Department of Mechanical Engineering
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LECTURE NOTES

Name of the Subject: Refrigeration & Air conditioning

Semester: 5th Year: 3rd

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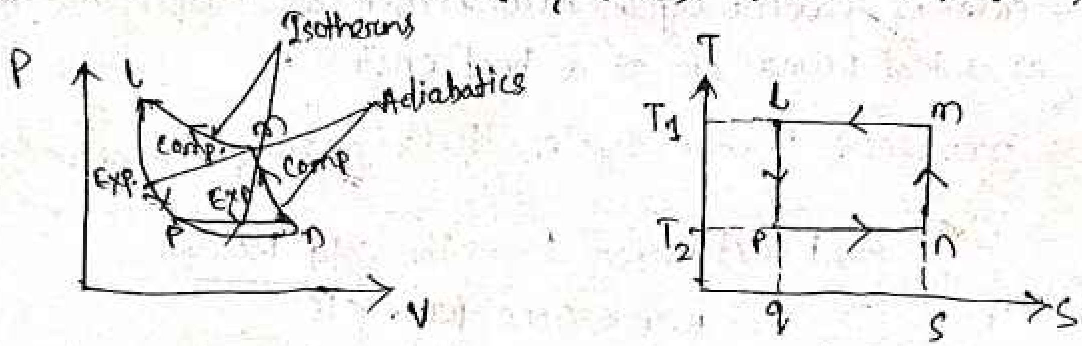
REFRIGERATION AND AIR-CONDITIONING

Refrigeration and Air-Conditioning

Refrigeration is the Science of the producing and maintaining temp. below that of the surrounding atmosphere.

- This means the removing of heat from a Substance to be cooled.
- Heat always passes downhill, from a warm body to a cooler one, until both bodies are at the same temperature.
- In simple, refrigeration means the cooling or removal of heat from a system. The equipment employed to maintain the system at a low temp. is termed as refrigerating System and the system which is kept at lower temp. is called refrigerated System.

Air Refrigerator Working on a Reversed Carnot Cycle



- Starting from point, the clearance space of the cylinder is full of air, the air is then expended adiabatically to point p during which its temperature falls from T_1 to T_2, the cylinder is put in contact with a cold body at temperature T_2.
- The air is then expanded isothermally to the point n, as a result of which heat is extracted from the cold body at temp. T_2.
- Now the cold body is removed; from n to m air undergoes adiabatic compression with the assistance of the some external power & temp. rises to T_1
- A hot body at temperature T_1 is put in contact with the cylinder.
- Finally the air is compressed isothermally during which process heat is rejected to the hot body.

Heat abstracted from the cold body = area 'npqs' = $T_2 \times pn$

Work Done per cycle = area 'lpnm' = $(T_1 - T_2) \times pn$

$\therefore$ Co-efficient of performance = $\dfrac{\text{Heat extracted the from the cold bod}}{\text{Work Done per cycle.}}$

$$= \frac{\text{Area 'npqs'}}{\text{Area 'lpnm'}}$$

$$= \frac{T_2 \times Pn}{(T_1 - T_2) \times Pn} = \frac{T_2}{T_1 - T_2}$$

→ Since the co-efficient of performance (COP) means the ratio of the desired effect in $\frac{kJ}{kg}$ to the energy supplied in $\frac{kJ}{kg}$, therefore C.O.P in case of Carnot cycle run either as refrigerating machine or a heat pump or as a heat engine.

(I) for a reversed Carnot cycle 'Refrigerating machine'

$$(COP)_{ref.} = \frac{\text{Heat extracted from the cold Body}}{\text{Work Done per cycle}}$$

$$= \frac{T_2\, PR}{(T_1 - T_2) PR} = \frac{T_2}{T_1 - T_2}.$$

(II) ~~for~~ for a carnot cycle 'Heat Pump'

$$(COP)_{heat\ Pump} = \frac{\text{Heat rejected to the hot body / cycle}}{\text{Work Done per cycle}}$$

[C.O.P. of heat pump is also called E.P.R i.e. Energy Performance Ratio.]

$$= \frac{T_1 \times lm}{(T_1 - T_2) pn} = \frac{T_1 \times PR}{(T_1 - T_2) \times PR}$$

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

$$= 1 + \frac{T_1}{T_1 - T_2} \left[\text{i.e. } 1 + (COP)_{ref.}\right]$$

(iii) For a Carnot cycle 'Heat Engine'

$$(C.O.P)_{\text{Heat Engine}} = \frac{\text{Work Done / Cycle}}{\text{Heat Supplied / Cycle}} = \frac{(T_1 - T_2) \times PD}{T \times ED}$$

$$= \frac{(T_1 - T_2) \times PD}{T_1 \times PD}$$

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

Q:- A certain m/c works on reversed Carnot cycle betn. temp. limits of $-10^\circ c$ & $27^\circ c$. Find its C.O.P. working as

(I) A refrigerating m/c
(II) As heat pump
(III) A heat engine.

Given $T_2 = 273 + (-10) = 263K$, $T_1 = 273 + 27 = 300K$

COP

(I) As a refrigerating m/c :
$$C.O.P. = \frac{T_2}{T_1 - T_2} = \frac{263}{300 - 263} = 7.11$$

(II) As a heat pump
$$C.O.P = \frac{T_1}{T_1 - T_2} = \frac{300}{300 - 263} = 8.11$$

(III) As a heat engine
$$C.O.P. = \frac{T_1 - T_2}{T_1} = \frac{300 - 263}{300} = 0.123$$

:- An inventor claims to have developed a refrigerating unit which maintains the refrigerated space at $-5^\circ c$ when operating in a room where temp. is $26^\circ c$ and has a C.O.P = 8.4. Find out whether he's claim is correct or not.

Given $T_2 = 273 + (-5) = 268K$, $T_1 = 273 + 26 = 299K$, $COP = 8.4$

The COP of the refrigerating unit will be maxm. when it is working on reversed Carnot cycle.

$$\therefore \text{Max}^m \text{ Possible COP} = \frac{T_2}{T_1 - T_2} = \frac{268}{299 - 268} = 8.64$$

Since the COP claimed by the inventor is less than the maxm. possible COP, hence the claim of the inventor is correct.

Q1- A refrigerating m/c working on reversed Carnot cycle consumes 5.5 kW for producing refrigerating effect of 940 kJ/min. for maintaining a region at −38°C. Determine :

(i) COP of refrigerating m/c

(ii) Higher temp. of cycle.

(iii) Amount of heat delivered in kJ/min, when this device is used as an heat pump.

Given $W = 5.5$ kW, $R_n = 940 \frac{kJ}{min}$, $t_2 = -38°C$

(i) COP of the m/c

$$T_2 = 273 + (-38) = 235K$$

$$\text{C.O.P of the m/c} = \frac{R_n(\text{net refrigerating effect})}{W(\text{work done})}$$

$$= \frac{(940/60)}{5.5} = 2.85$$

(ii) Higher temp. of the cycle

$$\text{Also COP} = \frac{T_2}{T_1 - T_2}$$

$$\therefore 2.85 = \frac{235}{T_1 - 235} \quad \text{or } T_1 = \frac{235}{2.85} + 235$$

$$= 317.5K$$

or 44.5°C.

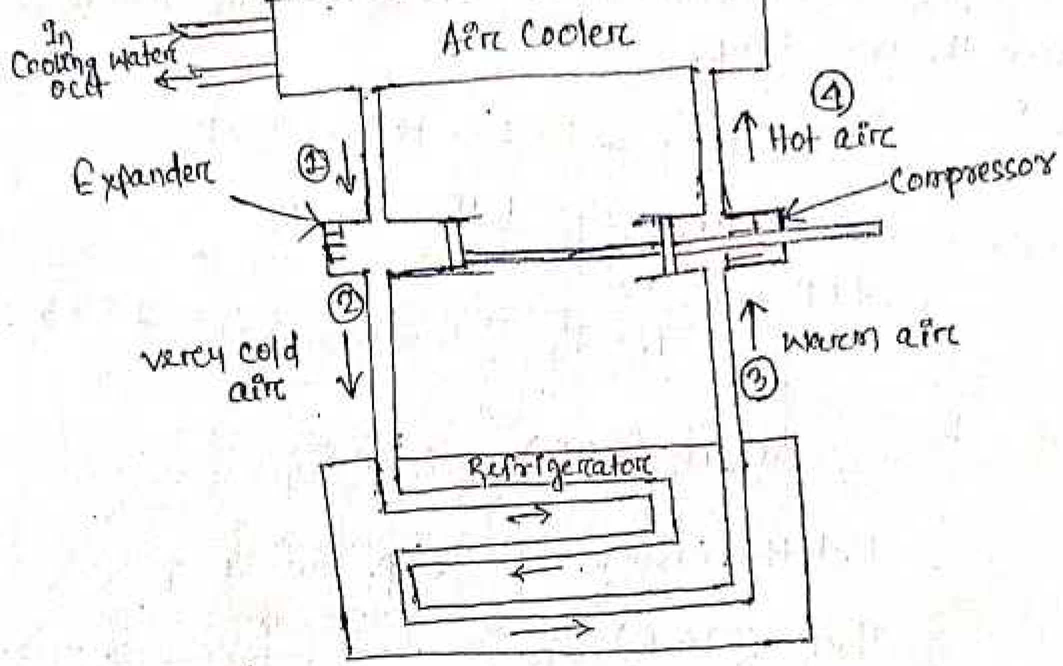
(iii) Heat delivered as heat pump

Heat Delivered = Heat absorbed + Work Done

$$= 940 + 5.5\left(\frac{kJ}{s}\right) \times 60$$

$$= 1270 \text{ kJ/min}.$$

Air-Refrigeration System Working on Reversed Brayton cycle (Or Bell-Coleman or Joule cycle)



Schematic diagram of an air-refrigeration System working on the reversed Brayton (or Bell-Coleman or Joule) cycle.

Elements of this System are

1. Compressor
2. Cooler (heat exchanger)
3. Expander
4. Refrigerator

In this System, work gained from expander is employed for compression of air, consequently less external work is needed for operation of the System. In practice it may not be done e.g. in some aircraft refrigeration Systems which employ air-refrigeration cycle the expansion work may be used for driving other devices.

This system uses reversed Brayton cycle, which is described below:

→ P-V & T-s diagrams for a reversed Brayton cycle are shown in fig..

→ Here it is assumed that (i) absorption & rejection of heat are constant pressure processes and (ii) Compression and expansion are Isentropic processes.

Q:- A carnot refrigerator operates betⁿ temp. of $-45°C$ & $45°C$. Determine c.o.p.. It is desired to make the c.o.p. equal to 3.5 by changing temperatures. The increase (or decrease) in upper temp. is to be equal to the decrease (or increase) in the lower temp. Determine the new temp.

Case :- I

Given $T_2 = 273 + (-45) = 228K$

$T_1 = 273 + 45 = 308K$

$$cop = \frac{T_2}{T_1 - T_2} = \frac{228}{318 - 228} = 2.533$$

Case :- II

Given $(COP)_{new} = 3.5$

Let the change of temp. be x.

Then $$(C.O.P)_{new} = \frac{228 - x}{(318 + x) - (228 - x)} = 3.5$$

or, $$\frac{228 - x}{90 + 2x} = 3.5$$ or, $(228 - x) = 3.5(90 + 2x)$

$\Rightarrow 228 - x = 315 + 7x$

$\Rightarrow x = -10.875°C$

$\therefore T_2 = 228 - (-10.875) = 238.875K$

$T_1 = 318 + (-10.875) = 307.125K$

$$(C.O.P)_{new} = \frac{T_2}{T_1 - T_2}$$

$$= \frac{238.875}{307.125 - 238.875} = 3.5$$

Q:- An ice plant produces 12 tonnes of ice per day at 0°c, using water at room temp. of 15°c. If the cop of plant is 2.6 & overall electro-mechanical efficiency is 87%. Determine the power rating of the comp. motor. Take specific heat of water as $4.18 \frac{kJ}{kg}$ & latent heat of ice $335 \frac{kJ}{kg}$

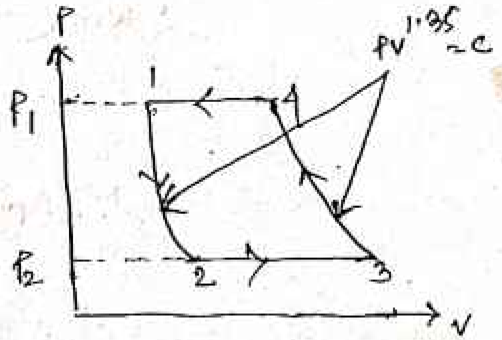
Dt. 19/11/2021

1.- A Bell-Coleman refrigerator operates betn pressure limits of 1 bar and 8 bar. Air is drawn from the cold chamber at 9°c, compressed & then it is cooled to 29°c before entering the expansion cylinder. Expansion & compression follow the law $pv^{1.35} = C$. Calculate the theoretical c.o.p of the system. for air take $\gamma = 1.4$, $C_p = 1.003\ \frac{kJ}{kgK}$

Sol:- Given

$P_2 = 1$ bar, $P_1 = 8$ bar, $T_3 = 9+273 = 282K$

$T_1 = 29+273 = 302K$



Considering polytropic compression 3-4, we get

$$\frac{T_4}{T_3} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= \left(\frac{8}{1}\right)^{\frac{1.35-1}{1.35}} = (8)^{0.259} = 1.71$$

$$\Rightarrow T_4 = T_3 \times 1.71$$

$$= 282 \times 1.71 = 482.2K$$

Considering polytropic expansion 1-2, we have

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{8}{1}\right)^{\frac{1.35-1}{1.35}} = 1.71$$

$$T_2 = \frac{T_1}{1.71} = \frac{302}{1.71} = 176.6K$$

Heat extracted from the cold chamber per kg of air

$$= C_p(T_3 - T_2) = 1.003(282 - 176.6)$$

$$= 105.7\ \frac{kJ}{kg}$$

Heat rejected in the cooling chamber per kg of air

$$= C_p(T_4 - T_1) = 1.003(482.2 - 302)$$

$$= 180.7\ \frac{kJ}{kg}$$

Since the compression & expansion are not isentropic, difference betn heat rejected & heat absorbed is not equal to the work done because there are heat transfers to the surroundings & from the surroundings during compression & expansion.

To find the work done, the area of the diagram '1234' is to be considered.

$$\text{work Done} = \frac{n}{n-1}(P_4V_4 - P_3V_3) - \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

$$= \frac{n}{n-1} R\left[(T_4 - T_3) - (T_1 - T_2)\right] \qquad (m = 1kg)$$

The value of R can be calculated as follows :

$$\frac{C_p}{C_v} = \gamma$$

$$\therefore C_v = \frac{C_p}{\gamma} = \frac{1.003}{1.4} = 0.716 \ \frac{KJ}{kgK}$$

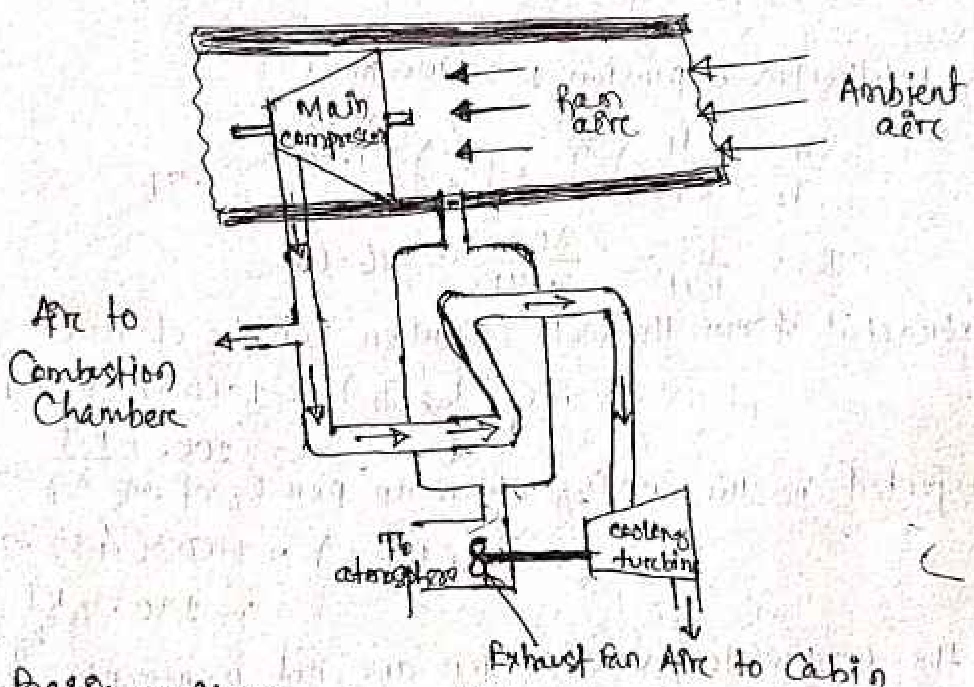
$$R = (C_p - C_v) = 1.003 - 0.716 = 0.287 \ \frac{KJ}{kgK}$$

$$\therefore \text{Work Done} = \frac{1.35}{0.35} \times 0.287 \left[(482.2 - 282) - (302 - 176.6) \right]$$

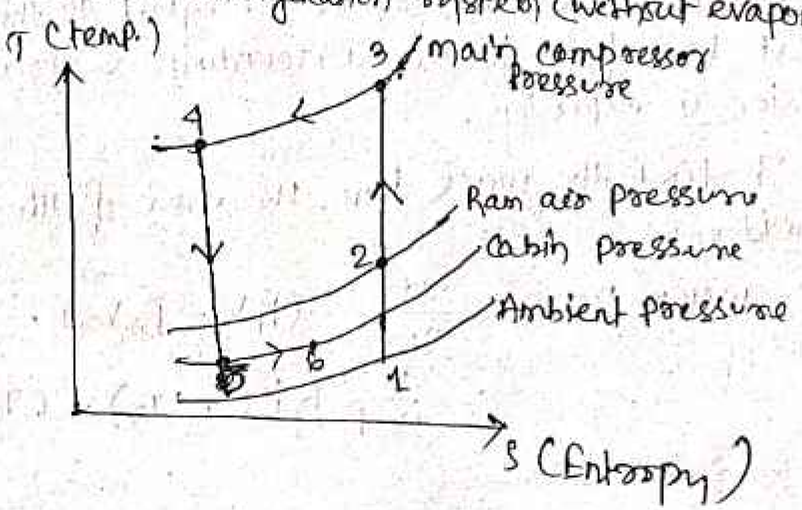
$$= 82.8 \ \frac{KJ}{kg}$$

$$\therefore COP = \frac{\text{Heat abstracted}}{\text{Work Done}} = \frac{105.7}{82.8} = 1.27$$

Air-craft Refrigeration Systems



Basic or simple air-refrigeration system (without evaporative cool



(ii) Process 1–2 : Ramming of air

If C is the aircraft velocity or velocity of air relative to the aircraft in metres per second, then kinetic energy (K.E.) of outside air relative to aircraft is

$$KE = \frac{C^2}{2}\frac{J}{kg} \quad \text{or} \quad \frac{C^2}{2000}\frac{kJ}{kg} \quad \longrightarrow (1)$$

from energy eqⁿ, we have

$$h_2 - h_1 = \frac{C^2}{2000}$$

$$C_pT_2 - C_pT_1 = \frac{C^2}{2000}$$

$$\therefore T_2 = T_1 + \frac{C^2}{2000\,C_p} \quad \text{or} \quad \frac{T_2}{T_1} = 1 + \frac{C^2}{2000\,C_p \times T_1} \quad \longrightarrow (2)$$

we know that $C_p - C_v = R$

$$\Rightarrow C_p\left(1 - \frac{1}{\gamma}\right) = R \quad \left(\because \frac{C_p}{C_v} = \gamma\right)$$

$$\Rightarrow C_p = \frac{\gamma R}{\gamma - 1}$$

Substituting the value of C_p in eqⁿ (2), we get

$$\frac{T_2}{T_1} = 1 + \frac{C^2(\gamma - 1)}{2000\,\gamma R\,T_1} \quad \longrightarrow (3)$$

$$= 1 + \frac{C^2(\gamma - 1)}{2a^2} \quad \longrightarrow (4)$$

where a = local sonic or acoustic velocity at the ambient air condition $= \sqrt{\gamma R T_1}$, where R is in $\frac{kJ}{kgK}$

Also $\frac{T_2}{T_1} = 1 + \frac{\gamma - 1}{2}M^2$

where M = mach number of the flight

Again, the stagnation pressure after isentropic compression (P_2) is given by : $\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}}$

(ii) Process 2-3: Compression in main Compressor.

The work done during this compression (isentropic) process,

$$W_{comp.} = m_a \times c_p (T_3 - T_2) \longrightarrow (5)$$

Where m_a = Mass of air bled from the air compressor for refrigeration purposes.

(iii) Process 3-4: Cooling of ram air in heat exchanger

The heat rejected in the heat exchanger during the cooling process is given by:

$$Q_{rej.} = m_a \times c_p \times (T_3 - T_4) \longrightarrow (6)$$

(iv) Process 4-5: Expansion in turbine

The work done by the cooling turbine during this expansion process is given by:

$$W_{exp} = m_a \times c_p \times (T_4 - T_5) \longrightarrow (7)$$

The work of this is used to drive the exhaust fan which dr the cooling air from the heat exchanger.

(v) Process 5-6: Air getting heated to cabin temperature.

The refrigerating effects produced or heat absorbed is given by:

$$Q_{ref.} = m_a \times c_p \times (T_6 - T_5) \longrightarrow (8)$$

Where T_6 = Inside temp. of the cabin

Now, C.O.P of air cycle, = $\frac{\text{Refrigerating effect Produced}}{\text{Work Done}}$

$$= \frac{m_a \times c_p \times (T_6 - T_5)}{m_a \times c_p \times (T_3 - T_2)} = \frac{T_6 - T_5}{T_3 - T_2} \rightarrow$$

If Q tonnes of refrigeration is the cooling load in the cabin then the air required for the refrigeration purpose,

$$m_a = \frac{14000 \times Q}{c_p \times (T_6 - T_5)} \text{ kg/h} \longrightarrow (10)$$

Power reqd. for the refrigeration system,

$$P = \frac{m_a \times c_p \times (T_3 - T_2)}{3600} \text{ kW} \longrightarrow (11)$$

Let T_3 = Temp. of air after isentropic compression in the compressor.

$T_{3'}$ = Actual temp. of the air leaving the compressor.

T_5 = Temp. of the air leaving the turbine after isentropic expansion

$T_{5'}$ = Actual temp of the air leaving the turbine

For the Isentropic compression process 2-3, we have

$$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4.8}{0.96}\right)^{\frac{1.4-1}{1.4}} = 1.584$$

$$\therefore T_3 = T_2 \times 1.584 = 302 \times 1.584 = 478.4K$$

The isentropic efficiency of the compressor,

$$\eta_c = \frac{\text{Isentropic increase in temp}}{\text{Actual increase in temp}} = \frac{T_3 - T_2}{T_{3'} - T_2}$$

$$\Rightarrow 0.9 = \frac{478.4 - 302}{T_{3'} - 302}$$

$$\Rightarrow T_{3'} = \frac{(478.4 - 302)}{0.9} + 302 = 498K$$

Now, for Isentropic expansion process 4-5, we have

$$\frac{T_4}{T_5} = \left(\frac{P_4}{P_5}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4.8}{1}\right)^{\frac{1.4-1}{1.4}} = 1.565$$

$$\therefore T_5 = \frac{T_4}{1.565} = \frac{339}{1.565} = 216.6K$$

The isentropic efficiency of the turbine,

$$\eta_T = \frac{\text{Actual increase in temp.}}{\text{Isentropic increase in temp}} = \frac{T_4 - T_{5'}}{T_4 - T_5}$$

$$\Rightarrow 0.9 = \frac{339 - T_{5'}}{339 - 216.6} \Rightarrow T_{5'} = 339 - 0.9(339 - 216.6)$$
$$= 228.8K$$

(i) Mass of air circulated, m_a

Mass of air circulated,

$$m_a = \frac{14000\,Q}{c_p(T_6 - T_{5'})} \text{ kg/hr.}$$

$$= \frac{14000 \times 25}{1.005(299 - 228.8)} = 4960.95 \frac{kg}{hr}$$
$$= 82.765 \text{ kg/min.}$$

(ii) $$COP = \frac{14000\,Q}{m_a \times c_p \times (T_{3'} - T_2)} = \frac{14000 \times 25}{4960.95 \times 1.005 \times (498 - 302)}$$
$$= 0.358$$

Q:- A piston & cylinder m/c contains a fluid system which passes thro a complete cycle of four processes. During a cycle, the sum of all heat transfers is -170 kJ. The system completes 100 cycles per min. Complete the following table showing the method for each item, and compute the net rate of work output in kW.

Q:- A refrigerating m/c of 6 tonnes capacity working on Bell-Coleman cycle has an upper limit of pressure of 5.2 bar. The pressure & temp. at the start of the compression are 1.0 bar and 16°C respectively. The compressed air cooled at constant pressure at a temp. of 41°C enters the expansion cylinder. Assuming both expansion & compression processes to be adiabatic with $\gamma = 1.4$, Calculate: (i) Co-efficient of performance.

(ii) Quantity of air in circulation per minute.

(iii) Piston displacement of compressor & expander.

(iv) Bore of compressor & expansion cylinders. The unit runs at 240 rpm & is double acting; stroke length = 200 mm

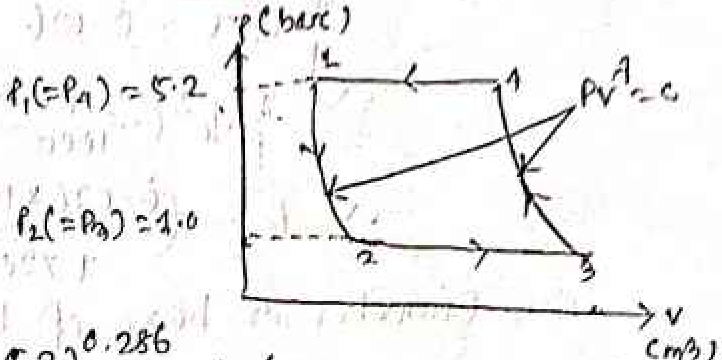
(v) Power reqd. to drive the unit

For air take $\gamma = 1.4$ & $c_p = 1.003 \frac{kJ}{kgK}$

Sol:- $T_3 = 16+273 = 298K$

$T_1 = 41+273 = 314K$

$P_1 = 5.2$ bar, $P_2 = 1.0$ bar



Considering the adiabatic compression process 3-4, we get

$$\frac{T_4}{T_3} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5.2}{1}\right)^{\frac{1.4-1}{1.4}} = (5.2)^{0.286} = 1.6$$

$\therefore T_4 = 1.6 \times T_3 = 1.6 \times 289 = 462.4K$

Considering the adiabatic expansion, process 1-2, we get

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow \frac{314}{T_2} = \left(\frac{5.2}{1}\right)^{\frac{0.4}{1.4}} = 1.6$$

$$\Rightarrow T_2 = \frac{314}{1.6} = 196.25K$$

(i) C.O.P

Since both the compression & expansion processes are isentropic/reversible adiabatic, therefore

$$\text{C.O.P. of the m/c} = \frac{T_2}{T_1 - T_2} = \frac{196.25}{314 - 196.25} = 1.67 \text{ Ans}$$

(ii) Mass of air in circulation:-

Refrigerating effect per kg of air $= C_p(T_3 - T_2)$

$= 1.003(289 - 196.25) = 93.03 \frac{kJ}{kg}$

Refrigerating effect produced by the refrigerating machine

$= 6 \times 14000 = 84,000 \frac{kJ}{hr.}$

Hence mass of air in circulation $= \frac{84000}{93.03 \times 60} = 15.05 \frac{kg}{min}$ Ans

C.O.P. of the refrigeration System $= \frac{14000 Q}{m_a \times C_p \times (T_3 - T_2)} = \frac{14000 Q}{1 \times 3600}$.

(iii) Piston Displacement & bore of Compressor :-

Piston Displacement of Compressor = Volume Corresponding to Point i.e. V_3

$$\therefore V_3 = \frac{mRT_3}{P_2} = \frac{15.05 \times 0.287 \times 1000 \times 289}{1.0 \times 10^5}$$

$$= 12.48 \ m^3/min \quad \text{Ans}$$

$\therefore$ Swept volume per Stroke $= \frac{12.48}{2 \times 240} = 0.026 \ m^3$

If d_c = Dia. of Compressor cylinder & l = Length of Stroke (= 200

then,

$$\frac{\pi}{4} d_c^2 \times l = 0.026$$

$$\Rightarrow \frac{\pi}{4} d_c^2 \left(\frac{200}{1000}\right) = 0.026$$

$$\Rightarrow d_c = \left(\frac{0.026 \times 1000 \times 4}{\pi \times 200}\right)^{1/2} = 0.407 \ m \text{ or } 407 \ mm$$

i.e. Diameter or bore of the Compressor cylinder = 407 mm Ans

Piston Displacement & bore of Expander

Piston displacement of expander = Vol. Corresponding to point 2. i.e.

$$\therefore V_2 = \frac{mRT_2}{P_2} = \frac{15.05 \times 0.287 \times 1000 \times 196.25}{1.0 \times 10^5} = 8.476 \ m^3$$

$\therefore$ Swept vol. per stroke $= \frac{8.476}{2 \times 240} = 0.0176 \ m^3$

If d_e = Dia. of the expander and l = Length of stroke (= 200 mm

then,

$$\frac{\pi}{4} d_e^2 \times \left(\frac{200}{1000}\right) = 0.0176$$

$$d_e = \left(\frac{0.0176 \times 1000 \times 4}{\pi \times 200}\right)^{1/2} = 0.335 \ m \text{ or } 335 \ mm$$

i.e. Diameter or bore of the expander cylinder = 335 mm

(v) Power reqd. to drive the unit :

$$C.O.P = \frac{\text{Refrigerating effect}}{\text{Work Done}} = \frac{R_n}{W}$$

$$\Rightarrow 1.67 = \frac{6 \times 14000}{W} \Rightarrow W = \frac{6 \times 14000}{1.67} = 50299.4 \ \frac{kJ}{h}$$

$$= 13.97 \ \frac{kJ}{s}$$

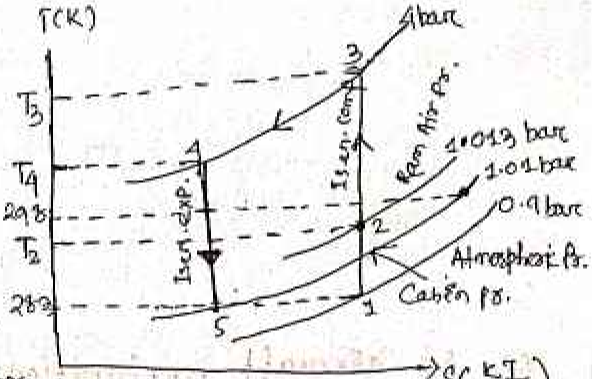
Hence power reqd. = 13.97 kW Ans

A simple air cooled system is used for an aeroplane having a load of 9 tonnes. The atmospheric pressure and temp. are 0.9 bar and 10°c respectively. During ramming pressure increases to 1.013 bar. In the heat exchanger, the temp. of air is reduced by 55°c. The pr. in the cabin is 1.01 bar and the temp. of air leaving the cabin is 25°c. Determine:

(i) Power reqd. to take the load of cooling in the cabin.

(ii) COP of the system.

Assume that all the expansions & compressions are isentropic. The pr. of the compressed air is 4 bar.



Given

Cooling Load, $Q = 9$ tonnes.

$P_1 = 0.9$ bar, $P_2 = 1.013$ bar

$P_3 = P_4 = 4$ bar, $P_5 = P_6 = 1.01$ bar

$T_1 = 10+273 = 283K$, $T_6 = 25+273 = 298K$

Let T_2 = Temp. of air at the end of ramming or entering the main compressor.

T_3 = Temp. of air leaving leaving the compressor.

T_4 = Temp. of air leaving the heat exchanger

T_5 = Temp. of air leaving the cooling turbine.

We know that, $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.013}{0.9}\right)^{\frac{1.4-1}{1.4}} = 1.034$

$\therefore T_2 = T_1 \times 1.034 = 283 \times 1.034 = 292.6K$

Similarly, $\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{1.013}\right)^{\frac{1.4-1}{1.4}} = 1.48$

$\therefore T_3 = T_2 \times 1.48 = 292.6 \times 1.48 = 433K = 160°c$

Since the temp. of air is reduced by 55°c in the heat exchanger, therefore temp. of air leaving the heat exchanger.

$T_4 = (160-55)+273 = 378K$

We know that, $\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.01}{4}\right)^{\frac{1.4-1}{1.4}} = 0.675$

$\therefore T_5 = T_4 \times 0.675 = 378 \times 0.675 = 255K$

The mass of air (m_a) reqd. for refrigeration purpose,

$$m_a = \frac{14000Q}{C_p(T_6 - T_5)} \ \frac{kg}{hr}$$

$$= \frac{14000 \times 9}{1(298 - 255)} = 2930.2 \ \frac{kg}{hr}$$ (Taking C_p for air $= 1 \frac{kJ}{kgK}$

∴ Power reqd. to take the load of cooling in the cabin :

$$P = \frac{m_a \times C_p \times (T_3 - T_2)}{3600} \ kW$$

$$= \frac{2930.2 \times 1 \times (433 - 292.6)}{3600} = 114.28 \ kW$$

(iii) COP of the system

$$COP = \frac{14000Q}{P \times 3600}$$

$$= \frac{14000 \times 9}{114.28 \times 3600} = 0.306 \ \text{Ans}$$

Q:- An aircraft refrigeration plant has to handle a cabin load of 25 tonnes. atmospheric temp is 16°C. The atmospheric air is compressed to a pressure of & temp. of 29°C. due to ram action. This air is then further compressed in a compressor to 4.8 bar, cooled in a heat exchanger to 66°C, expanded in a turbi 1 bar pressure & supplied to the cabin. The air leaves the cabin at a temp. The isentropic efficiencies of both compressor & turbine are 0.9. Calculate:

(i) The mass of air circulated per minute.

(ii) C.O.P

Take for air $C_p = 1.005 \frac{kJ}{kgK}$ & $\gamma = 1.4$

Given $Q = 25$ tonnes

$T_1 = 16 + 273 = 289K$

$P_2 = 0.96$ bar, $T_2 = 29 + 273 = 302K$

$P_3 = P_{3'} = P_4 = 4.8$ bar

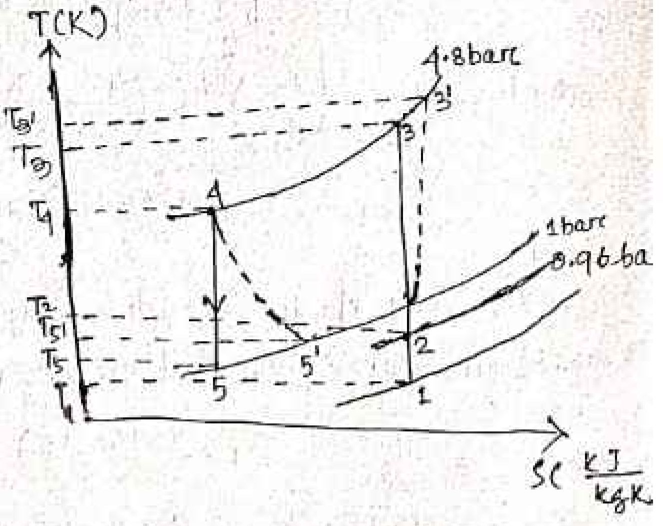
$T_4 = 66 + 273 = 339K$

$P_5 = P_{5'} = 1$ bar

$T_6 = 26 + 273 = 299K$

$\eta_c = \eta_T = 0.9$, $C_p = 1.005 \frac{kJ}{kgK}$

$\gamma = 1.4$



(ii) Power of the compressor

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

Heat reqd. to be removed $= 180 \frac{kJ}{min}$

Mass of refrigerant circulated/min

$$= \frac{180}{h_2 - h_4} = \frac{180}{(1151.18 - 115)} = 0.174 \frac{kg}{min}$$

Total work done by the compressor/min

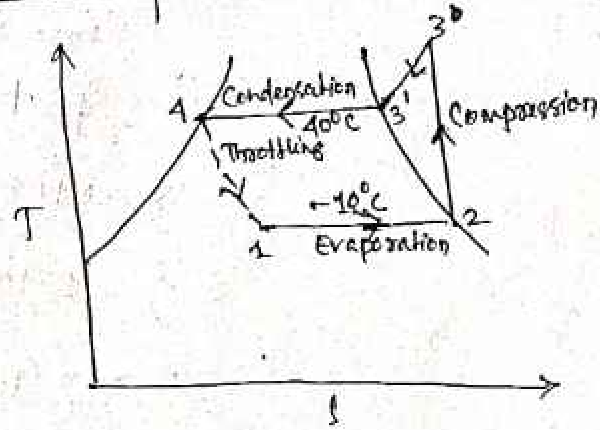
$$= 0.174(h_3 - h_2) = 0.174(1288 - 1157.18)$$
$$= 23.81 \frac{kJ}{min}$$
$$= \frac{23.81}{60} \frac{kJ}{s} = 0.397 \frac{kJ}{s} \text{ or kW}$$

:- In a standard vapour compression refrigeration cycle, operating betn an evaporator temp. of $-10°c$ and a condenser temp. of $40°c$, the enthalpy of the refrigerant, Freon-12, at the end of compression is $220 \frac{kJ}{kg}$. Show the cycle diagram on T-s plane.

Calculate : (i) The cop of the cycle.

(ii) The refrigerating capacity & the compressor power assuming a refrigerant flow rate of 1 kg/min. You may use the extract of Freon-12 property table given below:

t (°c)	P (MPa)	h_f ($\frac{kJ}{kg}$)	h_g (kJ/kg)
−10	0.2191	26.85	183.1
40	0.9607	74.53	203.1



Given : Evaporator temp. $= -10°c$

Condenser temp. $= 40°c$

Enthalpy at the end of compressor

$$h_3 = 220 \frac{kJ}{kg}$$

From the table (given), we have

$$h_2 = 183.1 \frac{kJ}{kg}$$
$$h_4 = h_{f4} = 26.85 \frac{kJ}{kg}$$

(i) The C.O.P. of the cycle:

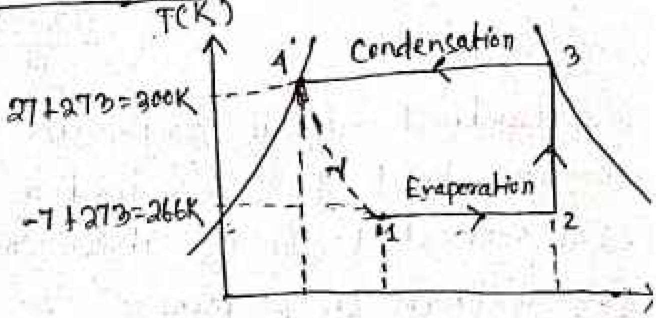
$$COP = \frac{R_n}{W} = \frac{h_2 - h_4}{h_3 - h_2} = \frac{183.1 - 26.85}{220 - 183.1} = 4.23$$

Q:- A refrigerator works betn. $-7^\circ c$ & $27^\circ c$. The vapour is dry at the end adiabatic compression. There is no under-cooling & evaporation is by throttle. Determine: (i) The co-efficient of performance.

(ii) Power of the compressor to remove 180 KJ/min.

The properties of the refrigerant are as under:

Temp ($^\circ c$)	Enthalpy (kJ/kg)		Entropy (kJ/kgK)	
	Liquid (h_f)	Latent heat (h_{fg})	Liquid (S_f)	Vapour (S_g)
-7	-30	1298	-0.108	4.75
27	115	1173	427	4.33



Given

$T_3 = T_4 = 300K$

$T_1 = T_2 = 266K$, $x_3 = 1$

Heat to be removed $= 180 \frac{kg}{min}$

(i) Co-efficient of performance, c.o.p:

$$c.o.p = \frac{Rn}{W} = \frac{h_2 - h_1}{h_3 - h_2}$$

Here $h_3 = h_{f_3} + h_{fg_3} = 115 + 1173 = 1288 \frac{kJ}{kg}$

$h_1 = h_{f_4} = 115 \frac{kJ}{kg}$

To find h_2, let us first find dryness fraction at point 2.

Entropy at '2' = Entropy at '3' (process 2-3 being isentropic)

i.e. $S_2 = S_3$

$$S_{f_2} + x_2 S_{fg_2} = S_{g_3}$$

$$\Rightarrow -0.108 + x_2 \times \frac{h_{fg_2}}{T_2} = 4.33$$

$$\Rightarrow -0.108 + x_2 \times \frac{1298}{266} = 4.33$$

$$\therefore x_2 = (4.33 + 0.108) \times \frac{266}{1298} = 0.91$$

$$\therefore h_2 = h_{f_2} + x_2 h_{fg_2}$$

$$= -30 + 0.91 \times 1298 = 1151.18 \frac{kJ}{kg}$$

Substituting the values in (i), we get

$$C.O.P = \frac{1151.18 - 115}{1288 - 1151.18} = 7.57$$

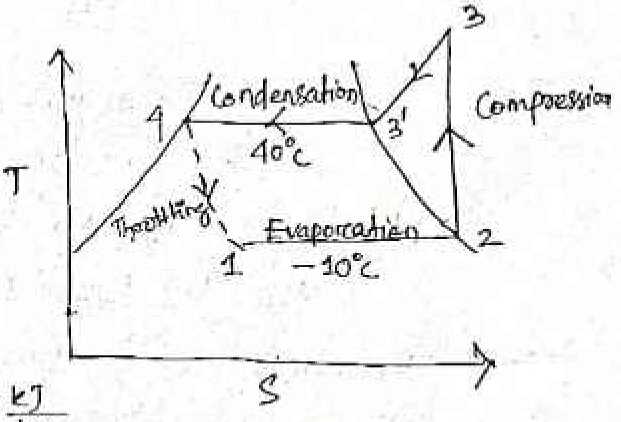
Dt. 18/12/2021

– In a standard vapour compression refrigeration cycle, operating bet? an evaporator temp. of −10°c and a condenser temp. of 40°c, the enthalpy of the refrigerant, Freon-12, at the end of compression is 220 $\frac{kJ}{kg}$. Show the cycle diagram on T-s plane. Calculate (i) The cop of the cycle.

(ii) The refrigerating capacity and the compressor power assuming a refrigerant flow rate of 1 kg/min. You may use the extract of freon-12 property table given below:

t(°c)	P(MPa)	$h_f(\frac{kJ}{kg})$	$h_g(\frac{kJ}{kg})$
−10	0.2191	26.85	183.1
40	0.9607	74.53	203.1

A:- The cycle is shown on T~s Diagram



Given: Evaporator temp. = −10°c.

Condenser temp. = 40°c

Enthalpy at the end of compressor

$h_3 = 220 \frac{kJ}{kg}$

from the table (given), we have

$h_2 = 183.1 \frac{kJ}{kg}$, $h_1 = h_{f_4} = 26.85 \frac{kJ}{kg}$

(i) The C.O.P. of the cycle:

$$C.O.P = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{183.1 - 26.85}{220 - 183.1} = 4.23$$

(ii) Refrigerating Capacity:

Refrigerating Capacity = $m(h_2 - h_1)$

[Where m = mass flow rate of refrigerant = 1 kg/min Given]

$= 1 \times (183.1 - 26.85) = 156.25 \frac{kJ}{min}$

Compressor power:

Compressor power = $m(h_3 - h_2) = 1 \times 220 - 183.1 = 36.9 \frac{kJ}{min}$

or $0.615 \frac{kJ}{s}$

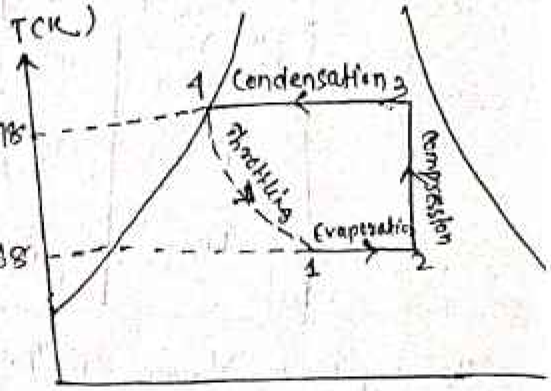
Given
$\dot{m} = 6\,kg/min$, $\eta_{relative} = 50\%$, $x_2 = 0.6$, $C_{pw} = 4.187\ \frac{kJ}{kgK}$.

Latent heat of ice $= 335\ \frac{kJ}{kg}$

$h_{f_2} = 31.4\ \frac{kJ}{kg}$, $h_{fg_2} = 154.0\ \frac{kJ}{kg}$, $h_{f_3} = 59.7\ \frac{kJ}{kg}$

$h_{fg_3} = 138\ \frac{kJ}{kg}$, $h_{f_4} = 59.7\ \frac{kJ}{kg}$

$h_2 = h_{f_2} + x_2 h_{fg_2} = 31.4 + 0.6 \times 154 = 123.8\ \frac{kJ}{kg}$



For isentropic compression 2-3,

$$s_3 = s_2$$

$$s_{f_3} + x_3 \frac{h_{fg_3}}{T_3} = s_{f_2} + x_2 \frac{h_{fg_2}}{T_2}$$

$$\Rightarrow 0.2232 + x_3 \times \frac{138}{298} = 0.1251 + 0.6 \times \frac{154}{268}$$

$$= 0.4698$$

$$\therefore x_3 = (0.4698 - 0.2232) \times \frac{298}{138}$$

$$= 0.5325$$

Now, $h_3 = h_{f_3} + x_3 h_{fg_3} = 59.7 + 0.5325 \times 138 = 133.2\ \frac{kJ}{kg}$

Also $h_1 = h_{f_4} = 59.7\ \frac{kJ}{kg}$

Theoretical C.O.P $= \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{123.8 - 59.7}{133.2 - 123.8} = 6.82$

Actual C.O.P $= \eta_{relative} \times (C.O.P)_{theoretical}$

$= 0.5 \times 6.82 = 3.41$

Heat extracted from 1kg of water at 20°c for the formation of 1kg ice at 0°c

$$= 1 \times 4.187 \times (20 - 0) + 335 = 418.74\ \frac{kJ}{kg}$$

Let m_{ice} = Mass of ice formed in kg/min

$$(C.O.P)_{Actual} = 3.41 = \frac{R_n(actual)}{W} = \frac{m_{ice} \times 418.74}{m\,(h_3 - h_2)}$$

$$= \frac{m_{ice} \times 418.74\ (kJ/min)}{6\,(133.2 - 123.8)\left(\frac{kJ}{min}\right)}$$

$$\therefore m_{ice} = \frac{6\,(133.2 - 123.8)}{418.74} = 0.459\ \frac{kg}{min}$$

$$= \frac{0.459 \times 60 \times 24}{1000}\ \text{tonne (in 24 hours)} = 0.661\ \text{ton}$$

Refrigerating effect (net), $R_n = h_2 - h_1$

$= 139.6 - h_{f4} = 139.6 - 81.17 = 58.43 \frac{kJ}{kg}$ $(\because h_4 = h_{f4})$

$\therefore (COP)_{theoretical} = \frac{R_n}{W} = \frac{58.43}{18} = 3.24$

Relative coefficient of performance $= \frac{(C.O.P)_{actual}}{(C.O.P)_{theoretical}}$

$\Rightarrow 0.45 = \frac{(C.O.P)_{actual}}{3.24}$

$\therefore (C.O.P)_{actual} = 0.45 \times 3.24 = 1.458$

Now heat extracted per minute $= (C.O.P)_{actual} \times [m \times (h_3 - h_2)]$

$= 1.458 \times [8.2(157.6 - 139.6)]$

$= 215.2 \frac{kJ}{min}$

Heat extracted per day $= 215.2 \times 60 \times 24 = 309888 \frac{kJ}{day}$

Heat extracted per kg of water (to produce ice at $0°C$)

$= 1 \times C_{pw} \times (15 - 0) + (h_{fg})_{ice}$

$= 1 \times 4.18 \times (15 - 0) + 335 = 397.7 \frac{kJ}{kg}$

$\therefore$ Mass of ice made per day $= \frac{309888}{397.7 \times 1000}$ tonne $= 0.779$ tonne Ans

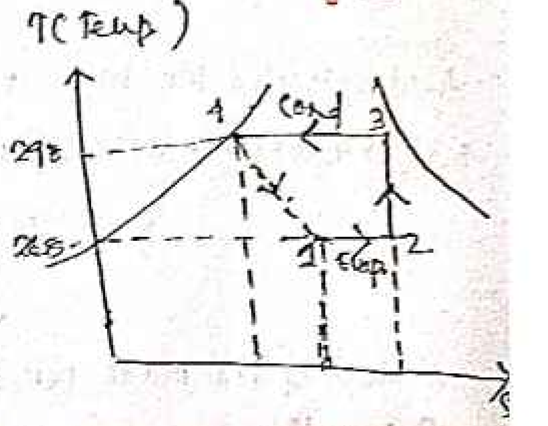
A refrigeration cycle is reqd to produce ice at $0°C$ from water at $20°C$. The m/c has a condenser temp. of 298K while the evaporator temp. is 268K. The relative efficiency of the m/c is 50%. & 6kg of Freon-12 refrigerant is circulated through the system per minute. The refrigerant enters the compressor with a dryness fraction of 0.6. Specific heat of water is $4.187 \frac{kJ}{kgK}$ & the latent heat of ice is $335 \frac{kJ}{kg}$. Calculate the amount of ice produced in 24 hours. The table of properties of Freon-12 is given below

Temp. (K)	Liquid Heat (kJ/kg)	Latent Heat (kJ/kg)	Entropy of Liquid (kJ/kgK)
298	59.7	138.0	0.2232
268	31.4	154.0	0.1251

Q:- Determine the theoretical C.O.P. for CO_2 machine working bet^n the temp. range of 25°c & −5°c. The dryness fraction of CO_2 gas during the suction stroke is 0.6. The following properties are given:

Temp. (°c)	Heat ($\frac{kJ}{kg}$)		Latent heat ($\frac{kJ}{kg}$)	Entropy ($\frac{kJ}{kg\,K}$)	
	Liquid	vapour		Liquid	vapour
25	81.17	202.5	121.34	0.251	0.644
−5	−7.53	236.8	245.2	−0.042	0.891

How many tonnes of ice would a machine working bet^n the same limit having a relative co-efficient of performance of 45% make in 24 hours? water for the ice is supplied at 15°c and the compressor takes 8.2 kg of CO_2 per minute. Specific heat of water may taken as 4.18 $\frac{kJ}{kg°c}$ and latent heat of ice as 335 $\frac{kJ}{kg}$.



Sol^n:- Given,

$T_3 = T_4 = 25+273 = 298\,K$

$T_1 = T_2 = -5+273 = 268\,K$

$x_2 = 0.6$

$(COP)_{relative} = 0.45$, $m = 8.2\,kg/min$

$C_{pw} = 4.18\,\frac{kJ}{kg°c}$, $h_{fg}(ice) = 335\,\frac{kJ}{kg}$

Let us first find dryness at point '3' (x_3)

Entropy at '2' = Entropy at '3'

$\Rightarrow s_{f2} + \frac{x_2 h_{fg2}}{T_2} = s_{f3} + \frac{x_3 h_{fg3}}{T_3}$

$\Rightarrow -0.042 + \frac{0.6\times245.2}{(-5+273)} = 0.251 + \frac{x_3\times121.34}{(25+273)}$

$\Rightarrow 0.042 + 0.549 = 0.251 + 0.407x_3$

$\Rightarrow 0.507 = 0.251 + 0.407\,x_3$

dryness fraction, $x_3 = \frac{(0.507-0.251)}{0.407} = 0.63$

Work Done, $W = h_3 - h_2$

$h_3 = h_{f3} + x_3 h_{fg3} = 81.17 + 0.63\times121.34 = 157.6\,\frac{kJ}{kg}$

$h_2 = h_{f2} + x_2 h_{fg2} = -7.53 + 0.6\times245.2 = 139.6\,\frac{kJ}{kg}$

$\therefore$ Work Done, $W = 157.6 - 139.6$

$= 18\,\frac{kJ}{kg}$

Enthalpy at point '2', $h_2 = h_{f2} + x_2 h_{fg2}$

$= 158.2 + 0.62 \times 1280.8 = 952.3 \ \frac{kJ}{kg}$.

Enthalpy at point '1' $\cdot$ $h_1 = h_{f4} = 298.9 \ \frac{kJ}{kg}$

Also entropy at point '2' = entropy at point '3'

i.e. $S_2 = S_3$

$s_{f2} + x_2 s_{fg2} = s_{f3} + x_3 s_{fg3}$

$0.630 + 0.62 \times \frac{1280}{(-5+273)} = 1.124 + x_3 \times \frac{1167.1}{(25+273)}$

i.e. $x_3 = 0.63$

$\therefore$ Enthalpy at point '3' $\Rightarrow h_3 = h_{f3} + x_3 h_{fg3}$

$= 298.9 + 0.63 \times 1167.1$

$= 1034.17 \ \frac{kJ}{kg}$.

$(C.O.P)_{theoretical} = \frac{h_2 - h_1}{h_3 - h_2}$

$= \frac{952.3 - 298.9}{1034.17 - 952.3} = \frac{653.4}{81.87} = 7.98$

$(COP)_{Actual} = (COP)_{relative} \times 7.98 = 0.55 \times 7.98 = 4.39$

Work done per kg of refrigerant $= h_3 - h_2 = 1034.17 - 952.3 = 81.87 \ \frac{kJ}{kg}$

Refrigerant in circulation, $m = 6.4$ kg/min

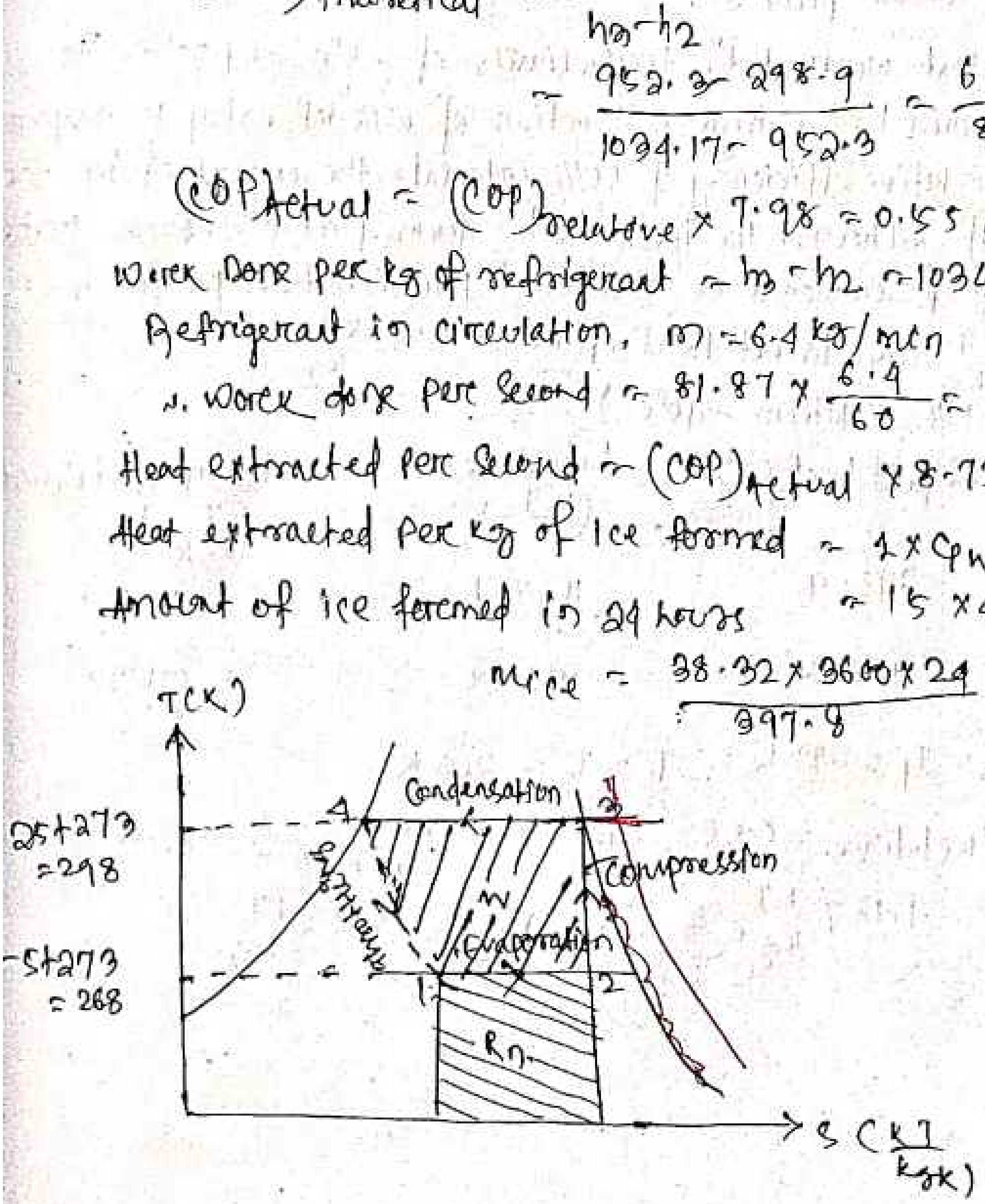
$\therefore$ Work done per second $= 81.87 \times \frac{6.4}{60} = 8.73 \ \frac{kJ}{s}$

Heat extracted per second $= (COP)_{Actual} \times 8.73 = 4.39 \times 8.73 = 38.32 \ \frac{kJ}{s}$

Heat extracted per kg of ice formed $= 1 \times C_{pw}(15-0) + h_{fg(ice)}$

Amount of ice formed in 24 hours $= 15 \times 4.187 + 335 = 397.8$ kJ

$m_{ice} = \frac{38.32 \times 3600 \times 24}{397.8} = 8322.9$ kg



2. When the vapour is superheated after compression

→ If the compression of the vapour is continued after it has become dry vapour will be superheated, its effect on T~s diagram is shown.

→ The vapour enters the compressor at condition '2' and is compressed to where it is superheated to temp. T_{sup}.

→ Then, it enters the condenser. Here firstly superheated vapour cools to te (represented by line 3-3') & then it condenses at constant temp. along line 3'-4, the remaining of the cycle, however, is the same as before.

Now, work done = area '2-3-3'-4-b-2'

& heat extracted / absorbed = area '2-1-g-f-2'

$$C.O.P = \frac{\text{Heat extracted}}{\text{Work Done}} = \frac{\text{area '2-1-g-f-2'}}{\text{area '2-3-3'-4-b-2'}} = \frac{h_2 - h_1}{h_3 - h_2}$$

In this case, $h_3 = h_3' + C_p (T_{sup} - T_{sat})$ & h_3' = enthalpy of dry & saturated vapour at the point 3'.

Q:- A refrigerating plant works bet^n temp. limits of $-5°C$ and $25°C$. The working fluid Ammonia has a dryness fraction of 0.62 at entry to compr. If the m/c has a relative efficiency of 55%, calculate the amount of ice fo during a period of 24 hours. The ice is to be formed at 0°C from wat at 15°C & 6.4 kg of ammonia is circulated per minute. Specific heat water is $4.187 \frac{kJ}{kg}$ and latent heat of ice is $335 \frac{kJ}{kg}$.

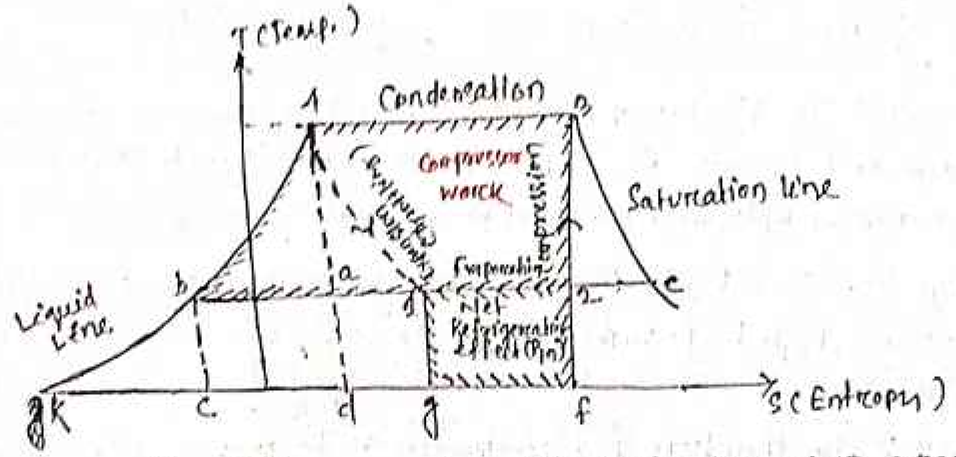
Properties of NH_3 (datum $-40°C$)

Temp (°C)	Liquid heat $\left(\frac{kJ}{kg}\right)$	Latent heat $\left(\frac{kJ}{kg}\right)$	Entropy of liqu $\left(\frac{kJ}{kgK}\right)$
25	298.9	1167.1	1.124
-5	158.2	1280.8	0.630

Sol^n:- Given $T_3 = T_4 = 298\,K$, $T_1 = T_2 = 268\,K$

$(COP)_{relative} = 0.55$, $m = 6.4 \frac{kg}{min}$, $x = 0.62$

$C_{pw} = 4.187 \frac{kJ}{kg°C}$, $(h_{fg})_{ice} = 335 \frac{kJ}{kg}$.



→ It then undergoes throttling expansion while passing through the expansion valve. and it again reduces to T_2, it is represented by the line 4-1 shown broken; throttling is an irreversible process. From the T~s Diagram it may noted due to this expansion the liquid partially evaporates, as its dryness fraction is represented by the ratio $\frac{b1}{b2}$.

→ At '1' it enters the evaporator where it is further evaporated at constant pressure & constant temp. to the point '2' & the cycle is completed.

Work done by the compressor = W = area '2-3-4-b-2'

Heat absorbed = area '2-1-g-f-2'

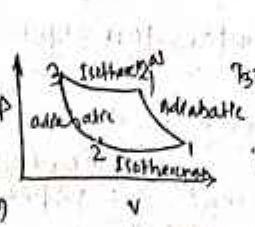
$$\therefore COP = \frac{\text{Heat extracted or Refrigerating effect}}{\text{Work Done}}$$

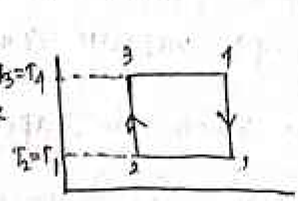
$$= \frac{\text{area '2-1-g-f-2'}}{\text{area '2-3-4-b-2'}}$$

$$\text{or, } COP = \frac{h_2 - h_1}{h_3 - h_2} = \frac{h_2 - h_{f4}}{h_3 - h_2}$$

(∵ $h_1 = h_{f4}$ since during the throttling expansion 4-1 the total heat content remains unchanged.)

Function of Evaporator





→ The function of an evaporator is to absorb heat from the surrounding location or medium, which is to be cooled.

→ It converts liquid refrigerant into vapour.

→ In the evaporator, the temp. of refrigerant must always less that of surrounding temp., so that heat flows to the refrigerant

Functions of parts of a simple vapour compression system

1. Compressor :- The function of a compressor is to remove the vapour from the evaporator and to raise its temp. & pr. to a point such that it (vapour) can be condensed with available condensing media.

2. Discharge line (or hot-gas line) :- A hot gas or discharge line delivers the high-pressure, high temp. vapour from the discharge of the compressor to the condenser.

3. Condenser :- The function of a condenser is to provide a heat transfer surface through which heat passes from the hot refrigerant vapour to the condensing medium.

4. Receiver tank : A receiver tank is used to provide storage for a condensed liquid so that a constant supply of liquid is available to the evaporator as required.

5. Liquid Line : A liquid line carries the liquid refrigerant from the receiver tank to the refrigerant flow control.

6. Expansion valve (Refrigerant flow control) :- Its function is to meter the proper amount of the refrigerant to the evaporator and to reduce the pressure of liquid entering the evaporator so that liquid will vaporise in the evaporator at the desired low temp. & take out sufficient amount of heat.

7. Evaporator : An evaporator provides a heat transfer surface through which heat pass from the refrigerated space into the vaporising refrigerant.

8. Suction Line : The suction line conveys low pressure vapour from the evaporator to the suction inlet of the compressor.

Simple vapour compression cycle on Temp.-Entropy (T-s) Diagram

1. When the vapour is dry & saturated at the end of Compression (partially condensed)

The fig. represents the vapour compression cycle on T~s Diagram. The points 1, 2, 3 and 4 corresponds to the points 1, 2, 3 and 4 in fig.

→ At point '2' the vapour which is at low temp. (T_2) & low pressure enters the compressor's cylinder and is compressed adiabatically to '3', where its temp. increases to the temp. T_3.

→ It is then condensed in the condenser (line 3-4) where it gives up its latent heat to the condensing medium.

Similarly, considering isentropic expansion 1-2, we get

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{5}\right)^{\frac{1.4-1}{1.4}} = \left(\frac{1}{5}\right)^{0.286} = 0.63$$

$$\therefore \frac{T_2}{301} = 0.63 \Rightarrow T_2 = 301 \times 0.63 = 189.6\,K$$

(i) Heat extracted from the cold chamber/hour

$$= m C_p (T_3 - T_2)$$
$$= 500 \times 1.003 (281 - 189.6)$$
$$= 45837\ kJ/h$$

(ii) Heat rejected in the cooling chamber (or to cooling water)

$$= m C_p (T_4 - T_1)$$
$$= 500 \times 1.003 (445.2 - 301)$$
$$= 72316\ kJ/h$$

Since both compression & expansion are reversible adiabatic, therefore, Work Done = Heat rejected − Heat abstracted

$$= 72316 - 45837 = 26479\ \frac{kJ}{h}$$

$$C.O.P. = \frac{\text{Heat abstracted in cold chamber}}{\text{Work Done}}$$

$$= \frac{45837}{26479} = 1.73$$

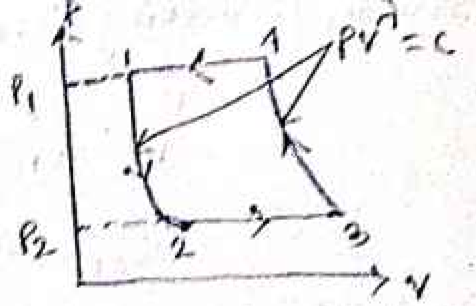
An open air cycle operated by air-refrigeration system is required to produce 6 tonnes of refrigerating effect with a cooler pressure of 11 bar abs. and a refrigerated space or region at a pressure of 1.05 bar. The temp. of air leaving the air cooler is 38°c and the air leaving the room is 16°c. Calculate.

(i) Mass of air circulated per minute.
(ii) Compressor displacement required per minute.
(iii) Expander displacement required per minute.
(iv) C.O.P.
(v) Power required per tonne of refrigeration.

Assume that theoretical cycle is operating with isentropic compression and expansion with no compressor clearance and no losses.

for air take $\gamma = 1.41$ & $C_p = 1.003\ \frac{kJ}{kg\,K}$

Sol^n:- $T_1 = 38+273 = 311K$



For isentropic expansion process 1-2, we have

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= \left(\frac{1.05}{11}\right)^{\frac{1.41-1}{1.41}}$$

$$= 0.505$$

$$\Rightarrow \frac{T_2}{311} = 0.505 \Rightarrow T_2 = 311 \times 0.505 = 157K$$

Also, $T_3 = 16+273 = 289K$

For isentropic compression process, 3-4, we have

$$\frac{T_4}{T_3} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow \frac{T_4}{289} = \left(\frac{11}{1.05}\right)^{\frac{1.41-1}{1.41}} = 1.98$$

$$\Rightarrow T_4 = 289 \times 1.98 = 572.2K$$

Refrigerating effect per kg of air circulated

$$= C_p(T_3 - T_2) = 1.003(289-157) = 132.4 \frac{kJ}{kg}$$

(i) Mass of air circulated per min

$$m = \frac{6 \times 14000}{132.4 \times 60} = 10.57 \text{ kg}$$

(ii) Compressor piston displacement,

$$v_3 = \frac{mRT_3}{p_2} = \frac{10.57 \times (0.287 \times 1000) \times 289}{1.05 \times 10^5}$$

$$= 8.35 \text{ m}^3/\text{min}$$

(iii) Expander Displacement

$$v_2 = \frac{mRT_2}{p_2} = \frac{10.57 \times (0.287 \times 1000) \times 157}{1.05 \times 10^5} = 4.5$$

For isentropic compression & expansion

$$COP = \frac{T_3 - T_2}{T_4 - T_3 + T_2 - T_1} = \frac{289-157}{572.2-289+157-311}$$

$$= \frac{132}{129.2} = 1.02$$

(iv) Total power reqd. $= \frac{mC_p(T_4 - T_3 + T_2 - T_1)}{60}$ kW

$$= \frac{10.57 \times 1.003(572.2-289+157-311)}{60}$$

$$= 22.83 \text{ kW}$$

$\therefore$ Power reqd. $= \frac{22.83}{6} = 3.8$ kW

But, $R = C_p \left(\frac{\gamma-1}{\gamma} \right) J$ (where $J = 1$ in S.I. units)

$$\therefore W_{comp} - W_{exp} = \left(\frac{n}{n-1} \right) \left(\frac{\gamma-1}{\gamma} \right) m C_p \left(T_4 - T_3 + T_2 - T_1 \right)$$

for isentropic compression & expansion,

$$W_{net} = m C_p \left(T_4 - T_3 + T_2 - T_1 \right)$$

According to law of conservation of energy the net work on the gas must be equivalent to the net heat rejected, therefore,

$$C.O.P = \frac{Q_{added}}{Q_{rejected} - Q_{added}} = \frac{Q_{added}\ (kJ)}{W_{net}\ (kJ)}$$

for the ~~cycle~~ air cycle assuming polytropic compression & expansion, the co-efficient of performance,

$$C.O.P = \frac{m C_p (T_3 - T_2)}{\left(\frac{n}{n-1} \right) \left(\frac{\gamma-1}{\gamma} \right) m C_p \left(T_4 - T_3 + T_2 - T_1 \right)}$$

$$= \frac{(T_3 - T_2)}{\left(\frac{n}{n-1} \right) \left(\frac{\gamma-1}{\gamma} \right) \left(T_4 - T_3 + T_2 - T_1 \right)} \rightarrow (6)$$

Note :- The reversed Brayton cycle is same as the Bell-Coleman cycle. Conventionally Bell-Coleman cycle refers to closed cycle with expansion taking place in reciprocating expander and compressor respectively and heat rejection and heat absorption taking place in condenser and evaporator respectively.

- Reversed Brayton cycle finds its application for air-conditioning of aeroplanes where air is used as refrigerant.

Merits and Demerits of Air-Refrigeration System

Merits:

(1) Since air is non-flammable, therefore there is no risk of fire as in the machine using NH_3 as the refrigerant.

(2) It is cheaper as air is easily available as compared to the other refrigerants.

(3) As compared to the other refrigeration systems the weight of air-refrigeration system per tonne of refrigeration is quite low, because of this reason this system is employed in aircrafts.

Since $P_1 = P_4$ & $P_2 = P_3$, therefore ~~eq^n (4)~~ from eq^n ② & eq^n

we have

$$C.O.P = \frac{T_4}{T_3} = \frac{T_1}{T_2} \quad \text{or} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Now substituting these values in eq^n ①, we get

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

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

where r_p = Compression or expansion ratio $= \frac{P_1}{P_2} = \frac{P_4}{P_3}$

If the process is considered to be polytropic, the steady flow work of compression is given by,

$$W_{comp.} = \frac{n}{n-1}(P_4 V_4 - P_3 V_3) \longrightarrow ④$$

Similarly work of expansion is given by,

$$W_{exp} = \frac{n}{n-1}(P_1 V_1 - P_2 V_2) \longrightarrow ⑤$$

Eq^n ④ & ⑤ may easily be reduced to the theoretical isentropic process shown in fig (b) by substituting $\gamma = n$ & the b relationship, $R = C_p\left(\frac{\gamma - 1}{\gamma}\right) J$

The net external work required for operation of the cycle.

= Steady flow work of Compression − Steady flow work expansion.

$= W_{comp.} - W_{exp.}$

$$= \left(\frac{n}{n-1}\right)(P_4 V_4 - P_3 V_3 - P_1 V_1 + P_2 V_2) \qquad \left[\begin{array}{l} \because P_1 V_1 = mRT \\ P_2 V_2 = mRT \\ P_3 V_3 = mR \\ P_4 V_4 = mR \end{array}\right.$$

$$= \left(\frac{n}{n-1}\right) mR\,(T_4 - T_3 - T_1 + T_2)$$

$$= \left(\frac{n}{n-1}\right) \frac{mR}{J}\,(T_4 - T_3 - T_1 + T_2) \text{ in heat units}$$

Consider m kg of air

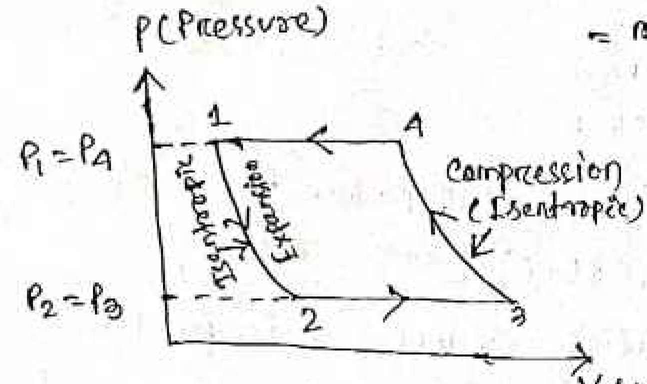
Heat absorbed in refrigerator $= Q_{added} = m \times c_p \times (T_3 - T_2)$

Heat rejected in cooler, $Q_{rejected} = m \times c_p \times (T_4 - T_1)$

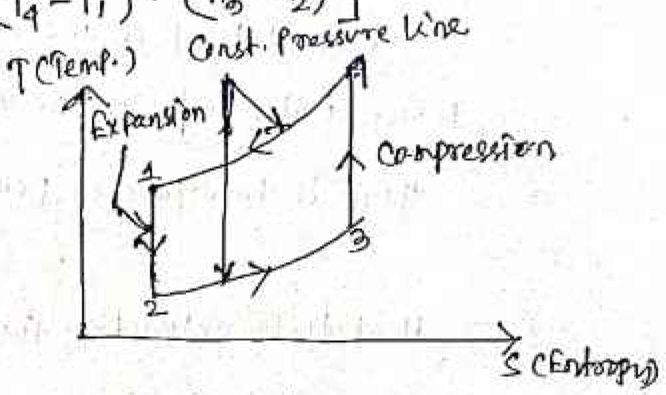
Work done during the cycle = Heat rejected − Heat absorbed

$$= m c_p (T_4 - T_1) - m c_p (T_3 - T_2)$$

$$= m c_p [(T_4 - T_1) - (T_3 - T_2)]$$



(a) P−V diagram



(b) (T−S diagram)

∴ Co-efficient of performance,

$$C.O.P = \frac{\text{Heat absorbed}}{\text{Work Done}}$$

$$= \frac{m c_p (T_3 - T_2)}{m c_p [(T_4 - T_1) - (T_3 - T_2)]}$$

$$= \frac{T_3 - T_2}{(T_4 - T_1) - (T_3 - T_2)}$$

$$= \frac{T_2 \left(\frac{T_3}{T_2} - 1 \right)}{T_1 \left(\frac{T_4}{T_1} - 1 \right) - T_2 \left(\frac{T_3}{T_2} - 1 \right)} \longrightarrow (1)$$

We know that for isentropic compression process 3−4,

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3} \right)^{\frac{\gamma - 1}{\gamma}} \longrightarrow (2)$$

Similarly, for isentropic expansion process 1−2,

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2} \right)^{\frac{\gamma - 1}{\gamma}} \longrightarrow (3)$$

Q:- An ice plant produces 12 tonnes of ice per day at 0°c, using water at room temperature of 15°c. If the COP of the plant is 2.6 and overall electro-mechanical efficiency is 87% determine the power rating of the compressor motor. Take specific heat of water as 4.18 $\frac{kJ}{kg}$ and latent heat of ice 335 $\frac{kJ}{kg}$.

Given mass of ice produced, m_{ice} = 12 tonnes

$t_{water} = 15°c$, $C.O.P = 2.6$, $\eta = 87\%$, $C_{pw} = 4.18 \frac{kJ}{kg}$

Latent heat of ice = 335 $\frac{kJ}{kg}$

Power rating of compressor motor :

Heat to be extracted per kg of water (to produce ice at 0°c)

$$= 1 \times 4.18(15-0) + 335 = 397.7 \frac{kJ}{kg}$$

Heat to be extracted for producing 12 tonnes of ice per day.

$$= 12 \times 1000 \times 397.7 \ kJ$$

$$\text{Refrigerating effect, } R_n = \frac{12 \times 1000 \times 397.7}{24 \times 3600} = 55.24 \frac{kJ}{s}$$

~~Ab~~ Also, $C.O.P. = \frac{R_n}{W}$

i.e. $2.6 = \frac{55.24}{W}$

$$\Rightarrow W = \frac{55.24}{2.6} = 21.2 \frac{kJ}{s} \ (= 21.2 kW)$$

∴ Power rating of the compressor motor

$$= \frac{W}{\eta} = \frac{21.2}{0.87} = 24.37 kW$$

Dt. 09/11/2021

Q1- A reserved carnot cycle is used for heating & cooling purpose. If the working supplied is 9.5kW & COP is 3.6 for cooling. find:

(i) T_2/T_1

(ii) The refrigerating effect in tonnes of refrigeration.

(iii) C.O.P. for heating

Given

Work Supplied, $W = 9.5kW$, $COP = 3.6$

(i) $\frac{T_2}{T_1} = ?$

We know that, $(C.O.P)_{cooling} = \frac{T_2}{T_1 - T_2}$ or $\frac{T_1 - T_2}{T_2} = \frac{1}{C.O.P} = \frac{1}{3.6}$

or, $\frac{T_1}{T_2} - 1 = \frac{1}{3.6}$ or $\frac{T_1}{T_2} = 1 + \frac{1}{3.6} = 1.278$

$\therefore \frac{T_2}{T_1} = 0.782$ Ans

(ii) Refrigerating Effect, R_n:

$C.O.P = \frac{R_n}{W}$ or $3.6 = \frac{R_n}{9.5}$

$\therefore R_n = 3.6 \times 9.5$ kJ/s

$= 3.6 \times 9.5 \times 3600$ $\frac{kJ}{h}$

$= \frac{3.6 \times 9.5 \times 3600}{14000} = 8.794$ tonnes

(iii) $(COP)_{heating} = \frac{T_1}{T_1 - T_2} = \frac{1}{1 - \frac{T_2}{T_1}} = \frac{1}{1 - 0.782} = 4.587$ Ans

:- A carnot Refrigerator requires 1.3kW per tonne of refrigeration to maintain a region at low temp. of $-35°c$. Determine:

(i) COP of Carnot refrigerator.

(ii) Higher temp of the cycle.

(iii) The heat delivered and COP, when this device is used as a heat pump.

Data Given

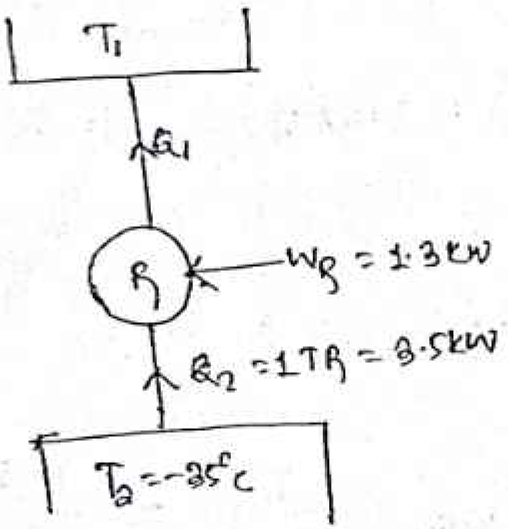
Heat extracted, $Q_2 = 1$ TR

$= 3.5kW$

Work Done on refrigerator

$W_R = 1.3kW$

Temp. of refrigerator, $T_2 = -35°c = 238K$



To find (i) $(COP)_R = ?$

(ii) Higher temp. of the cycle, $T_1 = ?$

(iii) Q_1 & $(COP)_P = ?$

(i) $(COP)_R = \frac{\text{Heat extracted}}{\text{Work Done}} = \frac{Q_2}{W_R}$

$= \frac{3.5}{1.3} = 2.7.$

(ii) But, we know $(COP)_R = \frac{T_2}{T_1 - T_2}$

$\Rightarrow 2.7 = \frac{238}{T_1 - 238}$

$\Rightarrow T_1 - 238 = \frac{238}{2.7} = 88.15$

$\Rightarrow T_1 = 326.15K = 53.15°c$

(iii) $(COP)_P = 1 + (COP)_R = 1 + 2.7 = 3.7$

$(COP)_P$ is unity greater than $(COP)_R$ when both are ope bet? same temp. limits.

But, $(COP)_P = \frac{\text{Heat Supplied}}{\text{Work Done}} = \frac{Q_1}{W_P} = \frac{Q_1}{W_R}$

$\Rightarrow 3.7 = \frac{Q_1}{1.3}$

$\Rightarrow Q_1 = 4.81kW$

Demerits:

(1) The C.O.P of this system is very low in comparison to other systems.

(2) The weight of air required to be circulated is more compared with refrigerants used in other systems. This is due to the fact that heat is carried by air in the form of sensible heat.

Q:- One kg of air at a pressure of 1.05 bar and a temperature of 20°c is compared to 6 bar. It is then cooled to 27°c in the cooler before entering the expansion cylinder. Assuming compression and expansion as isentropic processes, determine:

(i) Refrigerating effect per kg of air

(ii) Theoretical C.O.P.

Take $C_p = 1.0$ & $\gamma = 1.4$

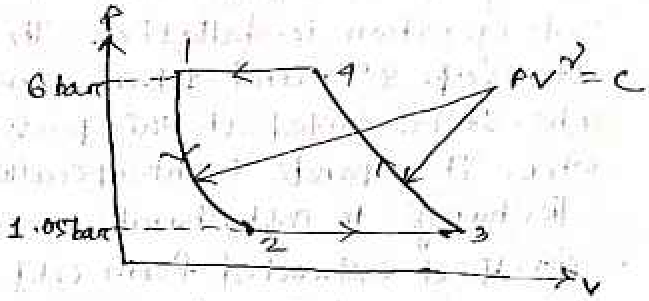
Given

$T_3 = 20 + 273 = 293K$

$T_1 = 27 + 273 = 300K$

$P_2 (= P_3) = 1.05$ bar

$P_1 (= P_4) = 6$ bar



(i) Refrigerating effect per kg of air

Consider the isentropic compression 3-4, we have

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\Rightarrow \frac{T_4}{293} = \left(\frac{6}{1.05}\right)^{\frac{1.4-1}{1.4}} = 1.645$$

$$\Rightarrow T_4 = 293 \times 1.645 = 482K$$

Similarly, considering isentropic expansion 1-2, we get

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

$$\Rightarrow \frac{T_2}{300} = \left(\frac{1.05}{6}\right)^{\frac{1.4-1}{1.4}} = 0.608$$

$$\Rightarrow T_2 = 300 \times 0.608 = 182.4K$$

New refrigerating effect per kg of air

$$= 1 \times C_p (T_3 - T_2)$$

$$= 1 \times 1 \times (293 - 182.4)$$

$$= 110.6 \frac{kJ}{kg}$$

(ii) Theoretical C.O.P.

Heat rejected to the cooler per kg of air

$= 1 \times C_p (T_4 - T_1)$

$= 1 \times 1 \times (482 - 300) = 182 \frac{kJ}{kg}$

$\therefore$ Work Done per kg of air

$=$ Heat rejected $-$ Heat abstracted

$= 182 - 110.6 = 71.4 \frac{kJ}{kg}$

$\therefore$ Theoretical C.O.P. $= \frac{\text{Heat abstracted}}{\text{Work Done}} = \frac{110.6}{71.4} = 1.55$

Q:- 500kg of atmospheric air is circulated per hour in an open type refrigeration installation. The air is drawn from the cold chamber at temp. 8°c and 1 bar, and then compressed isentropically to 5 bar abs. It is cooled at this pressure to 28°c and then led to the expander where it expands isentropically down to atmospheric pressure and is discharge to cold chamber. Determine :

(i) Heat extracted from cold chamber per hour.

(ii) Heat rejected to cooling water per hour.

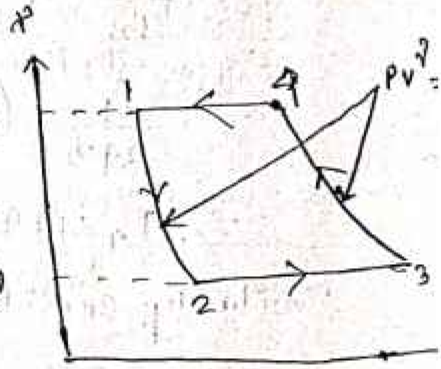
(iii) COP of the system.

for air, take $\gamma = 1.4$ & $C_p = 1.003 \frac{kJ}{kgK}$

Sol^n:- Given, $T_3 = 8 + 273 = 281K$

$T_1 = 28 + 273 = 301K$

$P_2 = 1$ bar, $P_1 = 5$ bar



Considering the isentropic compression 3-4, we have

$$\frac{T_4}{T_3} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\Rightarrow \frac{T_4}{281} = \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$= (5)^{0.286} = 1.58$$

$$\Rightarrow T_4 = 445.2K$$

Q:- A refrigerating system operates on the reversed carnot cycle. The higher temp. of the refrigerant in the system is 35°c & the lower temp. is -15°c. The capacity is to be 12 tonnes. Neglect all losses. Determine:

(i) Co-efficient of performance.

(ii) Heat rejected from the system per hour.

(iii) Power Reqd.

Given $T_1 = 273+35 = 308\,K$, $T_2 = 273-15 = 258\,K$

Capacity = 12 tonnes

(i) <u>COP</u>

$$COP = \frac{T_2}{T_1 - T_2} = \frac{258}{308-258} = 5.16$$

(ii) Heat rejected from the system per hour:

$$C.O.P = \frac{\text{Refrigerating effect}}{\text{Work I/P}}$$

$$\Rightarrow 5.16 = \frac{12 \times 14000 \frac{kJ}{hr}}{\text{Work I/P}}$$

$$\therefore \text{Work I/P} = \frac{12 \times 14000}{5.16} = 32558 \frac{kJ}{hr}.$$

Thus, Heat rejected per hour

= Refrigerating effect per hour + Work I/P per hour.

$$= 12 \times 14000 \frac{kJ}{hr} + 32558 \frac{kJ}{hr}.$$

$$= 200558 \frac{kJ}{h}$$

(iii) Power reqd.

$$= \frac{\text{Work I/P / hour}}{60 \times 60} = \frac{32558}{3600} = 9.04\,kW$$

Q:- A refrigerating system operates on the reversed carnot bet^n. temp. limits of 25°C & −10°C. The capacity is to be 8
Determine: (i) co-efficient of performance
(ii) Power rating of the compressor motor if the electro-mechanical efficiency is 85%
(iii) Heat rejected from the system per min.

Sol^n:- Given

$T_1 = 273 + 25 = 298K$

$T_2 = 273 + (-10) = 263K$

Capacity = 8 tonnes, $\eta = 85\%$

(i) co-efficient of performance (C.O.P)

$$C.O.P = \frac{T_2}{T_1 - T_2} = \frac{263}{298-263} = 7.51$$

(ii) Power rating of the compressor motor.

We know that, $COP = \frac{R_n \text{ (Refrigerating effect)}}{W \text{ (Work Done)}}$

i.e. $7.51 = \frac{8 \times 14000 \left(\frac{KJ}{hr}\right)}{W}$

$$\Rightarrow W = \frac{8 \times 14000}{7.51} \frac{KJ}{hr} = \frac{8 \times 14000}{7.51 \times 3600} \frac{KJ}{s} = 4.$$

∴ Power rating of the compressor motor = 4.

$$= \frac{4.14}{0.85} = 4.87 \text{ kW}$$

(iii) Heat rejected from the system:

= Refrigerating Effect + Work Done.

$$= \frac{8 \times 14000}{60}\left(\frac{KJ}{min}\right) + 4.14 \times 60 \left(\frac{KJ}{min}\right)$$

$= 1866.67 + 248.4$

$= 2115.07 \frac{KJ}{min}$

Concept of Psychrometry and Psychrometrics   Dt. 29/12/2021

- The art of measuring the moisture content of air is termed "Psychrometry".
- The science which investigates the thermal properties of moist air, considers the measurement and control of the moisture content of air and studies the effect of atmospheric moisture on material and human comfort may properly be termed "psychrometrics".

Air consists of fixed gases principally, nitrogen & oxygen with an admixture of water vapour in varying amounts.

Dry Air

→ The international Joint committee on psychrometric Data has adopted the following exact compositions of air expressed in mole fractions (volumetric): oxygen 0.2095, Nitrogen 0.7809, Argon 0.0093, carbon dioxide 0.0003.

→ Molecular weight of air for all air-conditioning calculations will be taken as 28.97. Hence the gas constant.

$$R_{air} = \frac{8.3143}{28.97} = 0.287 \frac{kJ}{kgK}$$

→ Dry air is never found in practice. Air always contains some moisture. Hence the common designation "air" usually means moist air.

→ The term 'dry air' is used to indicated the water free contents of air having any degree of moisture.

- Moist air 2. Saturated air:

→ Moist air is said to be saturated when its condition is such that it can co-exist in natural equilibrium with an associated condensed moisture phase presenting a flat surface to it.

→ For a given temp., a given quantity of air can be saturated with a fixed quantity of moisture.

- At saturation, vapour pressure of moisture in air corresponds to the saturation pressure given in steam tables corresponding to the given temp of air.
- Dry-bulb temperature (DBT) :- It is the temp. of air as registered by an ordinary thermometer (t_{db})
- Wet-bulb temp. (WBT) :- It is the temp. registered by a thermometer when the bulb is covered by a wetted wick & is exposed to a current of rapidly moving air (t_{wb}).

5. Adiabatic Saturation temp. :- It is the temp. at which the water or ice can saturate air by evaporating adiabatically into it. It is numerically equivalent to the measured wet bulb temp. (t_{wb})

6. Wet bulb depression :- It is the difference betn. dry-bulb and wet bu temp. ($t_{db} - t_{wb}$).

7. Dew Point Temperature (DPT) :- It is the temp. to which air must be cooled at constant pressure in order to cause condensation of any of water vapour. It is equal to steam table saturation temp. correspondi to the actual partial pressure of water in the air (t_{dp})

8. Dew point depression :- It is the difference betn. the dry bulb and dew point temp. ($t_{db} - t_{dp}$)

9. Specific humidity (Humidity ratio) :-

→ It is the ratio of the mass of water vapour per unit mass o dry air in the mixture of vapour and air. It is generally express as grams of water per kg of dry air.

→ for a given barometric pressure it is a function of dew point alone.

10. Relative Humidity (RH) (ϕ) :- It is the ratio of the partial pressur water vapour in the mixture to the saturated partial pressure at the bulb temp., expressed as percentage.

11. Sensible heat :- It is the heat that changes the temp. of a su when added to or abstracted from it.

12. Latent Heat :- It is the heat that does not affect the temp. but changes the state of a substance when added to or abstracte from it.

13. Enthalpy :- It is the combination energy which represents the of internal & flow energy in a steady flow process. It is deter from an arbitrary datum point for the air mixture & is expre as kJ per kg of dry air (h).

Psychrometric Relations

Dt. 31/12/2021

1. Pressure:

- Dalton's Law of Partial Pressure is employed to determine the pressure of a mixture of gases.

This law states that the total pressure of a mixture of gases is equal to the sum of partial pressures which the component gases would exert if each existed alone in the mixture vol. at the mixture temp.

- Precise measurements have shown that this law as well as Boyle's & Charles's Laws are only approximately correct. Modern tables of atmospheric air properties are based on the correct versions.

- For calculating partial pressure of water vapour in the air many equations have been proposed, probably Dr. Carrier's eq^n. is most widely used.

$$P_v = (P_{vs})_{wb} - \frac{[P_t - (P_{vs})_{wb}](t_{db} - t_{wb})}{1527.4 - 1.3 t_{wb}} \rightarrow ①$$

Where, P_v = Partial pressure of water vapour.

P_{vs} = Partial pressure of water vapour when air is fully saturated.

P_t = Total pressure of moist air.

t_{db} = Dry bulb temp. (°c) and

t_{wd} = Wet bulb temp (°c)

2. Specific Humidity W:

$$\text{Specific humidity} = \frac{\text{Mass of water vapour}}{\text{Mass of dry air}}$$

$$\text{or, } W = \frac{m_v}{m_a}$$

$$\text{Also, } m_a = \frac{P_a V}{R_a T} \rightarrow ②$$

$$\text{and, } m_v = \frac{P_v \times V}{R_v \times T} \rightarrow ③$$

Where, P_a = Partial pressure of dry air

P_v = Partial Pressure of water vapour.

V = Volume of mixture.

R_a = characteristic gas constant for dry air

and R_v = Characteristic gas constant for water vapour.

From Eq^n. ② & ③

$$W = \frac{P_v \times V}{R_v \times T} \times \frac{R_a T}{P_a V} = \frac{R_a}{R_v} \times \frac{P_v}{P_a}$$

$$\text{But } R_a = \frac{R_0}{M_a} \left(= \frac{8.3143}{28.97} = 0.287 \frac{kJ}{kg\,K} \text{ in SI units} \right)$$

5. Adiabatic Saturation temp. :- It is the temp. at

$$\therefore R_v = \frac{R_0}{M_v} \left(= \frac{8.3143}{18} = 0.462 \frac{kJ}{kgK} \text{ in SI units} \right)$$

where R_0 = Universal gas constant

M_a = Molecular weight of air, and

M_v = Molecular weight of water vapour

$$\therefore W = \frac{0.287}{0.462} \times \frac{P_v}{P_a} = 0.622 \frac{P_v}{P_t - P_v}$$

$$\text{i.e. } W = 0.622 \frac{P_v}{P_t - P_v} \longrightarrow (4)$$

The masses of air and water vapour in terms of specific volumes are given by expression as

$$m_a = \frac{V}{v_a} \text{ and } m_v = \frac{V}{v_v}$$

where v_a = Specific volume of dry air

v_v = Specific volume of water vapour

$$W = \frac{v_a}{v_v} \longrightarrow (5)$$

7. Degree of Saturation

Degree of Saturation = Mass of water vapour associated with unit of dry [air] / Mass of water vapour associated with [unit] of dry saturated air.

$$\mu = \frac{W}{W_s} \rightarrow (6)$$, where W_s = Specific humidity when air is fully saturated.

$$\mu = \frac{0.622\left(\frac{P_v}{P_t - P_v}\right)}{0.622\left(\frac{P_{vs}}{P_t - P_{vs}}\right)} = \frac{P_v(P_t - P_{vs})}{P_{vs}(P_t - P_v)}$$

$$= \frac{P_v}{P_s}\left[\frac{\left(1 - \frac{P_{vs}}{P_t}\right)}{\left(1 - \frac{P_v}{P_t}\right)}\right] \longrightarrow (7)$$

where P_{vs} = Partial Pressure of water vapour when air is fully saturated (P_{vs} can be calculated from steam tables corresponding to the dry bulb temp. of the air)

... compression cycle following are the properties of the refrigerant R-12 at various points. Dt. 04/01/2022

Compressor Inlet : $h_2 = 183.2 \frac{kJ}{kg}$, $v_2 = 0.0767\ m^3/kg$

Compressor Discharge : $h_3 = 222.6 \frac{kJ}{kg}$, $v_3 = 0.00164\ m^3/kg$

Condenser exit : $h_{f4} = 84.9 \frac{kJ}{kg}$, $v_4 = 0.00083\ m^3/kg$

The piston displacement volume for compressor is 1.5 ltrs per stroke and its volumetric efficiency is 80%. The speed of the compressor is 1600 rpm.

Find (i) Power rating of the compressor (kW)

(ii) Refrigerating effect (kW)

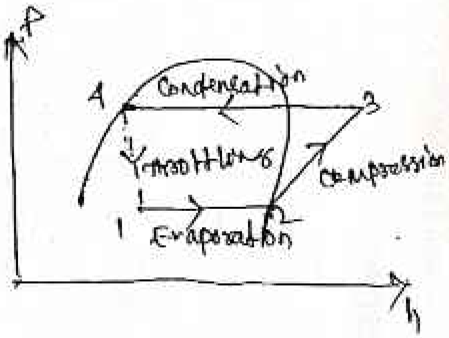
1 Ltr. = 0.001 m³

Solⁿ:- Piston Displacement vol.

$$= \frac{\pi}{4} D^2 \times L = 1.5\ ltr.$$

$$= 1.5 \times 1000 \times 10^{-6}\ m^3/stroke$$

$$= 0.0015\ m^3/revolution$$



(i) Power rating of the compressor (kW)

Compressor discharge

$$= 0.0015 \times 1600 \times 0.8 (\eta_{vol.})$$

$$= 1.92\ m^3/min$$

Mass flow rate of compressor

$$\dot{m} = \frac{\text{Compressor discharge}}{v_2}$$

$$= \frac{1.92}{0.0767} = 25.03 \frac{kg}{min}$$

Power rating of the compressor $= \dot{m}(h_3 - h_2)$

$$= \frac{25.03}{60}(222.6 - 183.2)$$

$$= 16.44\ kW \text{ Ans}$$

(ii) Refrigerating Effect (kW):

Refrigerating effect $= \dot{m}(h_2 - h_1) = \dot{m}(h_2 - h_{f4})$ $(\because h_1 = h_{f4})$

$$= \frac{25.03}{60}(183.2 - 84.9)$$

$$= 41\ kW$$

Find vapour pressure, specific humidity and the enthalpy of saturated air at a temp. of 32°C.

Sol. Since the air is saturated, therefore,

$$t_{db} = t_{wb} = t_{dp} = 32°c$$

From steam table corresponding to 32°c.

(i) Vapour pressure, $P_v (= P_{vs}) = 0.0476$ bar Ans

(ii) Specific humidity, $W = \frac{0.622 P_v}{P_t - P_v} = \frac{0.622 \times 0.0476}{1.0312 - 0.0476}$ ($\because P_t = 760$ mm Hg $= 1.01312$ bar ... atmospheric pressure)

$= 0.03$ kg/kg of dry air

(iii) Enthalpy of saturated air,

$$h = 1.005 t_{db} + W(2500 + 1.88 t_{db}) \frac{kJ}{kg} \text{ of dry air}$$

$$= 1.005 \times 32 + 0.03(2500 + 1.88 \times 32)$$

$$= 32.16 + 76.8 = 108.96 \frac{kJ}{kg} \text{ of dry air.}$$

Q. The atmospheric conditions are 20°c & specific humidity of 0.0095 kg/kg of dry air. Calculate the following:

(i) Partial pressure of vapour (ii) Relative humidity (iii) Dew point temp.

Sol.:- Dry bulb temp, $t_{db} = 20°c$

sp. humidity, $W = 0.0095$ kg/kg of dry air

(i) Partial pressure of vapour, P_v:

The sp. humidity is given by $W = \frac{0.622 P_v}{P_t - P_v}$

$$\Rightarrow 0.0095 = \frac{0.622 P_v}{1.0132 - P_v}$$

$$\Rightarrow 0.0095(1.0132 - P_v) = 0.622 P_v$$

$$\Rightarrow 0.009625 - 0.0095 P_v = 0.622 P_v$$

$$\Rightarrow P_v = 0.01524 \text{ bar.}$$

(ii) Relative Humidity, ϕ:

Corresponding to 20°c, from steam tables

$P_{vs} = 0.0234$ bar

$\therefore$ Relative Humidity, $\phi = \frac{P_v}{P_{vs}} = \frac{0.01524}{0.0234} = 0.65$ or 65%

(iii) Dew point temp, t_{dp}:

The dew point temp. is the saturation temp. of water vapour at a pr. of 0.01524 bar

t_{dp} [from steam tables by interpolation]

$$= 13 + \frac{(14-13)}{(0.01598 - 0.0150)} \times (0.01524 - 0.0150)$$

$$= 13 + \frac{0.00024}{0.00098} = 13.24°C$$

Dt-09/01/1

5. Enthalpy of moist air (h)

It is the sum of enthalpy of dry air and enthalpy of water vapour associated with dry air. It is expressed in $\frac{kJ}{kg}$ of dry air.

$$h = h_{air} + W \cdot h_{vapour}$$

$$= C_p t_{db} + W \cdot h_{vapour}$$

Where, h = Enthalpy of mixture / kg of dry air

h_{air} = Enthalpy of 1 kg of dry air.

h_{vapour} = Enthalpy of 1 kg of vapour obtained from steam tables

W = Specific humidity in kg / kg of dry air

and C_p = Specific heat of dry air normally assumed as 1.005

Also $h_{vapour} = h_g + C_{ps}(t_{db} - t_{dp})$

Where h_g = Enthalpy of saturated steam at dew point temp.

and $C_{ps} = 1.88 \frac{kJ}{kgK}$

$$\therefore h = C_p t_{db} + W[h_g + C_{ps}(t_{db} - t_{dp})] \longrightarrow (1)$$

$$= (C_p + C_{ps}W)t_{db} + W(h_g - C_p t_{dp}) \longrightarrow (2)$$

$$= C_{pm} t_{db} + W(h_g - C_{ps} t_{dp}) \longrightarrow (3)$$

Where

$C_{pm} = (C_p + C_{ps}W)$ is the Specific heat of humid a or humid specific heat.

The value of C_{pm} is taken as 1.002 ~~kg~~ kJ / kg dry air per

It is the heat capacity of $(1+W)$ kg of moisture per kg o dry air.

$h_{vapour} = h_g$ at dry bulb temp. So,

$$h = C_p t_{db} + W h_g \longrightarrow (3)$$

- However, a better approximation is given by the following relationship :

$$h_{vapour} = 2500 + 1.88\, t_{db} \text{ kJ/kg of water va}$$

Where t_{db} is dry bulb temp. in °C, and the datum state is liquid water at °C.

$$\therefore h = 1.005\, t_{db} + W(2500 + 1.88\, t_{db}) \text{ kJ/kg dry} \longrightarrow (5)$$

Given, $t_{db} = 32^\circ c$, $W = 13.1$ g/kg of air, $P_t = 758$ mm Hg.

(i) Partial pressure of vapour P_v:

$$W = \frac{0.622}{P_t - P_v} \quad \text{or,} \quad \frac{13.1}{1000} = \frac{0.622 P_v}{758 - P_v}$$

or, $0.131(758 - P_v) = 0.622 P_v$

or, $10.157 - 0.0134 P_v = 0.622 P_v$

$P_v = 15.98$ mm Hg.

(ii) Relative Humidity; ϕ:

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

Where P_{vs} = Saturation Pressure of dry bulb temp. of $32^\circ c$.

$= 0.0476$ bar

$= 0.0476 \times \frac{758}{1.0132} = 35.61$ mm Hg.

$\therefore \phi = \frac{15.98}{35.61} = 0.4487$ or 44.87%

iii) Dew point temp, t_{db}:

The dew point temp. is the saturation temp. of water vapour at a pressure of 15.98 mm Hg (0.0214 bar). from steam tables this temp. is found as follow.

t_{dp} (from steam tables by interpolation)

$$= 18 + \frac{19-18}{0.022-0.0206} \times [0.0214 - 0.0206]$$

$= 18.57^\circ C$

(ii) Refrigerating capacity :

Refrigerating capacity = $m(h_2 - h_1)$

[where m = mass flow rate of refrigerant = 1 kg/min, given]

$= 1 \times (183.1 - 26.85) = 156.25$ kJ/min

Compressor power :

Compressor power $= m(h_3 - h_2) = 1 \times (220 - 183.1)$

$= 36.9 \frac{kJ}{min}$

or, $0.615 \frac{kJ}{s}$

$= 0.615$ kW

Dt. 07/10/01/2022

Q:- The atmospheric conditions are 32°C & specific humidity of 13.4 g/kg of air. Determine :

(i) Partial pressure of vapour (ii) Relative humidity

(iii) Dew point temp.

~~Assume that condition of atmospheric air is 30°C & 55% RH & pressure is 1.0132 bar~~ Atmospheric Pr. = 758 Hg

Sol:- Corresponding to 30° c, from steam tables, $P_{vs} = 0.0425$ bar

$\therefore$ Relative humidity (RH), $\phi = \frac{P_v}{P_{vs}}$

i.e. $0.55 = \frac{P_v}{0.0425}$

$\therefore P_v = 0.02337$ bar

Also the specific humidity, $W = \frac{0.622 P_v}{P_t - P_v} = \frac{0.622 \times 0.02337}{1.0132 - 0.02337}$

$= 0.01468$ kg/kg of dry air.

The specific humidity after removing 0.04 kg of water vapour become, $0.01468 - 0.004 = 0.01068$ kg/kg of