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MECHANICAL ENGINEERING

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ESE-2019 : MAINS TEST SERIES

UPSC ENGINEERING SERVICES EXAMINATION

MECHANICAL ENGINEERING | Test No. 1

Section A : Thermodynamics

Section B : Refrigeration and Air Conditioning

Time Allowed : 3 hrs.

Maximum Marks: 300

Question Paper Specific Instructions

Please read each of the following instructions carefully before attempting questions:

- Answers must be written only in **ENGLISH**.
- There are **EIGHT** questions divided in **TWO** sections.
- Candidate has to attempt **FIVE** questions in all.
- Question no. **1** and **5** are **compulsory** and out of the remaining **THREE** are to be attempted choosing at least **ONE** question from each section.
- The number of marks carried by a question/part is indicated against it.
- Wherever any assumptions are made for answering a question, they must be clearly indicated. Diagrams/figures, wherever required, shall be drawn in the space provided for answering the question itself.
- Unless otherwise mentioned, symbols and notations carry their usual standard meanings. Attempt of questions shall be counted in sequential order. Unless struck off, attempt of a question shall be counted even if attempted partly. Any page or portion of the page left blank in the Question-cum-Answer Booklet must be clearly struck off.

DO NOT OPEN THIS BOOKLET UNTIL YOU ARE ASKED TO DO SO

For any query write to us, at: **info@madeeasy.in**

Section A : Thermodynamics

Q.1 (a) A piston cylinder device with air undergoes an expansion process for which pressure and volume are related as given below:

P(kPa)	100	95	89.5833
V(m³)	0.1	0.1043671	0.1096

Calculate the length of cylinder if mean effective pressure is 80 kPa and bore diameter is 200 mm.

[12 marks]

(b) The interior lighting of refrigerator is provided by incandescent lamps whose switches are actuated by the opening of the refrigerator door (30 times per day for an average 40 seconds). Consider a refrigerator whose 39 W light bulb remains ON continuously as a result of a malfunction of the switch. If the refrigerator has a coefficient of performance of 1.3 and the cost of electricity is ₹ 5 per kWh, calculate the increase in the energy consumption of the refrigerator and its cost per year if the malfunctioning of switch is not repaired or fixed. ₹ 1,684.475

[12 marks]

(c) A mixture of 60% CH_4 and 40% C_3H_8 by volume is compressed isentropically from 40°C, 0.4×10^6 N/m² to 1.2×10^6 N/m² in compressor. Find the final temperature of the mixture, the work required per unit mass assuming ideal gas behaviour with constant specific heats. Take specific heat at constant pressure (c_p) for CH_4 and C_3H_8 are 36 and 75 kJ/kmol-K respectively.

[12 marks]

(d) Explain the equivalence of the Clausius and Kelvin-Plank statements by showing diagram that violation of Clausius statements leads to the violation of Kelvin-Plank statement too and vice-versa.

[12 marks]

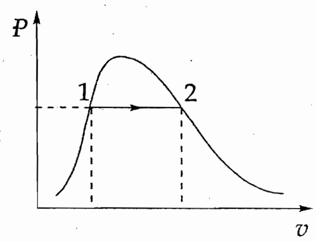
(e) Explain Perpetual motion machine of first kind (PMM1), second kind (PMM2) and third kind (PMM3).

[12 marks]

Q.2 (a) A tank containing 40 kg of liquid water initially at 50°C has one inlet and one exit with equal mass flow rates. Liquid water enters at 50°C and a mass flow rate of 250 kg/h. A cooling coil immersed in the water removes energy at a rate of 7.5 kW. The water is well mixed by a paddle wheel so that the water temperature is uniform throughout. The power input to the water from the paddle wheel is 0.5 kW. The pressure at inlet and exit are equal and all kinetic and potential energy change can be ignored. Determine and plot the variation of water temperature with time. (Specific heat of water, c = 4.2 kJ/kgK)

[20 marks]

(b) **(i)** Water, initially a saturated liquid at 150°C, is contained within a piston-cylinder assembly. The water undergoes a process to the corresponding saturated vapor state, during which the piston moves freely in the cylinder. There is no heat transfer with the surroundings. If the change of state is brought about by the action of a paddle wheel, determine the net work per unit mass and the amount of entropy produced per unit mass. Process is shown on the given diagram.



Use the following table:

Temp., T(°C)	Sat. press., P_{sat} (kPa)	Specific volume (m³/kg)		Internal Energy (kJ/kg)		Enthalpy (kJ/kg)		Entropy (kJ/kgK)	
		Sat. liquid, v_f	Sat. vapour, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. liquid, h_f	Evap. h_{fg}	Sat. liquid, s_f	Evap., s_{fg}
150	476.16	0.001091	0.39248	631.66	1927.4	632.18	2113.8	1.8418	4.9953
145	415.68	0.001085	0.44600	610.19	1944.2	610.64	2129.2	1.7908	5.0919

[6 marks]

(ii) A rigid tank, contains 5 kg of refrigerant 134a initially at 20°C and 140 kPa. The refrigerant is now cooled while being stirred until its pressure drops to 100 kPa. Determine the entropy change of the refrigerant during this process.

Note: At P_1 = 140 kPa, T_1 = 20°C; v_1 = 0.16544 m³/kg; s_1 = 1.0624 kJ/kgK

At P_2 = 100 kPa; v_f = 0.0007259 m³/kg; v_g = 0.19254 m³/kg ; s_f = 0.07188 kJ/kgK; s_{fg} = 0.87995 kJ/kgK

[6 marks]

(iii) Which is the more effective way to increase the efficiency of a Carnot engine, to increase T_1 keeping T_2 constant or to decrease T_2, keeping T_1 constant and also show it mathematically? [Assume, $T_1 > T_2$]

[8 marks]

(c) A tank having a volume of 0.85 m^3 initially contains water as a two-phase liquid-vapour mixture at 260°C and a quality of 0.7. Saturated water vapour at 260°C is slowly withdrawn through a pressure-regulating valve at the top of the tank as energy is transferred by heat to maintain the pressure constant in the tank. This continues until the tank is filled with saturated vapor at 260°C. Determine the amount of heat transfer. Neglect all kinetic and potential energy effects.
At P = 4.8 MPa, T_{sat} = 260°C, u_f = 1128.4 kJ/kg, u_g = 2599.0 kJ/kg, v_f = 0.0012755 m^3/kg, v_g = 0.04221 m^3/kg, h_g = 2796.6 kJ/kg

[20 marks]

Q.3 (a) Argon enters an insulated turbine operating at steady state, at 1000°C and 2 MPa and exhausts at 350 kPa. The mass flow rate is 0.5 kg/s and the turbine develops power at the rate of 120 kW. Determine:

1. the temperature of the argon at the turbine exit,
2. the irreversibility of the turbine,
3. the second law efficiency.

Neglect K.E. and P.E. effects. Take T_0 = 20°C, P_0 = 1 bar.
(The value of γ = 1.67 and c_p = 0.52 kJ/kgK)

[20 marks]

(b) A vertical 12 cm diameter piston cylinder device contains an ideal gas at the ambient conditions of 1 bar and 24°C. Initially, the inner face of the piston is 20 cm from the base of the cylinder. Now an external shaft connected to the piston exerts a force corresponding to a boundary work input of 0.1 kJ. The temperature of the gas remains constant during the process. Determine:

1. the amount of heat transfer
2. the final pressure in the cylinder, and
3. the distance that the piston is displaced

State the suitable assumptions, if any.

[20 marks]

(c) An ideal gas cycle consists of three reversible processes in the following sequence

- Constant volume heat addition
- Isentropic expansion to 'r' times the initial volume.
- Constant pressure process.

1. Sketch the cycle on the P-V and T-s diagrams.
2. Determine the expression of the efficiency of the cycle in terms of r and γ, where γ is adiabatic index.
3. If $r = 7$ and $\gamma = 1.4$, evaluate the efficiency of cycle.

[20 marks]

Q.4 (a) **(i)** Air at 10°C and 80 kPa enters the diffuser of a jet engine with a velocity of 200 m/s. The inlet area of the diffuser is 0.4 m^2. The air leaves the diffuser with a velocity that is very small compared with the inlet velocity. Determine:

1. mass flow rate of the air
2. the temperature of the air leaving the diffuser. (State the assumptions also.)

Use the following table:

T(K)	h(kJ/kg)
280	280
283	283
295	295
303	303

[8 marks]

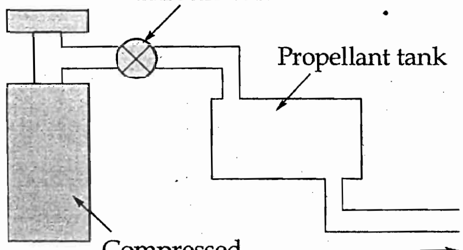
(ii) A rigid tank is divided into two equal parts by a partition. Initially, one side of the tank contains 5 kg of water at 200 kPa and 25°C, and the other side is evacuated. The partition is then removed, and the water expands into the entire tank. The water is allowed to exchange heat with its surroundings until the temperature in the tank returns to the initial value of 25°C. Determine:

1. the volume of the tank.
2. the final pressure.
3. the heat transfer for this process.

At $(P_{sat})_{25°C} = 3.1698$ kPa, $T_{sat} = 25°C$, $v_f = 0.001$ m^3/kg, $v_g = 43.340$ m^3/kg, $u_f = 104.83$ kJ/kg, $u_{fg} = 2304.3$ kJ/kg

[12 marks]

(b) Compressed air is used to expel the liquid propellent from the propellent tank as shown below. The initial pressure of the air is 200 bar. The propellant has a density of 1.12 gm/cc and the propellant tank is filled to capacity and contains 900 kg of propellent. The propellant leaves at a constant pressure of 200 N/cm^2 (20 bar). Considering the air as the system, determine the work done by the air in forcing the propellant from the propellant tank and the volume of compressed air tank necessary to pump all the fuel in the rocket.



[20 marks]

(c) A 5 kg block initially at 350°C is quenched in an insulated tank that contains 100 kg of water at 30°C. Assuming the water that vaporizes during the process condenses back in the tank and the surroundings are at 20°C and 100 kPa, Determine:
1. the final equilibrium temperature.
2. the exergy of the combined system at the initial and the final states,
3. the wasted work potential during this process.

Given: Specific heat of block, $c_{iron} = 0.45$ kJ/kgK, $c_{water} = 4.18$ kJ/kgK

[20 marks]

Section B : Refrigeration and Air Conditioning

Q.5 (a) What are the serious practical difficulties encountered in the application of reversed Carnot cycle with vapour as a refrigerant? Explain briefly.
Also mention drawbacks of reversed Carnot cycle with gas as a refrigerant.

[12 marks]

(b) Consider a reverse Carnot cycle with gas as a refrigerant. Prove that COP of the refrigerator is $\frac{1}{r^{\gamma-1}-1}$ where, r - Compression ratio, γ - Air adiabatic index. Also draw T-s and P-V diagram.

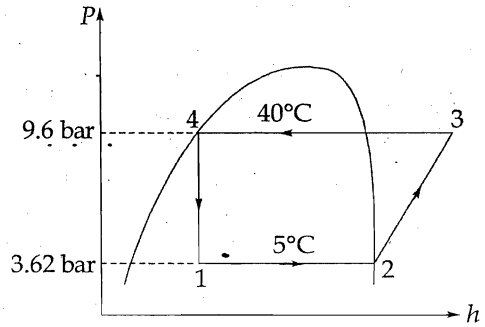
[12 marks]

(c) In an absorption type refrigerator, the heat is supplied to NH_3 generator by condensing steam at 2 bar and 90% dry. The temperature to be maintained in the refrigerator is –5°C. The temperature of the atmosphere is 30°C. Find the maximum COP possible of the refrigerator. If the refrigeration load is 20 tons and actual COP is 70% of maximum COP. Find the mass of steam required per hour.
At $P = 2$ bar, $T_{sat} = 119.62$°C, $h_{fg} = 2200$ kJ/kg

[12 marks]

(d) Two-cylinder reciprocating compressor is designed for 7.5 tons refrigeration capacity at 5°C evaporator and 40°C condenser temperatures. The clearance is 5% of stroke and mean piston speed is limited to 3 m/s. Take $\frac{L}{D} = 0.8$ and compression follows the law $PV^{1.15}$ = Constant. Pressure drops at suction and discharge valves may be taken as 0.2 bar and 0.4 bar, respectively. Determine rpm of the compressor and its dimensions.

Given: $v_2 = 0.0525\ m^3/kg$, Mass flow rate = 0.207 kg/s



[12 marks]

(e) Five grams of water vapour per kg of atmospheric air is removed and temperature of air after removing the water vapour becomes 25°C DBT. Determine:

1. Relative humidity
2. Dew point temperature
3. Degree of saturation

Assume condition of atmospheric air is 35°C and 60% RH and pressure is 1.013 bar.

Pressure (kPa)	Temperature (°C)
2.61	22
2.8105	23
3.1693	25
5.6280	35

[12 marks]

Q.6 (a) The air-conditioning unit of a pressurised jet aircraft receives its air from the compressor driven by the engine at a pressure of 1.4 bar. The pressure and temperature of the surrounding air at the height of the aircraft are 0.2 bar and 225 K, respectively. The air conditioning unit consists of a secondary compressor and a turbine mounted on the same shaft. The pressure and temperature of air leaving the turbine are 1 bar and 275 K. Calculate the pressure after the secondary compressor and temperature of air at the exit from the cooler. Assume that all processes are reversible adiabatic.

[20 marks]

(b) Saturated NH_3 at 2.3637 bar enters a 15 cm × 14 cm (bore × stroke) twin cylinder, single action compressor whose volumetric efficiency is 79% and speed is 250 rpm. The head pressure is 11.671 bar. Liquid NH_3 at 21°C enters the expansion valve for ideal vapour compression refrigeration cycle. Determine:

1. the NH_3 circulated in kg/min
2. the tons of refrigeration
3. the COP of refrigerating cycle.

P (bar)	T (°C)	v_g (m³/kg)	h_f (kJ/kg)	h_g (kJ/kg)	s_f (kJ/kg.K)	s_g (kJ/kg.K)	c_{pl} (kJ/kg.K)	c_{pv} (kJ/kg.K)
2.3637	–15	0.50905	131.28	1444.19	0.7424	5.8285	4.528	2.442
11.671	30	0.11048	342.08	1485.93	1.4892	5.2623	4.843	3.252

[20 marks]

(c) **(i)** Explain the basic principle of vapour absorption refrigeration system with the help of schematic diagram. Also explain functions of those devices which replace compressor.

[14 marks]

(ii) Explain working principle of electrolux refrigeration system.

[6 marks]

Q.7 (a) A building has the following calculated cooling loads:

RSH gain = 310 kW, RLH gain = 100 kW,

The space is maintained at the following conditions:

Room DBT = 25°C, Room RH = 50%

Outdoor air is at 38°C and 50% RH and 10% by mass of air supplied to the building is outdoor air. If the air supplied to the space is not to be at a temperature lower than 18°C, find:

1. Minimum amount of air supplied to space in m³/s.
2. Volume flow rates of return (recirculated room) air, exhaust air and outdoor air.
3. Volume flow rate of air entering the cooling coil.
4. Refrigeration capacity , ADP and BPF.

[20 marks]

(b) Explain the winter air conditioning system by showing all the processes on psychrometric chart.

[20 marks]

(c) An open cycle air-refrigeration system working between 1 bar and 12 bar produces 25 tons of refrigeration. The temperature of air leaving the cooler is 298 K and its temperature leaving the evaporator is 273 K. Assuming the expansion and compression follow the law $PV^{1.35} = C$ and $PV^{1.3} = C$ respectively. Determine:

1. Mass of the air circulated per minute.
2. COP of the system
3. kW power per ton of refrigeration.
4. The compressor and expander piston displacements

Neglect the clearance. [Take c_p of air as 1 kJ/kgK]

[20 marks]

Q.8 (a) 90 m^3 of air per minute at 5°C DBT and 2.5°C WBT is passed through a heating coil which gives 40.7 kW energy to the air. Saturated steam at 110°C with a rate of 40 kg/hr is mixed with the air leaving the heater. Determine DBT and WBT of the air after mixing with the steam. Take enthalpy of saturated steam at 110°C = 2691 kJ/kg. Use the following table:

T (°C)	2.5	5
P_{sat}(kPa)	0.73	0.874

Given: $P_v = (P_{vs})_{WBT} - \dfrac{\left[P_t - (P_{vs})_{WBT}\right]\left[T_{DB} - T_{WB}\right]}{1547 - 1.44T_{WB}}$

P_v - Vapour pressure at DBT.

$(P_{vs})_{WBT}$ - Vapour saturation pressure at WBT.

P_t - Total atmospheric pressure of air.

T_{DB} - Dry bulb temperature (in °C)

T_{WB} - Wet bulb temperature (in °C)

[20 marks]

(b) (i) Gas at 1 bar is compressed adiabatically from 3.157 m^3 to 1 m^3 in a reciprocating compressor with 8 percent clearance. If the exponent of the re-expansion curve is changes from 1.4 to 1.1, then find the percentage change in volumetric efficiency keeping the pressure ratio same.

[8 marks]

(ii) Explain the various important properties of refrigerant required for the vapour compression refrigeration system. Also explain why use of isentropic expander is economically not feasible in VCRS.

[12 marks]

(c) Explain the construction and working of vortex tube with a suitable schematic diagram. Also draw T-s diagram and mention points clearly.

[20 marks]

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Detailed Solutions

ESE-2019 Mains Test Series

Mechanical Engineering Test No : 1

Section A : Thermodynamics

Q.1 (a) Solution:

For polytropic index:

$$P_1V_1^n = P_2V_2^n = P_3V_3^n$$

$$n = \frac{\ln\left(\frac{P_1}{P_2}\right)}{\ln\left(\frac{V_2}{V_1}\right)} = \frac{\ln\left(\frac{100}{95}\right)}{\ln\left(\frac{0.1043671}{0.1}\right)} = 1.2$$

$$\text{Workdone} = \frac{P_1V_1 - P_3V_3}{n-1} = 0.90835 \text{ kJ}$$

Now, Workdone = (Mean effective pressure) × (Swept volume)

$$0.90835 = 80 \times \frac{\pi}{4} \times d^2 \times L$$

$$0.90835 = 80 \times \frac{\pi}{4} \times 0.2^2 \times L$$

$$L = 0.361 \text{ m}$$

$$L = 361 \text{ mm}$$

Q.1 (b) Solution:

(i) Assume no heat transfer through walls of refrigerator.

(ii) The life of the light bulb is more than 1 year.

Work input to refrigerator,

$$W_{\text{ref}} = \frac{Q_{ref}}{COP} = \frac{39}{1.3} = 30 \text{ W}$$

Therefore, the total power consumed by the refrigerator because of bulb,

$$W_{\text{total}} = W_{\text{light}} + W_{\text{refrig.}} = 39 + 30 = 69 \text{ W}$$

Now, total number of hours in a years,

$$\text{Annual hours} = 365\left(\frac{\text{days}}{\text{year}}\right)\times 24\left(\frac{h}{\text{day}}\right) = 8760 \text{ h/yr}$$

Now, normal operating hours

$$= \left(30\frac{\text{times}}{\text{day}}\right)\times\left(40\frac{s}{\text{time}}\right)\times\left(\frac{1h}{3600\,\text{s}}\right)\times\left(365\frac{\text{days}}{\text{year}}\right)$$

$$= 121.67 \text{ h/yr}$$

Then additional hour light remains ON as a result of the malfunction becomes,

$$\text{Additional operating hours} = \text{Annual hours} - \text{Normal operating hours}$$

$$= 8760 - 121.67 = 8638.33 \text{ h/year}$$

Now, additional electric power consumption $= W_{\text{total}} \times$ (additional operating hours)

$$= (0.069 \text{ kW}) \times (8638.33) \text{ h/year}$$

$$= 596.045 \text{ kWh/yr}$$

$$\text{Additional power cost} = (\text{Additional power consumption}) \times (\text{unit cost})$$

$$= 596.045 \times 5 = ₹\ 2980.225 \text{ /year}$$

Q.1 (c) Solution:

The specific heat of mixture, $\bar{C}_P = 0.6 \times 36 + 0.4 \times 75$

$$= 51.6 \text{ kJ/kmol-K}$$

Now,

$$\bar{C}_v = \bar{C}_P - \bar{R} = 51.6 - 8.314$$

$$= 43.286 \text{ kJ/kmol-K}$$

The molecular weight of the mixture,

$$W_m = 0.6 \times 16 + 0.4 \times 44$$

$$= 27.2 \text{ kg/kmol}$$

Now,
$$\gamma_m = \frac{\bar{C}_P}{\bar{C}_V} = 1.192$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\gamma_m - 1/\gamma_m} = (3)^{0.192/1.192}$$

$$T_2 = 373.59 \text{ K}$$

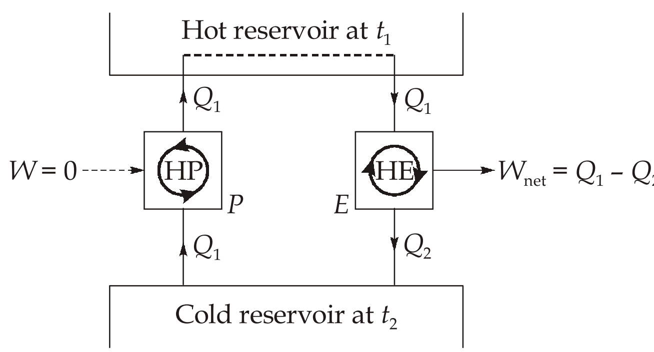
Work required per unit mass, $W = \bar{C}_P(T_2 - T_1)$

$$W = 51.6 \times (373.59 - 313) = 3126.444 \text{ kJ/kmol}$$

$$W = \frac{3126.444}{27.2}(\text{kJ/ kg}) = 114.943 \text{ kJ/kg}$$

Q.1 (d) Solution:

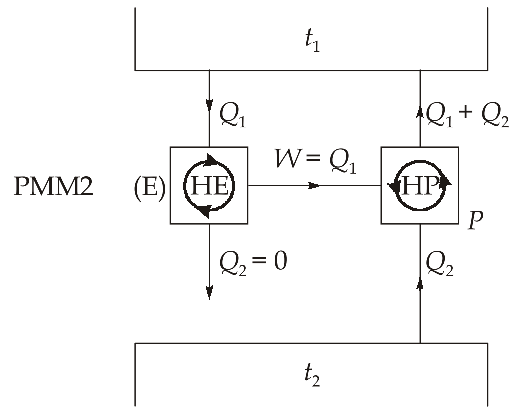
1. Let us first consider a cyclic heat pump P which transfers heat from a low temperature reservoir (t_2) to a high temperature reservoir(t_1) with not other effect, i.e., with no expenditure of work, violating Clausius statement.



Violation of the Clausius statement

Let us assume a cyclic heat engine E operating between the same thermal energy reservoirs, producing W_{net} in one cycle. The rate of working of the heat engine is such that it draws an amount of heat Q_1 from the hot reservoir equal to that discharged by the heat pump. Then the hot reservoir may be eliminated and the heat Q_1 discharged by the heat pump is fed to the heat engine. So we can see that the heat pump *P* and the heat engine E acting together constitute a heat engine operating in cycles and producing net work while exchanging heat only with one body at a single fixed temperature (t_2). This violates the Kelvin-Planck statement.

2. Let us now consider a perpetual motion machine of the second kind (E) which produces net work in cycle by exchanging heat with only one thermal energy reservoir (at t_1) and thus violates the Kelvin-Planck statement.

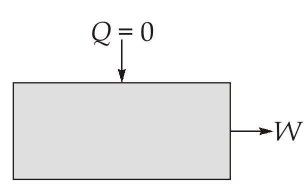


Violation of the Kelvin-Planck statement

Let us assume a cyclic heat pump P extracting heat Q_2 from a low temperature reservoir at t_2 and discharging heat to the high temperature reservoir at t_1 with the expenditure of work W equal to what the PMM2 delivers in a complete cycle. So E and P together constitute a heat pump working in cycle and producing the sole effect of transferring heat from a lower to a higher temperature body, thus violating the Clausius statement.

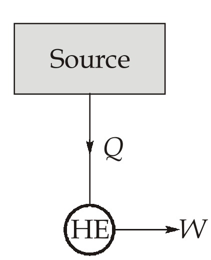
Q.1 (e) Solution:

PMM1: An imaginary device which would produce work continuously without absorbing any energy from its surrounding is called a perpetual motion machine of first kind (PMM1)



PMM2: Second law of thermodynamics restrict the thermal efficiency of a heat engine to less than one. It stipulates that some portion of the energy absorbs as heat from a source must always be rejected to a low temperature sink.

So, PMM2 violates this second law of thermodynamics. It is a device which would perform solely by absorbing energy as heat from a body without rejecting any heat to low temperature.



PMM3: It is impossible to construct a device which runs completely in the absence of friction.

Q.2 (a) Solution:

Assumptions:

(i) Heat transfer through tank walls is neglected.

(ii) The water in tank is incompressible and there is no change in pressure at inlet and exit.

(iii) Change in kinetic and potential energy is neglected.

(iv) Temperature of water in the tank is uniform.

The energy rate balance,

$$\frac{dU_{cv}}{dt} = \dot{Q} - W_{cv} + \dot{m}(h_1 - h_2) + \dot{m}\frac{(v_1^2 - v_2^2)}{2} + \dot{m}g(z_1 - z_2)$$

$$m_{cv}\frac{cdT}{dt} = \dot{Q} - \dot{W}_{cv} + \dot{m}(h_1 - h_2) \quad \text{...(i)}$$

Now enthalpy change, $h_1 - h_2 = (u_1 - u_2) + v(P_1 - P_2)$

$$h_1 - h_2 = c(T_1 - T_2)$$

Since the water is well mixed, the temperature at the exit equal the temperature of the overall quantity of liquid in the tank. So,

$$h_1 - h_2 = c(T_1 - T) \quad \text{...(ii)}$$

where, $T \rightarrow$ The uniform water temperature at time, t from equation (i),

$$\frac{dT}{dt} = \frac{(\dot{Q}_{cv} - \dot{W}_{cv})}{m_{cv} \cdot c} + \frac{\dot{m}}{m_{cv}}(T_1 - T) \quad \text{...(iii)}$$

$$\frac{dT}{\left(\frac{\dot{Q}_{cv} - \dot{W}_{cv}}{m_{cv}\, c}\right) + \frac{\dot{m}}{m_{cv}}(T_1 - T)} = dt$$

$$\frac{1}{\left(-\frac{\dot{m}}{m_{cv}}\right)}\ln\left[\frac{\dot{Q}_{cv} - \dot{W}_{cv}}{m_{cv} \cdot c} + \frac{\dot{m}}{m_{cv}}(T_1 - T)\right] = t + c_1$$

where, c_1 is constant.

Now,

$$\frac{\dot{Q}_{cv} - \dot{W}_{cv}}{m_{cv} \cdot c} + \frac{\dot{m}}{m_{cv}} T_1 - \frac{\dot{m}}{m_{cv}} T = c_2 e^{-\frac{\dot{m}}{m_{cv}} t}$$

$$T = -c_2 \frac{m_{cv}}{\dot{m}} e^{-\frac{\dot{m}}{m_{cv}} t} + \left(\frac{\dot{Q}_{cv} - \dot{W}_{cv}}{\dot{m} c} \right) + T_1 \quad \text{...(iv)}$$

Now boundary condition, at $t = 0, T = T_1$

$$c_2 = \left(\frac{\dot{Q}_{cv} - \dot{W}_{cv}}{\dot{m} c} \times \frac{\dot{m}}{m_{cv}} \right) = \frac{\dot{Q}_{cv} - \dot{W}_{cv}}{m_{cv} \cdot c}$$

Now putting value in equation (iv),

$$T = -\left(\frac{\dot{Q}_{cv} - \dot{W}_{cv}}{m_{cv} \cdot c} \times \frac{m_{cv}}{\dot{m}} \right) e^{\frac{-\dot{m}}{m_{cv}} t} + \left(\frac{\dot{Q}_{cv} - \dot{W}_{cv}}{\dot{m} c} \right) + T_1$$

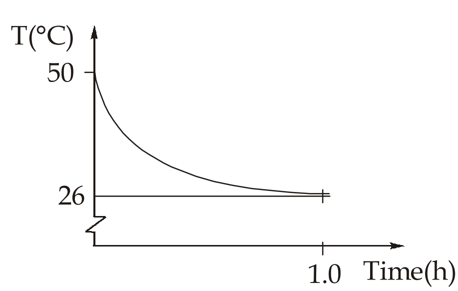
$$T = T_1 + \frac{\dot{Q}_{cv} - \dot{W}_{cv}}{\dot{m} c} \left[1 - e^{-\frac{\dot{m}}{m_{cv}} t} \right] \quad \text{...(v)}$$

Now putting values in equation (v) we get,

$$T = 50 + \frac{(-7.5 - (-0.5))}{\left(\frac{250}{3600} \right) \times 4.2} \left[1 - e^{\frac{-250}{40} t} \right]$$

$$T = 50 - 24 \, [1 - e^{-6.25t}]$$

where t is in hour and T is in °C.



Q.2 (b)(i) Solution:

(i) No heat transfer with the surroundings.

(ii) The system is at an equilibrium state initially and finally.

(iii) No change in kinetic energy and potential energy between these two states.

Writing energy balance equation,

$$\Delta U + \Delta KE + \Delta PE = Q - W$$

Now $\Delta KE = 0, \Delta PE = 0, Q = 0$

$$\frac{W}{m} = -(u_2 - u_1) = -(u_g - u_f) = -u_{fg}$$

$$\frac{W}{m} = -1927.4 \text{ kJ/kg}$$

Here –ve sign indicates that the work input by stirring is greater in magnitude than the work done by the water as it expands.

Now: For entropy produced,

$$\text{Change in entropy, } \Delta S = \int_1^2 \left(\frac{\Delta Q}{T}\right)_b + S_{gen}$$

$$\frac{S_{gen}}{m} = s_2 - s_1 = s_g - s_f$$

$$\frac{S_{gen}}{m} = s_{fg}$$

$$\frac{S_{gen}}{m} = 4.9953 \text{ kJ/kgK}$$

Q.2 (b)(ii) Solution:

(i) Volume of tank is constant.

$$(v_2 = v_1)$$

Now, $v_2 = v_f + x_2 v_{fg}$

$$0.16544 = 0.0007259 + x_2(0.19254 - 0.0007259)$$

$$x_2 = 0.859$$

Thus, $s_2 = s_f + x_2 s_{fg} = 0.07188 + 0.859(0.87995)$

$$= 0.8278 \text{ kJ/kgK}$$

Then the entropy change of the refrigerant during the process,

$$\Delta S = m(s_2 - s_1) = (5 \text{ kg})\,(0.8278 - 1.0624)$$

$$\Delta S = -1.173 \text{ kJ/K}$$

Q.2 (b)(iii) Solution:

The efficiency of a Carnot engine is given by:

$$\eta = 1 - \frac{T_2}{T_1}$$

Let T_2 be decreased by ΔT with T_1 remaining the same,

$$\eta_1 = 1 - \frac{T_2 - \Delta T}{T_1}$$

If T_1 is increased by the same ΔT, T_2 remaining the same.

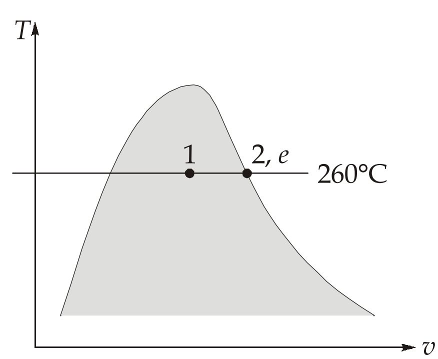
$$\eta_2 = 1 - \frac{T_2}{T_1 + \Delta T}$$

Then, $$\eta_1 - \eta_2 = \frac{T_2}{(T_1 + \Delta T)} - \frac{T_2 - \Delta T}{T_1} = \frac{(T_1 - T_2)\Delta T + \Delta T^2}{T_1(T_1 + \Delta T)}$$

Since, $T_1 > T_2$, $(\eta_1 - \eta_2) > 0$

So, the more effective way to increase the cycle efficiency is to decrease T_2.

Q.2 (c) Solution:



Assumptions:

1. For the control volume, $\dot{W}_{cv} = 0$ and kinetic and potential energy effects are neglected.
2. At the exit the state remains constant.
3. The initial and final states of the mass within the vessel are equilibrium states.

Since there is a single exit and not inlet, the mass rate balance, takes the form

$$\frac{dm_{cv}}{dt} = -\dot{m}_e$$

with assumption 1, the energy rate balance, reduces to

$$\frac{dU_{cv}}{dt} = \dot{Q}_{cv} - \dot{m}_e h_e$$

Combining the mass and energy rate balances results in,

$$\frac{dU_{cv}}{dt} = \dot{Q}_{cv} + h_e \frac{dm_{cv}}{dt} \quad \text{...(i)}$$

By assumption 2, the specific enthalpy at the exit is constant. Accordingly, integration of the equation (i) gives,

$$\Delta U_{cv} = Q_{cv} + h_e \Delta m_{cv}$$

Solving for the heat transfer Q_{cv},

$$Q_{cv} = \Delta U_{cv} - h_e \Delta m_{cv}$$

or

$$Q_{cv} = (m_2 u_2 - m_1 u_2) - h_e(m_2 - m_1) \quad \text{... (ii)}$$

where m_1 and m_2 denote respectively, the initial and final amounts of mass within the tank.

The terms u_1 and m_1 of the foregoing equation can be evaluated with property values at 260°C and the given value for quality. Thus

$$\begin{aligned} u_1 &= u_f + x_1(u_g - u_f) \\ &= 1128.4 + (0.7)\,(2599.0 - 1128.4) = 2157.82 \text{ kJ/kg} \end{aligned}$$

Also,

$$\begin{aligned} v_1 &= v_f + x_1(v_g - u_f) \\ &= 1.2755 \times 10^{-3} + (0.7)\,(0.04221 - 1.2755 \times 10^{-3}) \\ &= 29.93 \times 10^{-3} \text{ m}^3/\text{kg} \end{aligned}$$

Using the specific volume v_1, the mass initially contained in the tank is

$$m_1 = \frac{V}{v_1} = \frac{0.85 \text{ m}^3}{\left(29.93 \times 10^{-3} \text{ m}^3/\text{kg}\right)} = 28.4 \text{ kg}$$

The final state of the mass in the tank is saturated vapor at 260°C, so gives

$$\begin{aligned} u_2 &= u_{g(260°C)} = 2599.0 \text{ kJ/kg} \\ v_2 &= v_{g(260°C)} = 42.21 \times 10^{-3} \text{ m}^3/\text{kg} \end{aligned}$$

The mass contained within the tank at the end of the process is

$$m_2 = \frac{V}{v_2} = \frac{0.85\ \text{m}^3}{\left(42.21 \times 10^{-3}\ \text{m}^3/\text{kg}\right)} = 20.14\ \text{kg}$$

Also, $h_e = h_{g(260°C)} = 2796.6$ kJ/kg.

Substituting values into the equation (ii) for the heat transfer yields

$$Q_{cv} = (20.14)(2599.0) - (28.4)(2157.82) - 2796.6(20.14 - 28.4)$$
$$= 14161.688\ \text{kJ}$$

Q.3 (a) Solution:

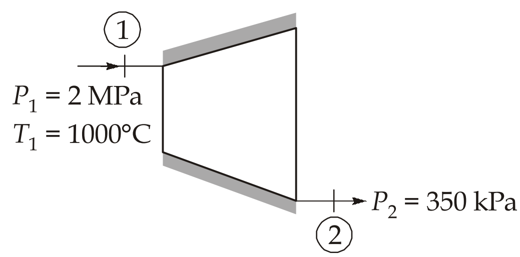
Given: Inlet temperature, $T_1 = 1273$ K

Mass flow rate, $\dot{m} = 0.5$ kg/s

Power developed, $\dot{W} = 120$ kW

$\gamma = 1.67$

$c_p = 0.52$ kJ/kgK



(a) Temperature of the argon at the turbine exit.

Using SFEE,

$$\dot{m}\left\{h_1 + (KE)_1 + (PE)_1\right\} + \dot{Q} = \dot{m}\left\{h_2 + (KE)_2 + (PE)_2\right\} + \dot{W} \quad \text{...(1)}$$

Neglecting change in KE and PE.

$$\dot{Q} = 0 \quad \{\because \text{insulated turbine}\}$$

From eq. (1)

$$h_1 = h_2 + \frac{\dot{W}}{\dot{m}}$$

$$c_p T_1 = c_p T_2 + \frac{\dot{W}}{\dot{m}}$$

$$T_2 = 1273 - \frac{120}{0.5 \times 0.52}$$

$$T_2 = 811.46\ \text{K}$$

We know that, $$c_p = \frac{\gamma R}{\gamma - 1}$$

$\therefore$ $$R = 0.2086\ \text{kJ/kgK}$$

(b) Irreversibility

Ideal work by turbine per unit mass,

$$w_I = \psi_1 - \psi_2$$
$$= (h_1 - T_0 s_1) - (h_2 - T_0 s_2)$$
$$= (h_1 - h_2) + T_0(s_2 - s_1)$$
$$= c_p[T_1 - T_2] + 293\left[c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)\right]$$

$$w_I = 0.52[1273 - 811.46] + 293\left[0.52\ln\left(\frac{811.46}{1273}\right) - 0.2086\ln\left(\frac{350}{2000}\right)\right]$$

$$w_I = 240 + 37.922$$
$$w_I = 277.922 \text{ kJ/kg}$$

Given, $\dot{m} = 0.5$ kg/s

Ideal power output, $\dot{W}_I = 0.5 \times 277.922 = 138.961$ kW

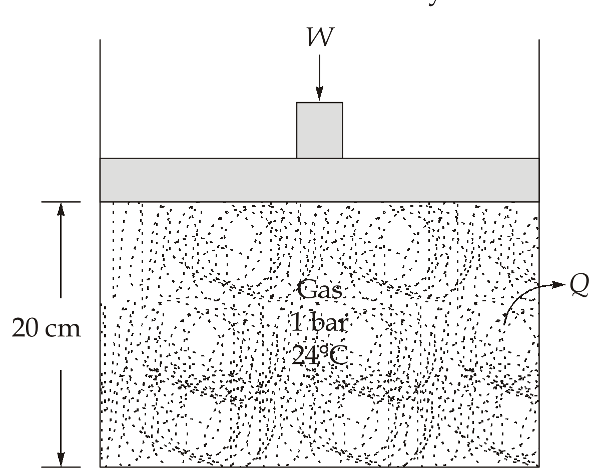
$\therefore$ Irreversibility, $I = \dot{W}_I - \dot{W}_{act} = 138.961 - 120 = 18.96$ kW

(c) Second law efficiency

$$\eta_{II} = \frac{\dot{W}_{act}}{\dot{W}_I} \times 100 = \frac{120}{138.961} \times 100 = 86.35\%$$

Q.3 (b) Solution:

- $\Delta PE \simeq 0$ and $\Delta KE \simeq 0$
- Absence of friction between piston and cylinder
- Closed system since no mass crosses the boundary.



The energy balance for the system can be expressed as,

$$E_{in} - E_{out} = \Delta E_{system}$$

Net energy transfer by heat, work and mass = change in internal energy, KE and PE

$$W_{in} - Q_{out} = \Delta PE + \Delta KE + \Delta U$$

$$= \Delta U = mC_v(T_2 - T_1) \qquad [\Delta KE \simeq 0 \text{ and } \Delta PE \simeq 0]$$

$$= 0 \qquad (\text{since } T_2 = T_1)$$

Thus, the amount of heat transfer is equal to the boundary work input,

$$Q_{out} = W_{in} = 0.1 \text{ kJ}$$ Answer

$$\text{Initial volume, } V_1 = \frac{\pi}{4} D_1^2 L_1$$

$$= \frac{\pi}{4} \times (0.12)^2 \times 0.2 = 0.002262 \text{ m}^3$$

The relation for isothermal work done can be found out as

$$W_{in} = -P_1 V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$0.1 = -100 \times 0.002262 \ln\left(\frac{V_2}{0.002262}\right)$$

$$V_2 = 0.001454 \text{ m}^3$$

final pressure can be obtained from following

$$P_1 V_1 = P_2 V_2$$

$$P_2 = \frac{P_1 V_1}{V_2} = \frac{100 \times 0.002262}{0.001454} = 155.57 \text{ kPa}$$ Answer

The final position of the piston can be determined from final volume.

$$V_2 = \frac{\pi}{4} D_2^2 L_2$$

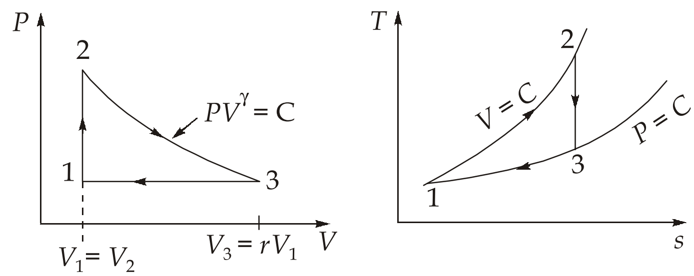
$$0.001454 = \frac{\pi}{4} \times (0.12)^2 \times L_2$$

$$L_2 = 0.1285 \text{ m}$$

$$\Delta L = L_1 - L_2 = 0.2 - 0.1285$$

$$= 0.07146 \text{ m} = 7.1 \text{ cm}$$ Answer

Q.3 (c) Solution:



$$Q_1 = \text{heat addition in process } 1 - 2$$

$$Q_2 = \text{heat rejection in process } 3 - 1$$

$$\eta = \frac{W}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

If it is ideal gas (given)

$$Q_1 = c_v(T_2 - T_1)$$

$$Q_2 = c_p(T_3 - T_1)$$

$$\eta = 1 - \frac{c_p(T_3 - T_1)}{c_v(T_2 - T_1)}$$

$$\eta = 1 - \gamma\left(\frac{T_3 - T_1}{T_2 - T_1}\right) \quad ...(1)$$

Process 3–1 (constant pressure),

$$\frac{T_3}{T_1} = \frac{V_3}{V_1} = r \quad ...(2)$$

Process 1 – 2 (constant volume),

$$\frac{T_2}{T_1} = \frac{P_2}{P_1} \quad ...(3)$$

We know that, $P_1 = P_3$

Process 2 – 3, $PV^{\gamma} = C$

$$\left(\frac{P_2}{P_3}\right) = \left(\frac{P_2}{P_1}\right) = \left(\frac{V_3}{V_2}\right)^{\gamma} = \left(\frac{rV_1}{V_1}\right)^{\gamma} = r^{\gamma} \quad ...(4)$$

From (1), (2), (3) and (4)

$$\eta = 1-\frac{\gamma[T_3-T_1]}{[T_2-T_1]} = 1-\frac{\gamma T_1\left[\frac{T_3}{T_1}-1\right]}{T_1\left[\frac{T_2}{T_1}-1\right]}$$

$$= 1-\frac{\gamma[r-1]}{\left[\frac{P_2}{P_1}-1\right]} = 1-\frac{\gamma[r-1]}{[r^{\gamma}-1]}$$

$$\eta = \frac{(r^{\gamma}-1)-\gamma(r-1)}{(r^{\gamma}-1)}$$

for $\gamma = 1.4$ and $r = 7$

$$\eta = \frac{7^{1.4}-1-1.4(7-1)}{7^{1.4}-1} = 0.41 \text{ or } 41\%$$

Q.4 (a)(i) Solution:

Assumptions:

(i) Steady state for the control volume.
(ii) Air is an ideal gas.
(iii) Change in potential energy is neglected.
(iv) Heat transfer through the diffuser wall is negligible.
(v) Kinetic energy at exit is neglected.
(vi) There are no work interactions.

(a) For mass flow rate,

$$v_1 = \frac{RT_1}{P_1} = \frac{0.287\times 283}{80} = 1.015 \text{ m}^3/\text{kg}$$

Then, $$\dot{m} = \frac{1}{v_1}\times C_1 A_1 = \frac{1}{1.015}\times 200\times 0.4 = 78.8 \text{ kg/s}$$

Since flow is steady, the mass flow rate through the entire diffuser remains constant.

(b) The energy balance equation,

$$\frac{dE}{dt} = \dot{m}(h_1 - h_2) + \dot{Q} - \dot{W} + \dot{m}\frac{(C_1^2 - C_2^2)}{2} + \dot{m}g(z_1 - z_2)$$

Now for steady state,

$$0 = \dot{m}(h_1 - h_2) + \dot{m}\frac{(C_1^2 - C_2^2)}{2}$$

$$h_2 = h_1 + \frac{C_1^2 - C_2^2}{2}$$

The exit velocity at diffuser is usually small compared with the inlet velocity,

Now $$h_2 = h_{1@283\text{K}} + \frac{C_1^2 - C_2^2}{2} = 283 + \frac{(200)^2 - 0^2}{2000}$$

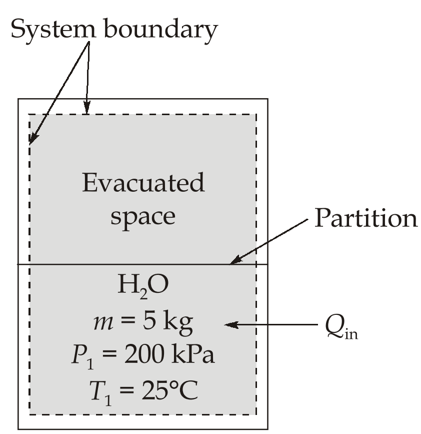
$$h_2 = 303 \text{ kJ/kg}$$

Now from table, the temperature corresponding to this enthalpy value is 303 K.

Temperature at exit, $T_2 = 303$ K

Q.4 (a)(ii) Solution:

(i) System is stationary neglect kinetic and potential energy changes.
(ii) The volume of rigid tank is constant.
(iii) The water temperature remains constant during the process.
(iv) There is no electrical, shaft or other work transfer.



(a) Initially the water in the tank is compressed liquid since its pressure (200 kPa) is greater than the saturation pressure at 25°C (3.1698 kPa). So approximate the compressed liquid as saturated liquid at the given temperature,

$$v_1 = v_{f@25°\text{C}} = 0.001 \text{ m}^3/\text{kg} \simeq 0.001 \text{ m}^3/\text{kg}$$

Then the initial volume of water,

$$V_1 = m.v_1 = 5 \times 0.001 = 0.005 \text{ m}^3$$

The total volume of the tank is twice this amount:

$$V_{tank} = 2(0.005) = 0.01 \text{ m}^3$$

(b) At the final state, the specific volume of the water,

$$v_2 = \frac{V_2}{m} = \frac{0.01}{5} = 0.002 \text{ m}^3/\text{kg}$$

Since, $v_{f@25°} < v_2 < v_g$, the water is a saturated liquid vapour mixture at the final state, and thus the pressure is the saturation pressure at 25°C,

$$P_2 = P_{sat@25°C} = 3.1698 \text{ kPa}$$

(c) The energy balance on the system,

$$E_{in} - E_{out} = \Delta E_{system}$$

$$Q_{in} = \Delta U + W$$

$$Q_{in} = \Delta U = m(u_2 - u_1) \quad \text{...(i)}$$

$$u_1 \simeq u_{f@25°C} = 104.83 \text{ kJ/kg}$$

The quality at the final state is determined from the specific volume,

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002 - 0.001}{43.34 - 0.001} = 2.3 \times 10^{-5}$$

$$u_2 = u_f + x_2 u_{fg}$$

$$= 104.83 + 2.3 \times 10^{-5}(2304.3) = 104.88 \text{ kJ/kg}$$

Now from equation (i),

$$Q_{in} = 5 \times (104.88 - 104.83)$$

$$Q_{in} = 0.25 \text{ kJ}$$

Q.4 (b) Solution:

Given:

$$m_p = 900 \text{ kg}$$

$$\rho_p = 1.12 \text{ gm/cc}$$

$$\Rightarrow \quad \rho_p = 1.12 \times \frac{10^{-3}}{10^{-6}} \text{ kg/m}^3$$

$$\frac{m_p}{V_p} = 1120 \text{ kg/m}^3$$

$$\Rightarrow \quad V_p = 0.8035 \text{ m}^3$$

$$\therefore \quad W_p = P \times V = 20 \times 100 \times 0.8035$$

$$= 1607 \text{ kJ}$$

1 - 2 air, $\quad PV = mRT$

$$\Rightarrow \quad \frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

$$\Rightarrow \quad \frac{200 \times V_a}{T_1} = \frac{20 \times (V_a + V_p)}{T_2}$$

$$\Rightarrow \quad \frac{T_2}{T_1} = \frac{20}{200}\left[\frac{V_a + V_p}{V_a}\right]$$

$$\Rightarrow \quad \frac{T_2}{T_1} = 0.1\left[\frac{V_a + V_p}{V_a}\right] \quad \text{... (i)}$$

$\Rightarrow$ For air, $\quad \delta Q = dU + \delta W$

$$0 = m_a c_V (T_2 - T_1) + (1607)$$

$$0 = \frac{P_1 V_a}{R T_1} \times \frac{R}{\gamma - 1}(T_2 - T_1) + 1607$$

$$-1607 = 200 \times 100 \frac{V_a}{\gamma - 1}\left\{\frac{T_2}{T_1} - 1\right\}$$

$$\Rightarrow \quad -0.08035\left(\frac{\gamma - 1}{V_a}\right) = \frac{T_2}{T_1} - 1$$

$$\therefore \quad \frac{T_2}{T_1} = 1 - 0.08035\left(\frac{\gamma - 1}{V_a}\right) \quad \text{... (ii)}$$

Equating (i) and (ii)

$$0.1\left[\frac{V_a + V_p}{V_a}\right] = 1 - 0.08035\left[\frac{\gamma - 1}{V_a}\right]$$

$$\frac{0.1 V_a + 0.1 \times 0.8035}{V_a} = \frac{V_a - 0.08035(\gamma - 1)}{V_a}$$

$$0.08035 + 0.08035(\gamma - 1) = V_a - 0.1 V_a$$

$$0.08035\,(1 + \gamma - 1) = 0.9 V_a$$

$$V_a = \frac{0.08035\gamma}{0.9} = \frac{0.08035}{0.9} \times 1.4$$

$$V_a = 0.125 \text{ m}^3$$

Q.4 (c) Solution:

Assumptions:

(i) Both water and the iron block are incompressible.

(ii) Constant specific heats at room temperature can be used for both the water and the iron.

(iii) Neglect change in kinetic energy and potential energy.

(iv) System is well insulated and thus there is no heat transfer.

System = (Iron block + Water)

(a) Now,

Heat given by iron block = Heat taken by water

$$\left[mc\left(T_i - T_f\right)\right]_{\text{iron}} = \left[mc\left(T_f - T_i\right)\right]_{\text{water}}$$

$$5 \times 0.45(350 - T_f) = 100 \times 4.18 \times (T_f - 30)$$

Final equilibrium temperature, T_f = 31.7°C

(b) Now for exergy,

$$X = (U - U_0) - T_0(S - S_0) + P_0(V - V_0)$$

$$= mc\left(T - T_0\right) - T_0\left[mc\ln\left(\frac{T}{T_0}\right)\right]$$

(Since incompressible. so, $V - V_0 = 0$)

$$= mc\left(T - T_0 - T_0 \ln\frac{T}{T_0}\right)$$

Total exergy at initial state, $X_{1,\text{ total}} = X_{1,\text{ iron}} + X_{1,\text{ water}}$

$$= 5\times 0.45\left(623 - 293 - 293\ln\frac{623}{293}\right) + 100\times 4.18\left(303 - 293 - 293\ln\frac{303}{293}\right)$$

$$= 245.2 + 69.8$$

$$X_{1,\text{ total}} = 315 \text{ kJ}$$

Similarly, total exergy at final state,

$$X_{2,\ total} = X_{2,\ iron} + X_{2,\ water}$$

$$= 5\times0.45\left(304.7-293-293\ln\left(\frac{304.7}{293}\right)\right)+100\times4.18\left(304.7-293-293\ln\left(\frac{304.7}{293}\right)\right)$$

$$= 0.5 + 95.1$$

$$X_{2,\ total} = 95.6 \text{ kJ}$$

(c) The water work potential is equivalent to the exergy destroyed.

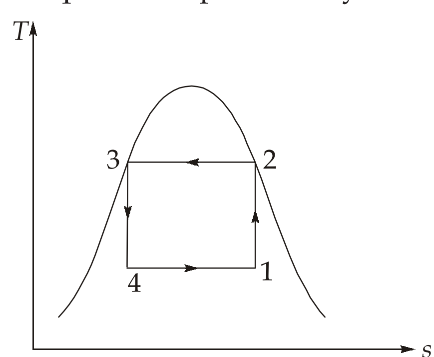
$$X_{destroyed} = X_{1,\ total} - X_{2,\ total}$$

$$X_{destroyed} = 219.4 \text{ kJ}$$

Section B : Refrigeration and Air Conditioning

Q.5 (a) Solution:

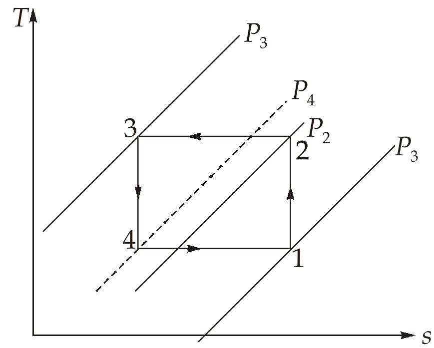
In the reversed Carnot cycle with vapour as refrigerant, the isothermal processes of condensation and evaporation are internally reversible process, and they are easily achievable in practice although there may be some problem in having only partial evaporation. However, isentropic compression and expansion processes have some limitations. In brief, it is difficult to design an expander to handle a mixture of largely liquid and partly vapour for the process 3-4. Also, because of the internal irreversibilities in the compressor and the expander, the actual COP of the Carnot cycle is very low, though the ideal cycle COP is the maximum. A cycle which is closest to the reversed Carnot vapour cycle is the vapour compression cycle.



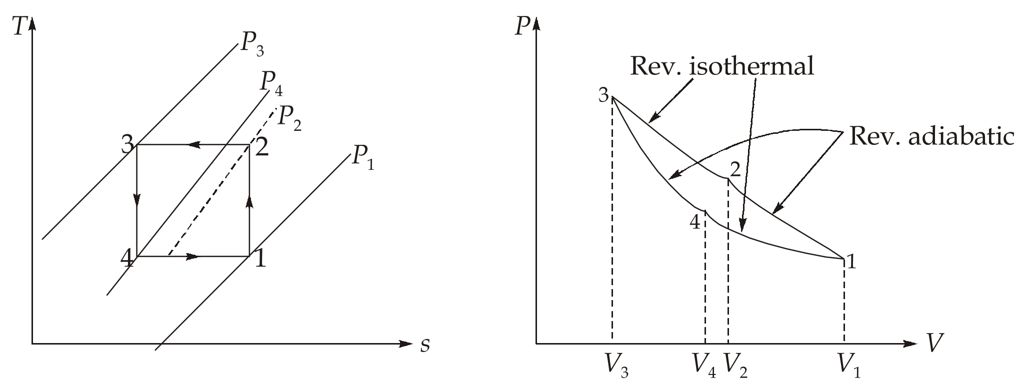
There are two drawbacks of reversed Carnot cycle with gas as a refrigerant.

(i) Firstly, it is not possible to devise, in practice, isothermal processes of heat absorption and rejection, 4-1 and 2-3 in figure with gas as the working substance. These are impractical as those will be infinitely slow.

(ii) Secondly, the cycle on P-V diagram is very narrow since the volume is changing both during the reversible isothermal and reversible adiabatic process. Drawn correctly to scale, the Carnot P-V diagram is much thin. As a result, the stroke volume of the cylinder is very large. The cycle, therefore, suffers from poor actual COP as a result of irreversibilities of the compressor and expander.



Q.5 (b) Solution:



Process 1-2: Isentropic compression,

$$Q = 0$$

$$W_{1\text{-}2} = \frac{P_2V_2 - P_1V_1}{\gamma - 1} = \frac{mR(T_2 - T_1)}{\gamma - 1}$$

Process 2-3: Isothermal compression and heat rejection.

$$Q_{2\text{-}3} = W_{2\text{-}3}$$

Assume air as an ideal gas. So, $\Delta U = 0$ for isothermal process.

$$Q_{2\text{-}3} = P_2V_2 \ln\left(\frac{V_2}{V_3}\right) = mRT_2 \ln\left(\frac{V_2}{V_3}\right)$$

Process 3-4: Isentropic expansion,

$$Q = 0$$

$$W_{3\text{-}4} = \frac{P_3V_3 - P_4V_4}{\gamma - 1} = \frac{mR(T_3 - T_4)}{\gamma - 1}$$

Process 4-1: Isothermal expansion and heat addition.

$$Q_{4\text{-}1} = W_{4\text{-}1}$$

$$= P_4V_4 \ln\left(\frac{V_1}{V_4}\right) = mRT_1 \ln\left(\frac{V_1}{V_4}\right)$$

Now refrigerating effect, $Q_{4\text{-}1} = mRT_1 \ln\left(\frac{V_1}{V_4}\right)$

Now for process 1-2 and 3-4,

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}$$

$$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\left(\frac{V_1}{V_2}\right)^{\gamma-1} = \left(\frac{V_4}{V_3}\right)^{\gamma-1}$$

Assume, $\frac{V_4}{V_3} = \frac{V_1}{V_2} = r$ (Compression ratio)

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = r^{\gamma-1}$$

Now, COP for refrigerator $= \frac{\text{Refrigerating effect}}{\text{Work input}} = \frac{mRT_1 \ln\left(\frac{V_1}{V_4}\right)}{mRT_2 \ln\left(\frac{V_2}{V_3}\right) - mRT_1 \ln\left(\frac{V_1}{V_4}\right)}$

$$= \frac{T_1}{T_2 - T_1} \qquad \left\{\frac{V_1}{V_4} = \frac{V_2}{V_3}\right\}$$

$$= \frac{1}{\frac{T_2}{T_1} - 1}$$

$$\text{COP} = \frac{1}{r^{\gamma-1} - 1}.$$ Hence proved.

Q.5 (c) Solution:

The maximum COP = $\frac{T_e}{T_g}\left(\frac{T_g - T_c}{T_c - T_e}\right)$

where,

$$
\begin{aligned}
T_g &= \text{Saturation temperature of steam at 2 bar} \\
&= 119.62°\text{C} = 392.62 \text{ K} \\
T_c &= 30 + 273 = 303 \text{ K} \\
T_e &= -5 + 273 = 268 \text{ K}
\end{aligned}
$$

$$\text{Maximum COP} = \frac{268}{392.62}\left(\frac{392.62 - 303}{303 - 268}\right) = 1.75$$

$$\text{Actual COP} = 1.75 \times 0.7 = 1.225$$

$$\text{20 tons load of refrigeration} = 20 \times 3.5 = 70 \text{ kJ/s}$$

$$\text{Actual COP} = \frac{\text{Refrigeration load}}{\text{Actual heat supplied}}$$

$$\therefore \quad \text{Actual heat supplied} = \frac{70}{1.225} = 57.15 \text{ kJ/ s} = 57.15 \text{ kW}$$

$$\text{Steam required per hour} = \frac{57.15 \times 3600}{x \cdot h_{fg}}$$

$$\text{Steam requirement/hour} = \frac{57.15 \times 3600}{0.9 \times 2200} = 103.91 \text{ kg/ hour}$$

Note: Only latent heat of steam is used for heating purposes.

Q.5 (d) Solution:

$$P_s \text{ (Suction pressure)} = 3.62 - 0.2 = 3.42 \text{ bar}$$

$$P_d \text{(Discharge pressure)} = 9.6 + 0.4 = 10 \text{ bar}$$

$$\text{Now, Volumetric efficiency, } \eta_v = 1 - C\left[\left(\frac{P_d}{P_s}\right)^{1/n} - 1\right] = 1 - 0.05\left[\left(\frac{10}{3.42}\right)^{1/1.15} - 1\right]$$

$$= 1 - 0.077 = 0.923 = 92.3\%$$

Piston displacement per cylinder,

$$V_P = \frac{\pi}{4} D^2 L N \eta_v = \dot{m} \times v_2 \quad \text{...(1)}$$

where, $\dot{m}$ is the mass flow of refrigerant per cylinder per minute

$$\dot{m} \times v_2 = \left(\frac{0.207}{2} \times 60\right) \times 0.0525$$

$$= 0.326 \text{ m}^3/\text{min-cylinder}$$

From eq. (1),

$$\therefore \quad \frac{\pi}{4} D^2 \times (0.8D) N \eta_v = 0.326 \quad \text{...(2)}$$

Given: $2LN = 3 \times 60 = 180$ m/min

$$\therefore \quad 2 \times 0.8DN = 180$$

$$N = \frac{180}{1.6D} = \frac{112.5}{D} \quad \text{...(3)}$$

From equation (2) and (3),

$$\frac{\pi}{4} D^2 \times 0.8D \times \frac{112.5}{D} \times 0.923 = 0.326$$

$$D^2 = \frac{0.326}{0.923 \times 112.5 \times 0.8} \times \frac{4}{\pi} = \frac{1}{200.132}$$

$$D = \frac{1}{14.1468} = 0.07068 \text{ m} = 7.068 \text{ cm}$$

$$N = \frac{112.5}{0.07068} = 1591.68 \text{ rpm}$$

$$L = 0.8D = 0.8 \times 7.068 = 5.6544 \text{ cm}$$

Q.5 (e) Solution:

For air at 35°C DBT and 60% RH

$$\phi = \frac{P_v}{P_{vs}}$$

$$0.6 = \frac{P_v}{5.6280}$$

$$P_v = 3.3768 \text{ kPa}$$

Now, Specific humidity, $\omega = \dfrac{0.622 \times P_v}{P_t - P_v} \times 1000$ grams/ kg of d.a.

$$= \frac{0.622 \times 0.033768}{1.01300 - 0.033768} \times 1000 = 21.45 \text{ g/kg of d.a.}$$

After removing 5 grams of water vapour, the specific humidity becomes:

$$\omega' = 21.45 - 5$$
$$= 16.45 \text{ g/kg of d.a.}$$

Now, the air is at 25°C DBT and 16.45 g/kg of d.a. of specific humidity. The partial pressure of water vapour at this condition can be calculated as follows:

$$\omega' = \frac{0.622 \times P_v'}{P_t - P_v'}$$

$$0.01645 = \frac{0.622 \times P_v'}{1.013 - P_v'}$$

$$P_v' = 0.0261 \text{ bar}$$
$$P_v' = 2.61 \text{ kPa}$$

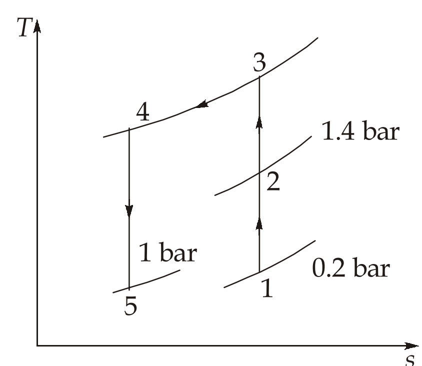
(i) Relative humidity, $\phi' = \dfrac{P_v'}{P_{vs}'} = \dfrac{2.61}{3.1693} = 0.8235 = 82.35\%$

(ii) Dew point temperature is the saturation temperature at the pressure,

$$P_v' = 2.61 \text{ kPa}$$
$$\text{DPT} = 22°\text{C} \quad \text{(from given table)}$$

(iii) Degree of saturation, $\mu = \phi\left(\dfrac{P_t - P_{vs}}{P_t - P_v'}\right) = 0.8235 \times \left(\dfrac{101.3 - 3.1693}{101.3 - 2.61}\right) = 0.8188$

Q.6 (a) Solution:



$$T_1 = 225 \text{ K}, P_1 = 0.2 \text{ bar}$$

Process 1 to 2,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = 225\left(\frac{1.4}{0.2}\right)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 392.32 \text{K}$$

Since secondary compressor and turbine are mounted on same shaft so

$$W_T = W_{c2}$$

Given, $W_{4-5} = W_{2-3}$

$$\dot{m}c_p(T_4 - T_5) = \dot{m}c_p(T_3 - T_2)$$

$$(T_4 - T_5) = (T_3 - T_2) \qquad \text{...(i)}$$

Now $$T_4 = T_5\left(\frac{P_4}{P_5}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_4 = 275\left(\frac{P_4}{1}\right)^{\frac{0.1-1}{1.4}} \qquad \text{...(ii)}$$

Similarly, $$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_3 = 392.32\left(\frac{P_3}{1.4}\right)^{\frac{1.4-1}{1.4}}$$

$$T_3 = 356.36(P_3)^{\frac{1.4-1}{1.4}}$$

Put value of T_3 and T_4 in eq. (i)

$$275(P_4)^{0.4/1.4} - 275 = 356.36(P_3)^{0.4/1.4} - 392.32$$

$$P_3 = P_4 = 3.6 \text{ bar}$$

Now from eq. (ii)

Temperature at the inlet of turbine,

$$T_4 = 275(3.6)^{0.4/1.4}$$

$$T_4 = 396.53 \text{ K}$$

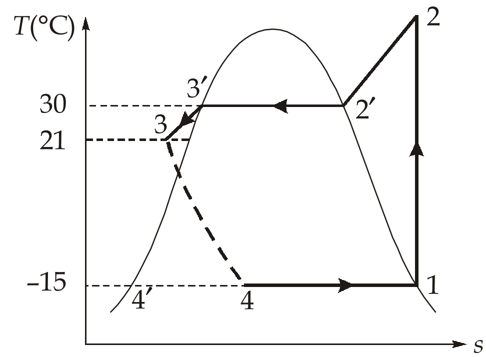
Q.6 (b) Solution:

Compression process 1 - 2 is isentropic

So, $$s_1 = s_2$$

$$5.8285 = 5.2623 + 3.252\ln\left(\frac{T_2}{T_2'}\right)$$

$$\ln\left(\frac{T_2}{303}\right) = 0.1740$$



$$T_2 = 360.587 \text{ K}$$

Now,
$$h_2 = h_2' + c_{P_v}(T_2 - T_2')$$
$$= 1485.93 + 3.252(360.587 - 303)$$
$$h_2 = 1673.20 \text{ kJ/kg}$$

Again,
$$h_3 = h_3' - c_{P_l}(T_3' - T_3)$$
$$= 342.08 - 4.843(303 - 294)$$
$$h_3 = 298.493 \text{ kJ/kg}$$

And expansion process 3 - 4 is isenthalpic so,

$$h_3 = h_4 = 298.493 \text{ kJ/kg}$$

(a)Displacement volume/min, $V_D = \frac{\pi}{4} \times D^2 \times L \times N \times \eta_{vol} \times \text{Number of cylinder}$

$$= \frac{\pi \times 15 \times 15 \times 14 \times 250 \times 0.79 \times 2}{4 \times 100 \times 100 \times 100} = 0.977 \text{ m}^3/\text{min}$$

Therefore, mass of refrigerant flowing per min,

$$\dot{m} = \frac{V_D}{v_1} = \frac{0.977}{0.50905} = 1.9193 \text{ kg/ min}$$

(b) The tons of Refrigeration,

$$\text{RE} = \frac{\dot{m}(h_1 - h_4)}{210} = \frac{1.9193 \times (1444.19 - 298.493)}{210}$$
$$= 10.47 \text{ tons of refrigeration}$$

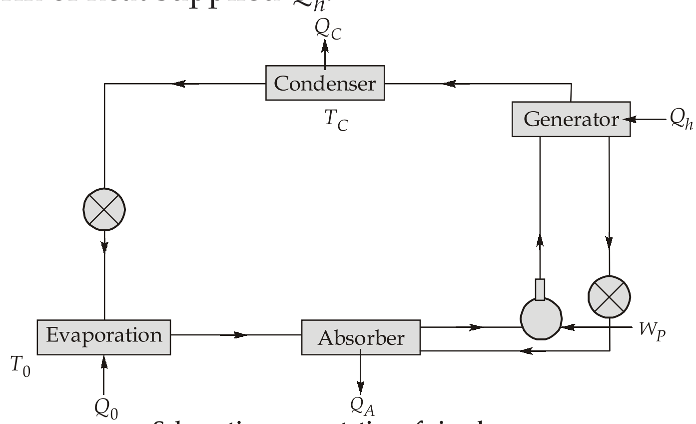
(c) Coefficient of performance, $\text{COP} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1444.19 - 298.493}{1673.20 - 1444.19} = \frac{1145.697}{229.01} = 5.00$

Q.6 (c)(i) Solution:

A simple vapour absorption system, consists of a condenser, an expansion device and an evaporator as in the vapour compression system and in addition, an absorber, a pump, a generator and a pressure reducing valve to replace the compressor. The schematic representation of the system is shown in figure in which various components of the system are arranged according to their pressures and temperatures. The refrigerating effect is shown as Q_0 at temperature T_0 and the heat rejected in the condenser as Q_C at temperature $T_C = T_K$ of the environment. The compressor work is replaced by the heat supplied in the generator Q_h plus pump work W_P. Cooling must be done in the absorber to remove the latent heat of the refrigerant vapour as it changes into the liquid state by absorption by the weak solution. Heat rejected in the absorber be Q_A at absorber temperature $T_A = T_K$. Then the energy balance of the system,

$$Q_0 + W_P + Q_h = Q_C + Q_A$$

The pump work $W_P = -\int vdP$ is very small compared to compressor work in the vapour compression system, as the specific volume v of the liquid is extremely small compared to that of the vapour ($v_f << v_g$). The energy consumption of the system is mainly in the generator in the form of heat supplied Q_h.



Schematic representation of simple vapour absorption system

In the vapour-absorption system, the function of the compressor is accomplished in a three step process by the use of the absorber, pump and generator. Functions of these devices are given below:

(i) **Absorber:** Absorption of the refrigerant vapour by its weak or poor solution in a suitable absorbent or adsorbent, forming a strong or rich solution of the refrigerant in the absorbent/adsorbent.

(ii) **Pump:** Pumping of the rich solution raising its pressure to the condenser pressure.

(iii) **Generator:** Distillation of the vapour from the rich solution leaving the poor solution for recycling.

Q.6 (c)(ii) Solution:

Electrolux principle works on 3-fluid system. There is no solution circulation pump. Total pressure is the same throughout the system. The third fluid remains mainly in the evaporator thus reducing partial pressure of refrigerant to enable it to evaporate at low pressure and hence low temperature.

Liquid NH_3 evaporates in the evaporator in the presence of H_2. Hydrogen is chosen as it is non-corrosive and insoluble in water. A thermosyphon bubble pump is used to lift the weak aqua from the generator to the separator. The discharge tube from the generator is extended down below the liquid level in the generator. The bubble rise and carry slugs of weak NH_3 – H_2O solution into the separator.

Two U-bends are provided as vapour-locks to prevent H_2 from getting into the high side or solution circuit. Partial pressure of H_2 provides the pressure difference of NH_3 between the condenser and the evaporator.

In condenser pure NH_3 vapour pressure = Total pressure

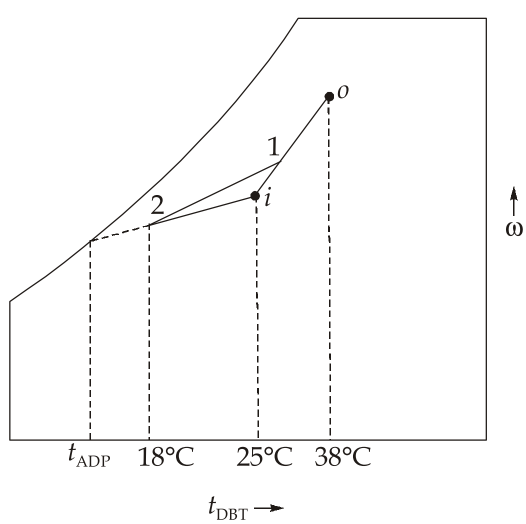
In evaporator NH_3 vapour pressure = Total pressure – Partial pressure of H_2.

Q.7 (a) Solution:

$$\text{RSHF} = \frac{RSH}{RSH + RLH} = \frac{310}{310+100} = 0.756$$

Steps:

1. Draw a line joining 0.756 SHF to alignment circle.
2. Draw a line passing through point (i) and parallel to previous line.
3. Draw a vertical give from 18°C DBT such that it intersects line passing through (i) at point 2.
4. The point 2 is the exit of cooling coil.



$$h_i = 50.5 \text{ kJ/kg d.a.,}$$
$$h_2 = 41.2 \text{ kJ/kg d.a.}$$
$$v_2 = 0.836 \text{ m}^3\text{/kg d.a.}$$
$$h_0 = 92 \text{ kJ/kg d.a.}$$

(a) Supply air quantity and volume flow rate

$$\dot{m}_{a_2} = \frac{RTH}{h_i - h_2} = \frac{410}{50.5 - 41.2} = 44.09 \text{ kg/ s}$$

$$\dot{Q}_{V_2} = \dot{m}_{a_2} \times v_2 = 44.09 \times 0.836$$
$$= 36.86 \text{ m}^3\text{/s}$$

(b) Quantity and volume flow rate of outdoor/exhaust air

$$\dot{m}_{a_0} = 0.1\dot{m}_{a_2} = 0.1 \times 44.09 = 4.41 \text{ kg/s}$$

$$\dot{Q}_{V_0} = \dot{m}_{a_0} v_0 = 44.09 \times 0.91 = 4.01 \text{ m}^3\text{/s}$$

Quantity and volume flow rate of return air,

$$\dot{m}_{a_i} = \dot{m}_{a_2} - \dot{m}_{a_0} = 44.09 - 4.41 = 39.68 \text{ kg/s}$$
$$\dot{Q}_{V_i} = \dot{m}_{a_i} \times v_i = 39.68 \times 0.86 = 34.05 \text{ m}^3\text{/s}$$

Here, $v_0 = 0.91 \text{ m}^3\text{/kg d.a.}$ and $v_i = 0.86 \text{ m}^3\text{/kg d.a.}$

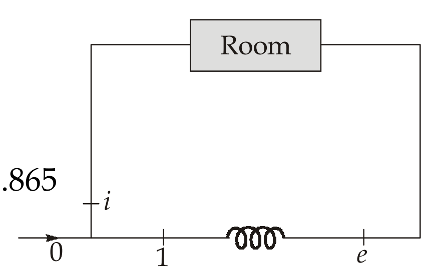
are the specific volumes of outdoor and indoor air respectively.

(c) State of air entering cooling coil.

$$t_1 = 0.9t_i + 0.1t_0 = 0.9(25) + 0.1(38)$$
$$= 26.3°\text{C}$$

$t_1' = 19.2°\text{C}$ at 26.3°C, DBT on the line joining i to 0.

$$v_i = 0.865$$
$$h_1 = 54.6 \text{ kJ/kg. d.a.}$$



Volume flow rate of air entering the cooling coil.

$$\dot{Q}_{V_i} = \dot{m}_{a_2} \times v_i = 44.09 \times 0.865$$
$$= 38.14 \text{ m}^3\text{/s}$$

(d) Refrigerating capacity of the coil.

$$\dot{Q}_{\text{coil}} = \text{GTH} = \dot{m}_{a_2}(h_1 - h_2) = 44.09(54.6 - 41.2)$$
$$= 591 \text{ kW}$$

Coil ADP is obtained by the intersection of the line joining 1 to 2 with the saturation curve. Thus,

$$t_{ADP} = 9°C$$

BPF of the coil,

$$BPF = \frac{t_2 - t_{ADP}}{t_1 - t_{ADP}} = \frac{18 - 9}{26.3 - 9} = 0.52$$

Q.7 (b) Solution:

In winter, the building sensible heat losses are partially compensated by the solar heat gains and the internal heat gains such as those from occupancy, lighting etc. Similarly, the latent heat loss due to low outside air humidity is more or less offset by the latent heat gains from occupancy. Thus in winter, the heating load is likely to be less than the cooling load in summer. However, the actual situation both in summer and winter depends on the swing of the outside temperature and humidity with respect to the inside conditions.

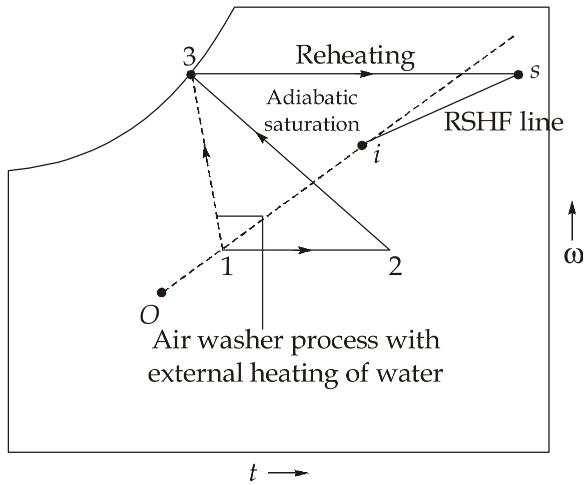
Further, certain sensible heat gains (negative loads) such as the solar heat may not be present at the time of peak load and hence they are not counted. On the other hand, latent heat gains from occupancy, etc. are always present and should be taken into account. As a result the design heating load for winter air conditioning is predominantly sensible.

In general the processes in the conditioning apparatus for winter air conditioning for comfort involve heating and humidifying. Two of the typical process combinations are:

(i) Preheating the air with steam or hot water in a coil followed by adiabatic saturation and reheat.

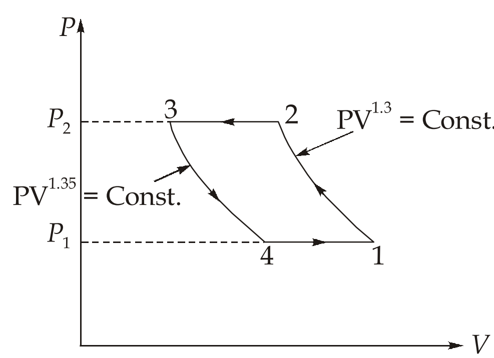
(ii) Heating and humidifying air in an air washer with pumped recirculation and external heating of water followed by reheat.

The process for the two systems are shown in figure. The first system with preheating and adiabatic saturation follows processes 1 - 2 and 2 - 3 respectively. The second system replaces the two processes with heated water spray in the air washer and the process line is 1 - 3. The leaving air state 3 from air washer may be affected by its saturation efficiency. The reheating process 3-s is common to both. The supply air states should lie on the room sensible heat factor line. It is therefore, determined by the room sensible heat factor and by the choice of supply air rate which is usually known from summer air conditioning calculations.



Winter air conditioning

Q.7 (c) Solution:



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = 273(12)^{0.3/1.3}$$

$$= 484.4 \text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{\frac{n-1}{n}} = 298\left(\frac{1}{12}\right)^{0.35/1.35}$$

$$= 156.47 \text{ K}$$

Refrigerating effect per kg of air circulated,

$$\text{RE} = c_P\,(T_1 - T_4) = 1\,(273 - 156.47)$$

$$= 116.53 \text{ kJ/kg}$$

$\therefore$ Mass of air circulated in the system per second,

$$\dot{m}_a = \frac{25\times 3.5}{116.53} = 0.7508 \text{ kg/ sec} = 45.05 \text{ kg/ min}$$

∴ Compressor displacement per min (V_1),

$$= \frac{\dot{m}_a R T_1}{P_1} = \frac{45.05 \times 287 \times 273}{1 \times 10^5} = 35.297 \text{ m}^3/\text{min}$$

Expander displacement per minute (V_4)

$$V_4 = V_1\left(\frac{T_4}{T_1}\right) = 35.297 \times \frac{156.47}{273} \quad [\because \text{Constant pressure}]$$

$$= 20.23 \text{ m}^3/\text{min}$$

Work done in the compressor per kg of air

$$W_C = \text{Area under 1 - 2 process}$$

$$= R\frac{n}{n-1}(T_2 - T_1) = 287 \times \frac{1.3}{0.3} \times (484.4 - 273)$$

$$= 262.91 \text{ kJ/kg}$$

Now work done by air in expander,

$$W_e = \text{Area under 3 - 4 process}$$

$$= R\frac{n}{n-1}(T_3 - T_4) = 287 \times \frac{1.35}{0.35} \times (298 - 156.47)$$

$$= 156.673 \text{ kJ/kg}$$

∴ Net work done on the refrigeration system,

$$W_{\text{net}} = W_C - W_e = 262.91 - 156.673$$

$$= 106.237 \text{ kJ/kg}$$

Coefficient of performance, COP $= \dfrac{\text{Refrigeration effect per kg}}{\text{Work done per kg}} = \dfrac{116.53}{106.237}$

$$= 1.0968$$

kW power to run the system $= m_a \times W_{\text{net}}$

$$= 0.7508 \times 106.237 = 79.76 \text{ kW}$$

kW power per ton of refrigeration $= \dfrac{79.96}{25} = 3.1984$ kW/ TR

For compressor and exapnder piston displacement,

Compressor piston displacement,

At point 1,

$$P_1 V_1 = mRT_1$$

$$1 \times 10^5 \times V_1 = 0.7509 \times 287 \times 273$$

$$V_1 = 0.5883 \text{ m}^3/\text{s}$$

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Now, expander piston displacement,

At point 4,

$$P_4V_4 = mRT_4$$
$$1 \times 10^5 \times V_4 = 0.7509 \times 287 \times 156.47$$
$$V_4 = 0.3372 \text{ m}^3/\text{s}$$

Q.8 (a) Solution:

$$P_v = (P_{vs})_{WBT} - \frac{\left[P_t - (P_{vs})_{WBT}\right][T_{DB} - T_{WB}]}{1547 - 1.44T_{WB}}$$
$$= 0.0073 - \frac{(1.01325 - 0.0073)(5 - 2.5)}{1547 - 1.44 \times 2.5}$$
$$= 0.00567 \text{ bar}$$
$$\omega = 0.622\frac{P_v}{P_t - P_v} = 0.622 \times \frac{0.00567}{1.01325 - 0.00567}$$
$$= 0.0035 \text{ kg/kg of air} = 3.5 \text{ gms/kg of dry air}$$

$\therefore$ h_1(Initial enthalpy of air) $= (1.005) \times 5 + 0.0035(2500 + 1.88 \times 5)$

$$= 13.808 \text{ kJ/kg}$$

The mass of dry air passed through heating coil per sec is given by:

$$\dot{m}_a = \frac{P_a v_a}{RT_a} = \frac{(1.01325 - 0.00567) \times 10^5 \times \frac{90}{60}}{287 \times (5 + 273)}$$
$$= 1.89 \text{ kg/s}$$

$\therefore$ q(Heat added by the coil per kg of dry air) $= \dfrac{40.7}{1.89} = 21.53 \text{ kJ/kg}$ of dry air

$\therefore$ h_2(Enthalpy of air leaving the coil) $= h_1 + q = 13.808 + 21.53$

$$= 35.338 \text{ kJ/kg of dry air}$$

After this, amount of steam mixed with air,

$$= \left(\frac{40}{3600}\right)\text{kg/s} \times \frac{1}{1.89 \text{ kg/s}}$$
$$= 0.00588 \text{ kg/kg of dry air}$$

$\therefore$ The specific humidity of air after mixing the steam

$$= 0.0035 + 0.00588$$
$$= 0.00938 \text{ kg/kg of dry air}$$

Amount of heat carried by the steam in the air per kg of dry air

$$= 0.00588 \times 2691$$
$$= 15.82 \text{ kJ/kg of dry air}$$

Enthalpy (h_3) of air after mixing the steam,

$$h_3 = h_2 + 15.82 = 35.338 + 15.82$$
$$= 51.158 \text{ kJ/kg of dry air}$$

This enthalpy is given by, $h_3 = (1.005)T_{DB_3} + \omega(2500 + 1.88T_{DB_3})$

$$51.158 = (1.005) \times T_{DB_3} + 0.00938\ (2500 + 1.88T_{DB_3})$$
$$= 1.02263T_{DB_3} + 23.45$$
$$T_{DB_3} = \frac{51.158 - 23.45}{1.032} = 27.095°C$$

From psychrometric chart,

At $\text{DBT}_3 = 27.095°\text{C}, \omega_3 = 0.00938 \text{ kg/kg of d.a.}$

$$\text{WBT}_3 = 18°\text{C}$$

Q.8 (b)(i) Solution:

$$\text{Clearance ratio} = 0.08$$
$$\left(\frac{P_2}{P_1}\right)^{1/n_c} = \frac{V_1}{V_2}$$
$$\frac{P_2}{P_1} = (3.157)^{1.4} = 5$$

Now, Volumetric efficiency $= 1 + C - C\left(\frac{P_2}{P_1}\right)^{1/n_e}$

[as, $n_c = n_e = 1.4$]

$$= 1 + 0.08 - 0.08 \times 3.157$$
$$\eta_{V_1} = 0.8274$$

Now, for new volumetric efficiency η_{V_2}.

As given in question exponent changes but pressure ratio should remain same:

$$\left(\frac{P_2}{P_1}\right)^{1/n'_e} = (5)^{1/1.1} \qquad [\text{Given, } n'_e = 1.1]$$

Now, $$\eta_{V_2} = 1 + C - C\left(\frac{P_2}{P_1}\right)^{1/n'_e}$$

$$= 1 + 0.08 - 0.08 \times 4.319$$

$$= 0.7345$$

% change in volumetric efficiency $= \dfrac{0.7345 - 0.8274}{0.8274} = -0.11228 \text{ or } -11.228\%$

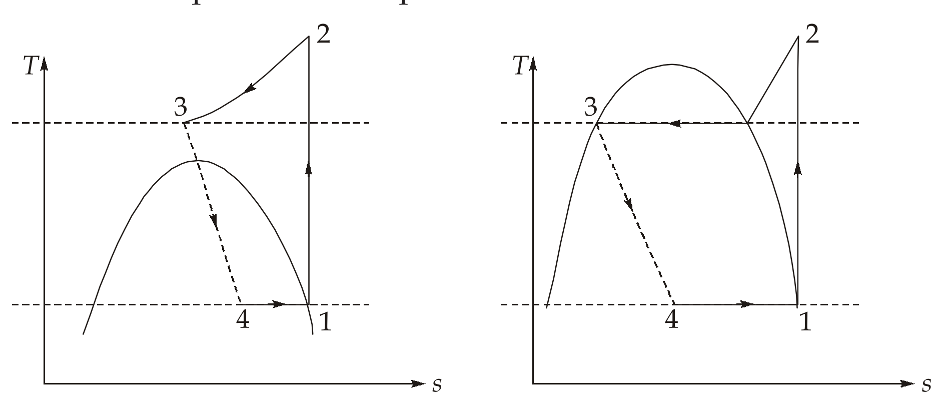
Negative sign indicates decrease in volumetric efficiency.

Q.8 (b)(ii) Solution:

There are various important properties of refrigerant required in the vapour compression refrigeration system. Some of them are given below:

(i) Normal boiling point: The boiling point of the substance corresponding to 1 atmospheric pressure is called normal boiling point. Generally low normal boiling point of refrigerants are preferred. Low normal boiling point refrigerants are called high pressure refrigerant and high normal boiling paint refrigerants are called low pressure refrigerants.

(ii) Critical temperature: As critical temperature decreases, refrigeration effect decreases and to keep the same refrigeration capacity mass flow rate should increase. This increase in mass flow rate increases the discharge volume of refrigerant which increases work input to the compressor.



So high critical temperature is required to get higher COP.

(iii) Freezing point: The freezing point of the refrigerant must be low to avoid freezing of refrigerant. Water has some very good properties but due to high freezing point it is not used in many applications.

(iv)Pressure ratios: Refrigerant with low pressure ratio is required.

(v) Specific volume: The specific volume at compressor inlet should be low because high specific volume results in larger size compressor. R-11 and R-113 have high specific volume at compressor inlet hence used with rotary compressor.

(vi) Latent heat and specific heat: The latent heat of the refrigerant should be high so as to have lower mass flow rates. Latent heat of NH_3 is very high.

The specific heat of liquid refrigerant should be low.

(vii) Compressor discharge temperature: The compressor discharge temperature should be low as high compressor discharge temperature results in overheating of compressor. NH_3 has high compressor discharge temperature hence NH_3 compressors are water cooled.

* Use of isentropic expander instead of throttling is practically not feasible instead of throttling: In VCRS throttling is preferred because with the use of isentropic expander very less work can be obtained (Since refrigerant remains mostly liquid during expansion). Hence use of isentropic expansion is not beneficial economically. But in gas refrigeration system isentropic expander is used because since air is used as the refrigerant which behaves as an ideal gas. Throttling will not result in any temperature drop.

Q.8 (c) Solution:

The vortex tube or Range-Hilsch tube, consists of a straight length of a tube with a concentric orifice located in a diaphragm near one end and a nozzle located tangentially near the outer radius adjacent to the orifice plate. Compressed gas enters the tube tangentially through a nozzle forming a vortex which, therefore, travels towards the right hand side of the tube called the hot end. A hot stream at temperature T_h which is above the temperature of supply, say, T_3 ejects from the hot end through the throttle value, while the cold stream at temperature T_c below the temperature of supply is received at the cold end through the orifice. The throttle valve opening controls the temperature and proportion of the cold stream with respect to the hot stream. The larger the throttle valve opening the lower the temperature of the cold stream and the smaller its fraction and vice-versa. The throttle value is placed sufficiently distant from the nozzle and the diaphragm immediately close to it.

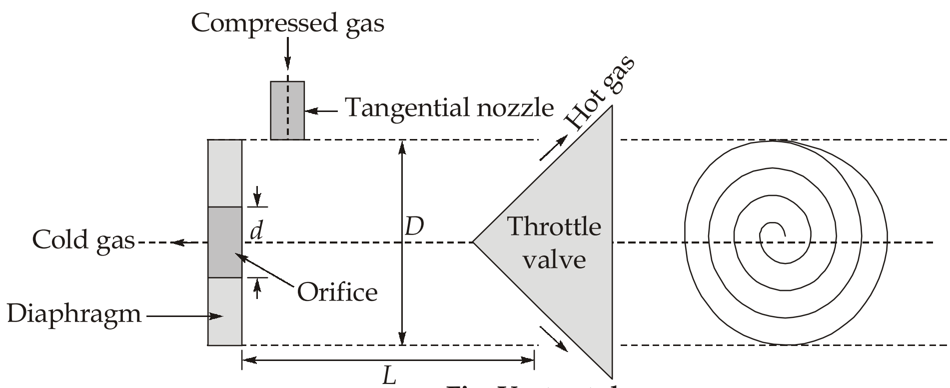
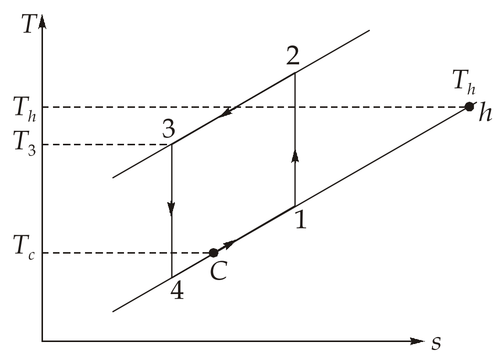


Fig. Vortex tube

The vortex tube system is a modification of the open-type air refrigeration system with the expander having been replaced by a vortex tube. In the Joule cycle, a temperature drop is obtained equal to the isentropic temperature drop (T_3 – T_4). The work of expansion is utilized to either run a cooling fan or a secondary compressor. The temperature drop obtained with the vortex tube, (T_3 – T_C) is smaller than the isentropic drop.



From the nozzle, the high-velocity gas travels from the periphery of the tube to the axis during which the separation of kinetic energy occurs. The kinetic energy is retained by the outer layers due to which they are heated and energy from the hot end of the tube at state *h*. The central core after having lost some kinetic energy emerges from the cold end at state *C*, i.e., at a temperature slightly above the static temperature of the expanded gas. The pressure of the cold gas stream is usually lowered further due to expansion in the vortex chamber.

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