 \, \ell n \left(\frac{T_{B}}{T_{A}}\right) - 0.287 \, \ell n \left(\frac{p_{B}}{p_{A}}\right) = 1.005 \, \ell n \left(\frac{273 + 13}{273 + 50}\right) - 0.287 \, \ell n \left(\frac{100}{130}\right)$$

$$= -0.1223 + 0.0753 = -0.047 \, \text{kJ/kg.K}$$

 $(\Delta s)_{system} = -0.047 \text{ kJ/kg.K}$ 

Since the duct is insulated  $(\Delta s)_{surr} = 0$ 

$$\therefore (\Delta s)_{univ} = -0.047 \text{ kJ/kg.K}$$

This is impossible. So the flow must be from B to A



#### **08(c).**

**Sol:** The cycle is shown on the T-s & P–v diagram



- (i) Temperature after high pressure heat exchanger =  $T_3$ Temperature after low pressure heat exchanger =  $T_1$ 
  - $\therefore T_1 = 5^{\circ} C = 278 K$

$$T_3 = 30^{\circ} C = 303 K$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \rightarrow (\text{For isetropic compression})$$
  
$$\therefore T_2 = 278 \times 4^{0.286} = 413.3 \text{ K} = 140.3^{\circ} \text{ C}$$

Similarly for isentropic expansion process :

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$
  
$$\therefore T_4 = \frac{303}{4^{0.286}} = 203.8 \text{ K} = -69.2^{\circ} \text{ C}$$

Refrigeration effect =  $\dot{m}_a c_{pa} (T_1 - T_4)$ 

 $6 \times 3.517 = \dot{m}_a \times 1.005(278 - 203.8)$ 

$$\Rightarrow \dot{m}_a = 0.283 \text{ kg/s}$$

(ii) (Heat gained by water in high pressure heat exchanger) = (Heat rejected by air in high pressure heat exchanger

$$\therefore \dot{m}_{w} \times c_{pw} \Delta T_{w} = \dot{m}_{a} \times c_{pa} \times (T_{2} - T_{3})$$

$$\dot{m}_{w} \times 4.2 \times 30 = 0.283 \times 1.005 \times (413.3 - 303)$$

$$\Rightarrow \dot{m}_{w} = 0.248 \text{ kg/s}$$

(iii)  $T_{max} = T_2 = 413.3 - K = 140.3^{\circ}C$ 

$$T_{min} = T_4 = 203.8 \text{ K} = -69.2^{\circ} \text{ C}$$

(iv) 
$$\text{COP} = \frac{\text{Re frigeratio n effect}}{\text{Net work}}$$
  
=  $\frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)}$   
=  $\frac{278 - 203.8}{(413.3 - 278) - (303 - 203.3)} = 2.098 \simeq 2.1$ 

:31: