ESE Mains 2023

Mechanical Engineering

Questions & Solutions

PAPER - 1

ESE ME Paper 1 : Marks Distribution					
S.No.	Subjects	Difficulty Level 2023	2023 Marks	2022 Marks	2021 Marks
1	**Thermodynamics**	Moderate	60	72	32
2	**Refrigeration and Air-Conditioning**	Moderate	72	64	24
3	**Power Plant**	Tough	44	136	112
4	**IC Engines**	Easy	56	32	64
5	**Heat Transfer**	Moderate	84	52	92
6	**Fluid Mechanics and Machinery**	Moderate	112	64	84
7	**Renewable Source of Energy**	Moderate	52	60	72
Total		**Moderate but lengthy**	**480**	**480**	**480**

Mechanical Engineering

Paper-1

SECTION - 'A'

1.

(a) **(i)** Differentiate between rotational and irrotational flows. Can there be any possibility of having zones possessing characteristics of both rotational and irrotational flows?

[6 Marks]

(ii) If the expression for the stream function is described by $\psi = x^3 - 3x^2y$, determine whether the flow is rotational or irrotational. Further, find out the correct expression of the velocity potential function of the following two, considering the flow is irrotational:

(1) $\phi = y^3 - 3x^2y$

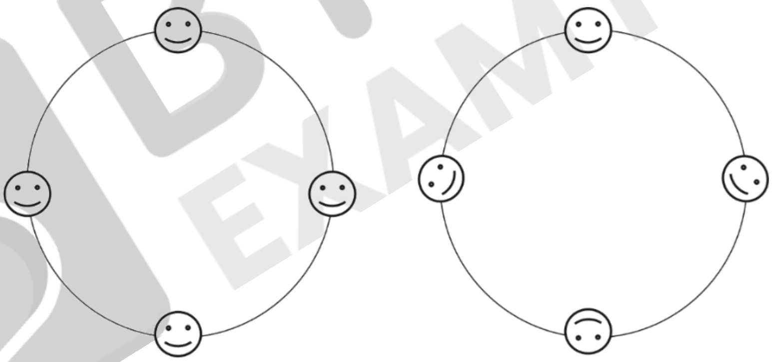
(2) $\phi = -7x^3y$

[6 Marks]

Sol.

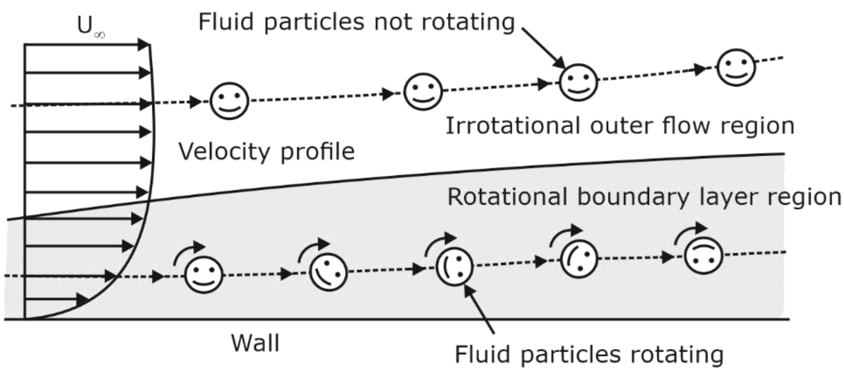
(i) **Rotational flow and irrotational flow**

- Rotational flow is the type of flow in which the fluid particles rotate about their axis while flowing along streamlines.
- On the other hand, a flow in which the fluid particles flow along the streamlines does not rotate about their axis is the irrotational flow.



(a) Irrotational flow (b) Rotational flow

When a viscous fluid flows over a flat plate, flow inside the boundary layer is rotational and flow outside the boundary layer is irrotational.



(ii) $\psi = x^3 - 3xy^2$

For irrotational flow, ψ must satisfy Laplace equation, i.e, $\frac{\partial^2\psi}{\partial x^2}+\frac{\partial^2\psi}{\partial y^2}=0$

LHS: $\frac{\partial}{\partial x}\left(\frac{\partial\psi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\partial\psi}{\partial y}\right)$

$=\frac{\partial}{\partial x}(3x^2-3y^2)+\frac{\partial}{\partial y}(-3x\times 2y)$

= 6x + (−6x)

= 0(RHS): Irrotational flow

Also, for 2-D, irrotational flow 'ϕ' must satisfy Laplace equation.

1. $\phi = y^3 - 3x^2y$

   To check: $\frac{\partial^2\phi}{\partial x^2}+\frac{\partial^2\phi}{\partial y^2}=0$

   LHS: $\frac{\partial}{\partial x}\left(\frac{\partial\phi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\partial\phi}{\partial y}\right)$

   $=\frac{\partial}{\partial x}(-6xy)+\frac{\partial}{\partial y}(3y^2-3x^2)$

   = −6y + (6y)

   = 0 (RHS)

   $\phi = y^3 - 3x^2y$ is a valid potential function.

2. $\phi = -7x^3y$

   To check: $\frac{\partial^2\phi}{\partial x^2}+\frac{\partial^2\phi}{\partial y^2}=0$

   LHS: $\frac{\partial}{\partial x}\left(\frac{\partial\phi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\partial\phi}{\partial y}\right)$

   $=\frac{\partial}{\partial x}(-7\times 3x^2y)+\frac{\partial}{\partial y}(-7x^3)$

   = (−21 × 2xy) + 0

   $= -42xy \neq 0$: $\phi = -7x^3y$ is not a valid potential function.

(b) A refrigerated truck whose dimensions are 12 m × 2.5 m × 3 m is to be precooled from 30 °C to an average temperature of 5 °C. The construction of the truck is such that a transmission heat gain occurs at the rate of 90 W/°C. If the ambient temperature is 30 °C, determine how long it will take for a system with a refrigeration capacity of 10 kW to precool this truck. The density of air may be taken as 1.2 kg/m³ and its specific heat at average temperature of 17.5 °C is C_p = 1 kJ/kg-°C. State the assumptions, if any.

[12 Marks]

Sol.

Mass of air inside refrigerated truck, $m = \rho \times Vol_{air}$

$= 1.2 \times (12 \times 2.5 \times 3)$

$= 108$ kg

Heat lost by air, $Q_{air} = mC_p\Delta T$

$$= 108 \times 1 \times 10^3 \times (30 - 5)$$
$$= 2700 \text{ kJ}$$
$$T_{avg} = \frac{30+5}{2} = 17.5\ °C$$

Heat transfer, $Q_{transfer} = UA\,(T_{ambiet} - T_{avg})$
$$= 90 \times (30 - T_{avg})$$
$$= 90 \times (30 - 17.5)$$
$$= 1125 \text{ W} = 1.125 \text{ kW}$$

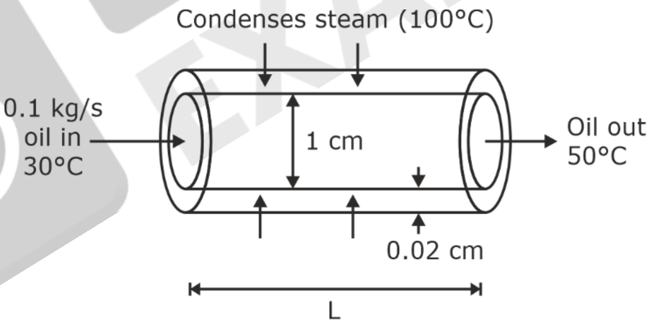
Refrigeration capacity, $Q_{ref} = 10$ kW
$$Q_{ref} = Q_{transfer} + \frac{Q_{air}}{t}$$

Where, t → time taken to precool the truck
$$10 = 1.125 + \frac{2700}{t}$$
Or t = 304.23 s = 5.07 min

(c) An engine oil flows through a copper tube of 1 cm internal diameter and 0.02 cm wall thickness at the flow rate of 0.1 kg/s. Consider that the temperature of the oil at the entry is 30 °C. If the oil is heated to 50 °C by steam condensing at atmospheric pressure, calculate the length of the copper tube. The properties of the oil are as follows:
C_p = 1964 J/kg-K, ρ = 876 kg/m³, k = 0.144 W/m-K,
μ = 0.210 N-s/m², Pr = 2870

[12 Marks]

Sol.



Heat transfer, $Q = \dot{m}C_p\,(T_e - T_i)$

$\dot{m}$ = mass flow rate
C_p = Specific heat of oil
T_e = Exit temperature of oil
T_i = Inlet temperature of oil

$$\dot{Q} = \dot{m}C_p\,(T_e - T_i)$$
$$= 0.1 \times 1964 \times (50 - 30) = 3928 \text{ J/s}$$

For oil,
$$Re = \frac{\rho \bar{V} d_i}{\mu}$$

d_i = inner diameter of pipe

$\bar{V}$ = average velocity of oil

ρ = density of oil

μ = viscosity of oil

A = cross-section area of pipe

$$Re = \frac{\rho \times \dfrac{\dot{m}}{\rho \times \frac{\pi}{4} d_i^2} \times d_i}{\mu} = \frac{4\dot{m}}{\pi d_i \mu}$$

$$Re = \frac{4 \times 0.1}{\pi \times 0.01 \times 0.210} = 60.63$$

Re < 2300, hence flow through pipe is laminar.

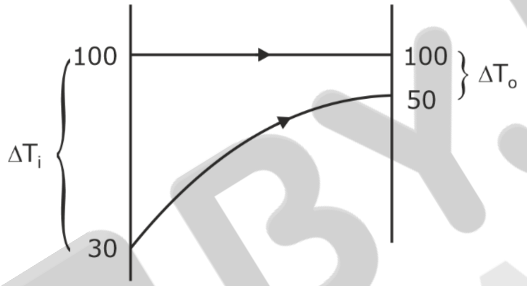
For fully developed laminar flow with constant wall temperature (as condenser steam)

$$Nu = \frac{h_i d_i}{k} = 3.66$$

$$h_i = 3.66 \times k/d_i$$

$$h_i = 3.66 \times 0.144/0.01$$

$$h_i = 52.704 \text{ W/m-K}$$



$$\Delta T_m = \frac{\Delta T_i - \Delta T_o}{\ln\left(\frac{\Delta T_i}{\Delta T_o}\right)} = \frac{70 - 50}{\ln\left(\frac{70}{50}\right)} = 59.44°C$$

$$Q = h_o A_o \Delta T_m$$

$$3928 = h_o \times \pi d_o \ell \Delta T_m$$

$$3928 = \frac{h_i d_i}{d_0} \times \pi d_o \ell \Delta T_m \qquad (\because h_i d_i = h_o d_o)$$

$$3928 = 52.704 \times 0.01 \times \pi \times \ell \times 59.44$$

$$\text{Or } \ell = 39.9 \text{ m}$$

(d) Explain the mechanism of NO_x formation and also the methods for its reduction in stationary gas turbine engines.

[12 Marks]

Sol. NO_x emission is one of the predominant emissions in stationary gas turbine engines. This emission is controlled by the standards. The most prevalent NO_x emission is nitric oxide or nitrogen monoxide, NO, and nitrogen dioxide, NO_2. Nitric oxide is the one which is mainly formed in the combustion chamber. Factors that influence the amount of NO formed are:

(i) Peak temperature,

(ii) Percentage of excess air,

(iii) Pressure,

(iv) Residence time at peak temperature and

(v) Fuel bound nitrogen

The peak temperature is attained when the fuel is burned with the stoichiometric (chemically correct) amount of air. Higher the temperature of the air at the inlet to the combustion chamber, higher the resulting equilibrium adiabatic flame temperature.

Burning the fuel with excess air lowers the maximum temperature but increases the availability of oxygen and nitrogen in the products of combustion. It is known that for a fixed air supply temperature and combustion chamber pressure, the amount of NO formed for equilibrium conditions increases from 0% excess air to 30% excess air, then starts to decrease even though the adiabatic equilibrium flame temperature decreases continuously. It is a known fact that increasing the combustion temperature, pressure, increase the equilibrium adiabatic flame temperature but decreases the amount of NO formed.

The preceding discussion assumes that equilibrium has been reached. The next important thing is to determine the rate at which at the products will reach equilibrium. The basic mechanism presently used to predict the formation of NO had its origin in the work of Zeldovich and coworkers around 1946.

NO_x Reduction in Stationary Engines:

It is known that the higher the temperature and longer the gases are at that temperature, more nitric oxide is formed. NO_x is the main pollutant from stationary gas turbine engines.

Prior to NO_x emission controls, gas turbine engine combustion chambers were designed so that the fuel-air ratio in the primary zone was approximately the stoichiometric value; that is, the percent excess air in the primary zone was 0%. This resulted in maximum temperature. The maximum temperature can be reduced by designing the combustion chamber so that the primary zone either operates fuel rich (insufficient air for complete combustion) or fuel lean (excess air). Both of these conditions can result in increased smoke (fuel rich) or increased carbon monoxide and total hydrocarbon emissions (fuel lean). Several methods can be used to reduce NO_x, emissions such as **water or steam injection or staged combustion or selective catalytic reduction.**

The most commonly used method of controlling NO_x, emissions is with water or steam injection into the primary zone of the combustion chamber. The water (or steam) injected acts as a heat sink, resulting in a lower maximum temperature, thereby reducing the amount of NO_x formed. The rate at which water is injected is approximately 50% of the fuel flow. Steam, rates are usually 100-200% of the fuel flow.

Staged combustion is currently being tested by a number of manufacturers. It provides a way of achieving NO_x emission levels of 25 ppm or less at 15% oxygen without using water or steam injection. Most of the systems being tested use a two-stage premixed combustor for use with natural gas. The resulting mixture is lean, so the amount of NO_x is low. Selective catalytic reduction involves injecting ammonia into the gas turbine engine exhaust steam. The exhaust gases then pass over a catalyst where the NO, reacts with the ammonia (NH_3), oxygen (O_2) and nitrogen, (N_2) to form water, (H_2O), and nitrogen (N_2). When combined with water or steam injection, it is reported that NO, levels of 10 ppm or less can be achieved. One major disadvantage is that the reaction is very much temperature dependent. For a vanadium pentoxide type catalyst, the exhaust gas temperature range for best operation is 600-750°F. For this reason, the selective catalytic reduction method for reducing NO, emission is limited to combined cycles only.

(e) **(i)** Why are higher heat transfer rates experienced in dropwise condensation than in film condensation?

[6 Marks]

(ii) Distinguish between nucleate boiling and film boiling.

[6 Marks]

Sol.

(i) Higher heat transfer rates are experienced in dropwise condensation compared to film condensation due to the following reasons:

Enhanced Surface Area: Dropwise condensation occurs when individual droplets form and grow on the condensing surface. These droplets act as separate condensation sites, creating a highly textured and rough surface. This roughness significantly increases the surface area available for heat transfer compared to a smooth film of condensed liquid in film condensation. The increased surface area allows for more efficient heat transfer.

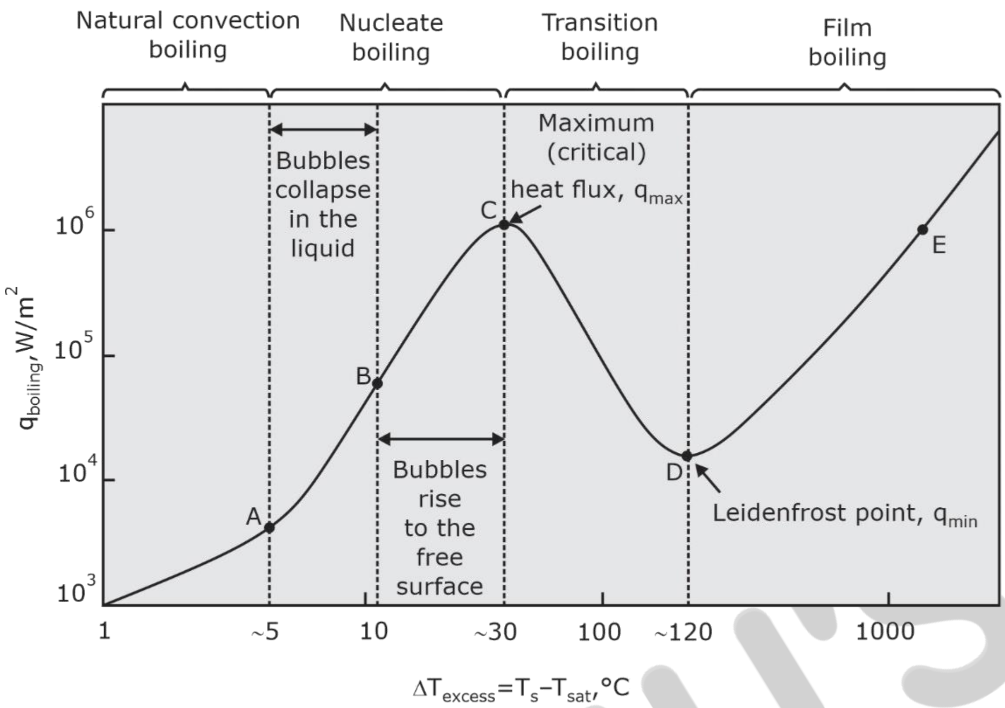
Reduced Thermal Resistance: In dropwise condensation, the formation of individual droplets on the surface creates a layer of vapor between the droplets and the surface. This vapor layer acts as an insulator, reducing the thermal resistance between the condensing surface and the droplets. Consequently, heat can more effectively transfer from the condensing surface to the droplets, leading to higher heat transfer rates.

Self-Cleaning Effect: Dropwise condensation exhibits a self-cleaning effect, where the droplets formed on the surface tend to coalesce and shed away, carrying heat with them. As the droplets detach, they leave behind fresh condensation sites, creating a continuous cycle of droplet formation. This self-cleaning mechanism prevents the accumulation of a stagnant film of condensate, which would otherwise hinder heat transfer. In film condensation, the continuous presence of a liquid film can lead to a decrease in heat transfer rates.

It's important to note that achieving and maintaining dropwise condensation can be challenging in practice, as the formation and retention of droplets can be disrupted by various factors such as surface contamination, impurities, and surface wetting characteristics. However, under ideal conditions, dropwise condensation offers superior heat transfer performance compared to film condensation.

(ii)

Nucleate boiling and film boiling are two different stages or regimes of boiling that occur during heat transfer processes. Here's how they can be distinguished:



Nucleate Boiling:

Mechanism: Nucleate boiling occurs when small bubbles of vapor form at discrete nucleation sites on a heated surface. These nucleation sites can be roughness elements, surface defects, or other imperfections. As the surface temperature increases, these bubbles grow and detach from the surface, causing agitation and mixing of the liquid.

Bubble Characteristics: In nucleate boiling, the bubbles formed are small and dispersed. They typically have a short lifespan and quickly detach from the surface due to buoyancy or fluid flow. The bubble formation and departure create a boiling noise and visual appearance known as "the Leidenfrost effect."

Heat Transfer: Nucleate boiling provides efficient heat transfer due to the direct contact between the hot surface and the bubbles. The bubbles act as carriers, removing heat from the surface as they rise to the bulk liquid. This mechanism enhances heat transfer rates compared to other modes of heat transfer.

Film Boiling:

Mechanism: Film boiling occurs when a continuous vapor film forms and blankets the heated surface. At high heat fluxes or surface temperatures, the liquid near the surface vaporizes rapidly, creating a vapor layer that insulates the surface from the bulk liquid. This film of vapor inhibits direct contact between the surface and the liquid, leading to reduced heat transfer efficiency.

Vapor Film Characteristics: In film boiling, a stable and continuous vapor film exists on the heated surface. The thickness of this vapor film is relatively large compared to the small bubbles observed in nucleate boiling. The presence of the vapor film creates a barrier to heat transfer.

Heat Transfer: Film boiling is associated with significantly reduced heat transfer rates compared to nucleate boiling. The insulating vapor film impedes heat transfer by introducing a large thermal resistance between the heated surface and the liquid. The heat transfer in film boiling primarily occurs through radiation and conduction across the vapor film.

In summary, nucleate boiling involves the formation and detachment of small bubbles from the heated surface, providing efficient heat transfer through direct contact between the surface and the liquid. On the other hand, film boiling occurs when a continuous vapor film forms on the surface, resulting in reduced heat transfer efficiency due to the presence of an insulating vapor layer.

2.

(a) **(i)** Find the distance from the pipe wall at which the local velocity is equal to the average velocity for turbulent flow in pipe.

[12 Marks]

(ii) Distinguish between hydrodynamically smooth and rough boundaries.

[8 Marks]

Sol.

(i) For turbulent flow in pipe ($Re \geq 4000$)

As we know $\frac{V_{avg} - u}{V^*} = 5.75 \log\left(\frac{R}{y}\right) - 3.75$

For $V_{avg} = u$,

$$5.75 \log\left(\frac{R}{y}\right) - 3.75 = 0$$

$$\log\left(\frac{R}{y}\right) = \frac{3.75}{5.75}$$

$$\frac{R}{y} = 4.489$$

$$y = \frac{R}{4.489}$$

$$y = 0.223\, R$$

Where y = distance from pipe wall

R = Radius of pipe

(ii) **HYDRODYNAMICALLY SMOOTH & ROUGH BOUNDARIES**

In turbulent flow in pipes, the region closes to the boundary, the effect of viscosity is maximum, i.e., flow is still laminar. This is known as the **laminar sublayer**.

The thickness of the laminar sublayer is directly proportional to the kinematic viscosity and inversely proportional to flow velocity. Thus, the thickness of the laminar sublayer decreases with an increase in the Reynold number.

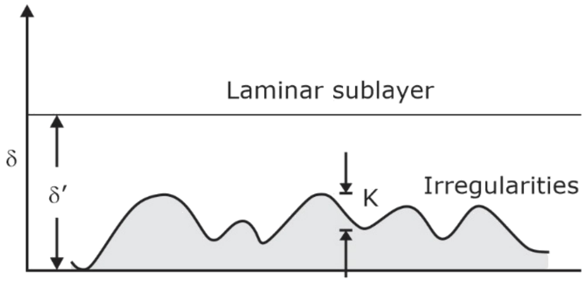
According to Nikuradse, the expression of the laminar sublayer is given by

$$\delta' = \frac{11.6\,\nu}{V_*}$$

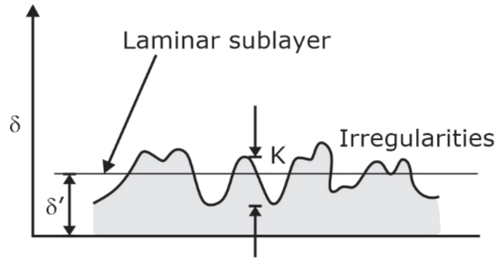
If the thickness of the laminar sublayer is large and eddies are not able to penetrate up to the boundary, then the boundary acts as hydrodynamically smooth.

If the thickness of the laminar sublayer is small and eddies are able to penetrate till the boundary, then the boundary acts as hydrodynamically rough.

Let K is the average height of the irregularities projecting from the surface of a boundary, as shown in figure.



(a) Smooth boundary



(b) Rough boundary

The boundary is said to be rough if the value of K is large and smooth if the value of K is low.

	Nikuradse's experiment	**Roughness Reynolds number**
Smooth boundary	$\frac{K}{\delta'} < 0.25$	$\frac{V_* K}{\nu} < 4$
Transition range	$0.25 < \frac{K}{\delta'} < 6$	$4 < \frac{V_* K}{\nu} < 100$
Rough boundary	$\frac{K}{\delta'} > 6$	$\frac{V_* K}{\nu} > 100$

(b) **(i)** In a closed system, 3 kg of air at initial conditions of 400 kPa and 90 °C adiabatically expands until its volume is 2.5 times the initial volume and temperature becomes equal to that of surroundings. If the conditions of the surroundings are 100 kPa and 25 °C, determine the following for this process:

(1) The maximum work

(2) The change in availability

(3) The irreversibility

[15 Marks]

(ii) Prove that for an ideal gas, the slope of an isochoric line on the T-s diagram is more than that of the isobaric line.

[5 Marks]

Sol.

(i) Given,

$m_{air} = 3$ kg

$P_1 = 400$ kPa

$T_1 = 90°C$

$V_2 = 2.5\ V_1$

$P_0 = 100$ kPa

$T_0 = 25°C$

1. $(\Delta s)_{sys} = c_V \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{V_2}{V_1}\right)$

$$= 0.718 \times \ln\left(\frac{25+273}{10+273}\right) + 0.287 \times \ln(2.5)$$

$(\Delta s)_{sys}$ = – 0.1417 + 0.2629

$(\Delta s)_{sys}$ = 0.1212 kJ/kg-K

$\Delta S_{sys} = 3 \times 0.1212 = 0.3637$ kJ/K

$W_{max} = T_0(\Delta S) - \Delta U$

$= 298 \times 0.3637 - mc_V \Delta T$

$= 108.38 - 3 \times 0.718\ (25 - 90)$

$= 108.38 + 140.01 = 248.39$ kJ

2. Change in availability = $\phi_1 - \phi_2$

Where $\phi = U + P_0V - T_0S$

$\Delta\phi = (\phi_1 - \phi_2) = (U_1 + P_0V_1 - T_0S_1) - (U_2 + P_0V_2 - T_0S_2)$

$= T_0(S_2 - S_1) - (U_2 - U_1) - P_0(V_2 - V_1)$

$\Delta\phi = W_{max} - P_0(V_2 - V_1)$

$= 248.39 - 100 \times (2.5V_1 - V_1)$

$\Delta\phi = 248.39 - 100 \times 1.5\ V_1$ ...(i)

$$V_1 = \frac{mRT_1}{P_1} = \frac{3 \times 0.287 \times 363}{400}\ m^3$$

$V_1 = 0.7813\ m^3$

Put in equation (i)

$\Delta\phi = 248.39 - 100 \times 1.5 \times 0.7813$ kJ

$\Delta\phi = 131.186$ kJ

3. Irreversibility (I)

$I = T_0(\Delta S)_{universe} = T_0(\Delta S_{system} + \Delta S_{surr})$

$I = T_0(\Delta S)_{sys}$

$I = 298 \times 0.3637$ kJ = 108.38 kJ

(ii) For isochoric process (V = C)

TdS = dU + PdV (PdV = 0)

TdS = dU = $mc_V dT$ (For ideal gas)

$$\left(\frac{dT}{dS}\right)_V = \frac{T}{mc_V}$$ (Slope of isochoric process on T-S curve)

For isobaric process (P = constant)

TdS = dH – VdP (VdP = 0)

TdS = dH = $mc_p dT$ (For ideal gas)

$$\left(\frac{dT}{dS}\right)_P = \frac{T}{mc_p}$$ (Slope of isobaric process on T-S curve)

As $c_p > c_v \Rightarrow \left(\frac{dT}{dS}\right)_P < \left(\frac{dT}{dS}\right)_V$

(c) A square plate heater (15 cm × 15 cm) is inserted between two slabs. Slab A is 2 cm thick (k = 50 W/m-°C) and slab B is 1 cm thick (k = 0.2 W/m-°C). The outside heat transfer coefficients on side of A and side of B are 200 W/m²-°C and 50 W/m²-°C respectively. The temperature of surrounding air is 25 °C. If the rating of heater is 1 kW, find the-

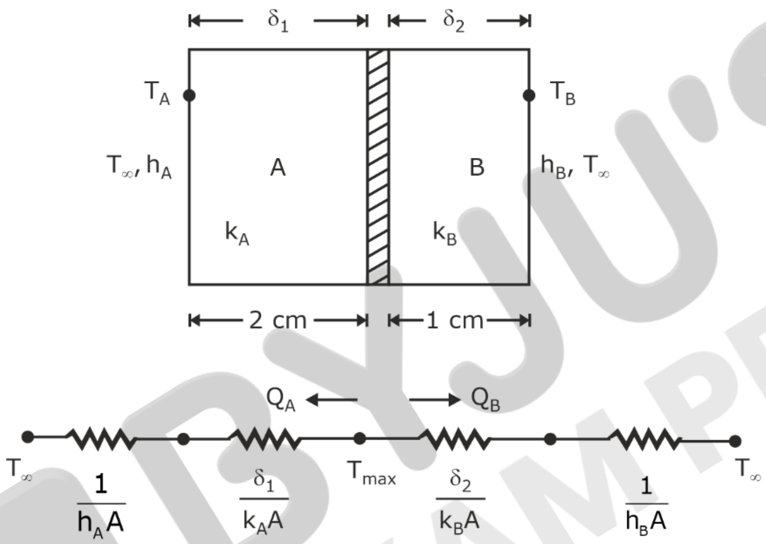
(i) maximum temperature of the system.

(ii) outer surface temperature of two slabs.

Assume steady-state heat flow.

[20 Marks]

Sol.



k_A = 50 W/m-°C, k_b = 0.2 W/m-°C

h_A = 200 W/m²-°C, h_B = 50 W/m²-°C

Q = 1000 W

$$R_{th_1} = \frac{\delta_1}{k_A A} = \frac{0.02}{50 \times 0.15 \times 0.15} = 0.0178 \text{ K/W}$$

$$R_{th_2} = \frac{1}{h_A A} = \frac{1}{200 \times 0.15 \times 0.15} = 0.22 \text{ K/W}$$

These resistances are in series and accordingly for slab A.

$R_A = R_{th_1} + R_{th_2}$ = 0.0178 + 0.22 = 0.24 K/W

$$R_{th_3} = \frac{\delta_2}{k_B A} = \frac{0.01}{0.2 \times 0.15 \times 0.15} = 2.22 \text{ K/W}$$

$$R_{th_4} = \frac{1}{h_B A} = \frac{1}{50 \times 0.15 \times 0.15} = 0.889 \text{ K/W}$$

These resistances are in series and accordingly for slab B.

$R_B = R_{th_3} + R_{th_4}$ = 2.22 + 0.889 = 3.109 K/W

Rating of heater, Q = $Q_A + Q_B$

$$Q = \frac{T_{max} - T_\infty}{R_A} + \frac{T_{max} - T_\infty}{R_B}$$

$$Q = (T_{max} - T_\infty)\left(\frac{1}{R_A} + \frac{1}{R_B}\right)$$

$$1000 = (T_{max} - 25)\left(\frac{1}{0.24} + \frac{1}{3.109}\right)$$

$$T_{max} = 247.8\ °C$$

Considering left side branch of circuit (slab A)

$$Q_A = \frac{T_{max} - T_\infty}{R_A} = \frac{247.8 - 25}{0.24} = 928.34\ W$$

If T_A is the temperature at exposed surface of slab A, then,

$$Q_A = \frac{T_A - T_\infty}{R_{th_2}}$$

$$928.34 = \frac{T_A - 25}{0.22}$$

$$T_A = 229.23\,°C$$

Considering right side branch of the circuit (slab B)

$$Q_B = \frac{T_B - T_\infty}{R_{th_4}}$$

$$Q - Q_A = \frac{T_B - T_\infty}{R_{th_4}}$$

$$1000 - 928.34 = \frac{T_B - 25}{0.889}$$

$$T_B = 88.71\,°C$$

3.

(a) A centrifugal pump discharges 2000 litres/s of water developing a head of 20 m when running at 300 rpm. The impeller diameter at the outlet and outlet flow velocity are 1.5 m and 3.0 m/s respectively. If the blades are set back at an angle of 30° at the outlet, determine the

(i) manometric efficiency.

(ii) power required by the pump.

(iii) minimum speed to start the pump if the inner diameter is 750 mm.

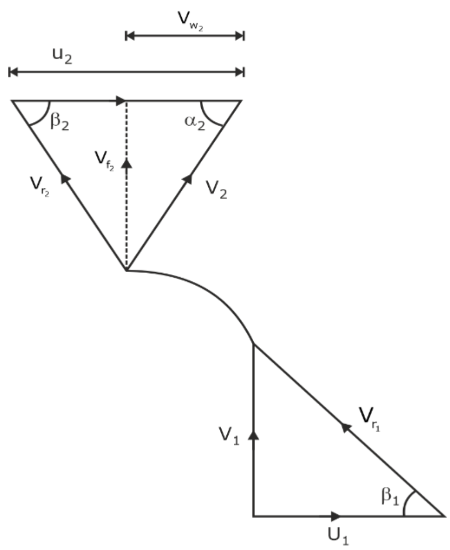
[20 Marks]

Sol. Given,

$Q = 2000$ lt/s $= 2\ m^3/s$, $H_m = 20$ m, $N = 300$ rpm

$D_2 = 1.5$ m, $V_{f2} = 3$ m/s, $\beta_2 = 30°$ (backward angle vanes)

Outlet velocity triangle



$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 1.5 \times 300}{60} = 23.56 \text{ m/s}$$

$$\tan\beta_2 = \frac{V_{f_2}}{u_2 - V_{w_2}}$$

$$\tan 30^\circ = \frac{3}{23.56 - V_{w_2}}$$

$$V_{w_2} = 18.36 \text{ m/s}$$

i. Manometric efficiency (hydraulic efficiency)

$$\eta_h = \frac{\text{Hydraulic Power}}{\text{Rotor Power}} = \frac{m g H_m}{m\left(V_{w_2} u_2\right)}$$

$$\eta_h = \frac{9.81 \times 20}{18.36 \times 23.56} = 0.4535 = 45.35\%$$

ii. Power required by pump = shaft power

SP = T × ω = RP – Mechanical losses

Assuming mechanical losses as zero.

$SP = RP = \dot{m}(V_{w2}u_2) = \rho Q(V_{w2}u_2)$

$= 10^3 \times 2 \times (18.36 \times 23.56) = 865123.2 \text{ W} = 865.123 \text{ kW}$

iii. Minimum speed to start the pump if D_1 = 750 mm

$$\omega_{min} = \sqrt{\frac{2gH_m}{\left(r_2^2 - r_1^2\right)}} = \sqrt{\frac{2 \times 9.81 \times 20}{\left(\frac{1.5}{2}\right)^2 - \left(\frac{0.75}{2}\right)^2}}$$

$$\omega_{min} = 30.498 \text{ rad/s} = \frac{2\pi N_{min}}{60}$$

$$N_{min} = 291.23 \text{ rpm}$$

(b) Air flows at 12 m/s past a smooth rectangular flat plate 0.4 m wide and 3 m long. Assuming that the transition occurs at Re =5.5 × 10^5, calculate the total drag force when-

(i) the flow is parallel to the length of the plate.

(ii) the flow is parallel to the width of the plate.

Assume,

Density of air, ρ = 1.24 kg/m³

Kinematic viscosity, v = 0.15 stokes

[20 Marks]

Sol. Given,

u_∞ = 12 m/s, ρ_{air} = 1.24 kg/m^3

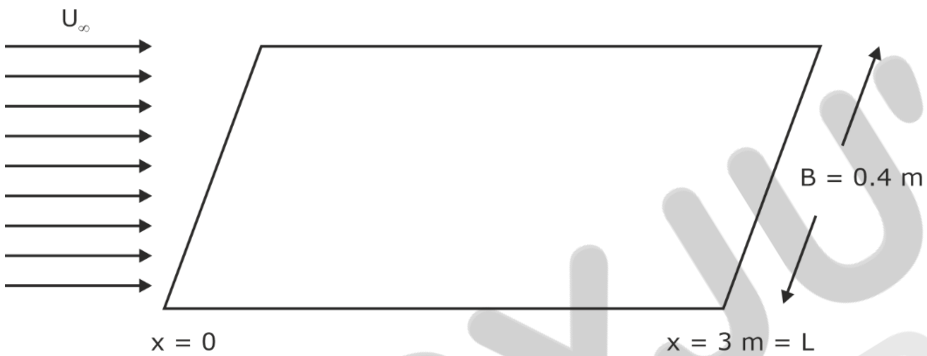
L = 3 m, ν = 0.15 stokes = 0.15 × 10^{-4} m^2/s

B = 0.4 m

Re_{cr} = 5.5 × 10^5

To find total drag force (F_D)

i. Flow parallel to length of plate



$$Re_L = \frac{\rho u_\infty L}{\mu} = \frac{u_\infty L}{\nu}$$

$$Re_L = \frac{12\times 3}{0.15\times 10^{-4}} = 2.4\times 10^6$$

$$Re_{Cr} = \frac{u_\infty x_{Cr}}{\nu} = 5.5\times 10^5$$

$$x_{Cr} = \frac{5.5\times 10^5 \times 0.15\times 10^{-4}}{12} = 0.6875 \text{ m}$$

Up to x_{Cr} = 0.6875, boundary layer will be laminar.

$$C_D = 2C_{f,x_{Cr}}$$

$$C_D = \frac{2\times 0.664}{\sqrt{Re_{Cr}}} = \frac{1.328}{\sqrt{5.5\times 10^5}} = 1.79\times 10^{-3}$$

$$F_{D_I} = C_D A_I\left(\frac{\rho u_\infty^2}{2}\right)$$

$$F_{D_I} = 1.79\times 10^{-3}\times(0.6875\times 0.4)\times\frac{1.24\times 12^2}{2}$$

$$F_{D_I} = 0.0439 \text{ N}$$

For IInd Region (Turbulent BL region)

$$C_D = \frac{5}{4}\times C_{f_{x=L}} = \frac{5}{4}\times\frac{0.059}{(Re_L)^{1/5}}$$

$$C_D = \frac{5}{4} \times \frac{0.059}{(2.4 \times 10^6)^{1/5}} = 3.9 \times 10^{-3}$$

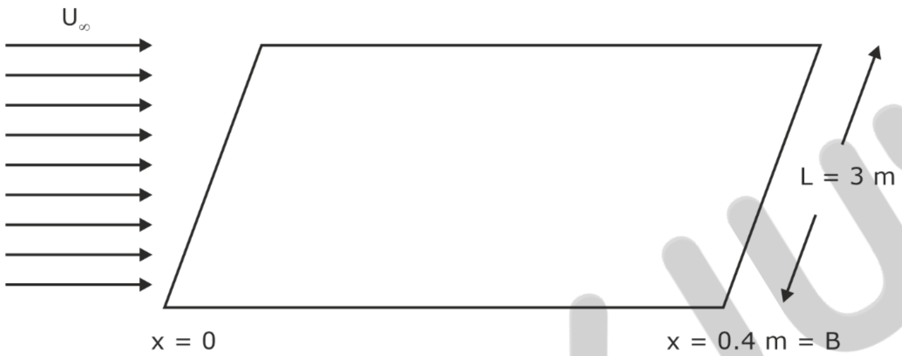
$$F_{D_{II}} = C_D A \left(\frac{\rho u_\infty^2}{2} \right)$$

$$F_{D_{II}} = 3.9 \times 10^{-3} \times (3 - 0.6875) \times 0.4^2 \times \frac{1.24 \times 12^2}{2}$$

$$F_{D_{II}} = 0.322 \text{ N}$$

$$F_{total} = FD_I + FD_{II} = 0.0439 + 0.322 = 0.366 \text{ N}$$

ii. Flow parallel to plate width



$$Re_x = \frac{u_\infty \times 0.4}{\nu} = \frac{12 \times 0.4}{0.15 \times 10^{-4}}$$

$$Re = 3.2 \times 10^5 < 5.5 \times 10^5$$

Laminar Boundary layer over the width plate

$$C_D = 2 \times C_{f,x=0.4} = \frac{2 \times 0.664}{\sqrt{Re_{x=0.4}}} = \frac{2 \times 0.664}{\sqrt{3.2 \times 10^5}}$$

$$C_D = 2.347 \times 10^{-3}$$

$$F_{D_{total}} = C_D A \left(\frac{\rho u_\infty^2}{2} \right) = 2.347 \times 10^{-3} \times (0.4 \times 3) \times \frac{1.24 \times 12^2}{2}$$

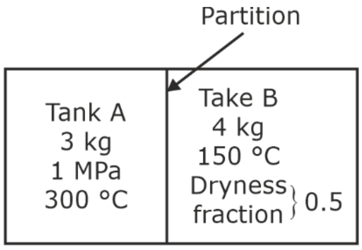
$$F_{D_{total}} = 0.2515 \text{ N}$$

(c) Two tanks, tank A and tank B, are separated by a partition as shown in the figure. Tank A contains 3 kg of steam at 1 MPa and 300 °C. Tank B contains 4 kg of saturated liquid-vapour mixture at 150 °C with a dryness fraction of 0.5. The partition is removed, and two fluids are allowed to mix until the thermal equilibrium and mechanical equilibrium are acquired. If the pressure of the final state is 300 kPa, determine-

(i) the temperature of the final state.

(ii) the quality of the steam at final state.

(iii) the amount of heat lost from the tanks.



[20 Marks]

Steam Table

Temp. °C T	Pressure kPa, MPa P	Specific Volume, m³/kg: Sat. Liquid v_f	Sat. Vapour v_g	Internal Energy, kJ/kg: Sat. Liquid u_f	Evap. u_{fg}	Sat. Vapour u_g	Enthalpy, kJ/kg: Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Entropy, kJ/kg-K: Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapour s_g
105	0.12082	0.001047	1.4194	440.00	2072.3	2512.3	440.13	2243.7	2683.8	1.3629	5.9328	7.2958
110	0.14328	0.001052	1.2102	461.12	2057.0	2518.1	461.27	2230.2	2691.5	1.4184	5.8202	7.2386
115	0.16906	0.001056	1.0366	482.28	2041.4	2523.7	482.46	2216.5	2699.0	1.4733	5.7100	7.1832
120	0.19853	0.001060	0.8919	503.48	2025.8	2529.2	503.69	2202.6	2706.3	1.5275	5.6020	7.1295
125	0.2321	0.001065	0.77059	524.72	2009.9	2534.6	524.96	2188.5	2713.5	1.5812	5.4962	7.0774
130	0.2701	0.001070	0.66850	546.00	1993.9	2539.9	546.29	2174.2	2720.5	1.6343	5.3925	7.0269
135	0.3130	0.001075	0.58217	567.34	1977.7	2545.0	567.67	2159.6	2727.3	1.6869	5.2907	6.9777
140	0.3613	0.001080	0.50885	588.72	1961.3	2550.0	589.11	2144.8	2733.9	1.7390	5.1908	6.9298
145	0.4154	0.001085	0.44632	610.16	1944.7	2554.9	610.61	2129.6	2740.3	1.7906	5.0926	6.8832
150	0.4759	0.001090	0.39278	631.66	1927.9	2559.5	632.18	2114.3	2746.4	1.8417	4.9960	6.8378
155	0.5431	0.001096	0.34676	653.23	1910.8	2564.0	653.82	2098.6	2752.4	1.8924	4.9010	6.7934
160	0.6178	0.001102	0.30706	674.85	1893.5	2568.4	675.53	2082.6	2758.1	1.9426	4.8075	6.7501
165	0.7005	0.001108	0.27269	696.55	1876.0	2572.5	697.32	2066.2	2763.5	1.9924	4.7153	6.7078
170	0.7917	0.001114	0.24283	718.31	1858.1	2576.5	719.20	204.5	2768.7	2.0415	4.6244	6.6663
175	0.8920	0.001121	0.21680	740.16	1840.0	2580.2	741.16	2032.4	2773.6	2.0909	4.5347	6.6256
180	1.0022	0.001127	0.19405	762.08	1821.6	2583.7	763.21	2015.0	2778.2	2.1395	4.4461	6.5857
185	1.1227	0.001134	0.17409	784.08	1802.9	2587.0	785.36	1997.1	2782.4	2.1878	4.3586	6.5464
190	1.2544	0.001141	0.15654	806.17	1783.8	2590.0	807.61	1978.8	2786.4	2.2358	4.2720	6.5078
195	1.3978	0.001149	0.14105	828.36	1764.4	2592.8	829.96	1960.0	2790.0	2.2835	4.1863	6.4697
200	1.5538	0.001156	0.12736	850.64	1744.7	2595.3	852.43	1940.7	2793.2	2.3308	4.1014	6.4322
205	1.7230	0.001164	0.11521	873.08	1724.5	2597.5	875.03	1921.0	2796.0	2.3779	4.0172	6.3951
210	1.9063	0.001173	0.10441	895.51	1703.9	2599.4	897.75	1900.7	2798.5	2.4247	3.9337	6.3584
215	2.1042	0.001181	0.09479	918.12	1682.9	2601.1	950.61	1879.9	2800.5	2.4713	3.8507	6.3221
220	2.3178	0.001190	0.08619	940.85	1661.5	2602.3	943.61	1858.5	2802.1	2.5177	3.7683	6.2860

Steam Table

T	v	u	h	s	v	u	h	s	v	u	h	s
	P = 200 kPa (120.23)				P = 300 kPa (133.55)				P = 400 kPa (143.63)			
900	2.70643	3854.5	4395.8	9.4565	1.80406	3854.2	4395.4	9.2691	1.35288	3853.9	395.1	9.1361
1000	2.93740	4052.5	4640.0	9.6563	1.95812	4052.3	4639.7	9.4689	1.46847	4050.0	4639.4	9.3360
1100	3.16834	4257.0	4890.7	9.8458	2.11214	4256.8	4890.4	9.6585	1.58404	4256.5	4890.1	9.5255
1200	3.39927	4467.5	5147.3	10.0262	2.26614	4467.2	5147.1	9.8389	1.69958	4467.0	5146.8	9.7059
1300	3.63018	4683.2	5409.3	10.1982	2.42013	4683.0	5409.0	10.0109	1.81511	4682.8	5408.8	9.8780
	P = 500 kPa (151.86)				P = 600 kPa (158.85)				P = 800 kPa (170.43)			
Sat.	0.37489	2561.2	2748.7	6.8212	0.31567	2567.4	2756.8	6.7600	0.24043	2576.8	2769.1	6.6627
200	0.42492	2642.9	2855.4	7.0592	0.35202	2638.9	2850.1	6.9665	0.26080	2630.6	2839.2	6.8158
250	0.47436	2723.5	2960.7	7.2708	0.39383	2720.9	2957.2	7.1816	0.29314	2715.5	2950.0	7.0384
300	0.52256	2802.9	3064.2	7.4598	0.43437	2801.0	3061.6	7.3723	0.32411	2797.1	3056.4	7.2372
350	0.57012	2882.6	31.67.6	7.6328	0.47424	2881.1	3165.7	7.5463	0.35439	2878.2	3161.7	7.4088
400	0.61728	2963.2	3271.8	7.7937	0.51372	2962.0	3270.2	7.7078	0.38426	2959.7	3267.1	7.5715
500	0.71093	3128.4	3483.8	8.0872	0.59199	3127.6	3482.7	8.0020	0.44331	3125.9	3480.6	7.8672
600	0.80406	3299.6	3701.7	8.3521	0.66974	3299.1	3700.9	8.2673	0.50184	3297.9	3699.4	8.1332
700	0.89691	3477.5	3926.0	8.5952	0.74720	3477.1	3925.4	8.5107	056007	3476.2	3924.3	8.3770
800	0.98959	3662.2	4157.0	8.8211	0.82450	3661.8	4156.5	8.7367	0.61813	3661.1	4155.7	8.6033
900	1.08217	3853.6	4394.7	9.0329	0.90169	3853.3	4394.4	8.9485	0.67610	3852.8	4393.6	8.8153
1000	1.17469	4051.8	4639.1	9.2328	0.97883	4051.5	4638.8	9.1484	0.73401	4051.0	4638.2	9.0153
1100	1.6718	4256.3	4889.9	9.4224	1.05594	4256.1	4889.6	9.3381	0.79188	4255.6	4889.1	9.2049
1200	1.35964	4466.8	5146.6	9.6028	1.13302	4466.5	5146.3	9.5185	0.84974	4466.1	5145.8	9.3854
1300	1.45210	4682.5	5408.6	9.7749	1.21009	4682.3	5408.3	9.6906	0.90758	4681.8	5407.9	9.5575
	P = 1.00 MPa (179.91)				P = 1.20 MPa (187.99)				P = 1.40 MPa (195.07)			
Sat.	0.19444	2583.6	2778.1	6.5864	0.16333	2588.8	2784.8	6.5233	0.14084	2592.8	2790.0	6.4692
200	0.20596	2621.9	2827.9	6.6939	0.16932	2612.7	2815.9	6.5898	0.14302	2603.1	2803.3	6.4975
250	0.23268	2709.9	2942.6	6.9246	0.19235	2704.2	2935.0	6.8293	0.16350	2698.3	2927.2	6.7467
300	0.25794	2793.2	3051.2	7.1228	0.21382	2789.2	3045.8	7.0316	0.18228	2785.2	3040.4	6.9533
350	0.28247	2875.2	3157.7	7.3010	0.23452	2872.2	3153.6	7.2120	0.20026	2869.1	3149.5	7.1359
400	0.30659	2957.3	3263.9	7.4650	0.25480	2954.9	3260.7	7.3773	0.21780	2952.5	3257.4	7.3025
500	0.35411	3124.3	3478.4	7.7621	0.29463	3122.7	3476.3	7.6758	0.25215	3121.1	3474.1	7.6026

Steam Table

		Specific Volume, m^3/kg		Internal Energy, kJ/kg			Enthalpy, kJ/kg			Entropy, kJ/kg-K		
Pressure MPa P	Temp. °C T	Sat. Liquid v_f	Sat. Vapour v_g	Sat. Liquid u_f	Evap. u_{fg}	Sat. Vapour u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapour s_g
0.275	130.60	0.001070	0.6573	548.57	1992.0	2540.5	548.87	2172.4	2721.3	1.6407	5.3801	7.0208
0.300	133.55	0.001073	0.6058	561.31	1982.4	2543.6	56145	2163.9	2725.3	1.6717	5.3201	6.9918
0.325	136.30	0.001076	0.5620	572.88	1973.5	2546.3	573.23	2155.8	2729:0	1.7005	5.2646	6.9651
0.350	138.88	0.001079	0.5243	583.93	1965.0	2548.9	584.31	2148.1	2732.4	1.7274	5.2130	6.9404
0.375	141.32	0.001081	0.4914	594.38	1956.9	2255.13	594.79	2140.8	2735.6	1.7527	5.1647	6.9174
0.40	143.63	0.001084	0.4625	604.29	1949.3	2553.6	604.73	2133.8	2738.5	1.7766	5.1193	6.8958
0.45	147.93	0.001088	0.4140	622.75	1934.9	2557.6	623.24	2120.7	2743.9	1.8206	5.0359	6.8565
0.50	151.86	0.001093	0.3749	639.66	1921.6	2561.2	640.21	2108.5	2748.7	1.8606	4.9606	6.8212
0.55	155.48	0.001097	0.3427	655.30	1909.2	2564.3	655.91	2097.0	2752.9	1.8972	4.8920	5.7892
0.60	158.85	0:001101	0.3157	669.88	1897.5	2567.4	670.54	2086.3	2756.8	1.9311	4.8289	5.7600
0.65	162.01	0:001104	0.2927	683.55	1886.5	2570.1	684.26	2076.0	2760.3	1.9627	4.7704	6.7330
0.70	164.97	0:001108	0.2729	696.43	1876.1	2672.5	697.20	2066.3	2763.5	1.9922	4.7158	6.7080
0.75	167.77	0.001111	0.2556	708.62	1866.1	2574.7	709.45	2057:0	2766.4	2.0199	4:6647	6.6846
0.80	170.43	0.001115	0.2404	720.20	1856.6	2576.8	721.10	2048.0	2769.1	2.0461	4.6166	6.6627
0.85	172.96	0:001118	0.2270	731.25	1847.4	2578.7	732:20	2039.4	2771.6	2.0709	4.5711	6.6421
0.90	175.38	0.001121	0.2150	741.81	1838.7	2580.5	742.82	2031.1	2773.9	2.0946	4.5280	6.6225
0.95	177.69	0.001124	0.2042	751.94	1830.2	2582.1	753.00	2023.1	2776.1	2.1171	4.4869	6.6040
1.00	179.91	0.001127	0.19444	761.67	1822.0	2583.6	762.79	2015.3	2778.1	2.1386	4.4478	6.5864
1.10	184.09	0.001133	0.17753	780.08	1806.3	2586.4	781:32	2000.4	2781.7	2.1791	4.3744	6.5535
1.20	187.99	0.001139	0.16333	797.27	1791.6	2588.8	798.64	1986.2	2784.8	2.2165	4.3067	6.5233
1.30	191.64	0.001144	0.15125	813.42	1777.5	2590.9	814.91	1972.7	2787.6	2.2514	4.2438	6.4953
1.40	195.07	0.001149	0.14084	828.68	1764.1	2592.8	830.29	1959.7	2790.0	2.2842	4.1850	6.4692
1.50	198.32	0:001154	0.13177	843.14	1751.3	2594.5	844.87	1947.3	2792.1	2.3150	4.2198	6.4448
1.75	205.76	0.001166	0.11349	876.44	1721.4	2597.8	878.48	1918.0	2796.4	2.3851	4.0044	6.3895
2.00	212.42	0.001177	0.09963	906.42	1693.8	2600.3	908.77	1890.7	2799.5	2.4473	3.8935	6.3408
2.25	218.45	0.001187	0.08875	933.81	1668.2	2602.0	936.48	1865.2	2801.7	2.5034	3.7938	6.2971

Sol. Given,

m_A = 3 kg(steam), m_B = 4 kg (saturates liquid-vapour)

P_{1A} = 1 MPa, T_{1B} = 150°C, T_{1A} = 300°C, x_{1B} = 0.5

After equilibrium: P_{mix} = 300 kPa

From steam table (For tank A)

u_{1A} = 2793.7 kJ/kg

v_{1A} = 0.25799 m^3/kg

$V_A = m_A \times v_{1A} = 3 \times 0.25799 = 0.77397\ m^3$

For Tank B (At 150°C)

$v_{1B} = v_f + x_{1B} \times (v_g - v_f)$

$v_{1B} = 0.001090 + 0.5(0.39248 - 0.001090)$

$v_{1B} = 0.196935\ m^3/kg$

$V_B = m_B \times v_{1B} = 4 \times 0.196935 = 0.78774\ m^3$

As $V_{mix} = V_A + V_B$

$V_{mix} = (0.77397 + 0.78774)\ m^3$

$V_{mix} = 1.562\ m^3$

$m_{mix} = m_A + m_B = 3 + 4 = 7$ kg

$$v_{mix} = \frac{V_{mix}}{m_{mix}} = \frac{1.562}{7}\ m^3/kg$$

$v_{mix} = 0.2231\ m^3/kg$

At P_{mix} = 300 kPa

At P_{sat} = 300 kPa, $v_f = 0.001073\ m^3/kg$; $v_g = 0.60582\ m^3/kg$

As $v_f < v_{mix} < v_g$

Saturation mixture

$T_2 = T_{sat} = 133.52°C$ (At P_{sat} = 300 kPa)

ii. $v_{mix} = v_f + x_{mix}(v_g - v_f)$

$$x_{mix} = \frac{v_{mix} - v_f}{v_g - v_f} = \frac{0.2231 - 0.001073}{0.60582 - 0.001073}$$

$x_{mix} = 0.367$

iii. $u_{mix} = u_f + x_{mix}(u_g - v_f)$

$u_{mix} = 561.11 + 0.367\ (2543.21 - 561.11)$

u_{mix} = 1286.5 kJ/kg

First law of thermodynamic

$Q = \Delta U + W$ (W = 0)

$Q = U_{mix} - (U_A + U_B)$

$= m_{mix} \times u_{mix} - (m_A \times u_{1A} + m_B \times u_{1B})$

$= (7 \times 1286.5) - [(3 \times 2793.7) + (4 \times 1595.36)]$

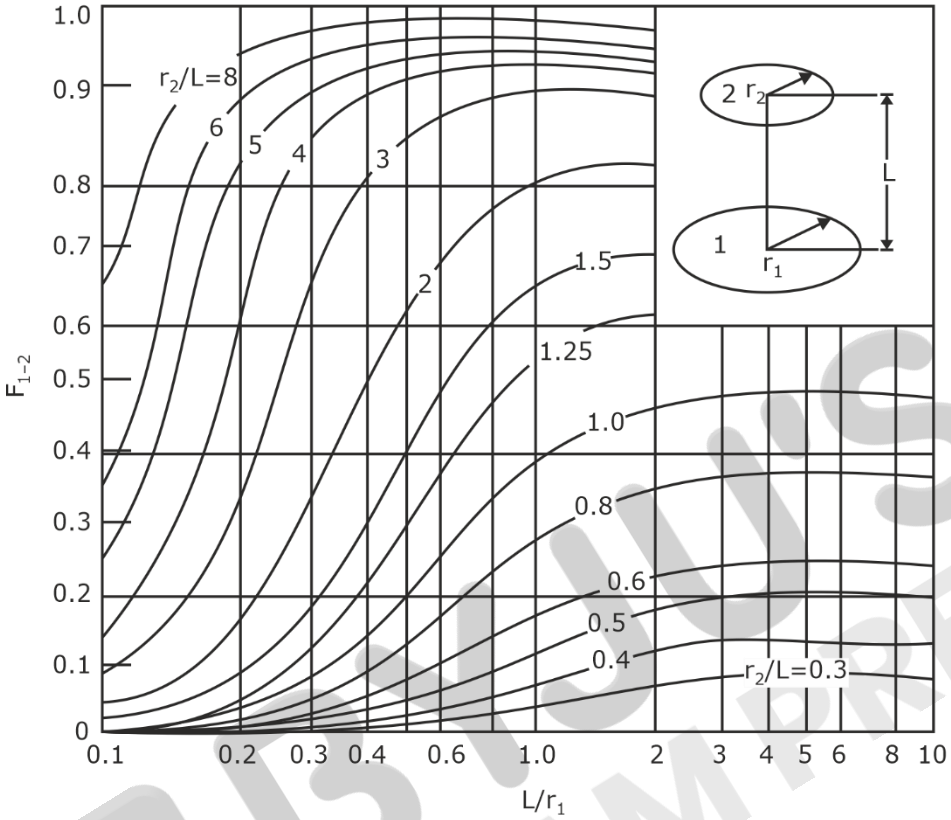
= 9005.5 – 14762.54 kJ

Q = – 5757.04 kJ

Negative sign means heat loss.

4.

(a) A truncated cone has top and bottom diameters of 10 cm and 20 cm respectively, and a height of 10 cm. Calculate the shape factor between the top surface and the side, and also the shape factor between the side and itself. Use the figure showing the radiation shape factor for radiation between two parallel coaxial disks:



[20 Marks]

Sol.

The geometry has three surfaces,

Bottom surface (1), Top surface (2) and Side surface (3)

The shape factor for each surface is given by,

For surface 1,

$$F_{1-1} + F_{1-2} + F_{1-3} = 1$$

Since bottom surface is flat, $F_{1-1} = 0$

$$F_{1-2} + F_{1-3} = 1$$

For surface 2,

$$F_{2-1} + F_{2-2} + F_{2-3} = 1$$

Since top surface is flat, $F_{2-2} = 0$

$$F_{2-1} + F_{2-3} = 1$$

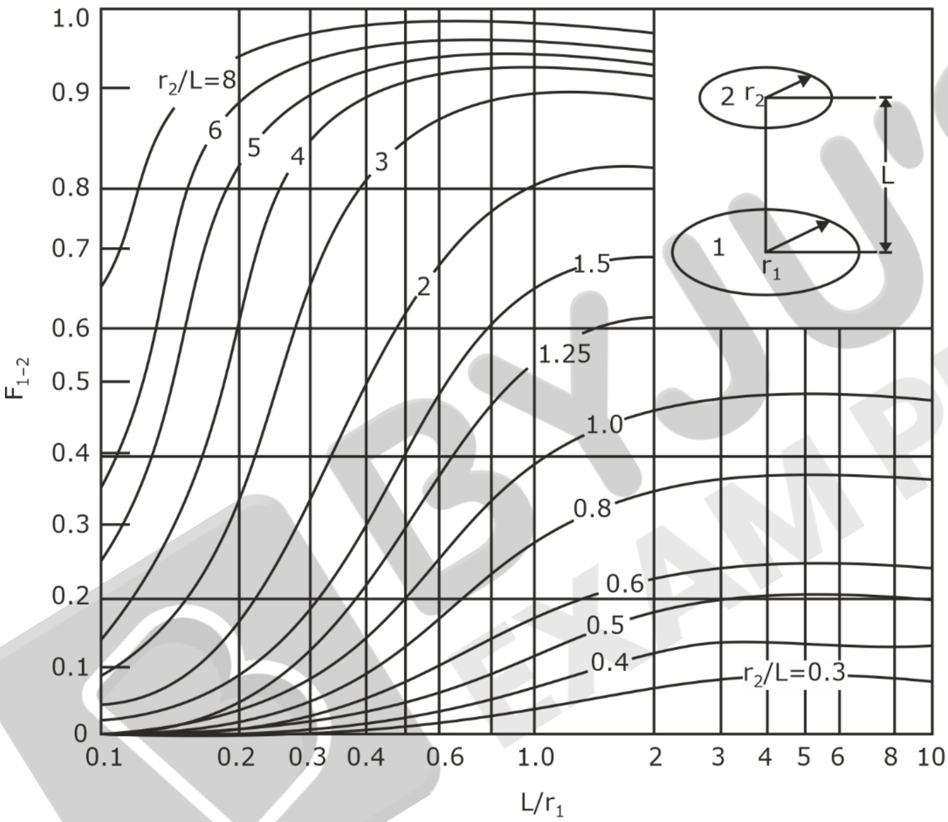
Also, $A_1F_{1-2} = A_2F_{2-1}$ this factor is given from the graph,

The radius of bottom surface $r_1 = 10$ cm, radius of top surface $r_2 = 5$ cm, and height L is 10 cm.

$$\frac{L}{r_1} = \frac{10}{10} = 1$$

$$\frac{r_2}{L} = \frac{5}{10} = 0.5$$

Based on these values, the shape factor F_{1-2} is found out as shown in figure



$F_{1-2} = 0.1$

$A_1F_{1-2} = A_2F_{2-1}$

$$F_{2-1} = \frac{A_1F_{1-2}}{A_2} = \frac{\pi r_1^2 \times F_{1-2}}{\pi r_2^2} = \frac{\pi \times 10^2 \times 0.1}{\pi \times 5^2} = 0.4$$

Therefore, the shape factor for top surface is given by,

$F_{2-1} + F_{2-3} = 1$

$F_{2-3} = 1 - F_{2-1} = 1 - 0.4 = 0.6$

Therefore, the shape factor for top surface to side surface is 0.6.

The shape factor for bottom surface is given by

$F_{1-2} + F_{1-3} = 1$

$F_{1-3} = 1 - F_{1-2} = 1 - 0.1 = 0.9$

Also

$A_1F_{1-3} = A_3F_{3-1}$

$$F_{3-1} = \frac{A_1F_{1-3}}{A_3}$$

Area of curved surface area is given by,

$$A_3 = \pi(r_1 + r_2)\left[(r_1 - r_2)^2 + L^2\right]^{0.5}$$

$$A_3 = \pi(10 + 5)\left[(10 - 5)^2 + 10^2\right]^{0.5} = 526.86\ \text{cm}^2$$

$$F_{3-1} = \frac{A_1F_{1-3}}{A_3} = \frac{\pi r_1^2 F_{1-3}}{A_3} = \frac{\pi \times 10^2 \times 0.9}{526.86} = 0.54$$

Also

$A_2F_{2-3} = A_3F_{3-2}$

$$F_{3-2} = \frac{A_2F_{2-3}}{A_3}$$

$$F_{3-2} = \frac{A_2F_{2-3}}{A_3} = \frac{\pi r_2^2 F_{2-3}}{A_3} = \frac{\pi \times 5^2 \times 0.6}{526.86} = 0.09$$

For surface 1, side surface

$F_{3-1} + F_{3-2} + F_{3-3} = 1$

$F_{3-3} = 1 - F_{3-1} - F_{3-2} = 1 - 0.54 - 0.09 = 0.37$

Therefore, the shape factor for side surface to itself is 0.37.

(b) A Francis turbine supplied through an 80 m diameter penstock has the following particulars :
Output power = 65000 kW
Speed = 150 rpm.
Hydraulic efficiency = 90%
Flow rate = 120 m^3/s
Mean diameter of turbine at entry = 5 m
Mean blade height at entry = 1.5 m
Entry diameter of draft tube 4.5 m
Velocity in tailrace = 2.5 m/s

The static pressure head in the penstock measured just before entry to the runner is 60 m. The point of measurement is 3.2 m above the level of the tailrace. The loss in the draft tube is equivalent to 30% of the velocity head at entry to it. The exit plane of the runner is 2 m above the tailrace and the flow leaves the runner without swirl. Calculate:

(i) The overall efficiency

(ii) The direction of flow relative to the runner at inlet

(iii) The pressure head at entry to draft tube

[20 Marks]

Sol. Given,

Output power i.e., SP = 65000 kW

D_P = 8 m

N = 150 rpm $\quad \frac{P_1}{\rho g} = 60$ m

Q = 120 m^3/s $\quad Z_1$ = 3.2 m

D_1 = 5 m $\quad h_{LD} = 30\% \times \frac{V_i^2}{2g}$

B_1 = 1.5 m $\quad$ Z = 2 m

D_i = 4.5 m $\quad \alpha_2 = 90^0$ (V_{w2} = 0)

V_0 = 2.5 m/s

i. Overall efficiency $\quad \eta_0 = \frac{SP}{HP} = \frac{P}{\rho Q g h}$ …. (i)

Gross head, $\quad H_g = \frac{P_1}{\rho g} + z_1 + \frac{V_1^2}{2g}$

Velocity of flow in penstock, $V_1 = \frac{4 \times 120}{\pi \times 8^2} = 2.387$ m/s

$$H_g = 60 + 3.2 + \frac{2.387^2}{2 \times 9.81} = 63.49 \text{ m}$$

Net head at turbine inlet, $H = H_g - \frac{V^2}{2g}$

$$H = 63.49 - \frac{(2.5)^2}{2 \times 9.81} = 63.17 \text{ m}$$

Put in equation (1)

$$\eta_0 = \frac{65000 \times 10^3}{10^3 \times 120 \times 9.81 \times 63.17}$$

$$\eta_0 = 0.874 = 87.4\%$$

ii. Direction of flow relative to runner at inlet

As $\quad \eta_h = \frac{RP}{HP} = \frac{m(V_{w_1}u_1 - V_{w_2}u_2)}{mgH}$

$$\eta_h = \frac{V_{w_1}u_1}{gH} = 0.9$$

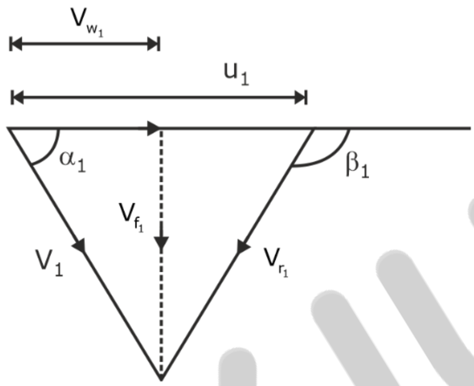
Where, $u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 5 \times 150}{60} = 39.27$ m/s

$$0.9 = \frac{V_{w_1} \times 39.27}{9.81 \times 63.17}$$

$$V_{w1} = 14.2 \text{ m/s}$$

As, $u_1 > v_{w_1} \Rightarrow \beta_1 > 90^\circ$

Inlet velocity triangle



$$Q = \pi D_1 B_1 \times V_{f1}$$

$$V_{f_1} = \frac{Q}{\pi D_1 B_1} = \frac{120}{\pi \times 5 \times 1.5} = 5.093 \text{ m/s}$$

$$\tan(180^\circ - \beta_1) = \frac{V_{f_1}}{u_1 - V_{w_1}} = \frac{5.093}{(39.27 - 14.2)}$$

$$180 - \beta_1 = \tan^{-1}\left(\frac{5.093}{39.27 - 14.2}\right)$$

$$180 - \beta_1 = 11.483^\circ$$

$$\beta_1 = 168.52^\circ$$

iii. Pressure head at entry to draft tube $\left(\frac{P_i}{\rho g}\right)$

$i \rightarrow$ Inlet to draft tube

$0 \rightarrow$ outlet to draft tube

Apply energy equation between inlet and outlet of draft tube,

$$\frac{P_i}{\rho g} + z_i + \frac{V_i^2}{2g} = \frac{P_0}{\rho g} + z_0 + \frac{V_0^2}{2g} + h_{LD}$$

Where, $V_i = \frac{4Q}{\pi D_i^2} = \frac{4 \times 120}{\pi (4.5)^2} = 7.545$ m/s

$$\frac{P_i}{\rho g} = \frac{V_0^2}{2g} + h_{LD} - \frac{V_i^2}{2g} - z_i$$

$$\frac{P_i}{\rho g} = \frac{V_0^2}{2g} + 0.3 \times \frac{V_i^2}{2g} - \frac{V_i^2}{2g} - z_i$$

$$\frac{P_i}{\rho g} = \frac{V_0^2}{2g} - 0.7 \times \frac{V_i^2}{2g} - z_i$$

$$\frac{P_i}{\rho g} = \frac{(2.5)^2}{2 \times 9.81} - \frac{0.7 \times (7.545)^2}{2 \times 9.81} - 2$$

$$\frac{P_i}{\rho g} = 0.318 - 2.031 - 2 = -3.713 \text{ m}$$

Pressure head at inlet of draft tube is -3.173 m.

(c) Two contains are connected with a pipe having a closed valve. One container contains a 5 kg mixture of 62.5% CO_2 and 37.5% O_2 on a mole basis at 30 °C and 125 kPa. The second container contains 10 kg of N_2 at 15 °C and 200 kPa. The valve in the pipe is opened and gases are allowed to mix. During the mixing process, 100 kJ of heat energy is supplied to the combined tank. Determine the volume of the mixture and write an energy balance equation. [Required property tables are attached]

[20 Marks]

Molar mass, gas constant, and critical-point properties

Substance	Formula	Molar mass, M kg/kmol	Gas constant, RkJ/kg-K*	Critical-point properties		
				Temperature, K	Pressure, MPa	Volume, m^3/kmol
Air	–	28.97	0.2870	132.5	3.77	0.0883
Ammonia	NH_3	17.03	0.4882	405.5	11.28	0.0724
Argon	Ar	39.948	0.2081	151	4.86	0.0749
Benzene	C_6H_6	78.115	0.1064	562	4.92	0.2603
Bromine	Br_2	159.808	0.0520	584	10.34	0.1355
n-Butane	C_4H_{10}	58.124	0.1430	425.2	3.80	0.2547
Carbon dioxide	CO_2	44.01	0.1889	304.2	7.39	0.0943
Carbon monoxide	CO	28.011	0.2968	133	3.50	0.0930
Carbon tetrachloride	CCl_4	153.82	0.05405	556.4	4.56	0.2759
Chlorine	Cl_2	70.906	0.1173	417	7.71	0.1242
Chloroform	$CHCl_3$	119.38	0.06964	536.6	5.47	0.2403
Dichlorodifluoromethane (R-12)	CCl_2F_2	120.91	0.06876	384.7	4.01	0.2179
Dichlorofluoromethane (R-21)	$CHCl_2F$	102.92	0.08078	451.7	5.17	0.1973
Ethane	C_2H_6	30.070	0.2765	305.5	4.48	0.1480
Ethyl alcohol	C_2H_3OH	46.07	0.1805	516	6.38	0.1673
Ethylene	C_2H_4	28.054	0.2964	282.4	5.12	0.1242
Helium	He	4.003	2.0769	5.3	0.23	0.0578
n-Hexane		86.179	0.09647	507.9	3.03	0.3677
Hydrogen (normal)	C_6H_{14}	2.016	4.1240	33.3	1.30	0.0649
Krypton	H_2	83.80	0.09921	209.4	5.50	0.0924
Methane	Kr	16.043	0.5182	191.1	4.64	0.0993
Methyl alcohol	CH_4	32.042	0.2595	513.2	7.95	0.1180
Methyl chloride	CH_3OH	50.488	0.1647	416.3	6.68	0.1430
Neon	CH_3Cl	20.183	0.4119	44.5	2.73	0.0417
Nitrogen	Ne	28.013	0.2968	126.2	3.39	0.0899
Nitrous oxide	N_2	44.013	0.1889	309.7	7.27	0.0961
Oxygen	N_2O	31.999	0.2598	154.8	5.08	0.0780
Propane	O_2	44.097	0.1885	370	4.26	0.1998
Propylene	C_3H_8	42.081	0.1976	365	4.62	0.1810
Sulfur dioxide	SO_2	64.063	0.1298	430.7	7.88	0.1217
Tetrafluoroethane (R-134a)	CF_3CH_2F	102.03	0.08149	374.2	4.059	0.1993
Trichlorofluoromethane (R-11)	CCl_3F	137.37	0.06052	471.2	4.38	0.2478
Water	H_2O	18.015	0.4615	647.1	22.06	0.0560
Xenon	Xe	131.30	0.06332	289.8	5.88	0.1186

*The unit kJ/kg·K is equivalent to kPa-m^3/kg-K. The gas constant is calculated from $R=R_u/M$, where R_u=8.31447 kJ/kmol-K and M is the molar mass.

Ideal-gas specific heat of varous common gases

At 300 K

Gas	Formula	Gas constant, R kJ/kg-K	c_p kJ/kg-K	c_v kJ/kg-K	k
Air	–	0.2870	1.005	0.718	1.400
Argon	Ar	0.2081	0.5203	0.3122	1.667
Butane	C_4H_{10}	0.1433	1.7164	1.5734	1.091
Carbon dioxide	CO_2	0.1889	0.846	0.657	1.289
Carbon monoxide	CO	0.2968	1.040	0.744	1.400
Ethane	C_2H_6	0.2765	1.7662	1.4897	1.186
Ethylene	C_2H_4	0.2964	1.5482	1.2518	1.237
Helium	He	2.0769	5.1926	3.1156	1.667
Hydrogen	H_2	4.1240	14.307	10.183	1.405
Methane	CH_4	0.5182	2.2537	1.7354	1.299
Neon	Ne	0.4119	1.0299	0.6179	1.667
Nitrogen	N_2	0.2968	1.039	0.743	1.400
Octane	C_8H_{18}	0.0729	1.7113	1.6385	1.044
Oxygen	O_2	0.2598	0.918	0.658	1.395
Propane	C_3H_8	0.1885	1.6794	1.4909	1.126
Steamt	H_2O	0.4615	1.8723	1.4108	1.327

Note: The unit kJ/kg.K is equivalent to kJ/kg.°C.

Sol.

In contains 1: CO_2 + O_2 Mixture

m_{mix} = 5 kg

P_1 = 125 kPa

T_1 = 30°C

For 1 mole of mixture = 0.625 moles of CO_2 + 0.375 moles of O_2

Molar mass of mixture

$$M_1 = x_{CO_2} \times M_{CO_2} + x_{O_2} \times M_{O_2}$$

$$= (0.625 \times 44) + (0.375 \times 32)$$

$$M_1 = 39.5\ \frac{kg}{kmol}$$

Gas constant of mixture (R_1)

$$R_1 = \frac{\bar{R}}{M_{mix}} = \frac{8.314\ kJ/k\,mol\text{-}K}{39.5\ kg/k\,mol}$$

$$R_1 = 0.2105\ \frac{kJ}{kg\text{-}K}$$

Volume of mixture of tank 1

$$V_1 = \frac{m_{mix}R_1T_1}{P_1} = \frac{5 \times 0.2105 \times (273+30)}{125}\ m^3$$

V_1 = 2.551 m^3

In Tank 2: N_2 at 15°C, 200 kPa, 10 kg

$$V_2 = \frac{m_2R_2T_2}{P_2} = \frac{10 \times 0.2968 \times (273+15)}{200}$$

V_2 = 4.274 m^3

Total volume after mixing

V_{total} = V_1 + V_2 = (2.551 + 4.274) m^3

V_{total} = 6.825 m^3

Mass fraction of CO_2

$$(m_f)_{CO_2} = \frac{m_{CO_2}}{m_{mix}} = \frac{n_{CO_2} \times M_{CO_2}}{n_{mix} \times M_{mix}} = \frac{0.625 \times 44}{39.5} = 0.6962$$

$$m_{CO_2} = 0.6962 \times m_{mix} = 0.6962 \times 5 = 3.481\,\text{kg}$$

$$(m_f)_{O_2} = \frac{m_{O_2}}{5} = \frac{0.375 \times 32}{39.5} = 0.3038$$

$$m_{O_2} = 0.3038 \times m_{mix} = 0.3038 \times 5 = 1.519\,\text{kg}$$

Mass fractions of combined mixture (m_{total} = 15 kg)

$$(m_f)_{CO_2} = \frac{m_{CO_2}}{m_{total}} = \frac{3.481}{15} = 0.232$$

$$(m_f)_{O_2} = \frac{m_{O_2}}{m_{total}} = \frac{1.519}{15} = 0.1012$$

$$(m_f)_{N_2} = \frac{m_{total} - \left(m_{CO_2} + m_{O_2}\right)}{m_{total}} = \frac{10}{15} = 0.6667$$

$$R_{mix} = m_{f_{CO_2}} \times R_{CO_2} + m_{f_{O_2}} \times R_{O_2} + m_{f_{N_2}} \times R_{N_2}$$

R_{mix} = (0.232 × 0.1889) + (0.1012 × 0.2598) + (0.6667 × 0.2969)

R_{mix} = 0.2680 kJ/kg K

Energy balance equation ($T_f \rightarrow$ Final temperature)

$$m_{CO_2} \times C_{V_{CO_2}}(T_f - T_1) + m_{O_2} C_{V_{O_2}}(T_f - T_1) + m_{N_2} C_{V_{N_2}}(T_f - T_2) + 100 = \text{Final Energy } (E_{final})$$

SECTION - 'B'

5.

(a) A six-cylinder SI engine operates on a four-stroke cycle. The bore of each cylinder is 75 mm and the stroke is 100 mm. The clearance volume per cylinder is 60 cc. At a speed of 4000 r.p.m., the fuel consumption is 18 kg/h and the torque developed is 140 N-m. Calculate the-

(i) brake thermal efficiency

(ii) relative efficiency on the basis of brake power.

The calorific value of the fuel can be taken as 45000 kJ/kg.

[12 Marks]

Sol. Given,

No. of cylinders (K) = 6

4-stroke cycle

D = 75 mm

L = 100 mm

V_C = 60 cm^3

N = 4000 rpm

Fuel consumption (m_f) = 8 kg/hr

T = 140 N-m

CV = 45000 kJ/kg

Brake power, BP = T × ω= $T \times \frac{2\pi N}{60} = \frac{140 \times 2\pi \times 4000}{60}$

BP = 58643.06 W = 58.64 kW

i. Brake thermal efficiency (η_{bth})

$$\eta_{bth} = \frac{BP}{m_f \times CV}$$

$$\eta_{bth} = \frac{58.64}{\frac{18}{3600} \times 45000} = 0.2606 \text{ or } 26.06\%$$

ii. Relative efficiency,

$$\eta_r = \frac{\eta_{bth}}{h_{airstd}} \qquad ...(i)$$

Where, $\eta_{air\,std} = 1 - \frac{1}{r^{\gamma-1}}$

And $r = \frac{V_1}{V_2} = \frac{V_C + V_S}{V_C} = 1 + \frac{V_S}{V_C}$

$$r = 1 + \frac{\frac{\pi}{4} D^2 \times L}{V_C} = \left\{1 + \frac{\frac{\pi}{4} \times (0.075)^2 \times 0.1}{60 \times 10^{-6}}\right\}$$

r = 1 + 7.363 = 8.363

$$\eta_{air\,std} = 1 - \frac{1}{(8.363)^{1.4-1}} = 0.5724$$

Put in equation (i)

$$\eta_r = \frac{\eta_{bth}}{\eta_{air\,std}} = \frac{0.2606}{0.5724}$$

$$\eta_r = 0.4553 = 45.53\%$$

(b) Draw the T-s and h-s diagrams for steam jet refrigeration system and write the expressions for the following:

(i) Nozzle efficiency

(ii) Entrainment efficiency

(iii) Compression efficiency

[12 Marks]

Sol.

Steam Jet Refrigeration System:

The working of the steam jet refrigeration system is represented on T-s and h-s chart as shown in figure below.

P_D = Pressure of steam supplied from boiler

P_C = Pressure in the condenser

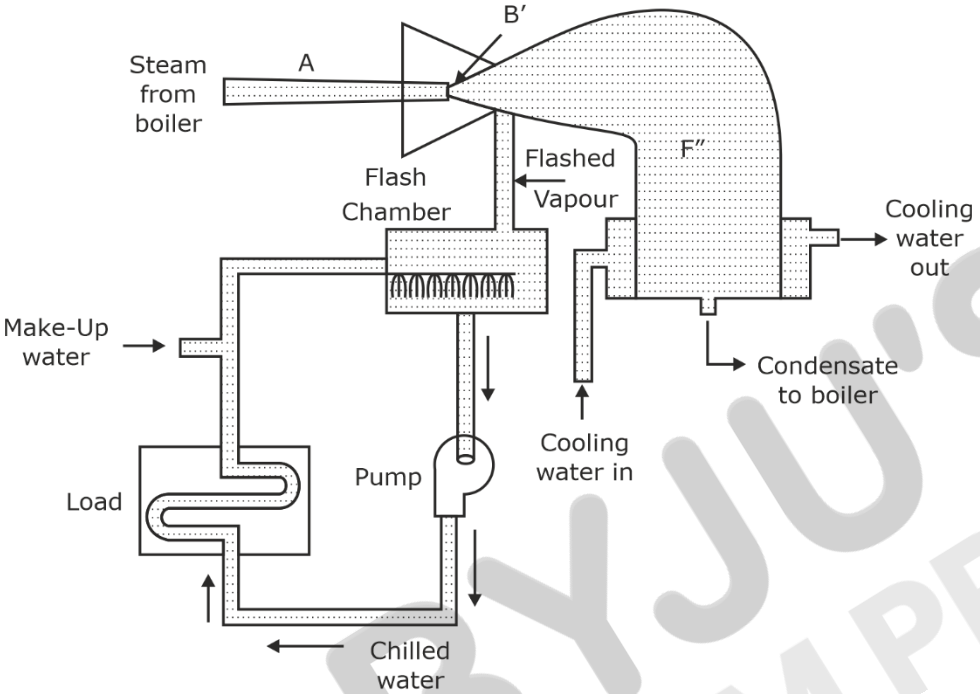
P_e = Pressure in the evaporator in the flash chamber

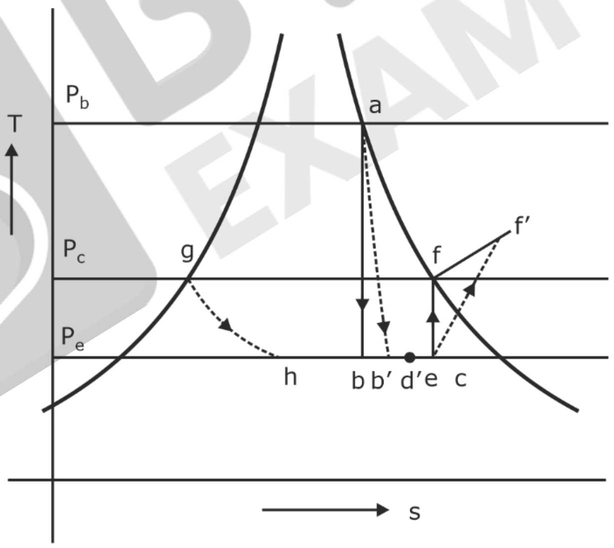
ab = Isentropic expansion of steam through nozzle

ab' = Actual expansion of steam through nozzle

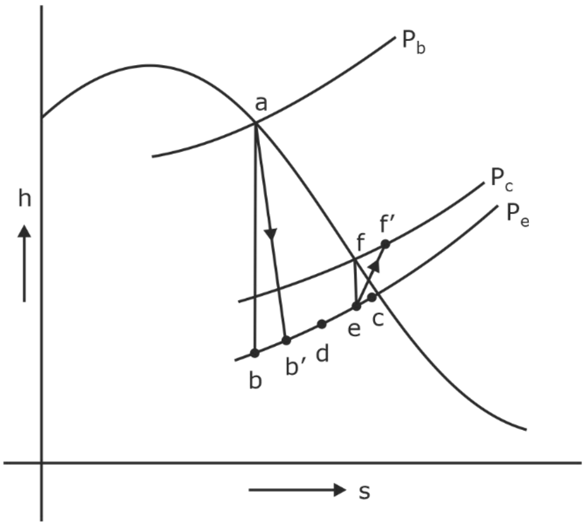
a = Condition of steam supplied

c = Condition of water vapour formed in the flash chamber





T-s diagram for steam jet refrigeration



h-s diagram for steam jet refrigeration

b' = Condition of steam coming out of nozzle

d = Condition of steam just before mixing with water vapour

c = Condition of water vapour formed in the evaporator chamber

e = Condition of mixture of steam at d and water vapour at c after mixing and just before starting the compression in the booster ejector

f' = Condition of the mixture entering into the condenser

ef = Isentropic compression in booster ejector

ef' = Actual compression in booster ejector

The actual expansion through nozzle does not follow isentropic process, so that actual drop is taken into account by nozzle efficiency, and it is given by

$$\eta_n \left(\text{Nozzle efficiency}\right) = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}$$

$$\eta_n = \frac{h_a - h_b'}{h_a - h_b}$$

The water vapour formed in the flash-chamber has negligible velocity compared with the velocity of the steam coming out of nozzle which is equivalent to $\sqrt{2gJ\left(h_a - h_b'\right)}$ m/sec.

The quantity $\left(h_a - h_b'\right)$ is equivalent to K.E. of motive steam, now available for entrainment of the vapour in the flash-chamber. (The process of giving the momentum of the water vapour formed in the flash-chamber by high velocity steam is known as entrainment of vapour). During the entrainment, steam will lose some energy. The process of entrainment is very inefficient and part of original motive force available for compression is reduced and it is taken into account by a factor known as entrainment efficiency and it is given by

$$\eta_e = \frac{\left(h_a - h_d\right)}{\left(h_a - h_b'\right)}$$

The actual compression of the mixture does not follow the isentropic compression, so it is taken into account by a factor known as compression efficiency and it is given by

$$\eta_c \text{ (compression efficiency)} = \frac{h_a - h_e}{h_f' - h_e}$$

(c) Briefly describe a natural draught cooling tower. Explain why it is hyperbolic in shape.

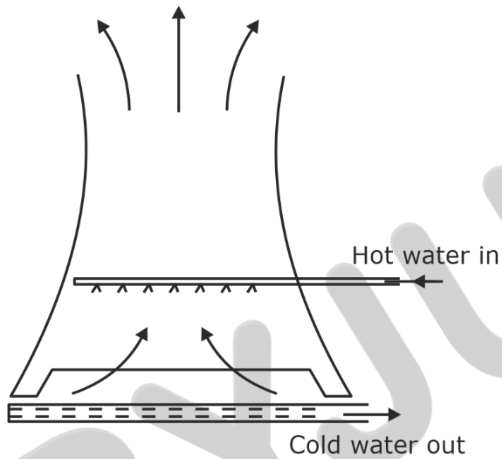
[12 Marks]

Sol.

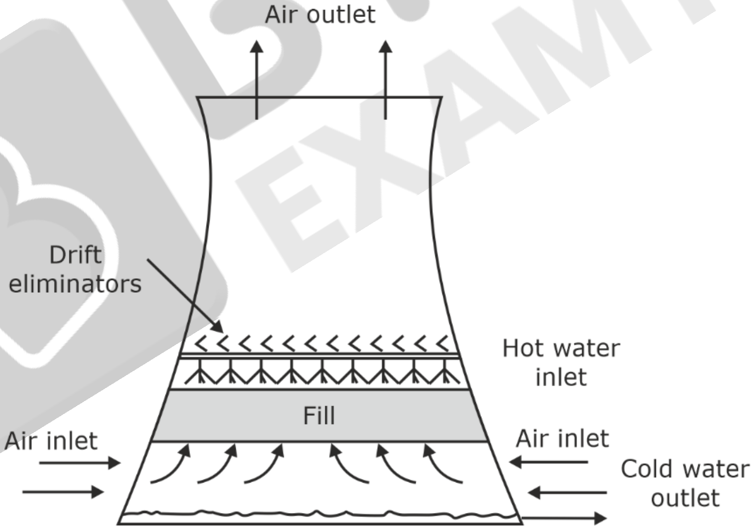
In natural draught cooling towers, the flow of air occurs due to the natural pressure head caused by the difference in density between the cold outside air and the hot humid air inside. Thus, the pressure head developed is

$$\Delta P_d = (\rho_o - \rho_i)\, gH$$

where H = height of the tower above the fill, ρ_o = density of outside air, and ρ_i = density of inside air.



(a) Natural draught coding tower



(b) Counterflow hyperbolic natural draught cooling tower

Because of relatively small density difference, $\rho_o - \rho_i$, H must be large so as to result in the desired ΔP_d, which must balance the air pressure losses in the tower. Natural draught cooling towers are, therefore, very tall. The tower body, above the water distribution system and the fill, is an empty shell of circular cross-section, but with a hyperbolic vertical profile. The hyperbolic profile offers superior strength and the greatest resistance to outside wind loading compared to other forms. Natural draught cooling towers are, therefore, often termed as hyperbolic towers. Made of reinforced concrete, they are an imposing sight and are conspicuous from a distance.

Some additional advantages of natural draught cooling tower are:

(i) Power cost and auxiliary equipment are eliminated as fans are not needed. Resulting in operating and maintenance costs are consequently reduced.

(ii) Chimney shape creates its own draft, ensuring efficient operation even when the wind is absent.

(iii) Cooling capacity of this type of tower is quite comparable with that of multicell installation of induced draft-type towers.

(iv) Local fogging and warm air circulation, which is there in mechanical draft installations, are also avoided.

(d) Distinguish among the following:

(i) Renewable energy

(ii) Green energy

(iii) Clean energy

Also, mention the relative environmental effects of the above.

[12 Marks]

Sol. These three terms — green, renewable, and clean energy are often used interchangeably in eco-friendly content, but they don't always mean the same thing. While some overlap exists, the subtle differences and nuances can impact funding, allocation, and the creation of government credits during and after the production of these sustainable energy sources. Thus, it’s important to understand that the true definitions of renewable, clean, and green energy depend upon how they’re created, how they’re refreshed, and their overall environmental impact.

Creation

- Renewable energy comes from sources that occur naturally and can be replaced naturally and completely within the span of an average human life.
- Green energy comes from natural sources that meet current energy needs without compromising future generations. It is a subset of renewable energy representing resources with the smallest environmental footprint.
- Clean energy releases zero or minuscule amounts of carbon dioxide and chemical contaminants during production. Although not necessarily renewable by definition, clean energy doesn't create large amounts of greenhouse gases or air pollutants.

Renewal

- Renewable energy sources never run out and are naturally replenished. However, renewable energy is flow-limited, meaning there is a limit to what can be captured over time (i.e. you can’t make more wind than what already exists).
- Green energy comes from renewable energy resources that can be renewed naturally and have the least environmental impact.
- Clean energy is created without emitting greenhouse gases, though it isn’t necessarily naturally renewable.

Impact

- Renewable energy can have an ecological impact, depending upon the process used to create that electricity.
- Green energy is considered the most environmentally friendly resource available to us today, with little to no ongoing environmental impact.

- Clean energy is power generation without creating adverse environmental impacts like carbon dioxide or greenhouse gases. Most clean energy sources are also renewable, including hydro energy, solar power, and wind power.

(e) Describe the emission norms for Indian vehicles if they have to comply with Bharat Stage (BS) Emission Standards-VI. Mention the devices and technology introduced to meet the BS-VI norms.

[12 Marks]

Sol. To regulate the pollution emitted by cars and two-wheelers, the government has put forth regulations known as Bharat stage emission standard (BSES). The central government has mandated that all vehicle manufacturers, both two-wheelers and four-wheelers, should manufacture, sell and register only BS6 (BS-VI).

Emission norms for BS6

- BS6 emission norms allow a motorcycle to emit not more than 60 mg/km of NO_x, (Nitrogen oxides).
- The particulate matter (PM) for petrol vehicles has been restricted to 4.5 mg/km.
- The limit of NO_x for diesel engines is 80 mg/km. The HC + NO_x limit has been set to 170 mg/km and PM level limit has been set to 4.5 mg/km.
- The BS6 fuel has less sulphur and NO_x. The content of sulphur in BS6 fuel is 10 ppm.

Devices and technology introduced to meet BS-VI norms.

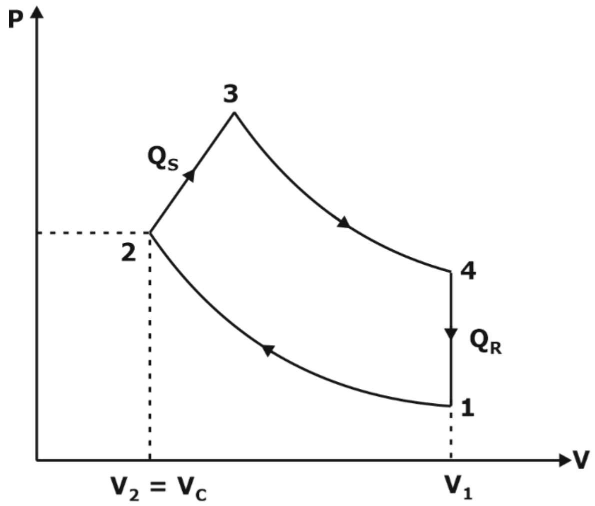
- Selective catalytic reduction technology: It reduces oxides of nitrogen by injecting an aqueous urea solution into the system. Hence, NOx from diesel cars can be brought down by nearly 70%. In the petrol cars, they can be reduced by 25%.
- Mandatory on-board diagnostics: Which inform the vehicle owner or the repair technician about how efficient the systems in the vehicles are.
- RDE (Real driving emission): It is introduced for the first time that will measure the emission in real-world conditions and not just under test conditions.

6.

(a) A gasoline engine has a stroke volume of 0.002 m^3 and a compression ratio of 6. At the end of the compression stroke, the pressure is 10 bar and the temperature is 400 °C. Ignition is set so that the pressure rises along a straight line during combustion and attains its highest value of 30 bar after the piston has travelled (1/40) of the stroke. The charge consists of a gasoline-air mixture in proportion of 1:18 by mass. Calculate the heat lost per kg of charge during combustion. Take R = 287 J/kg-K, calorific value of the fuel = 45 MJ/kg, C_p = 1 kJ/kg.

[20 Marks]

Sol.



Given,

$V_s = 0.002\ m^3$

$$r = 6 = \frac{V_1}{V_2} = 1 + \frac{V_s}{V_C}$$

$P_2 = 10$ bar

$T_2 = 400°C = 673$ K

$P_3 = 30$ bar

$$V_C = \frac{V_S}{r-1} = \frac{0.002}{6-1}$$

$\Rightarrow V_C = 4 \times 10^{-4}\ m^3 = V_2$

$$V_3 = V_2 + \frac{V_S}{40} = 4\times10^{-4} + \frac{0.002}{40}$$

$\Rightarrow V_3 = 4.5 \times 10^{-4}\ m^3$

$$\text{As } \frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} \Rightarrow \frac{10\times4\times10^{-4}}{673} = \frac{30\times4.5\times10^{-4}}{T_3}$$

$\Rightarrow T_3 = 2271.375$ K

W_{23} = Area under P-V curve on volume axis

$$= \frac{1}{2}(P_2 + P_3)\times(V_3 - V_2)$$

$$= \frac{1}{2}(10+30)\times10^5\times(4.5-4)\times10^{-4}$$

$W_{23} = 100$ J

Mass of mixture at the end of compression point 2

$$m_2 = \frac{P_2V_2}{RT_2} = \frac{10\times10^5\times4\times10^{-4}}{287\times673}$$

$\Rightarrow m_2 = 2.0709 \times 10^{-3}$ kg

$$(\Delta U)_{23} = m_2C_V(T_3 - T_2)$$

$C_V = C_P - R$

C_V = 0.713 kJ/kg K

$(\Delta U)_{23} = 2.0709 \times 10^{-3} \times 0.713 \times (2271.4 - 673)$

$(\Delta U)_{23}$ = 2.3604 kJ

1st law:

$Q_{23} = W_{23} + (\Delta U)_{23}$

$\Rightarrow Q_{23} = Q_S = (0.1 + 2.3604)$ kJ

$Q_{23} = Q_s$ = 2.4604 kJ (Head supplied)

To find heat lost

$$\frac{m_a}{m_f} = 18 \;\Rightarrow\; \frac{m_a}{m_f} + 1 = 18 + 1 \;\Rightarrow\; \frac{m_a + m_f}{m_f} = 19$$

$$m_f = \frac{1}{19} \times m_{mix} = \frac{1}{19} \times 2.0709 \times 10^{-3} \text{ kg}$$

$m_f = 1.09 \times 10^{-4}$ kg

Heat loss = $m_f \times CV = 1.09 \times 10^{-4} \times 45000$ kJ/kg

Heat loss = 4.905 kJ

Net heat loss = (4.905 – 2.4604) kJ

Heat lost/kg charge = $\dfrac{2.4446}{2.0709 \times 10^{-3}} = 1180.45$ kJ/kg charge

(b) A room is designed for air conditioning as per the following data:

Room sensible heat gain 30 kW Room latent heat gain = 10 kW

Inside design conditions are: 25 °C DBT and 50% RH

Outside conditions are: 40 °C DBT and 27 °C WBT

Bypass factor of the cooling coil = 0.10

The return air from the space is mixed with the outside air before entering the cooling coil in the ratio of 4:1 by weight. Determine the following:

(i) Apparatus dew point

(ii) Condition of air leaving the cooling

(iii) Quantity of dehumidified air coil

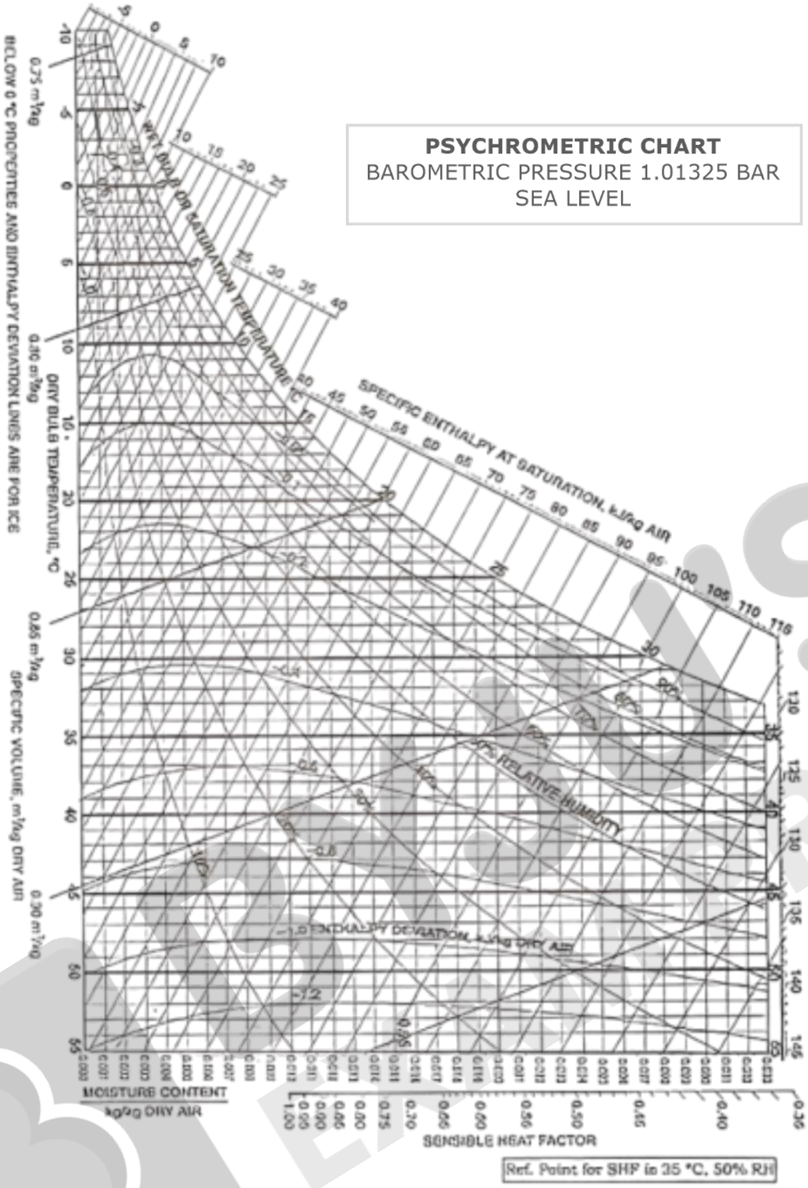
(iv) Mass of ventilation air

(v) Volume flow rate of fresh air

(vi) Total refrigeration load

[Psychrometric chart is attached]

[20 Marks]



Sol.

Given,

RSH = 30 kW

RLH = 10 kW

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{30}{30 + 10} = 0.75$$

Inside design conditions (DBT = 25°C; R_H = 50%)

From psychrometric chart,

ω_i = 0.01 kg vap/kg dry air

h_i = 50 kJ/kg-da

v_i = 0.86 m^3/kg-da

Outside conditions (DBT = 40°C, WBT = 27°C)

From psychrometric chart

ω_0 = 0.0172 kg vap/kg-da

h_0 = 85 kJ/kg-da

v_0 = 0.912 m^3/kg-da

Draw points i and o on psychrometric chart,

$$\frac{\text{Recirculated air}}{\text{Fresh outside air}} = \frac{4}{1}$$

Divide i – o in ratio 4 : 1

Point 1 i.e., $\frac{i-1}{1-o} = \frac{4}{1}$

$$t_1 = \frac{4t_i + 1t_o}{5} = \frac{(4\times 25)+(1\times 40)}{5} = 28°C \text{ DBT}$$

$$\omega_1 = \frac{4\omega_i + 1\omega_o}{5} = \frac{(4\times 0.01)+(1\times 0.0172)}{5} = 0.01144 \text{ kg vap / kg – da}$$

$$h_1 = \frac{4h_i + 1h_o}{5} = \frac{(4\times 50)+(1\times 85)}{5} = 57 \text{ kJ/kg – da}$$

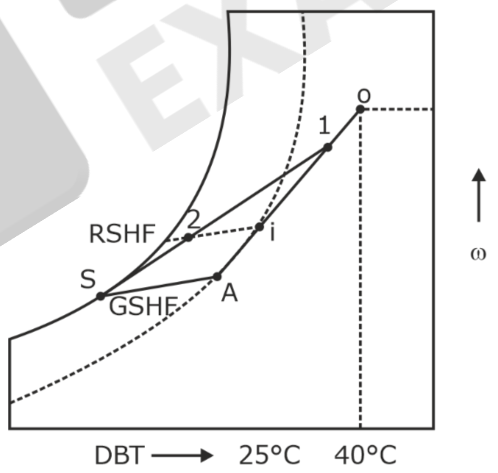
$$v_1 = \frac{4v_i + 1v_0}{5} = \frac{(4\times 0.86)+(1\times 0.91)}{5} = 0.8702 \text{ m}^3\text{/kg – da}$$

Draw RSHF line through point i and 2 and the saturation curve,

Extend line 1 – i to point A such that

$\frac{A-i}{1-A} = 0.1$ (As BPF = 0.1 for cooling coil)

From point A; draw a line parallel to RSHF intersecting saturation curve at point (s)



Now joint point 1 and S i.e., GSHF

Which intersects RSHF line through point i at point 2

t_{ADP} = 9.8°C

ω_{ADP} = 0.0078 kg vap/kg-da

h_{ADP} = 29.5 kJ/kg-da

$$\frac{t_2 - t_{ADP}}{t_1 - t_{ADP}} = 0.1 = \text{BPF}$$

$$\frac{t_2 - 9.8}{28 - 9.8} = 0.1$$

$$t_2 = 11.62°C$$

$$\frac{\omega_2 - \omega_{ADP}}{\omega_1 - \omega_{ADP}} = 0.1 \ (BPF)$$

$$\frac{\omega_2 - 0.0078}{0.01144 - 0.0078} = 0.1$$

$$\omega_2 = 0.00816 \text{ kg vap/kg-da}$$

Finally

i. ADP temperature = 9.8°C

ii. Condition of air leaving the cooling coil

$t_2 = 11.62°C$; $\omega_2 = 0.00816$ kg vap/kg-da

iii. Quantity of Dehumidified air

$$= \frac{RSH}{0.0204(t_i - t_2)} = \frac{30}{0.0204(25 - 11.62)}$$

= 109.9 m^3/min Or = 109.9 cmm

Mass of ventilation air = 0.2 × $m_{supplied\ air}$

Where $m_{supplied\ air} = \frac{cmm}{60 \times v_2} = \frac{109.9}{60 \times 0.819}$

$m_{supplied\ air}$ = 2.23 kg-da/sec = m_{as}

m_{ao} = 0.2 × 2.23 = 0.44 kg-da/sec

iv. Volume flow rate of fresh air (outside air)

= m_{ao} × 60 × v_0

= 0.44 × 60 × 0.912 = 24.47 cmm

v. Total refrigeration load (Grand total heat)

GTH = RSH + RLH + OATH

Where OATH (outside air total heat) = $m_{ao\,air}(h_o - h_i)$

= 0.44(85 – 50) = 15.4 kJ/sec

GTH = 30 + 10 + 15.4 = 55.4 kW

(c) The angles at inlet and discharge of the blading of a 50% reaction turbine are 35° and 20° respectively. The speed of rotation is 1500 rpm and at a particular stage, the mean ring diameter is 0.67 m and the steam condition is at 1.5 bar, 0.96 dry. Determine

(i) the required height of blading to pass 3.6 kg/s of steam;

(ii) the power developed by the ring.

[Saturated steam table is attached at the end of booklet]

[20 Marks]

Sol. Given,

β_1 = 35°, β_2 = 20°, D_m = 0.67 m, N = 150 rpm

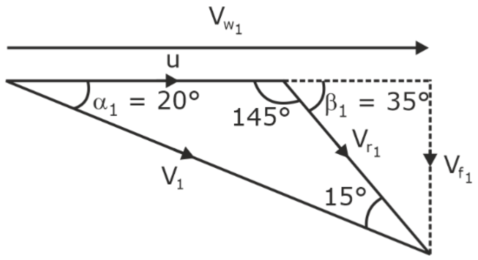
Blade speed, $u = \frac{\pi D_m N}{60}$

$$u = \frac{\pi \times 0.67 \times 1500}{60} = 52.62 \text{ m/s}$$

For degree of reaction, R = 0.5,

$\beta_1 = \alpha_1 = 35°$, $\beta_2 = \alpha_1 = 20°$

$V_2 = V_{r1}$, $V_1 = V_{r2}$



Using sine rule

$$\frac{V_1}{\sin 145°} = \frac{u}{\sin 15°} = \frac{V_{r_1}}{\sin 20°}$$

$$V_1 = \frac{u \times \sin 145°}{\sin 15°} = \frac{52.62 \times \sin 145°}{\sin 15°}$$

$$V_1 = 116.61 \text{ m/s} = V_{r_2}$$

$$V_{r_1} = \frac{u \times \sin 20°}{\sin 15°}$$

$$V_{r_1} = 69.54 \text{ m/s} = V_2$$

Also,

$$V_{w_1} = V_1 \cos \alpha_1 = 116.61 \times \cos 20° = 109.58 \text{ m/s}$$

$$V_{w_2} = V_2 \cos \alpha_2 = 65.54 \times \cos 35° = 56.96 \text{ m/s}$$

As, $\alpha_2 < 90° \Rightarrow V_{w_2} < 0$

$$\Delta V_w = V_{w_1} - \left(-V_{w_2}\right) = \left[109.58 - (-56.96)\right] \text{m/s}$$

$$\Delta V_w = 166.54 \text{ m/s}$$

Specific volume of steam, $v = v_f + x\, v_{fg}$ …. (i)

From steam table

v_f = 0001053 m³/kg; v_g = 1.159 m³/kg (At P = 1.5 bar)

Put in equation (i)

v = 0.001053 + 0.96 (1.159 – 0.001053)

$$v = 1.1127 \text{ m}^3/\text{kg}$$

Also, $\dot{m}_s = \rho Q = \frac{1}{v} \times \left(\pi D_m h\right) \times V_{f_1}$

Or $3.6 = \frac{1}{1.1127} \times \pi \times 0.67 \times h \times \left(V_1 \sin \alpha_1\right)$

Or $3.6 = \frac{\pi \times 0.67 \times h \times 116.61 \times \sin 20°}{1.1127}$

Or $h = 0.04771 \text{ m} = 47.71 \text{ mm}$

Power developed by the ring

$P = \dot{m}(\Delta V_w)u$

$= 3.6 \times 166.54 \times 52.67 = 31577.98$ W $= 31.577$ kW

7.

(a) The following data refer to a boiler unit consisting of an economizer, a boiler and a superheater:

Mass of water evaporated per hour = 5940 kg

Mass of coal burnt per hour = 675 kg

Lower calorific value of coal = 31600 kJ/kg

Pressure of steam at boiler stop valve = 14 bar

Temperature of feedwater entering economizer = 32 °C

Temperature of feedwater leaving economizer = 115 °C

Dryness fraction of steam leaving boiler and entering superheater = 0.96

Temperature of steam leaving superheater = 260 °C

Specific heat of superheater steam = 2.3 kJ/kg-K

Determine the following:

(i) Percentage of heat in coal utilized in economizer, boiler and superheater

(ii) Overall efficiency of the boiler unit

Assume specific heat of water = 4.187 kJ/kg-K

[Saturated steam table is attached at the end of booklet]

[20 Marks]

Sol.

Mass of water evaporated = 5940 kg/hr

Mass of coal burnt = 675 kg/hr

Lower calorific value of cool = 31600 kJ/kg

P_1 = 14 bar, $c_{p,\,water}$ = 4.187 kJ/kg-K, $t_{e_1} = 32°C$

$t_{e_2} = 115°C$, x = 0.96, t_{SH} = 260°C

$c_{p,\ superheated\ steam}$ = 2.3 kJ/kg-K

Heat utilized by 1 kg of feed water,

(a) In Economizer, $h_{f_1} = 1 \times c_{p,water}\left(t_{e_2} - t_{e_1}\right)$

$h_{f1} = 1 \times 4.187\ (115 - 32)$

$h_{f_1} = 347.521$ kJ

(b) In boiler, $h_{boiler} = \left(h_f + xh_{fg}\right) - h_{f_1}$ ..... (i)

At, P = 14 bar (use steam table)

t_s = 195°C; h_f = 830.1 kJ/kg; $h_{fg} = 1957.7$ kJ/kg

Put in equation (i)

$$h_{boiler} = \left[830.1 + (0.96 \times 1957.7)\right] - 347.521$$

$$h_{boiler} = 2361.97 \text{ kJ}$$

(c) In superheater

$$h_{superheater} = (1 - x)h_{fg} + c_{p,SH}\left(T_{SH} - T_s\right)$$

$$= \left[(1-0.96)\times 1957.7\right] + 2.3(260-195)$$

$$h_{superheater} = 227.808 \text{ kJ}$$

Also, per kg of coal burnt; mass of water evaporated = $\frac{5940}{675} = 8.8$ kg/hr

(i) Percentage of heat utilized,

(a) In economizer = $\frac{347.521}{31600 \text{ kJ/kg}} \times 8.8 \times 100 = 9.678\%$

(b) In boiler $= \frac{2361.97 \times 8.8}{31600} \times 100 = 65.77\%$

(c) In superheater = $\frac{227.808 \times 8.8 \times 100}{31600} = 6.344\%$

(ii) Overall efficiency of boiler plant

$$\eta_o = \frac{\text{Total heat absorbed}}{(CV)_{fuel}} \times 100$$

$$= \frac{(347.521 + 2361.97 + 227.808) \times 8.8}{31600} \times 100 = 81.798\%$$

(b) **(i)** Explain the various factors affecting anaerobic digestion process. Why do anaerobic microbes normally grow at a much lower rate than aerobic bacteria?

[10 Marks]

(ii) A family biogas plant is required to be designed to utilize the cow dung of five cows. The hydraulic retention time is 30 days. The temperature of the digester is to be maintained at 30 °C. The dry matter consumption per day is 2 kg. The biogas yield is 0.25 m³/kg. The efficiency of the burner is 60%. The heat of combustion of methane is 26 MJ/m³. The methane proportion is 70%. The density of feedstock material may be taken as 50 kg/m³. Find (1) the volume of biogas digester and (2) its thermal power.

[10 Marks]

Sol.

(i) Anaerobic digestion is the breakdown of organic matter by microbes without oxygen. It comprises four stages: Hydrolysis, acidogenesis, Acetogenesis, and methanogenesis. The syntropic nature of this process makes each stage uniquely important. For example, acidogenesis is a prerequisite for methanogenesis, but if high amounts of acids are formed during the acidogenesis stage, it will affect the methanogens and hinder digestion. Biogas is produced as a by-product, with methane and carbon dioxide as major constituents during the process. Due to the high energy potential of methane, anaerobic digestion of solid wastes is widely studied by researchers all over the world. Various factors affecting anaerobic digestion were identified over the years during the studies. The main factors are C/N ratio, F/M ratio, pH, temperature, Organic loading rate, hydraulic loading rate, and presence of toxins (inherent components or a by-product of the metabolism). Each of these nevertheless depends heavily on the type of substrate and inoculum employed. In field conditions, it is seldom practical to maintain all these factors, a few of these being uncontrollable in nature. Hence, the trade-off between the factors could often be brought within the desirable range. C/N ratio, OLR, and Detention time belong to this group. Mixing of the feedstock is yet another aspect that affects the efficacy of the digestion process. Anaerobes tend to grow slower than obligate aerobes because their energy yield from oxidizing organic molecules is smaller than that of aerobes.

(ii) Given data,

No. of cows = 5

Hydraulic retention time, HRT = 30 days

Temperature of digester = 30°C

Dry matter consumption = 2 kg/day/cow

η_{burner} = 60%

Biogas yield = 0.25 m^2/kg

Heat consumption of methane = 26 MJ/m^3

Methane proportion = 70%

$\rho_{feedstock}$ = 50 kg/m^3

Volume of biogas digester = Daily feed (m^3/day) × Rotational time (days)

Total mass of dry matter per day = 2 kg/day × 5 cows = 10 kg/day

Equal amount of water is added to make the slurry,

Mass of slurry = $(10\ \text{kg/day})_{\text{dry methane}} + (10\ \text{kg/day})_{\text{water}}$

Mass of slurry = 20 kg/day

$$\text{Volume of slurry} = \frac{\text{mass}}{\rho_{\text{feedstock}}} = \frac{20}{50}\frac{m^3}{\text{day}} = 0.4\ m^3/\text{kg}$$

With rotational time of 30 days

Total slurry in digester = 0.4 × 30 = 12 m^3

Volume of biogas digester = 12 m^3

$$\text{Biogas produced} = 0.25\frac{m^3}{\text{kg}} \times 10\ \text{kg/day} = 2.5\ m^3/\text{day}$$

$$\text{Thermal energy available} = 2.5\frac{m^3}{\text{day}} \times 26 \times 0.6 \times 0.7$$

$$= 2.5 \times 26 \times 0.6 \times 0.7\ \text{MJ/day}$$

$$\text{Thermal power} = \frac{2.5 \times 26 \times 0.6 \times 0.7}{24 \times 3600}\ \text{MJ/s or MW}$$

$$= 3.159 \times 10^{-4}\ \text{MJ/s}$$

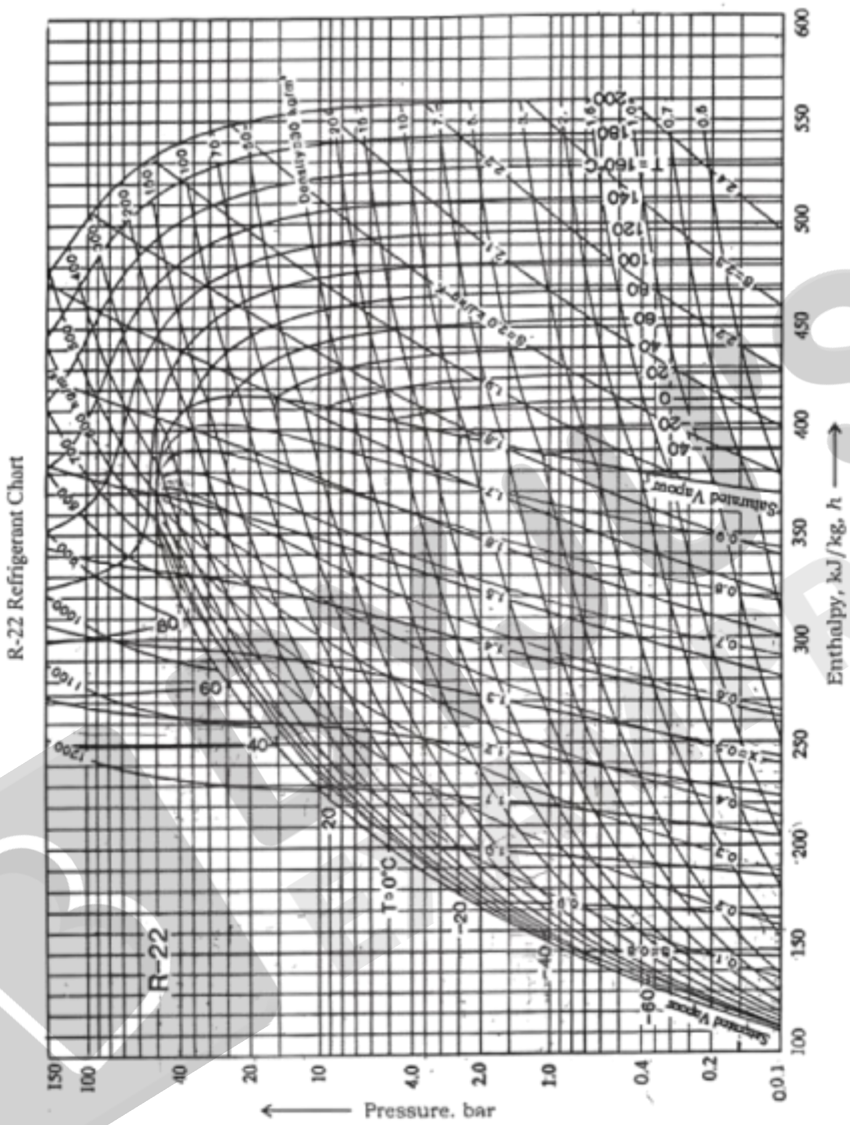
Thermal power = 0.3159 kJ/s or kW

(c)

(i) A refrigeration system with R-22 as refrigerant operates with an evaporating temperature of –10 °C and a condensing temperature of 35 °C. If the vapour leaves the evaporator saturated and is compressed isentropically, what is the COP of the cycle– (1) if saturated liquid enters the expansion device and (2) if the refrigerant entering the expansion device is with 10% vapour?

[R-22 refrigerant chart is attached]

[10 Marks]



Sol.

i.

Case 1: If saturated liquid enters the expansion device,

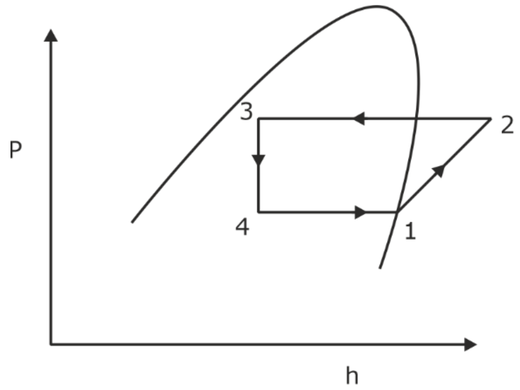
From chart (Approximate values)

h_1 = 400 kJ/kg

h_2 = 440 kJ/kg

$h_3 = h_4$ = 250 kJ/kg

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{400 - 250}{440 - 400} \approx 3.75$$



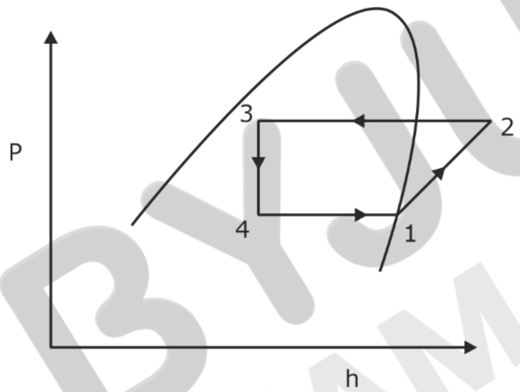
Case 2: Entry to expansion device is 10% vapour and 90% liquid.

From chart (Approximate values)

h_1 = 400 kJ/kg

h_2 = 440 kJ/kg

h_3 = h_4 = 265 kJ/kg



$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{400 - 250}{440 - 400} = 3.375$$

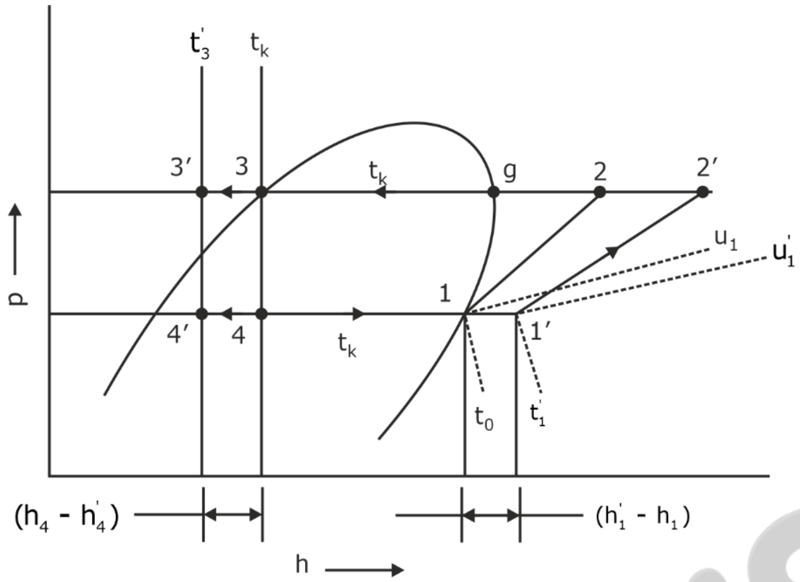
(ii) What is a liquid-to-suction heat exchanger in refrigeration and air conditioning? Illustrate the benefits of liquid-to-suction heat exchanger.

[10 Marks]

Sol.

If we combine superheating of vapour with liquid subcooling, we have a liquid vapour regenerative heat exchanger.

A liquid–vapour heat exchanger may be installed as shown in Figure. In this, the refrigerant vapour from the evaporator is superheated in the regenerative heat exchanger with consequent subcooling of the liquid from the condenser. The effect on the thermodynamic cycle is shown in Figure. Since the mass flow rate of the liquid and vapour is the same, we have from energy balance of the heat exchanger = $h'_1 - h_1 = h_3 - h'_3$



Vapour compression cycle with liquid-vapour regenerative heat exchange

The degree of superheat $\left(t_1^{'} - t_0\right)$ and the degree of subcooling $\left(t_k - t_3^{'}\right)$ need not be the same as the specific heats of the vapour and liquid phases are different. The effect on the capacity, power requirement per unit refrigeration and COP is expressed as follows:

$$\frac{Q_0^{'}}{Q_0} = \frac{h_1 - h_4^{'}}{h_1 - h_4} \cdot \frac{u_1}{u_1}$$

$$\frac{W^{*'}}{W^{*}} = \frac{h_1 - h_4}{h_1 - h_4^{'}} \cdot \frac{h_2^{'} - h_1^{'}}{h_2 - h_1}$$

$$E_c^{'} = \frac{\left(h_1 - h_4\right) + \left(h_1^{'} - h_1\right)}{\left(h_2 - h_1\right) + \left[\left(h_2^{'} - h_1^{'}\right) - \left(h_2 - h_1\right)\right]}$$

In all the above expressions, both numerators and denominators increase. The net effect, whether positive, negative or zero, depends on the refrigerant used and the operating conditions. In practice, the suction volume per ton and horsepower per ton are reduced for Freon 12 and R134a. Calculations show a slight increase in the suction volume and horsepower per ton for Freon 22 and ammonia. Experiments show, however, that the volumetric efficiency of most reciprocating compressors improves with superheat.

In particular, it must be stated that superheating of the vapour in a liquid-vapour regenerative heat exchanger is preferable to superheating in the evaporator itself, as the increased refrigerating effect $\left(h_1^{'} - h_1\right)$ due to superheating taking place from temperature t_0 to $t_1^{'}$ is transferred as $\left(h_4 - h_4^{'}\right)$ at temperature t_0, which lowers mean refrigeration temperature. Thus, we obtain the same increase in refrigerating effect at a lower temperature by the use of a liquid-vapour regenerative heat exchanger.

8.

(a) **(i)** Describe the working principle of hydrogen fuel cell. Also, comment on the reversible energy conversion efficiency of fuel cells.

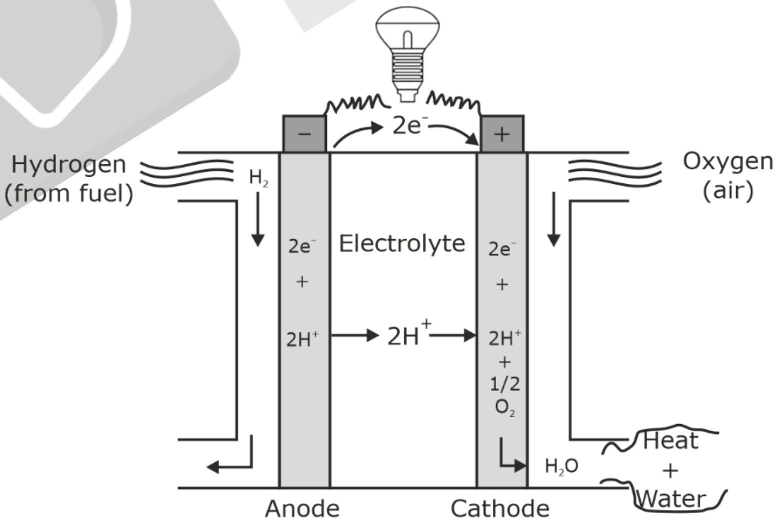
[10 Marks]

(ii) A flat plate solar collector measuring 2 m × 1.2 m has a loss resistance of 0.13 m² K/W and a plate transfer efficiency of 0.85. The glass cover has transmittance of 0.9 and the absorptance of the plate is also 0.9. Water enters at a temperature of 35 °C. The ambient temperature is 20 °C and the irradiance in the plane of the collector is 750 W/m². Calculate the flow rate needed to produce a temperature rise of 10 °C. The density of water and its specific heat at mean film temperature may be taken as 1000 kg/m³ and 4.2 J/g-°C respectively.

[10 Marks]

Sol.

(i) A hydrogen fuel cell has two electrodes where the reactions take place and an electrolyte which carries the charged particles from one electrode to the other. In order for a fuel cell to work, it needs hydrogen (H_2) and oxygen (O_2). The hydrogen enters the fuel cell at the anode. A chemical reaction strips the hydrogen molecules of their electrons and the atoms become ionized to form H^+. The electrons travel through wires to provide a current to do work. The oxygen enters at the cathode, usually from the air. The oxygen picks up the electrons that have completed their circuit. The oxygen then combines with the ionized hydrogen atoms (H^+), and water (H_2O) is formed as the waste product which exits the fuel cell. The electrolyte plays an essential role as well. It only allows the appropriate ions to pass between the anode and cathode. If other ions were allowed to flow between the anode and cathode, the chemical reactions within the cell would be disrupted.



The reaction in a single fuel cell typically produces only about 0.7 volts. Therefore, fuel cells are usually stacked or connected in some way to form a fuel cell system that can be used in cars, generators, or other products that require power.

The reactions involved in a fuel cell are as follows:

Anode side (an oxidation reaction): $2H_2 \rightarrow 4H^+ + 4e^-$

Cathode side (a reduction reaction): $O_2 + 4H^+ + 4e^- \rightarrow 2H_2O$

Net reaction (the "redox" reaction): $2H_2 + O_2 \rightarrow 2H_2O$

EFFICIENCY OF A FUEL CELL

In a fuel cell, electrochemical reactions occur whereby reactants are converted to products in a steady flow process. If the temperature and pressure of the flow stream from the entrance to the exit (during the reaction) remain unchanged, from the first law of thermodynamics:

$$\Delta Q - \Delta W = \Delta H + \Delta\,(KE) + \Delta(PE)$$

Where

ΔQ = Heat transferred to the steady flow stream from the surrounding

ΔW = Net work done by the flow stream on the surrounding

ΔH = Change in enthalpy of the flow stream from the entrance to exit (of the cell)

The change in KE and PE of the stream is usually negligible. Thus:

$$\Delta W = \Delta Q - \Delta H$$

For ΔW to be the maximum, the process must be reversible.

Entropy is an indicator of heat per kelvin temperature (T). For a reversible process, from the second law of thermodynamics, we have,

$$\Delta Q = \int T ds$$

But the surrounding temperature is constant. Thus, reversible heat transfer occurs at temperature T, the prevailing temperature at the inlet and exit. Thus,

$$\Delta Q_{rev} = T\,\Delta S \qquad \text{...... (1)}$$

where T is the temperature of the process, and it remains constant, ΔS in the change in entropy.

Thus, $\Delta W_{max} = -\,(\Delta H - T\Delta S)$

The energy available to perform useful work is called 'Gibbs Free Energy' G. It is given by:

$$G = H - TS$$

$$\Delta G = \Delta H - (T\Delta S - S\Delta T)$$

As there is no change in temperature, $\Delta T = 0$, and thus

$$\Delta G = \Delta H - T\Delta S \qquad \text{...... (2)}$$

Therefore: $\Delta W_{max} = -\Delta G$

Combining Eq. (1) and Eq. (2), we can write :

$$\Delta G = \Delta H - \Delta Q$$

or $\Delta Q = \Delta H - \Delta G$

The efficiency of energy conversion of a fuel cell:

$$\eta = \frac{\Delta W}{-\Delta H}$$

Maximum efficiency, $\eta_{max} = \frac{\Delta W_{max}}{-\Delta H} = \frac{\Delta G}{\Delta H}$

(ii) Given

Plate area (A_p) = 2 × 1.2 m^2 = 2.4 m^2

Loss resistance = 0.13 m^2-K/W = R_l

Transmittance of plate glass cover (τ) = 0.9

Plate transfer efficiency (η_p) = 0.85

Absorptance (α) = 0.9

Inlet temperature of water (T_s) = 35°C

Ambient temperature (T_a) = 20°C

Irradiance (G) = 750 W/m^2

ΔT = 10°C

Radiant heat flux striking the plate,

$Q_r = \tau \times A_p \times G = 0.9 \times 2.4 \times 750$

Q_r = 1620 W/m^2

Net heat flow into the plate

$$Q_n = (\alpha \times Q_r) - \left[\frac{T_s - T_a}{R_l} \times A_p\right]$$

$$Q_n = (0.9 \times 1620) - \left[\frac{35-20}{0.13} \times 2.4\right] \text{ W} = [1458 - 276.92]$$

Q_n = 1181.08 W

Useful power from the collector

$Q_u = \eta_p \times Q_n = 0.85 \times 1181.08 = 1003.91$ W

Now $Q_u = \dot{m}C_p\Delta T \Rightarrow 1003.91 = \dot{m} \times 4.2 \times 10^3 \times 10$

$$\dot{m}_{water} = 0.0239 \text{ kg/s}$$

Volume flow rate $V = \frac{\dot{m}}{\rho} = \frac{0.0239}{1000} = 2.39 \times 10^{-5} \frac{m^3}{s}$

(b) A two-pass surface condenser is required to handle the exhaust from a turbine developing 15 MW with specific steam consumption of 5 kg/kWh. The condenser vacuum is 660 mm of mercury when the barometer reads 760 mm of mercury. The mean velocity of water is 3 m/s and the water inlet temperature is 24 °C. The condensate is saturated water and the outlet temperature of cooling water is 4 °C less than the condensate temperature. The quality of exhaust steam is 0.9 dry. The overall heat transfer coefficient based on outer area of tubes is 4000 W/m²-°C. The water tubes are 38.4 mm in outer diameter and 29.6 mm in inner diameter. Calculate the following:

(i) Mass of cooling water circulated in kg/min

(ii) Condenser surface area

(iii) Number of tubes required per pass

(iv) Tube length

Assume atmospheric pressure to be 760 mm of mercury or 1.01325 bar and specific heat of water = 4.187 kJ/kg-K.

[Saturated steam table is attached at the end of booklet]

[20 Marks]

Sol. Given,

A two tube pass surface

Power (P) = 15 MW = 15 × 10^3 kW

Specific steam consumption, $\dot{m}_s$ = 5 kg/kWh

Barometer reads, P_{atm} = 760 mm of Hg

Condenser vacuum, P_g = –660 mm of Hg

Mean velocity, V_m = 3 m/s

Water inlet temperature, T_{ci} = 24 °C

Condensate is saturated water and outlet temperature of cooling water is 4 °C less than the condensate temperature, $T_{co} = T_{ho} - 4$ °C

Quality of exhaust steam x = 0.9

Overall heat transfer coefficient, U_0 = 400 W/m^{2}C

Outer diameter of water tube, d_o = 38.4 mm

Inner diameter of water tube, d_i = 29.6 mm

The absolute pressure, $P_{abs} = P_{atm} + P_g$

$$= 760 - 660 \text{ mm of Hg}$$

$$= 100 \text{ mm of Hg}$$

$$\therefore h = 100 \text{ mm}$$

$$\therefore \quad P_{abs} = \rho g h$$

$$= 13.6 \times 10^3 \times 9.81 \times \frac{100}{100}$$

$$P_{abs} = 133416 \text{ Pa} = 0.133 \text{ bar (or)} = 13.34 \text{ kPa}$$

At, P = 0.133 bar

$$T_h = T_{sat} = 51°C$$

$$h_{fg} = 2592 \text{ kJ/kg}$$

$$\therefore T_{co} = 51 - 4 = 47°C$$

(i) Mass of steam condensed per minute

$$m_h = P \times \dot{m}_s$$

$$= 15 \times 10^3 \times 5 = 75000 \text{ kg/h}$$

$$= \frac{75000}{60} = 1250 \text{ kg/min} \approx 20.83 \text{ kg/s}$$

Heat rejected by steam per minute,

$$Q = m_h \times x.h_{fg}$$

$$= 20.83 \times (0.9 \times 2592) = 48600 \text{ kW}$$

By energy balance

$$Q = m_c c_{pc} (T_{co} - T_{ci})$$

$$48600 = m_c \times 4.187 \times (47 - 24)$$

$$m_c = \frac{48600}{4.187 \times (47 - 24)} = 504.67 \text{ kg/s}$$

$$m_c \approx 30280 \text{ kg/min}$$

(ii) Condenser surface area

It operates in counter flow, so

$$\Delta T_1 = T_h - T_{co} = 51 - 47 = 4\ °C$$

$$\Delta T_2 = T_h - T_{ci} = 51 - 24 = 27\ °C$$

Logarithmic mean temperature difference (LMTD)

$$\Delta T_{ln} = \frac{\Delta T_1 - \Delta T_2}{1n\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{4 - 27}{ln\left(\frac{4}{27}\right)} = 12.04\ °C$$

We know, $Q = U_o A_o \Delta T_{ln}$

$$48600 \times 10^3 = 4000 \times A_o \times 12.04$$

$$A_o = \frac{48600 \times 10^3}{4000 \times 12.04} = 1009.136 \text{ m}^2$$

(iii) By continuity equation

$$m_c = \left(\frac{\pi}{4}\right) d_i^2 \rho V_m n$$

Where, $\rho = 1000$ kg/m^3

n = Number of pass of tubes

$$\therefore \quad 504.69 = \frac{\pi}{4} \times (0.0296)^2 \times 1000 \times 3 \times n$$

$$n = \frac{504.67}{2.064} = 244.46$$

$$\boxed{n \approx 244 \text{ tubes}}$$

(iv) Length of tube per pass

$$A_o = \pi d_o L \times n \times 2 \text{ passes}$$

Because it is 2 tube pass surface,

$$A_o = 2\pi d_o L n$$

$$L = \frac{A_o}{2\pi d_o n} = \frac{1009.136}{\pi \times 0.0384 \times 244 \times 2} = 17.14 \text{ m}$$

(c) The total pressure maintained in an Electrolux refrigerator is 15 bar. The temperature obtained in the evaporator is –15 °C. The quantities of heat supplied to the generator are (i) 420 kJ to dissociate one kg of vapour and (ii) 1460 kJ/kg for increasing the total enthalpy of NH_3. The enthalpy of NH_3 entering the evaporator is 330 kJ/kg. Take the following properties of NH_3 at -15 °C:

Pressure = 2.45 bar

Enthalpy of vapour = 1666 kJ/kg

Specific volume = 0.5 m^3/kg

The hydrogen enters the evaporator at 25 °C

Gas constant for H_2 = 4218 kJ/kg-°C

C_p (for H_2) = 12.77 kJ/kg-°C

Find the COP of the system assuming NH_3 leaves the evaporator in saturated condition.

[20 Marks]

Saturated Steam Pressure Table

P bar	t °C	v_f m^3/kg	v_g m^3/kg	h_f kJ/kg	h_g kJ/kg	h_{fg} kJ/kg	s_f kJ/kg-K	s_g kJ/kg-K
0.010	6.9828	.0010001	129.21	29.34	2514.4	2485.0	.1060	8.9767
0.015	13.036	.0010006	67.982	54.71	2525.5	2470.7	.1957	8.8288
0.020	17.513	.0010012	67.006	73.46	2533.6	2460.2	.2607	8.7246
0.025	21.096	.0010020	54.256	88.45	2540.2	2451.7	.3119	8.6440
0.030	24.100	.0010027	45.667	101.00	2545.6	2444.6	.3544	8.5785
0.035	26.694	.0010033	39.479	111.85	2550.4	2438.5	.3907	8.5232
0.040	28.963	.0010040	34.802	121.41	2554.5	2433.1	.4225	8.4755
0.045	31.035	.0010046	31.141	129.99	2558.2	2428.2	.4507	8.4335
0.050	32.896	.0010052	28.194	137.77	2561.6	2423.8	.4763	8.3960
0.055	34.605	.0010058	25.771	144.91	2564.7	2419.8	.4995	8.3621
0.060	36.183	.0010064	23.741	151.50	2567.5	2416.0	.5209	8.3312
0.065	37.651	.0010069	22.016	157.64	2570.2	2412.5	.5407	8.3029
0.070	39.025	.0010074	20.531	163.38	2572.6	2409.2	.5591	8.2767
0.075	40.316	.0010079	19.239	168.77	2574.9	2406.2	.5763	8.2523
0.080	41.534	.0010084	18.105	173.86	2577.1	2403.2	.5925	8.2296
0.085	42.689	.0010089	17.100	178.69	2579.2	2400.5	.6079	8.2082
0.090	43.787	.0010094	16.204	183.28	2581.1	2397.9	.6224	8.1881
0.095	44.833	.0010098	15.400	187.65	2583.0	2395.3	.6361	8.1691
0.100	45.833	.0010102	14.675	191.83	2584.8	2392.9	.6493	8.1511
0.11	47.710	.0010111	13.416	199.68	2588.1	2388.4	.6738	8.1177
0.12	49.446	.0010119	12.362	206.94	2591.2	2384.3	.6963	8.0872
0.13	51.062	.0010126	11.466	213.70	2594.0	2380.3	.7172	8.0592
0.14	52.574	.0010133	10.694	220.02	2596.7	2376.7	.7367	8.0334
0.15	53.997	.0010140	10.023	225.97	2599.2	2373.2	.7549	8.0093
0.16	55.341	.0010147	9.4331	231.59	2601.6	2370.0	.7721	7.9869
0.17	56.615	.0010154	8.9110	236.92	2603.8	2366.9	.7883	7.9658
0.18	57.826	.0010160	8.4452	241.99	2605.9	2363.9	.8036	7.9460
0.19	58.982	.0010166	8.0272	246.83	2607.9	2361.1	.8182	7.9272
0.20	60.086	.0010172	7.6498	251.45	2609.9	2358.4	.8321	7.9094
0.21	61.145	.0010178	7.3073	255.88	2611.7	2355.8	.8453	7.8925
0.22	62.162	.0010183	6.9951	260.14	2613.5	2353.3	.8581	7.8764
0.23	63.139	.0010189	6.7093	264.23	2615.2	2350.9	.8702	7.8611
0.24	64.082	.0010194	6.4467	268.18	2616.8	2348.6	.8820	7.8464
0.25	64.992	.0010199	6.2045	271.99	2618.3	2346.4	.8932	7.8323
0.26	65.871	.0010204	5.9803	275.67	2619.9	2344.2	.9041	7.8188
0.27	66.722	.0010209	5.7724	279.24	2621.3	2342.1	.9146	7.8058
0.28	67.547	.0010214	5.5788	282.69	2622.7	2340.0	.9248	7.7933
0.29	68.347	.0010219	5.3982	286.05	2624.1	2338.1	.9346	7.7812
0.30	69.124	.0010223	4.2293	289.30	2625.4	2336.1	.9441	7.7695
0.32	70.615	.0010232	4.9223	295.55	2628.0	2332.4	.9623	7.7474
0.34	72.029	.0010241	4.6504	301.48	2630.4	2328.9	.9795	7.7266
0.36	73.374	.0010249	4.4078	307.12	2632.6	2325.5	.9958	7.7070
0.38	74.658	.0010257	4.1900	312.50	2634.8	2322.3	1.0113	7.6884
0.40	75.886	.0010265	3.9934	317.65	2636.9	2319.2	1.0261	7.6709
0.45	78.743	.0010284	3.5762	329.64	2641.7	2312.0	1.0603	7.6307
0.50	81.345	.0010301	3.2402	340.56	2646.0	2305.4	1.0912	7.5947
0.55	83.737	.0010317	2.9636	350.61	2649.9	2299.3	1.1194	7.5623
0.60	85.954	.0010333	2.7318	359.93	2653.6	2293.6	1.1454	7.5327
0.65	88,021	.0010347	2.5346	368.62	2656.9	2288.3	1.1696	7.5055
0.70	89.959	.0010361	2.3647	376.77	2660.1	2283.3	1.1921	7.4804
0.75	91.785	.0010375	2.2169	384.45	2663.0	2278.6	1.2131	7.4570
0.80	93.512	.0010387	2.0870	391.72	2665.8	2274.1	1.2330	7.4352
0.85	95.152	.0010400	1.9719	396.63	2668.4	2269.8	1.2518	7.4147
0.90	96.713	.0010412	1.8692	405.21	2670.9	2265.6	1.2696	7.3954
0.95	98.204	.0010423	1.7770	411.49	2673.2	2261.7	1.2865	7.3771
1.0	99.632	.0010434	1.6937	417.51	2675.4	2257.9	1.3027	7.3598
1.1	102.32	.0010455	1.5492	428.84	2679.6	2250.8	1.3330	7.3277
1.2	104.81	.0010476	1.4281	439.36	2683.4	2244.1	1.3609	7.2984
1.3	107.13	.0010495	1.3251	449.19	2687.0	2237.8	1.3868	7.2715
1.4	109.32	.0010513	1.2363	458.42	2690.3	2231.9	1.4109	7.2465
1.5	111.37	.0010530	1.1590	467.13	2693.4	2226.2	1.4336	7.2234
1.6	113.32	.0010547	1.0911	475.38	2696.2	2220,9	1.4550	7.2017
1.7	115.17	.0010563	1.0309	483.22	2699.0	2215.7	1.4752	7.1813
1.8	116.93	.0010579	.97723	490.70	2701.5	2210.8	1.4944	7.1622
1.9	118.62	.0010594	.92900	497.85	2704.0	2206.1	1.5127	7.1440

Saturated Steam Pressure Table

P bar	t °C	v_f m^3/kg	v_g m^3/kg	h_f kJ/kg	h_g kJ/kg	h_{fg} kJ/kg	s_f kJ/kg-K	s_g kJ/kg-K
2.0	120.23	.0010608	.88544	504.70	2706.3	2201.6	1.5301	7.1268
2.1	121.78	.0010623	.84500	511.28	2708.5	2197.2	1.5468	7.1105
2.2	123.27	.0010636	.80964	517.62	2710.6	2193.0	1.5627	7.0949
2.3	124.71	.0010650	.77681	523.73	2712.6	2188.9	1.5781	7.0800
2.4	126.09	.0010683	.74845	529.63	2714.5	2184.9	1.5929	7.0667
2.5	127.43	.0010675	.71844	535.34	2716.4	2181.0	1.6071	7.0620
2.6	128.73	.0010888	.09251	540.87	2718.2	2177.3	1.6209	7.0380
2.7	129.98	.0010700	.66844	546.24	2719.9	2173.8	1.6342	7.0282
2.8	131.20	.0010712	.64604	551.44	2721.5	2170.1	1.6471	7.0140
2.9	132.39	.0010724	.62513	556.50	2723.1	2166.8	1.6695	7.0023
3.0	133.54	.0010735	.60556	561.43	2724.7	2163.2	1.6716	6.9909
3.1	134.66	.0010748	.58722	566.23	2726.1	2159.9	1.8834	6.9799
3.2	135.75	.0010757	.56999	570.90	2727.8	2156.7	1.6948	6.9603
3.3	136.82	.0010768	.55376	575.46	2729.0	2153.5	1.7069	6.9689
3.4	137.86	.0010779	.53846	579.92	2730.3	2150.4	1.7168	6.9489
3.5	138.87	.0010789	.52400	584.27	2731.6	2147.4	1.7273	6.9392
3.6	139.86	.0010799	.51032	588.53	2732.9	2144.4	1.7376	6.9297
3.7	140.83	.0010809	.49736	592.69	2734.1	2141.4	1.7476	6.9206
3.8	141.78	.0010819	.48506	596.76	2735.3	2138.6	1.7674	6.9116
3.9	142.71	.0010829	.47336	600.76	2736.5	2135.7	1.7670	6.9028
4.0	143.62	.0010839	.46222	604.67	2737.6	2133.0	1.7764	6.8943
4.1	144.52	.0010848	.45162	608.51	2738.7	2130.2	1.7856	6.8860
4.2	145.39	.0010658	.44150	612.27	2739.8	2127.5	1.7945	6.8779
4.3	146.25	.0010867	.43184	615.97	2740.9	2124.9	1.8033	6.8700
4.4	147.09	.0010876	.42260	619.60	2741.9	2122.3	1.8120	6.8623
4.5	147.92	.0010885	.41375	623.16	2742.9	2119.7	1.8204	6.8547
4.6	148.73	.0010894	.40528	626.67	2743.9	2117.2	1.8287	6.8473
4.7	149.53	.0010903	.39716	630.11	2744.8	2114.7	1.8388	6.8401
4.8	150.31	.0010911	.38936	633.50	2745.7	2112.2	1.8448	6.8330
4.9	151.08	.0010920	.38188	636.83	2746.6	2109.8	1.5227	6.8260
5.0	151.84	.0010928	.37468	640.12	2747.5	2107.4	1.8604	6.8192
5.2	153.33	.0010945	.36108	646.53	2749.3	2102.7	1.8754	6.8069
5.4	154.76	.0010961	.34846	652.76	2750.9	2098.1	1.8899	6.7932
5.6	156.16	.0010977	.33671	658.81	2752.8	2093.7	1.9040	6.7809
5.8	157.52	.0010993	.32574	664.69	2754.0	2089.3	1.9176	6.7600
6.0	158.84	.0011009	.31547	670.42	2755.5	2085.0	1.9308	6.7575
6.2	160.12	.0011024	.30585	676.01	2756.9	2080.9	1.9437	6.7464
6.4	161.38	.0011039	.29681	681.46	2758.2	2076.8	1.9562	6.7357
6.6	162.60	.0011063	.28830	688.78	2759.5	2072.7	1.9684	6.7252
6.8	163.79	.0011068	.28027	691.98	2760.8	2068.8	1.9802	6.7150
7.0	164.96	.0011082	.27268	607.06	2762.0	2064.9	1.9918	6.7062
7.2	166.10	.0011096	.26550	702.03	2763.2	2061.1	2.0031	6.8956
7.4	167.21	.0011110	.25870	708.90	2764.3	2037.4	2.1041	6.6862
7.6	168.30	.0011123	.25224	711.67	2765.4	2053.7	2.0249	6.6771
7.8	169.37	.0011137	.24610	716.36	2766.4	2050.1	2.0354	6.6683
8.0	170.41	.0011150	.24026	720.94	2767.5	2046.5	2.0457	6.6506
8.2	171.44	.0011163	.23469	725.43	2768.5	2043.0	2.0558	6.6511
8.4	172.45	.0011176	.22938	729.85	2769.4	2039.6	2.0657	6.6429
8.6	173.44	.0011188	.22430	734.19	2770.4	2036.2	2.0753	6.6348
8.8	174.41	.0011201	.21945	738.45	2771.3	2032.8	2.0848	6.6269
9.0	175.36	.0011213	.21481	742.64	2772.1	2029.5	2.0941	6.6192
9.2	176.29	.0011226	.21036	746.76	2773.0	2026.2	2.1033	6.6116
9.4	177.21	.0011238	.20610	750.82	2773.8	2023.0	2.1122	6.6042
9.6	178.12	.0011250	.20201	754.81	2774.8	2019.8	2.1210	6.5969
9.8	179.01	.0011262	.19807	758.74	2775.4	2016.7	2.1297	6.5898
10.0	179.88	.0011274	.19429	762.61	2776.2	2013.6	2.1382	6.5828
10.5	182.02	.0011303	.18545	772.03	2778.0	2005.9	2.1588	6.5650
11.0	184.07	.0011331	.17738	781.12	2779.7	1998.5	2.1786	6.5497
11.5	186.05	.0011350	.16909	789.92	2781.3	1991.3	2.1977	6.5342
12.0	187.96	.0011386	.16320	798.43	2782.7	1964.3	2.2161	6.5194
12.5	189.81	.0011412	.15693	806.69	2784.1	1977.4	2.2338	6.5061
13.0	191.61	.0011433	.15113	814.70	2785.4	1970.7	2.2510	6.4913
13.5	193.35	.0011464	.14574	822.40	2786.6	1964.2	2.2676	6.4780
14.0	195.04	.0011489	.14072	830.07	2787.8	1957.7	2.2837	6.4651
14.5	196.69	.0011514	.13604	837.46	2788.9	1951.4	2.2902	6.4526

Saturated Steam Pressure Table

P bar	t °C	v_f m^3/kg	v_g m^3/kg	h_f kJ/kg	h_g kJ/kg	h_{fg} kJ/kg	s_f kJ/kg-K	s_g kJ/kg-K
15.0	198.29	.0011539	.13166	844.66	2789.9	1945.2	2.3145	6.4406
15.5	199.85	.0011563	.12755	851.69	2790.8	1939.2	2.3292	6.4289
16.0	201.37	.0011586	.12369	858.56	2791.7	1933.2	2.3436	6.4175
16.5	202.86	.0011610	.12005	865.28	2792.6	1927.3	2.3576	6.4065
17.0	204.31	.0011633	.11662	871.84	2793.4	1921:5	2.3713	6.3957
17.5	205.72	.0011656	.11338	878.27	2794.1	1915.9	2.3846	6.3853
18.0	207.11	.0011678	.11032	884.57	2794.8	1910.9	2.3976	6.3751
18:5	208.47	.0011701	.10741	890.75	2795.5	1904.7	2.4103	6.3651
19.0	209.80	.0011723	.10465	896.81	2796.1	1899.3	2.4228	6.3554
19.5	211.10	.011744	.10203	902.75	2796.7	1893.9	2.4349	6.3459
20,0	212.37	.011766	.099536	908.59	2797.2	1888.6	2.4469	6.3367
20.5	213.63	.0011787	.097158	914.32	2797:7	1883.4	2.4585	6.3276
21.0	214.85	.0011809	.094890	919.96	2798.2	1878.2	2.4700	6.3187
21.5	216.06	.0011830	.092723	925.50	2798.6	1873.1	2.4812	6.3100
22.0	217.24	.0011850	.090652	930.95	2799.1	1868.1	2.4922	6.3015
22.5	218.41	.0011871	.088669	936.32	2799.4	1863.1	2.5030	6.2931
23.0	219.55	.0011892	.086769	941.60	2799.8	1858.2	2.5136	6.2849
23.5	220.68	.0011912	.084048	946.80	2800.1	1853.3	2.5241	6.2769
24.0	221.78	.011932	.083199	951.93	2800.4	1848.5	2.5343	6.2600
24.5	222.87	.0011962	.081520	956.98	2800.7	1843.7	2.5444	6.2612
25.0	223.94	.0011972	.079905	961.96	2800.9	1839.0	2.5543	6.2536
25.5	225.00	.0011991	.078352	966.87	2801.2	1834.3	2.5640	6.2461
26.0	226.04	.0012011	.076856	971.72	2801.4	1829.6	2.5736	6.2387
26.5	227.06	.0012031	.075415	976.50	2801.6	1825.1	2.5831	6.2315
27.0	228.07	.0012050	.074025	981.22	2801.7	1820.5	2.5924	6.2244
27.5	229.07	.0012069	.072684	985.88	2801.9	1816.0	2.6016	6.2173
28.0	230.05	.0012088	.071389	990.48	2802.0	1811.5	2.6106	6.2104
28.5	231.01	.0012107	.070138	995.03	2802.1	1807.1	2.6195	6.2036
29.0	231.97	.0012126	.068928	999.52	2802.2	1802.6	2.6283	6.1969
29.5	232.91	.0012145	.067758	1003.96	2802.2	1798.3	2.6370	6.1903
30.0	233.84	.0012163	.066626	1008.35	2802.3	1793.9	2.6455	6.1837
31.0	235.67	.0012200	.064467	1016.99	2802.3	1785.4	2.6623	6.1709
32.0	237.45	.0012237	.062439	1025.43	2802.3	1776.9	2.6786	6.1585
33.0	239.18	.0012274	.060629	1033.70	2802.3	1768.6	2.6945	6.1463
34.0	240.88	.0012310	.058728	1041.81	2802.1	1760.3	2.7101	6.1344
35.0	242.54	.0012345	.057025	1049.76	2802.0	1752.2	2.7253	6.1228
36.0	244.16	.0012381	.055415	1067.56	2801.7	1744.2	2.7401	6.1115
37.0	245.75	.0012416	.053881	1065.21	2801.4	1736.2	2.7547	6.1004
38.0	247.31	.0012451	.052438	1072.74	2801.1	1728.4	2.7689	6.0896
39.0	248.84	.0012486	.051061	1080.13	2800.8	1720.6	2.7829	6.0789
40.0	250.33	.0012521	.049749	1087.40	2800.3	1712.9	2.7965	6.0685
40.1	251.80	.0012565	.048500	1094.56	2799.9	1705.3	2.8099	6.0583
42.0	253.24	.0012589	.047307	1101.60	2799.4	1697.8	2.8231	6.0482
43.0	254.86	.0012623	.046168	1108.54	2798.9	1690.3	2.8360	6.0383
44.0	256.05	.0012657	.045079	1116.38	2798.3	1682.9	2.8487	6.0286
45.0	257.41	.0012691	.044037	1122.11	2797.7	1675.6	2.8612	6.0191
46.0	258.75	.0012725	.043038	1128.76	2797.0	1668.3	2.8735	6.0007
47:0	260.07	.0012758	.042081	1135.31	2796.4	1661.1	2.8855	6.0004
48.0	261.37	.0012792	.041161	1141.78	2795.7	1653.9	2.8974	5.9913
49.0	262.65	.012825	.040278	1148.16	2794.9	1646.8	2.9091	5.9824
50.0	263.91	.0012858	.039429	1154.47	2794.2	1639.7	2.9206	5.9735
51.0	265.15	.0012891	.038611	1160.69	2793.4	1632.7	2.9313	5.9648
52.0	266.37	.0012924	.037824	1166.85	2792.6	1625.7	2.9431	5.9561
53.0	267.58	.0012957	.037068	1172.93	2791.7	1618.8	2.9541	5.9476
54.0	268.76	.0012990	.036334	1178.94	2790.8	1611.9	2.9650	5.9392
55.0	269.93	.0013023	.035628	1184.89	2789.9	1605.0	2.9757	5.9309
56.0	271.09	.0013056	.034946	1190.77	2789.0	1598.2	2.9863	5.9227
57.0	272.22	.0013089	.034288	1196.59	2788.0	1591.4	2.9968	5.9146
58.0	273.35	.0013121	.033661	1202.35	2787.0	1584.7	3.0071	5.9066
59.0	274.46	.0013154	.033034	1208.05	2786.0	1578.0	3.0172	5.8986
60.0	275.55	.0013187	.032438	1213.69	2785.0	1571.3	3.0273	5.8908
61.0	276.63	.0013219	.031860	1219.28	2784.0	1564.7	3.0372	5.8830
62.0	277.70	.0013252	.031300	1224.82	2782.9	1558.0	3.0471	5.8753
63.0	278.75	.0013285	.030757	1230.31	2781.8	1551.5	3.0568	5.8677
64.0	279.79	.0013317	.030230	1235.75	2780.6	1544.9	3.0664	5.8601

Sol.

Given,

Total pressure of Electrolux Refrigerator (P_T) = 15 bar

Evaporator temperature (T_e) = 15°C = 285 K

$$Q_{G_1} = 420 \text{ kJ/kg}$$

$$Q_{G_2} = 1460 \text{ kJ/kg}$$

For NH_3; h_1 = 330 kJ/kg

P_2 = 2.45 bar

h_2 = 1666 kJ/kg

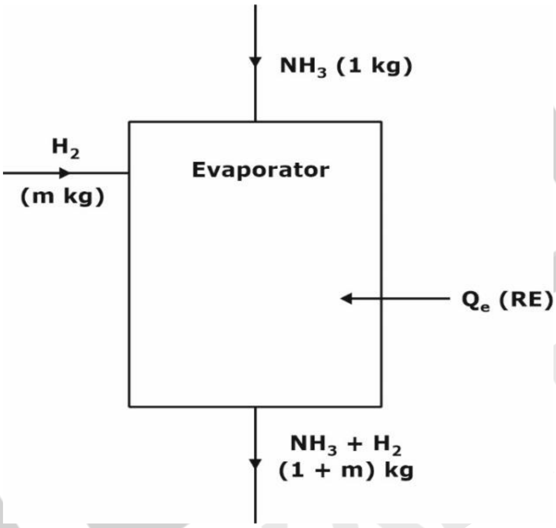
v = 0.5 m^3/kg

For hydrogen (H_2) : T_3 = 25°C = 298 K

R = 4.218 kJ/kg-°C

C_p = 12.77 kJ/kg-°C

Electrolux Refrigerator



Total heat given to generator,

$$Q_G = Q_{G_1} + Q_{G_2} = (420 + 1460) \text{ kJ/kg}$$

$$Q_G = 1880 \text{ kJ/kg}$$

Since in evaporator (P_T = 15 bar) and $P_2(P_{NH_3}) = 2.45$ bar

$$\Rightarrow \quad P_{H_2} = P_T - P_2 = 15 - 2.45$$

$$P_{H_2} = 12.55 \text{ bar}$$

For m kg H_2 flow per kg of NH_3

$$\upsilon_{NH_3} = 0.5 \text{ m}^3/\text{kg} = \upsilon_{H_2} \text{ at } -15^\circ\text{C} \ (258 \text{ K})$$

$$m_{H_2} = \frac{P_{H_2} \times \upsilon_{H_2}}{R_{H_2} \times T_{H_2}} = \frac{1255 \times 0.5}{4.218 \times 285}$$

$$\Rightarrow \ m_{H_2} = 0.5766 \text{ kg}$$

Using energy balance equation for evaporator

$$1 \times (h_2 - h_1) = m_{H_2}.C_{P_{H_2}}(T_3 - T_2) + Q_e$$

$$1 \times (1666 - 330) = 0.576 \times 12.77\ [25 - (-15)] + Q_e$$

$\Rightarrow$ Q_e = 1041.47 kJ/kg of NH_3

$$COP = \frac{Q_e}{Q_G} = \frac{1041.47}{1880}$$

COP = 0.554

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