Vapour compression Refrigeration System

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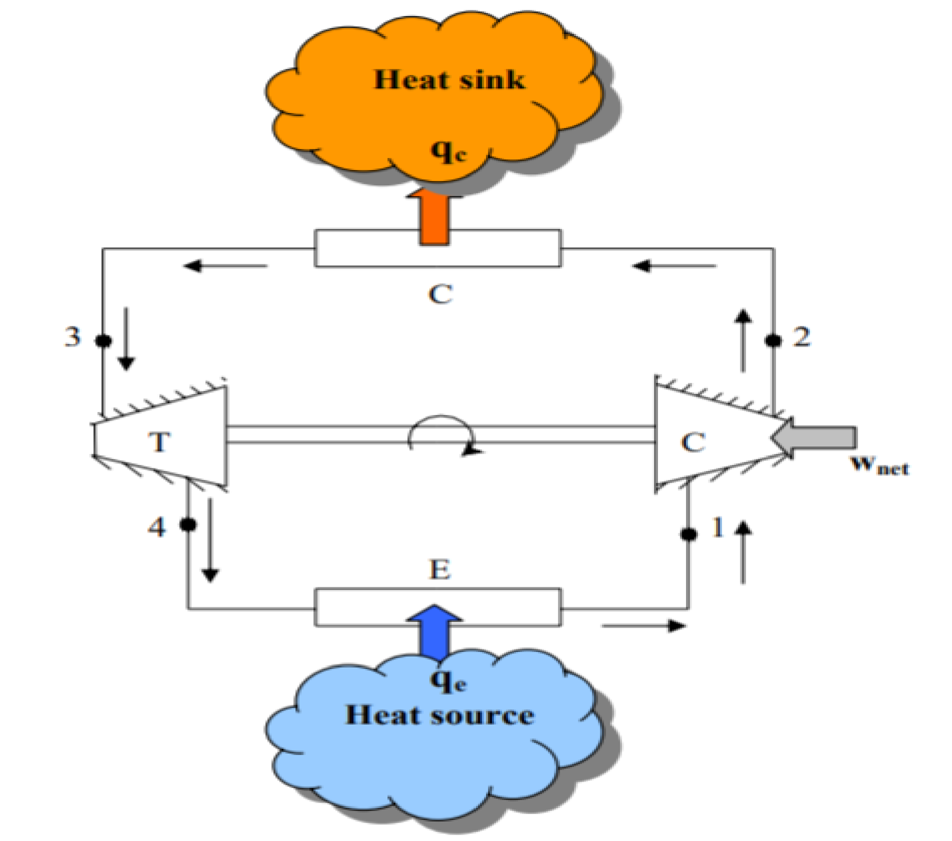
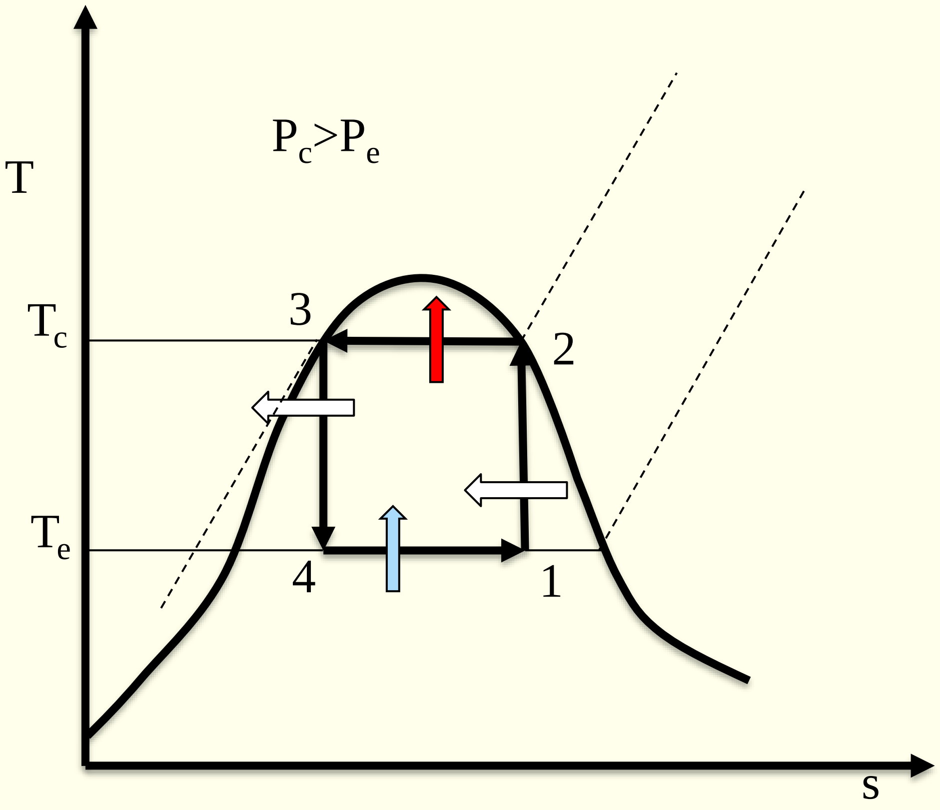
Carnot vapor compression refrigeration cycle

- Carnot refrigeration cycle is a completely reversible cycle
- It is used as a model of perfection for a refrigeration cycle operating between a constant temperature heat source and sink
- It is used as reference against which the real cycles are compared

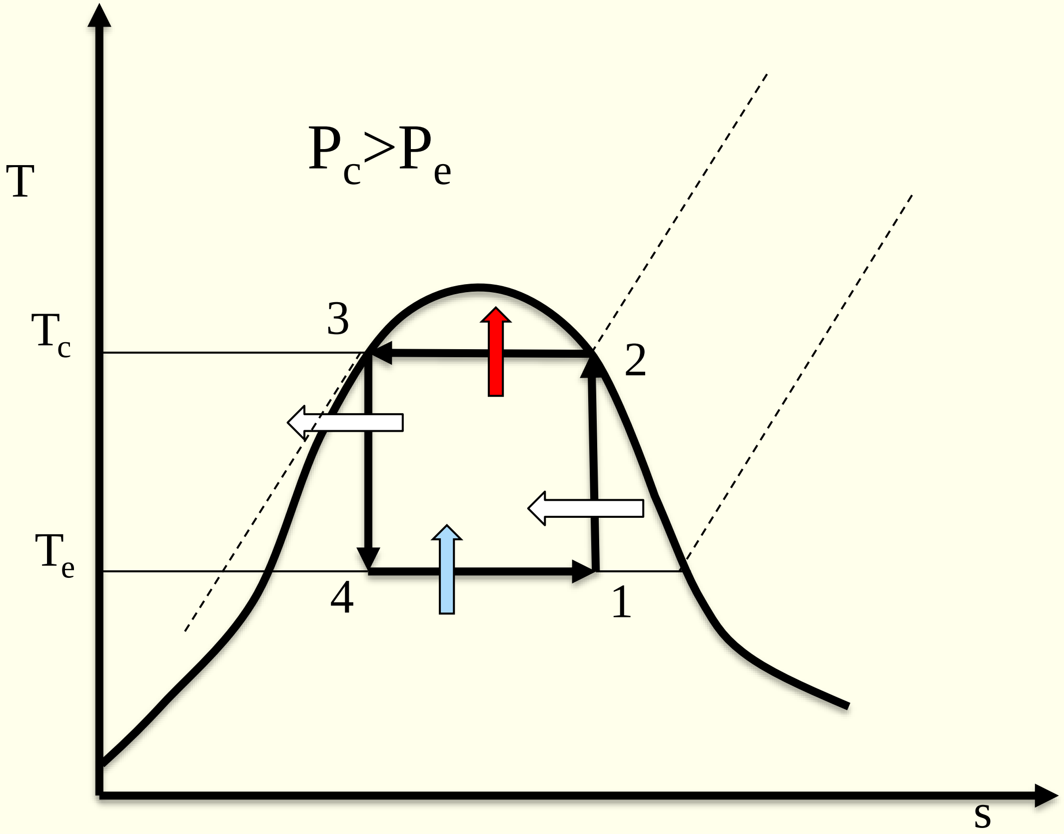
Carnot VCRS





Carnot VCRS

- From 1st and 2nd law
- $\oint \delta q = \oint \delta w$
- $\oint \delta q = q_{4-1} - q_{2-3} = q_e - q_c$
- $\oint \delta w = w_{3-4} - w_{1-2}$
- $= w_T - w_C = w_{net}$

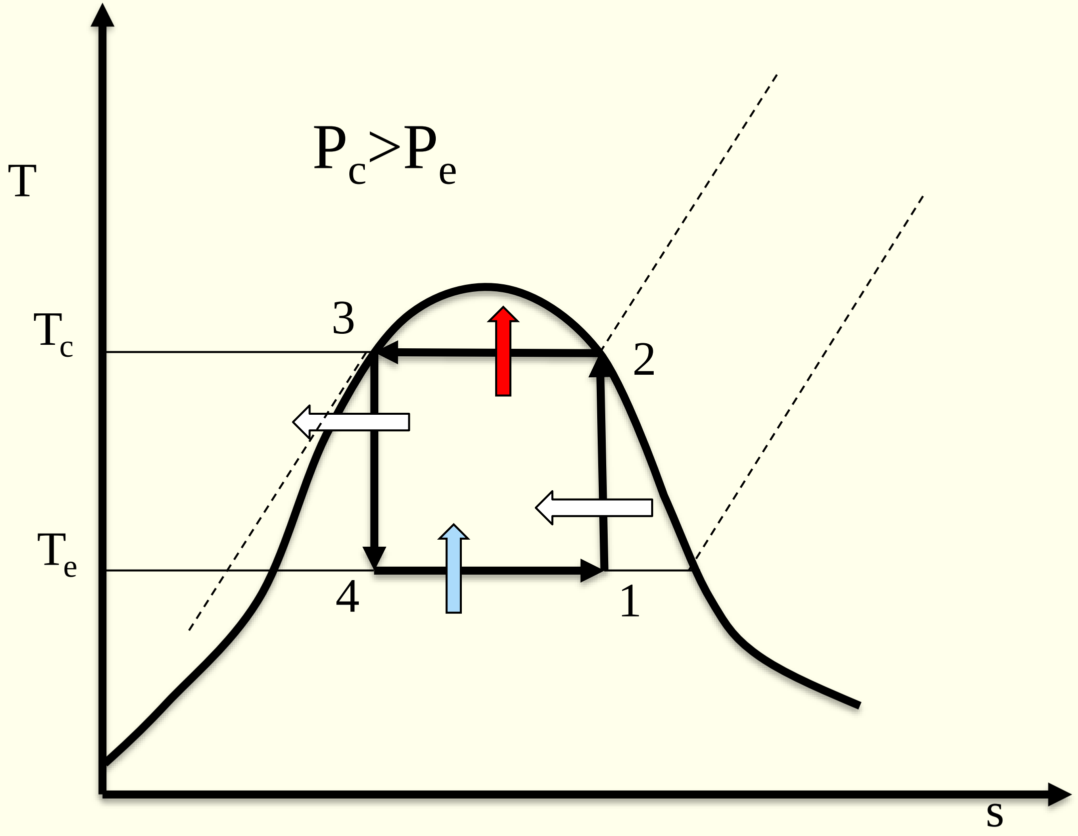


Carnot VCRS

- $(q_e - q_c) = w_{net}$

- $q_c = -q_{2-3} = -\int_2^3 T ds$
  $= T_c\,(s_2 - s_3)$

- $q_e = -q_{4-1} = -\int_4^1 T ds$
  $= T_e(s_1 - s_4)$



Carnot VCRS

- Since process 1-2 & 3-4 isentropic

$$s_1 = s_2 \quad \& \; s_3 = s_4$$

- The COP of Carnot system given by

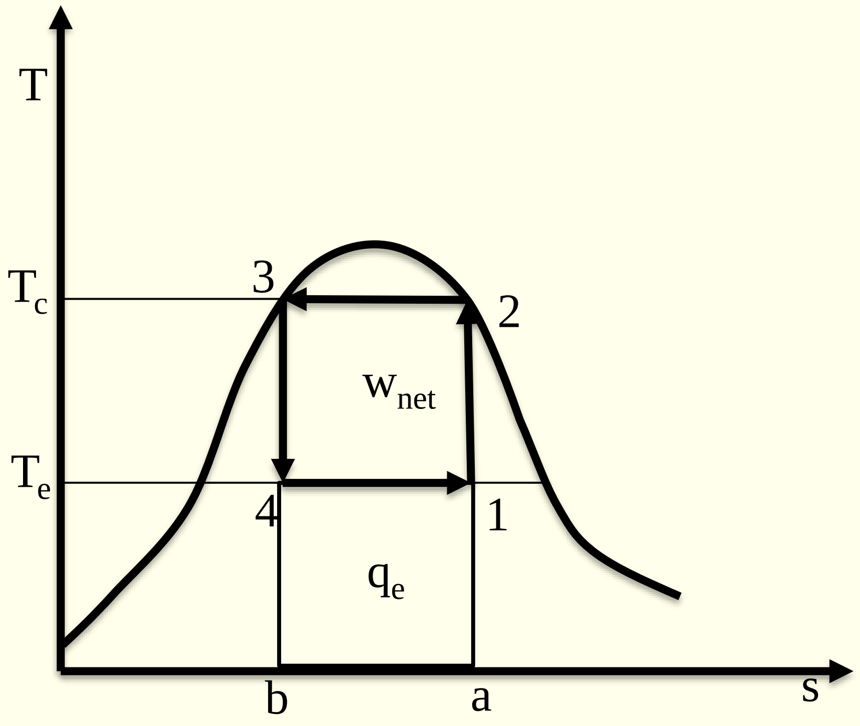
$$COP_{carnot} = \frac{RE}{net\ work\ input} = \frac{q_e}{w_{net}} = \frac{T_e(s_1 - s_4)}{T_c(s_2 - s_3) - T_e(s_1 - s_4)} = \frac{T_e}{T_c - T_e}$$

$$\Rightarrow COP_{carnot} = f(T_e, T_c)$$

COP of Carnot cycle

- COP of Carnot cycle is a function of heat source and sink temperatures only.
- COP is independent of the nature of the working fluid .
- Between the same heat source and sink, Carnot COP is the maximum possible COP.
- COP of Carnot cycle increases as evaporator temperature increases and condenser temperature decreases

COP of Carnot cycle



$q_e \uparrow$ as $T_e \uparrow$

$w_{net} \downarrow$ as $T_e \uparrow$ and $T_c \downarrow$

$COP \uparrow$ as $T_e \uparrow \quad T_c \downarrow$

1-

A Carnot refrigerator operates with Refrigerant R-134a as a refrigerant condensing at 50°C and evaporating at -15°C. Heat required to be pumped is 100MJ/hr. Find:

1. **COP using Carnot expression and properties of R134a.**
2. **mass flow rate of refrigerant**
3. **Determine power consumption per ton of refrigeration.**

1.

$T_L = 258K \qquad T_H = 323K$

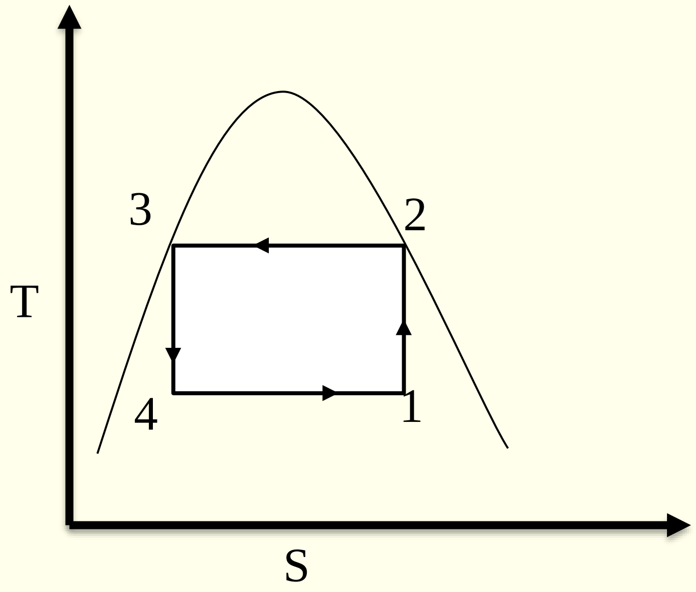
$$COP_{Carnot} = \frac{T_L}{T_H - T_L} = 3.9692$$

COP from Carnot expression

COP from refrigerant properties

Temperature (°C)	**Pressure(MPa)**	**Specific Enthalpy(kJ/kg)**		**Specific Entropy (kJ/kg K)**	
		Liquid	**Vapor**	**Liquid**	**Vapor**
-15	0.16405	180.135	389.63	0.92555	1.7371
50	1.3179	271.62	423.44	1.2375	1.7072

$$COP = \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)}$$



$$x_1 = \frac{s_2 - s_{f1}}{s_{g1} - s_{f1}} = 0.96315 \qquad x_4 = \frac{s_3 - s_{f4}}{s_{g4} - s_{f4}} = 0.3843$$

$$h_1 = h_{f1} + x_1(h_{g1} - h_{f1}) = 381.9101 kJ / kg$$

$$h_4 = h_{f4} + x_4(h_{g4} - h_{f4}) = 260.6439 kJ / kg$$

Putting values in above expression of COP gives

$$COP = 3.96894$$

2. $Q = \frac{100 * 10^6}{3600} = 27{,}777.7778 W \qquad q = h_1 - h_4 = 121.2662 kJ / kg$

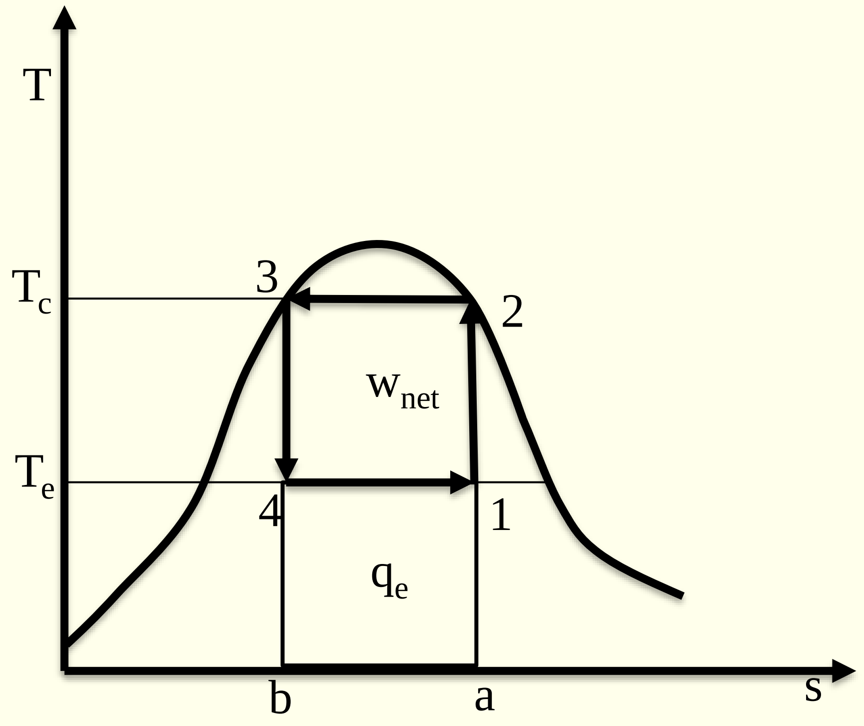
$$\dot{m} = \frac{Q}{q} = 0.22906 g / s$$

3.

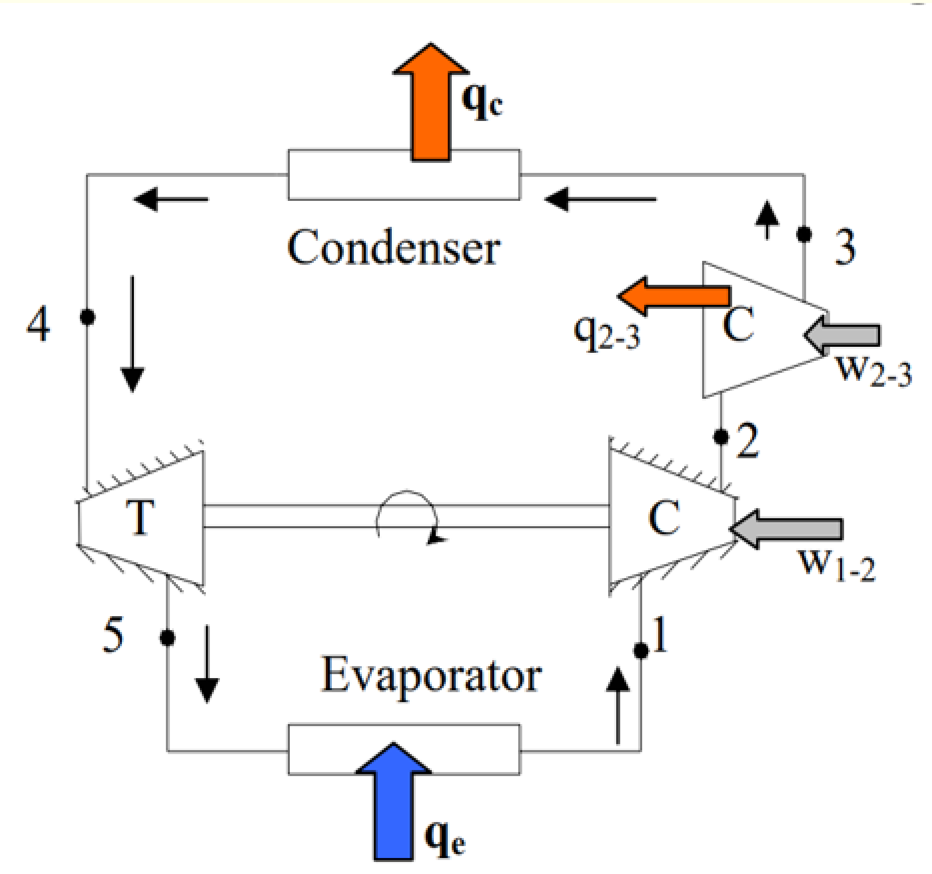
$$W = \frac{Q}{COP} = \frac{3.5167}{3.96894} = 0.88605kW$$

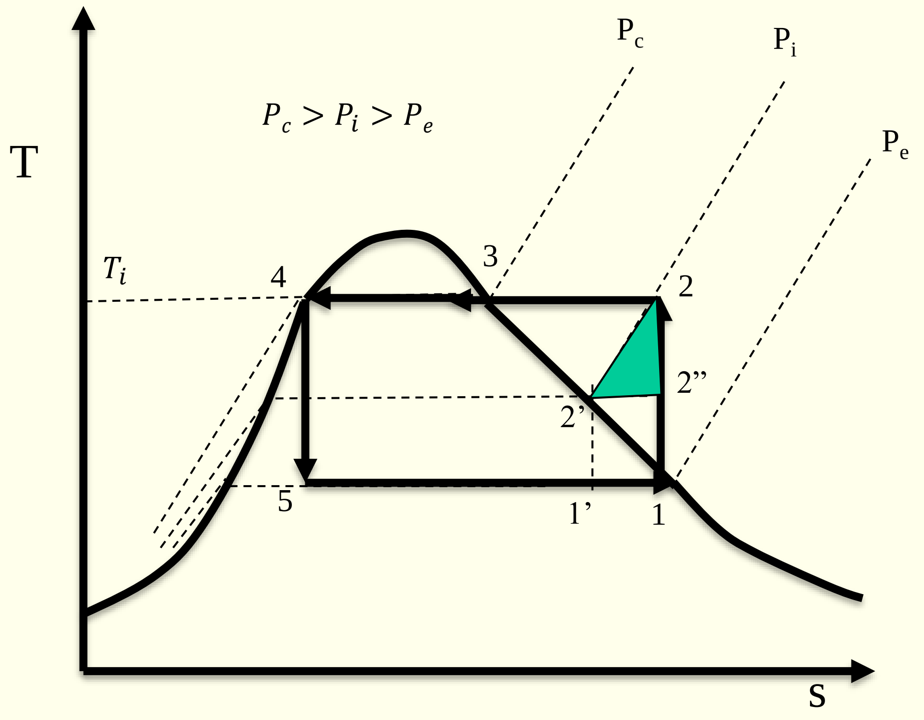
Practical difficulties with Carnot refrigeration system

- Wet compression may damage the compressor
- Extraction of work by expanding saturated liquid in a turbine is not economically justified, for smaller systems.
- Dry compression is possible with two compressors - 1 isentropic & 1 isothermal
- Isothermal compression is difficult to achieve
- Higher cost due to two compressors



Carnot cycle with 2 compressor –Dry compression





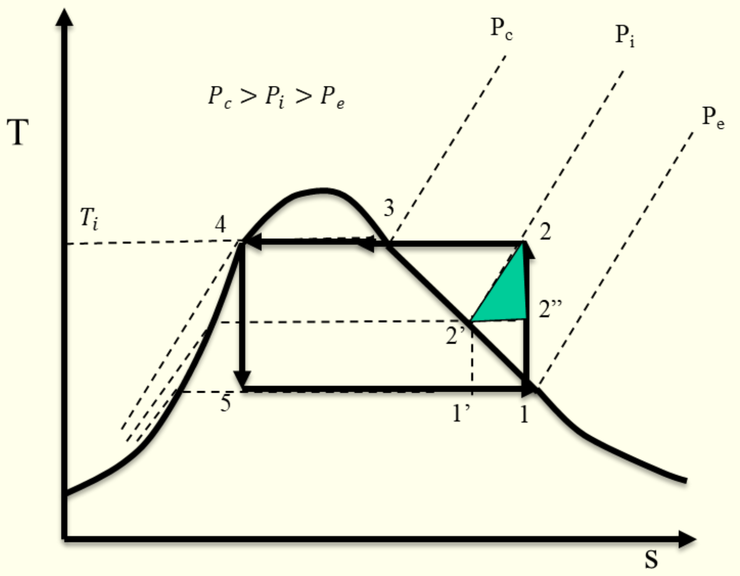
Dry Versus Wet Compression

- It is desirable to have dry compression with vapour initially as dry saturated or preferably superheated for a reciprocating compressor.

- State of vapour at the end of the compression will be at 2 at pressure P_i, saturation pressure of the refrigerant at T_i instead of 2'' which could be in Carnot cycle.

- Results in discharge temperature T_2 higher than $T_{2'}$ – Refrigerant leaves the compressor in superheated condition.

- Increase in work due to dry compression, area $2 - 2' - 2''$ – Superheat horn



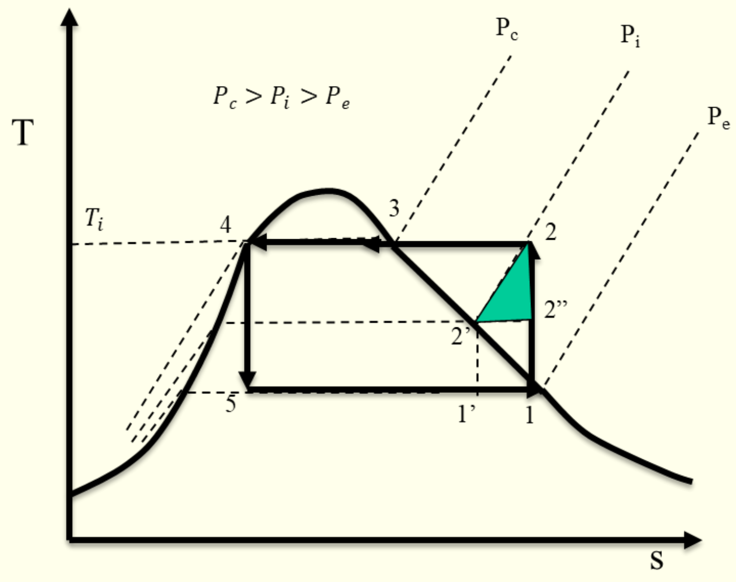
Dry Versus Wet Compression

- Wet compression : compression of wet refrigerant vapour at $1'$ to dry-saturated vapour at $2'$.

- Reciprocating compressor – wet compression is not possible :

a) liquid refrigerant may be trapped in the head of the cylinder and may damage the compressor valves and the cylinder itself.

b) liquid-refrigerant droplets may wash away the lubricating oil from the walls of the compressor cylinder, thus increasing wear.



Dry Versus Wet Compression

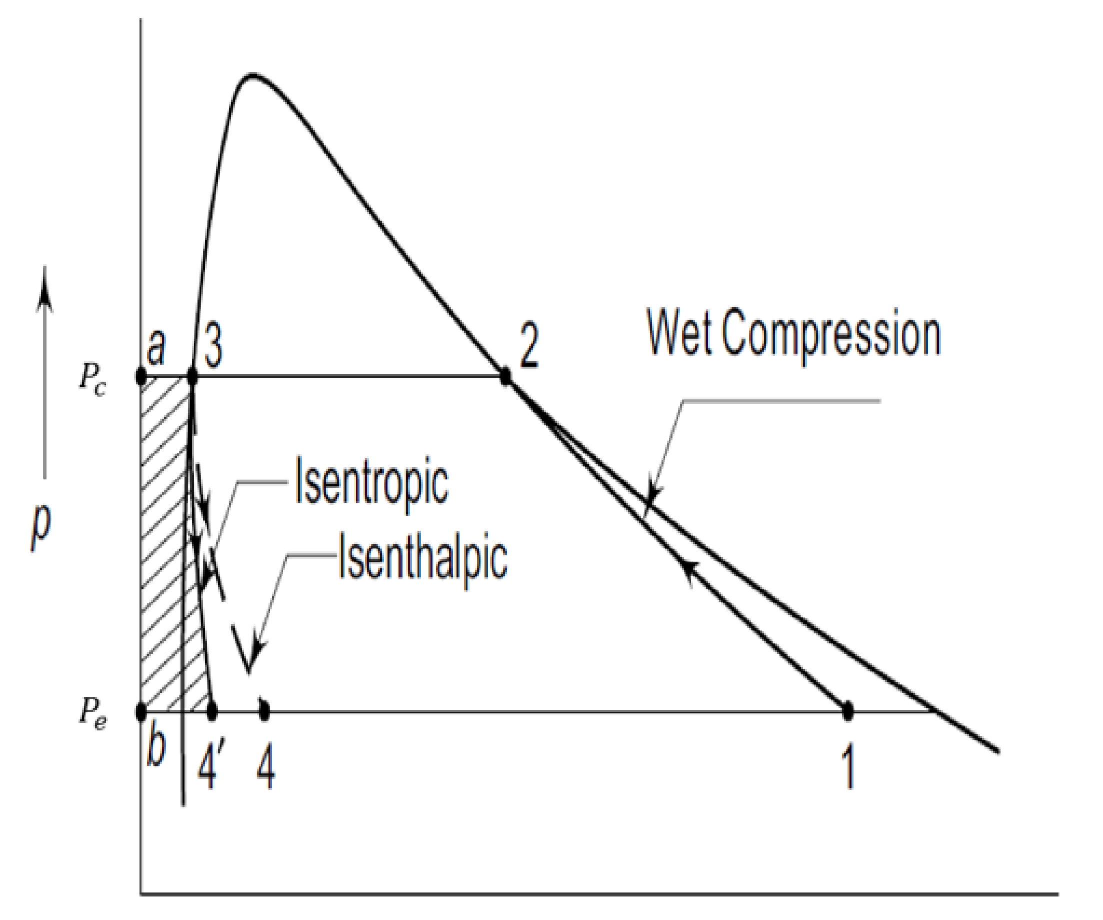
- Sometimes wet compression is desirable and also practicable with centrifugal or screw compressors with no valves.

- Improvement with wet compression is always desirable.

- Power consumption per ton refrigeration with wet compression could be 10-20 % less as compared to dry compression.

Throttling Versus Isentropic Expansion

- Refrigerators require very less net power input.
- The work recovered during isentropic expansion (3-a-b-4') is much smaller as compared to compression work (1-2-a-b).
- The thermodynamic and friction losses of an expander, if employed, may even exceed the gain in work.
- There are practical difficulties in expanding a liquid of a highly wet vapour in an expander

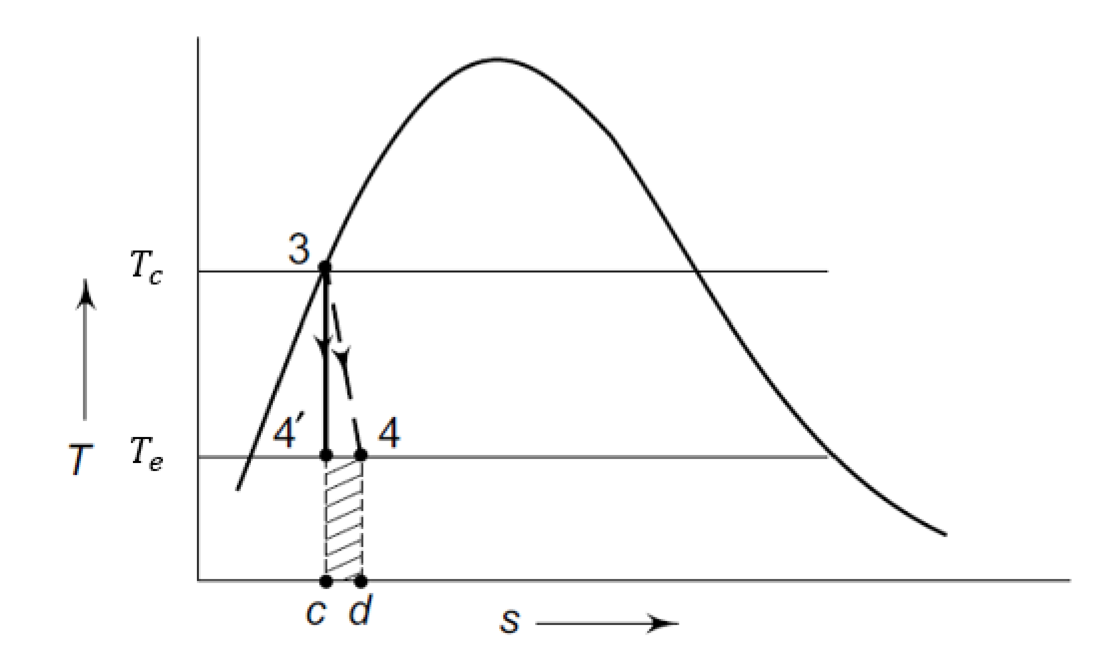


Throttling Versus Isentropic Expansion

- The isentropic expansion process of the Carnot cycle may be replaced by a simple throttling process.

- The process is an irreversible one and is accompanied by increase of entropy

- This results in a loss of work represented by area 3-a-b-4 on the *p-v* diagram and a decrease in the refrigerating effect represented by area 4'-c-d-4 on the *T-s* Diagram – both are same



Vapour Compression Refrigeration Systems(VCRS)

- This system is a modification over Carnot system:
- Isothermal heat rejection process is replaced by isobaric heat rejection
- Isentropic expansion of liquid is replaced by isenthalpic throttling
- This cycle is known as Evans-Perkins cycle or reverse Rankine cycle

VCR Cycle

- Exit conditions of evaporator and condenser are saturated
- The cycle consists of one low-side pressure and one high-side pressure
- Compression is isentropic
- Expansion is isenthalpic

Comparison between Carnot and VCRS

- For the same heat source and sink temperatures, compared to Carnot cycle:

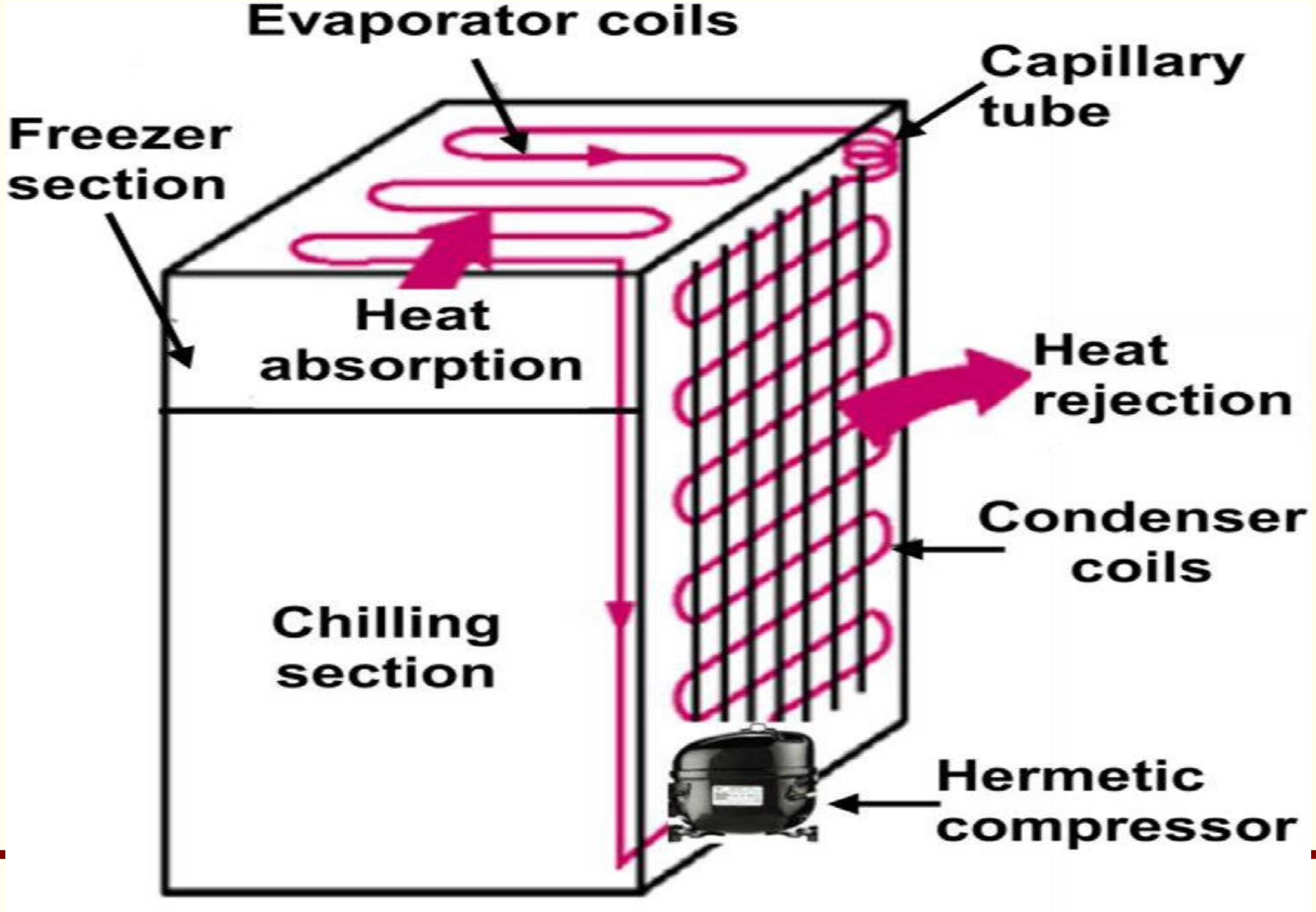
a) Refrigeration effect of VCR cycle decreases

b) Heat rejection increases

c) Compressor work input increases

d) $COP_{VCR} < COP_{carnot}$

Vapor Compression Refrigeration



Vapour Compression Refrigeration (VCR)

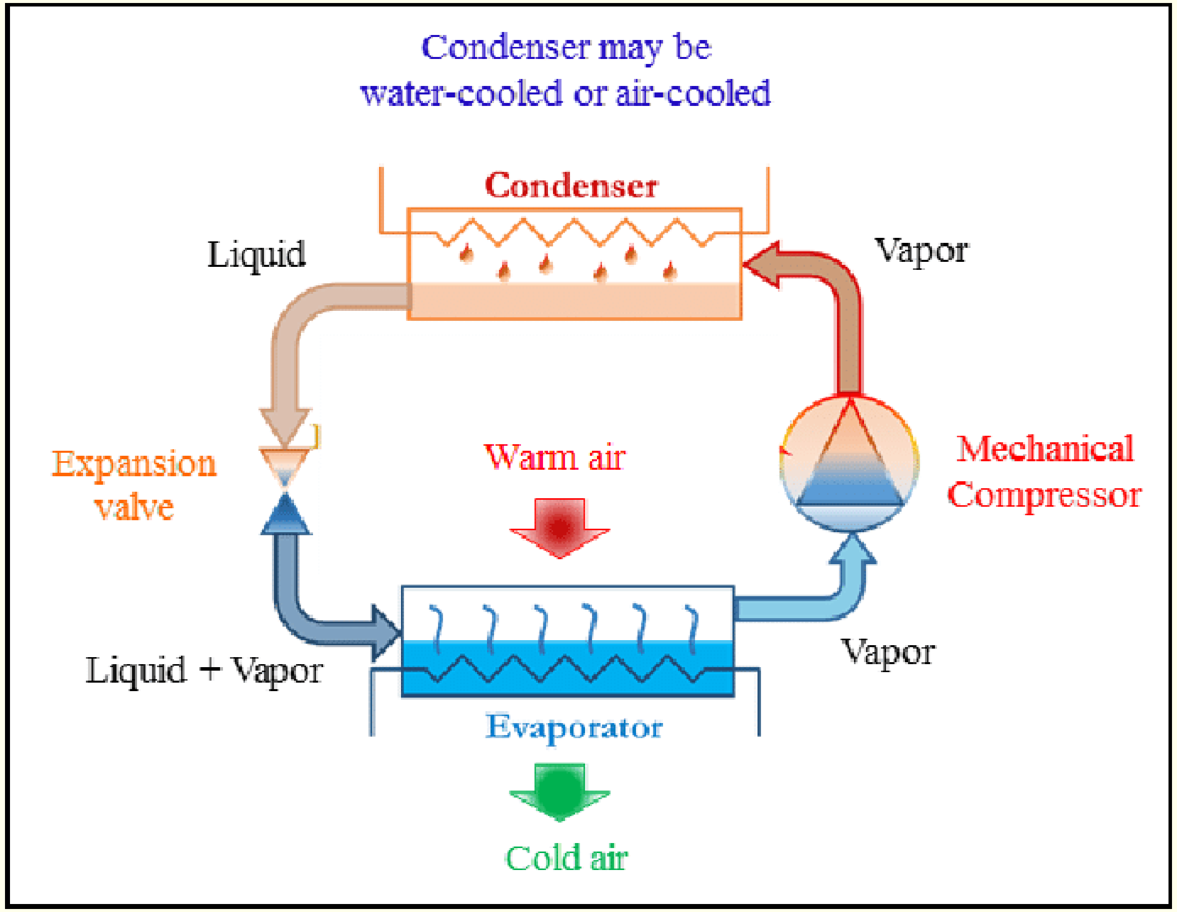
- In a vapour compression refrigeration system:

a) Refrigeration is obtained as the refrigerant evaporates at low temperatures

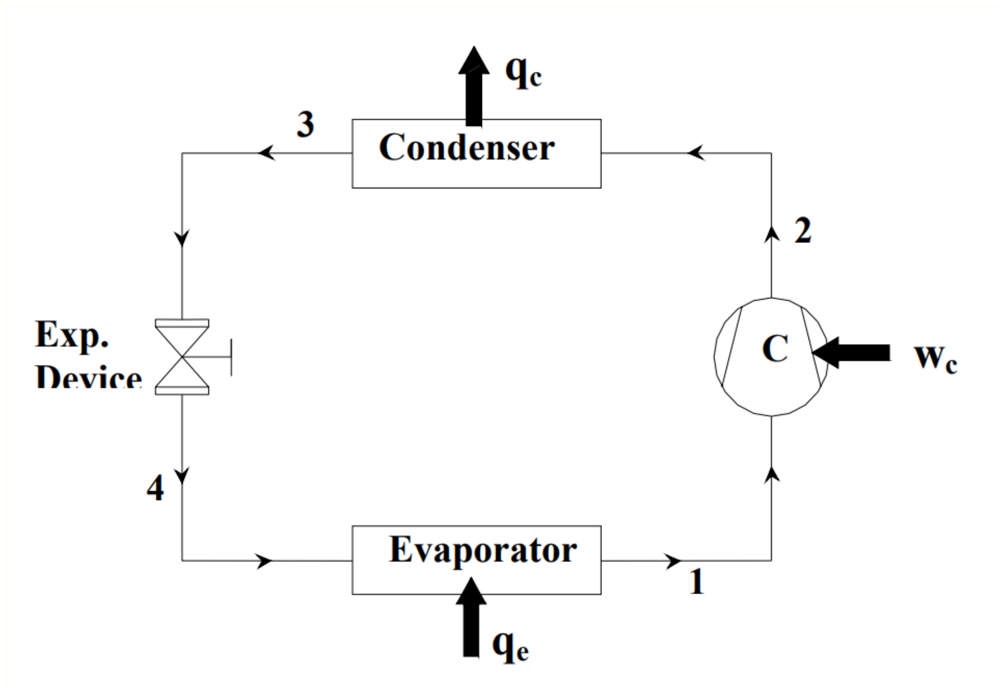
b) The system input is in the form of mechanical energy required to run the compressor

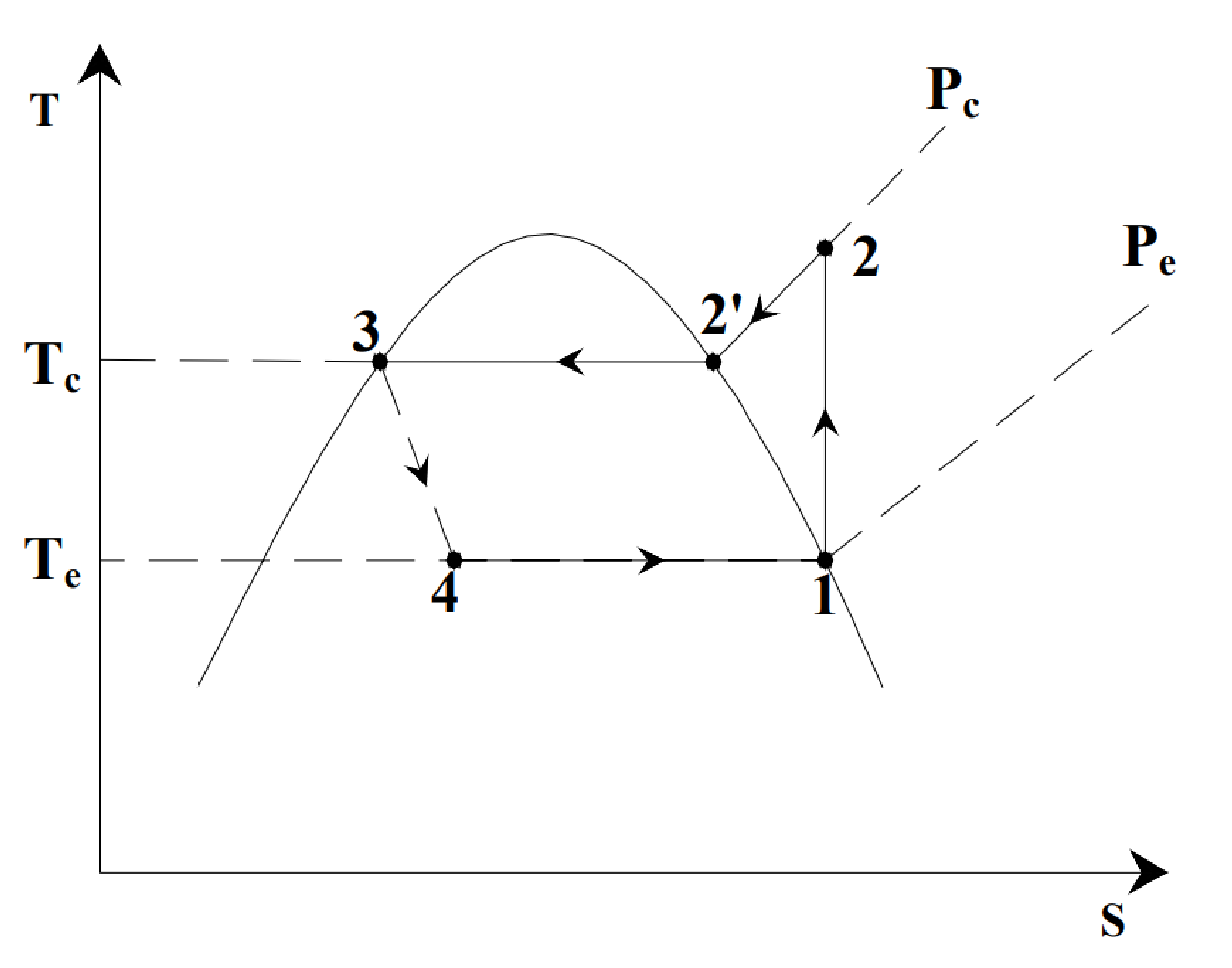
c) Suits almost all applications : with refrigeration capacities ranging from few watts to few megawatts

Vapour Compression system



Standard VCRS cycle





Analysis of VCRS cycle

- Assumptions:

a) Steady flow process through each component

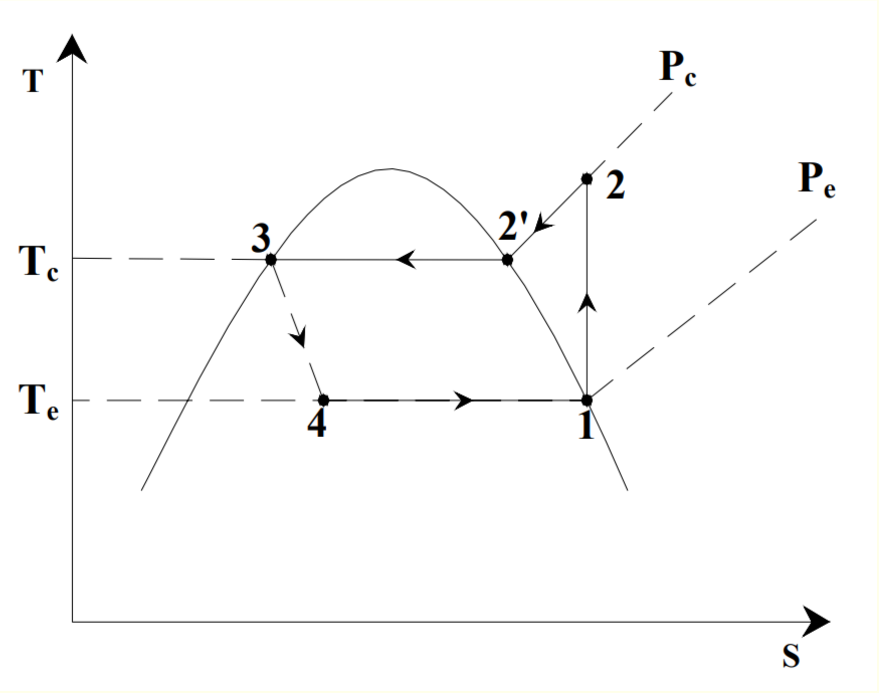
a) Negligible kinetic and potential energy changes across each component

b) No heat transfer or pressure drops in connecting pipe lines

c) Application of steady flow energy equation to each component

Analysis of VCRS cycle - Evaporator

- Evaporator pressure $P_e = P_{sat}(T_e)$
- Refrigeration capacity : $Q_e = m_r(h_1 - h_4)$
- $(h_1 - h_4)$ is specific refrigeration effect in kJ/kg
- Power input to compressor
- $W_c = m_r(h_2 - h_1)$



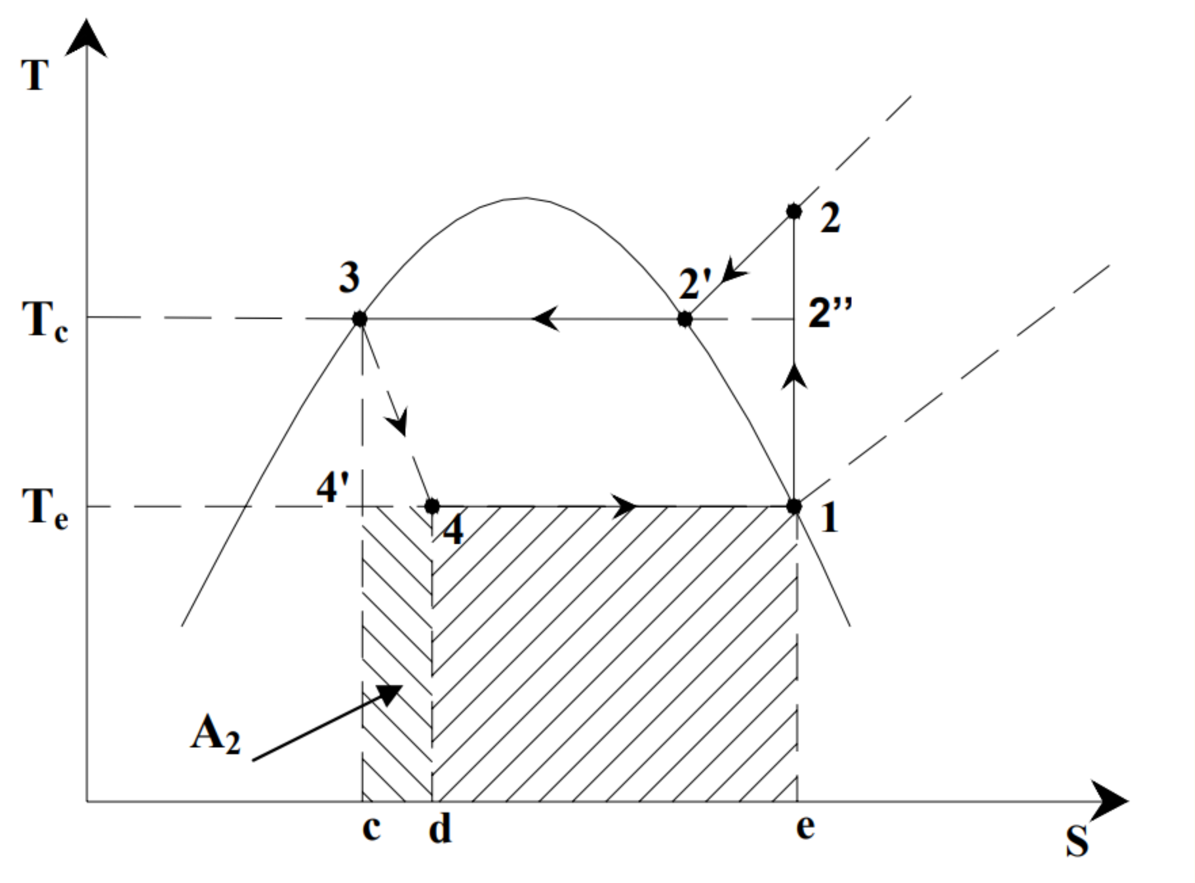
Analysis of VCRS cycle - Compressor

- Condenser pressure, $P_c = P_{sat}(T_c)$
- Heat rejection rate at condenser
- $Q_c = m_r(h_2 - h_3)$
- Expansion Device
- $h_4 = (1 - x_4)h_{fe} + x_4 h_{ge} = h_{4fe} + x_4 h_{fg}$

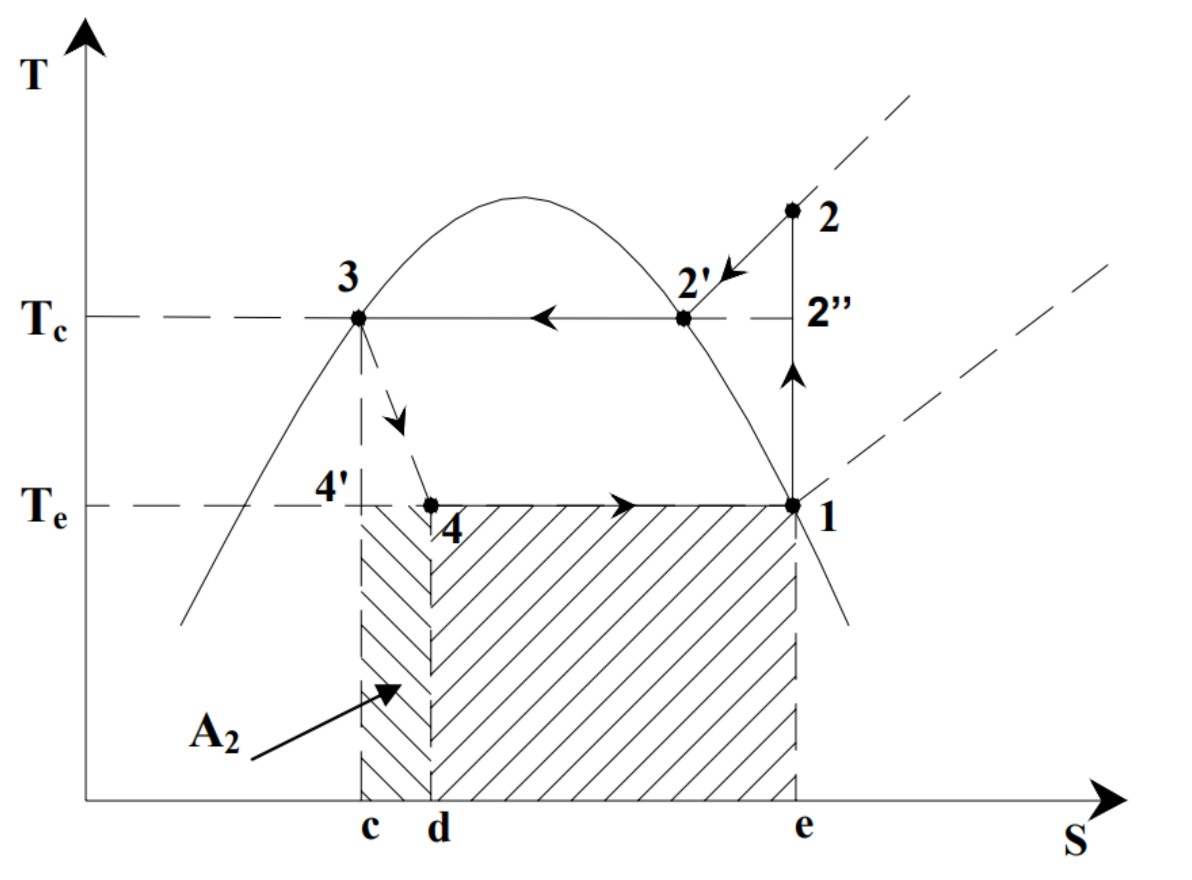
Analysis of VCRS cycle - COP

- COP of SSS cycle $COP = \frac{Q_e}{W_c} = \frac{\dot{m}_r(h_1 - h_4)}{\dot{m}_r(h_2 - h_1)} = \frac{(h_1 - h_4)}{(h_2 - h_1)}$
- At any point in cycle $\dot{m}_r = \frac{\dot{V}}{v}$, at compressor inlet $\dot{m}_r = \frac{\dot{V}_1}{v_1}$
- $\dot{Q}_e = \dot{m}_r(h_1 - h_4) = \dot{V}_1 \left(\frac{h_1 - h_4}{v_1}\right)$,
- $\left(\frac{h_1 - h_4}{v_1}\right)$ = volumetric refrigeration effect kJ/m^3

Comparison between Carnot and VCRS cycle



Comparison between Carnot and VCRS cycle



- $q_{e,Carnot} = q_{4'-1} = \int_{4'}^{1} Tds$
  $= T_e(s_1 - s_{4'}) = area\ e - 1 - 4' - c - e$

- $q_{e,VCRS} = q_{4-1} = \int_{4}^{1} Tds$
  $= T_e(s_1 - s_4) = area\ \ e - 1 - 4 - d - e$

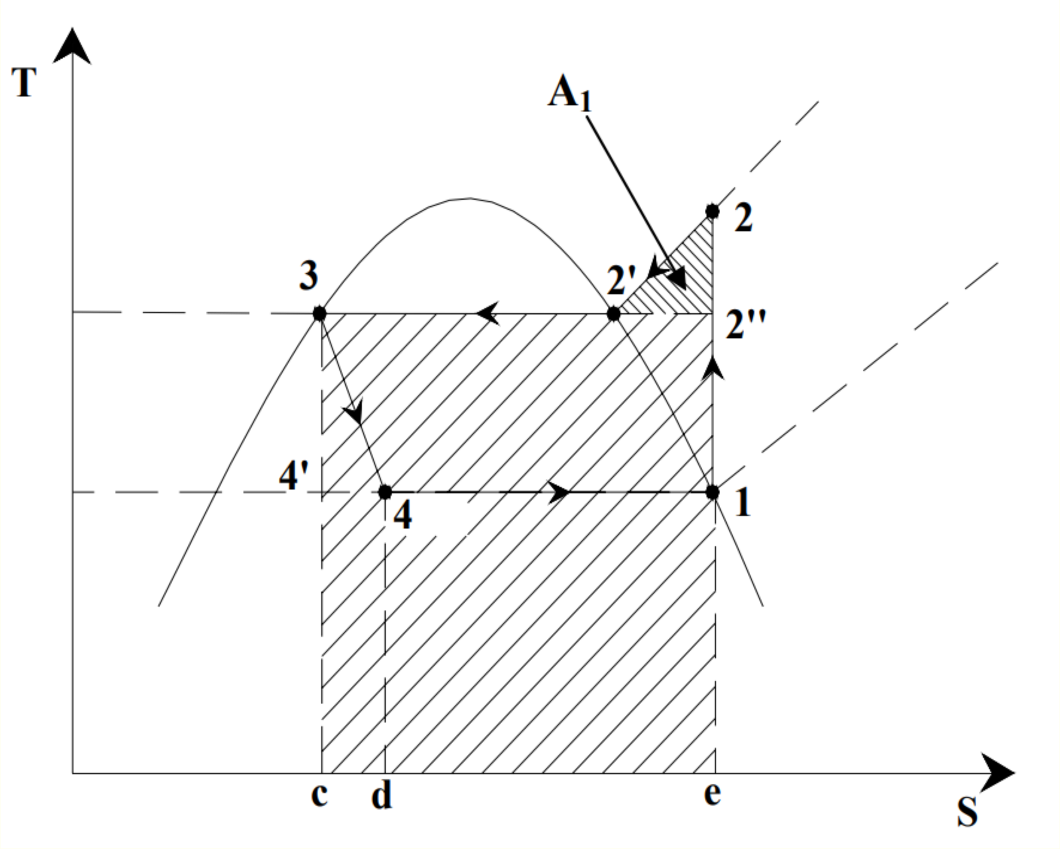
- $q_{e,Carnot} - q_{e,VCRS}$
  $= area\ d - 4 - 4' - c - d = (h_3 - h_{4'})$
  $= (h_4 - h_{4'}) = area\ A_2$

Comparison between Carnot and VCRS cycle

- $q_{c,Carnot} = -q_{2''-3} = -\int_{2''}^{3} Tds = T_c(s_{2''} - s_3)$
  $= area\ e - 2'' - 3 - c - e$

- $q_{c,VCRS} = -q_{2-3} = -\int_{2}^{3} Tds = T_c(s_2 - s_3)$
  $= area\ e - 2 - 3 - c - e$

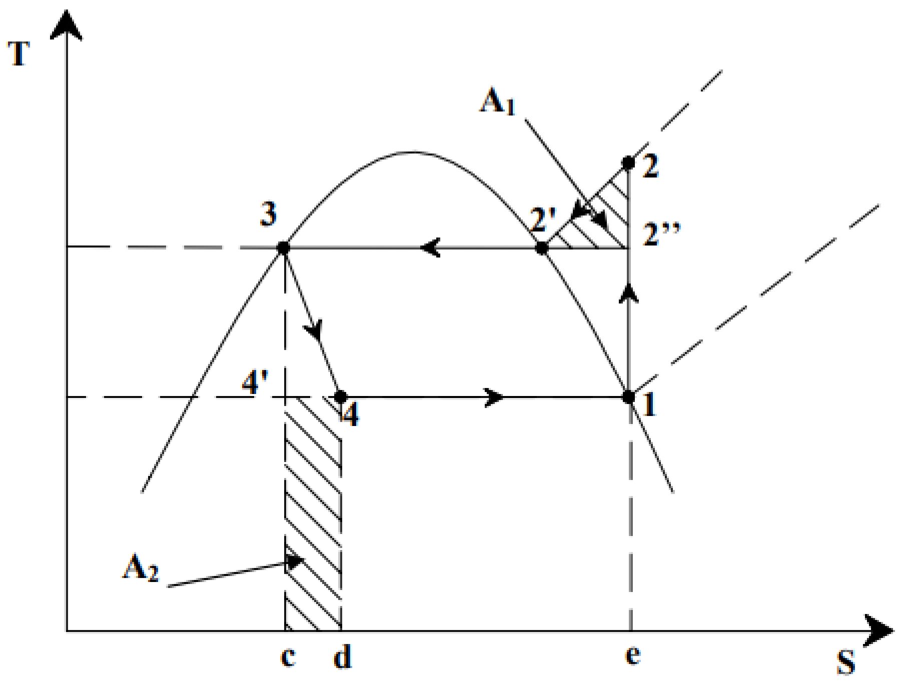
- $w_{net,Carnot} = (q_c - q_e)_{Carnot}$
  $= area\ \ 1 - 2'' - 3 - 4' - 1$



Comparison between Carnot and VCRS cycle

- $w_{net,VCRS} = (q_c - q_e)_{VCRS} = area\ \ 1 - 2 - 3 - 4' - c - d - 4 - 1$

- $w_{net,VCRS} - w_{net,Carnot} = area\ \ 2'' - 2 - 2' + area\ c - 4' - 4 - d - c = area\ \ A_1 + area\ A_2$



Comparison between Carnot and VCRS cycle

- COP of VCRS

- $C\frac{q_{e,VCRS}}{w_{net,VCRS}} = \frac{q_{e,Carnot} - area\ A_2}{w_{net,Carnot} + area\ A_1 + area\ A_2}$

- COP vcr < COPcarnot

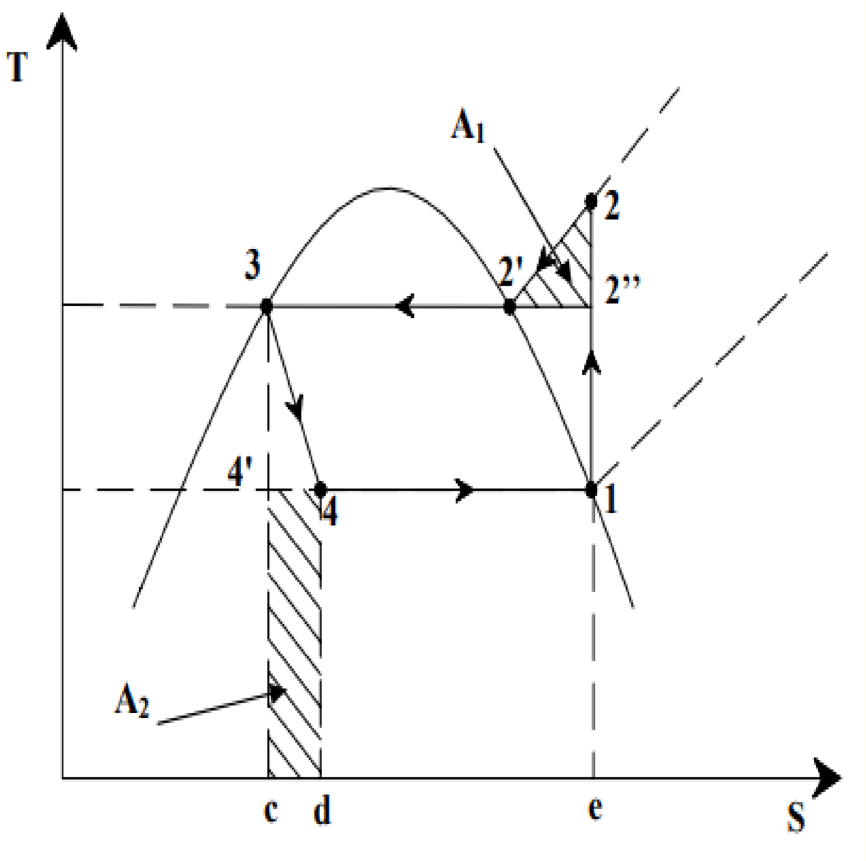
- Cycle efficiency

- $\eta_R = \frac{COP_{VCRS}}{COP_{Carnot}} = \left[\frac{1 - \left(\frac{area\ A_2}{q_{e,Carnot}}\right)}{1 + \left(\frac{area\ A_1 + area\ A_2}{w_{net,Carnot}}\right)}\right]$

Superheat vs throttling losses

- The superheat loss:
  a) Increases only the work input
  b) Does not affect refrigeration effect
  c) In heat pumps, superheat is a part of the useful heating effect

- The throttling loss (irreversible):
  a) Increases the work input, and also
  b) Reduces the refrigeration effect



Unit of Refrigeration capacity (TR)

- The rate of heat transfer that required to make 1 short ton (907 kg) of pure ice per day from water at 0 °C.

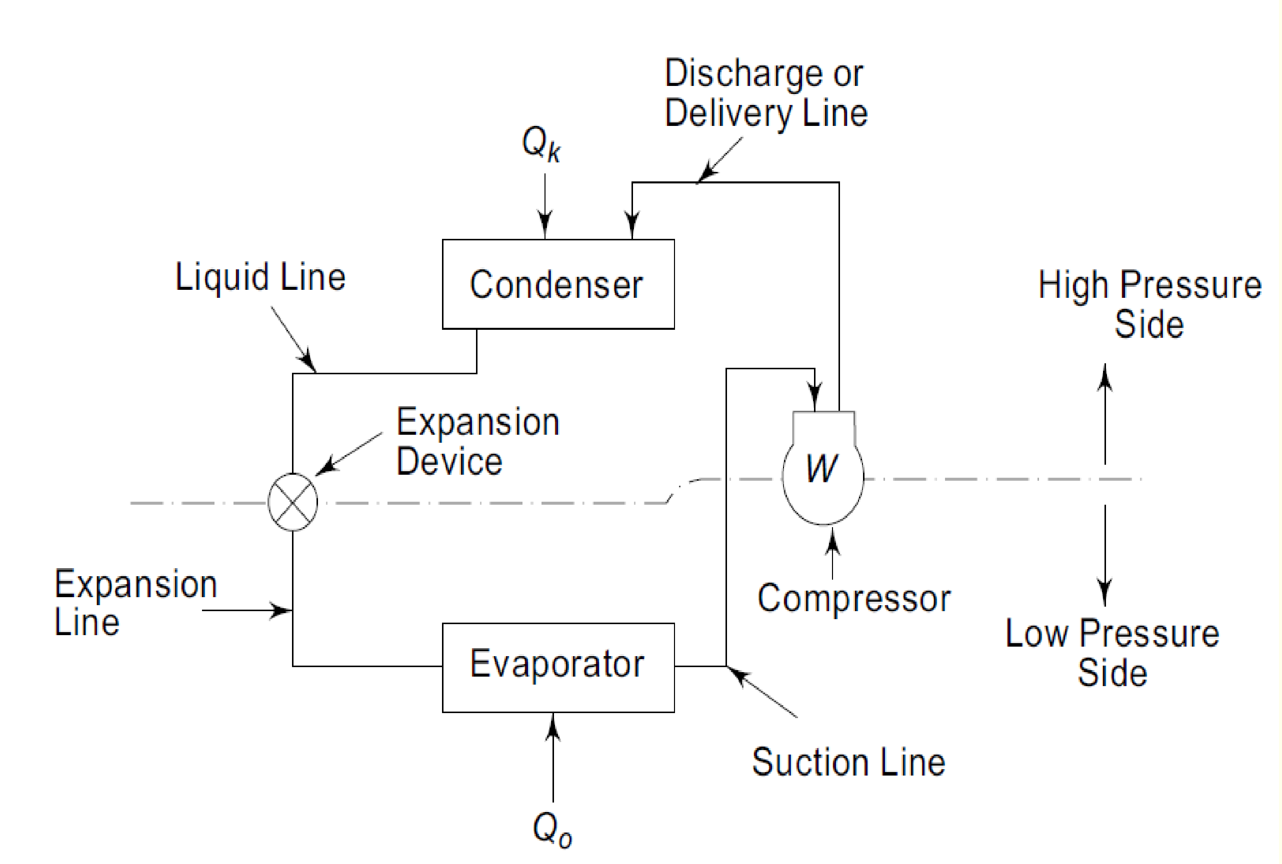
- Latent heat of fusion of ice = 334 kJ/kg

- Heat extracted per 24 hour $= 907 \text{ kg} * 334 \frac{kJ}{kg} = \frac{3{,}02{,}938}{24\ hr}\ kJ = \frac{12{,}622.416}{3600} \frac{kJ}{sec} = 3.506 \frac{kJ}{s} = 3.506\ kW$

- 1 ton of Refrigeration = 3.506 kJ/s =3.506 kW =4.701 HP

Vapour compression system

- VCR consists of compressor, a condenser, an expansion device and an evaporator
- High pressure side : compressor-delivery head, discharge line, condenser and liquid line.
- Low pressure side : expansion line, evaporator, suction line and compressor-suction head.
- Receiver and Drier in liq line.



Vapour compression system

- 1-2 : isentropic compression, Q=0

$$w = -\int vdp = -\int dh = -(h_2 - h_1)$$

- 2-3 : De-superheating and condensation,

$p_c = const$, heat rejected, $q_c = h_2 - h_3$

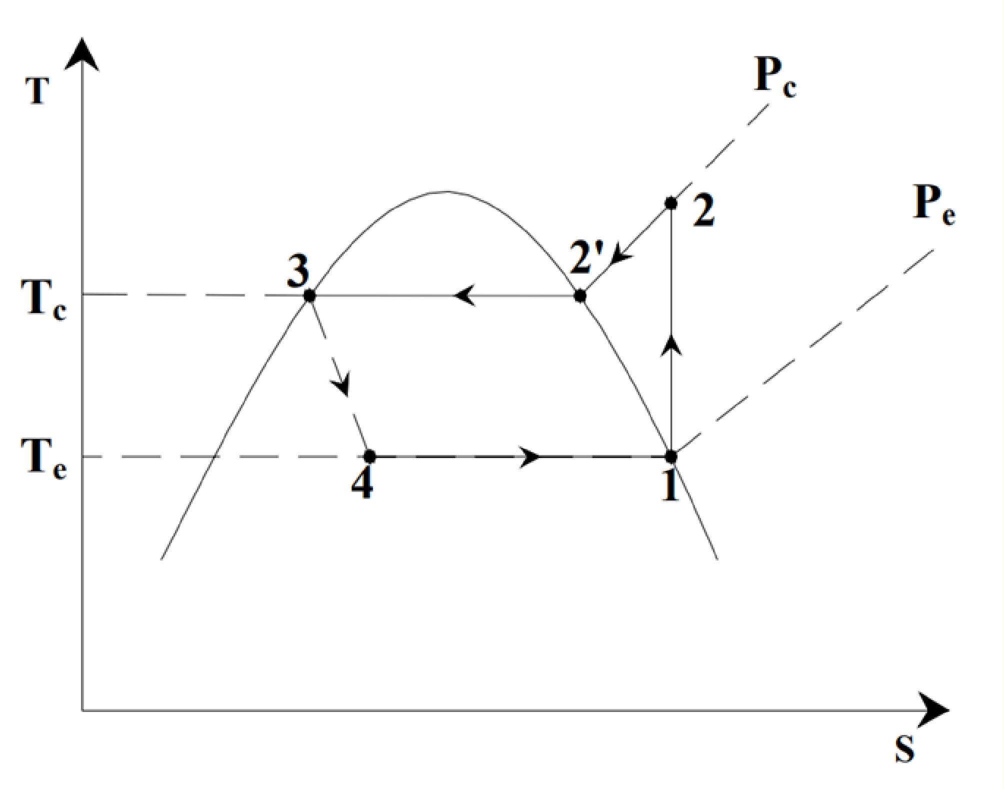
- 3-4 : Isenthalpic expansion,

$$h_3 = h_4 = h_{f_4} + x_4\left(h_1 - h_{f_4}\right)$$

$$x = \frac{h_3 - h_{f_4}}{h_1 - h_{f_4}}$$

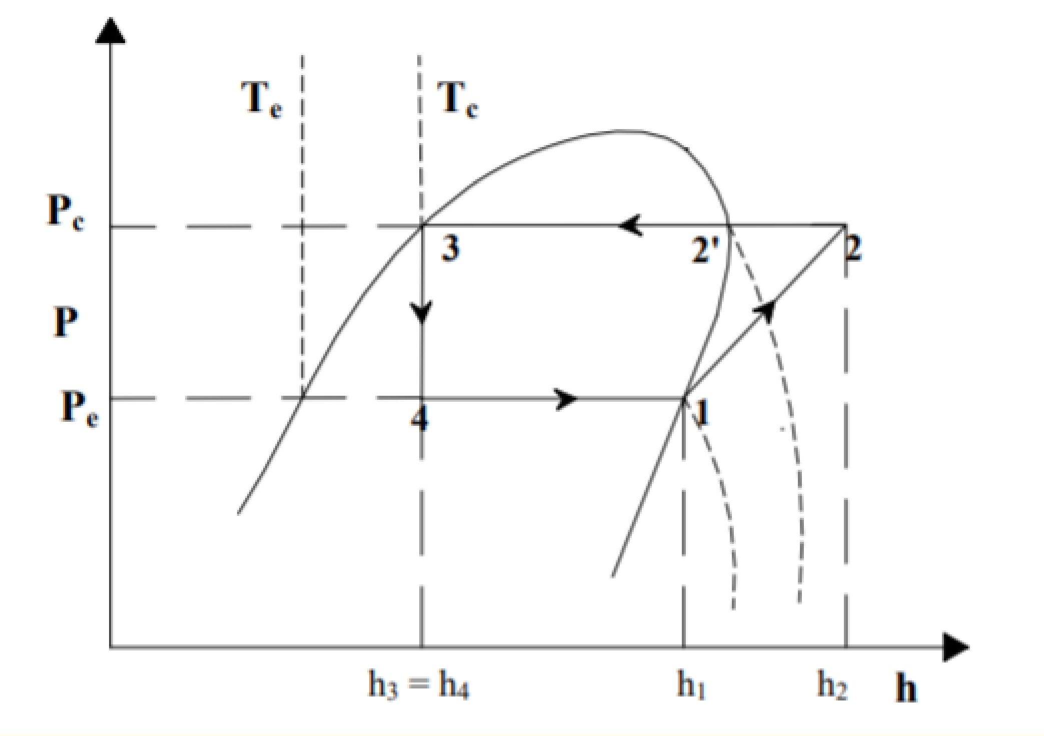
- 4-1 : Evaporation, $p_e = const$.

Refrigeration effect, $q_e = h_1 - h_4$



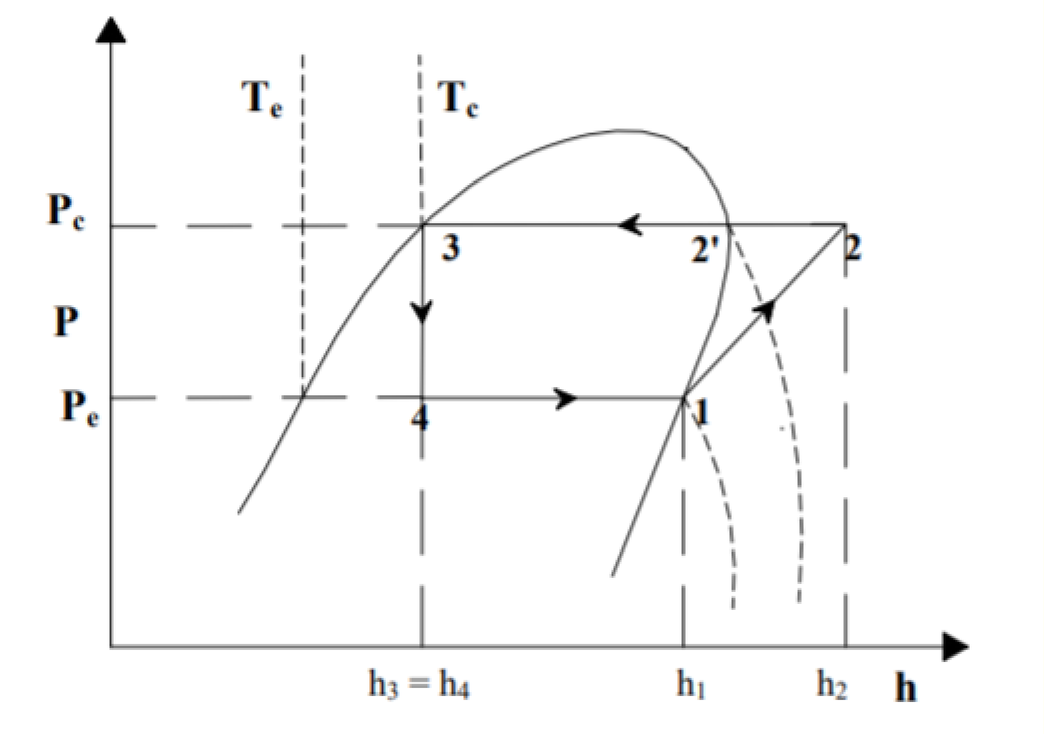
Vapour Compression Cycle on P-h Diagram

- Simple saturation cycle
- $q_c = q_e + w = h_2 - h_3$
- COP for cooling, $E_c = \frac{h_1 - h_4}{h_2 - h_1}$
- COP for heating, $E_h = \frac{h_2 - h_3}{h_2 - h_1}$



Vapour Compression Cycle on P-h Diagram

- Refrigeration circulation rate $\dot{m} = \frac{\dot{Q}_e}{q_e}$
- Theoretical piston displacement $\dot{V} = \dot{m}v_1$
- Actual piston displacement $\dot{V_p} = \frac{\dot{m}v_1}{\eta_v}$
- $\eta_v = 1 + \varepsilon - \varepsilon \left(\frac{P_c}{P_e}\right)^{\frac{1}{\gamma}}$
- Power consumption $\dot{W} = \dot{m}w = \dot{m}(h_2 - h_1)$
- Heat rejected $\dot{Q}_c = \dot{m}q_c = m(h_2 - h_3)$



Effect of Operating Conditions – Evaporator pressure

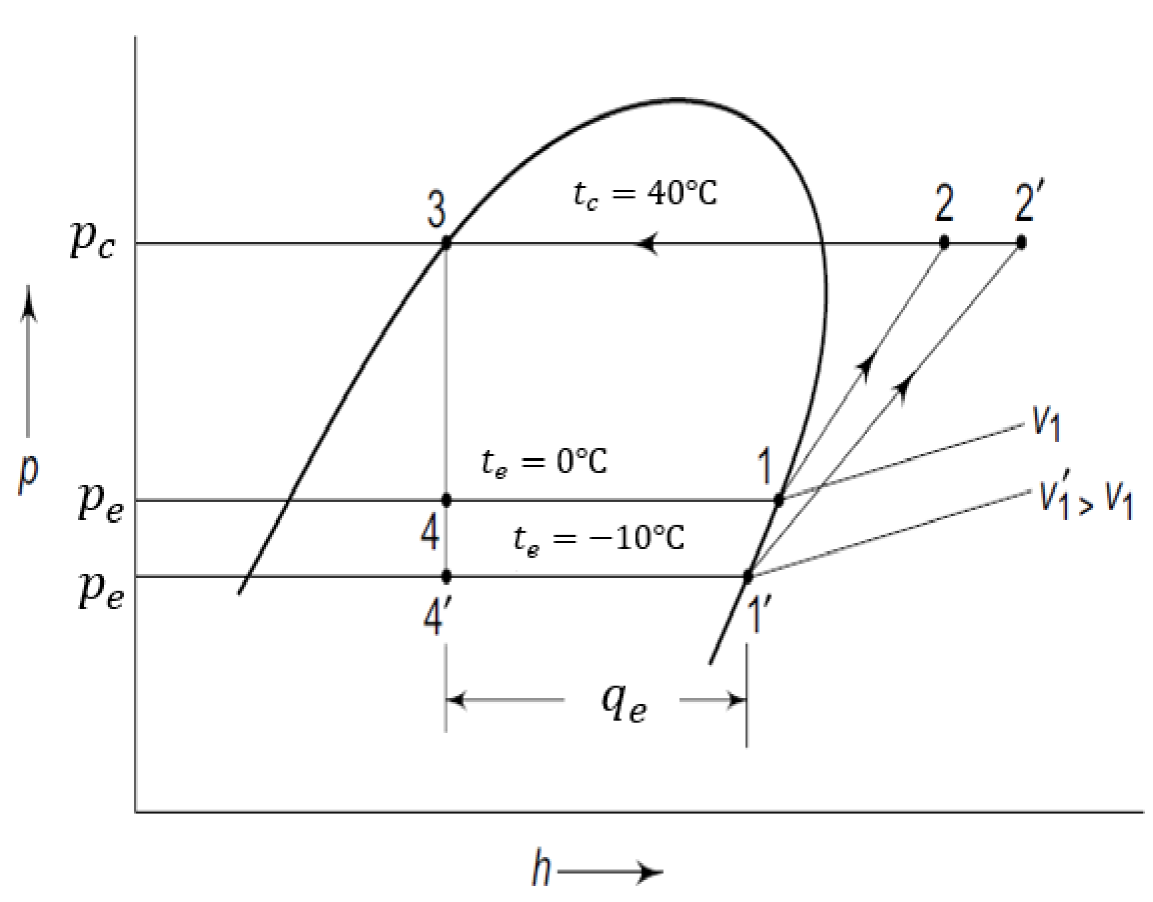
- Decrease in Evaporator temperature results in:

  a) Decrease in refrigerating effect

  b) Increase in the specific volume of suction vapor

  c) Decrease in volumetric efficiency

  d) Increase in compressor work



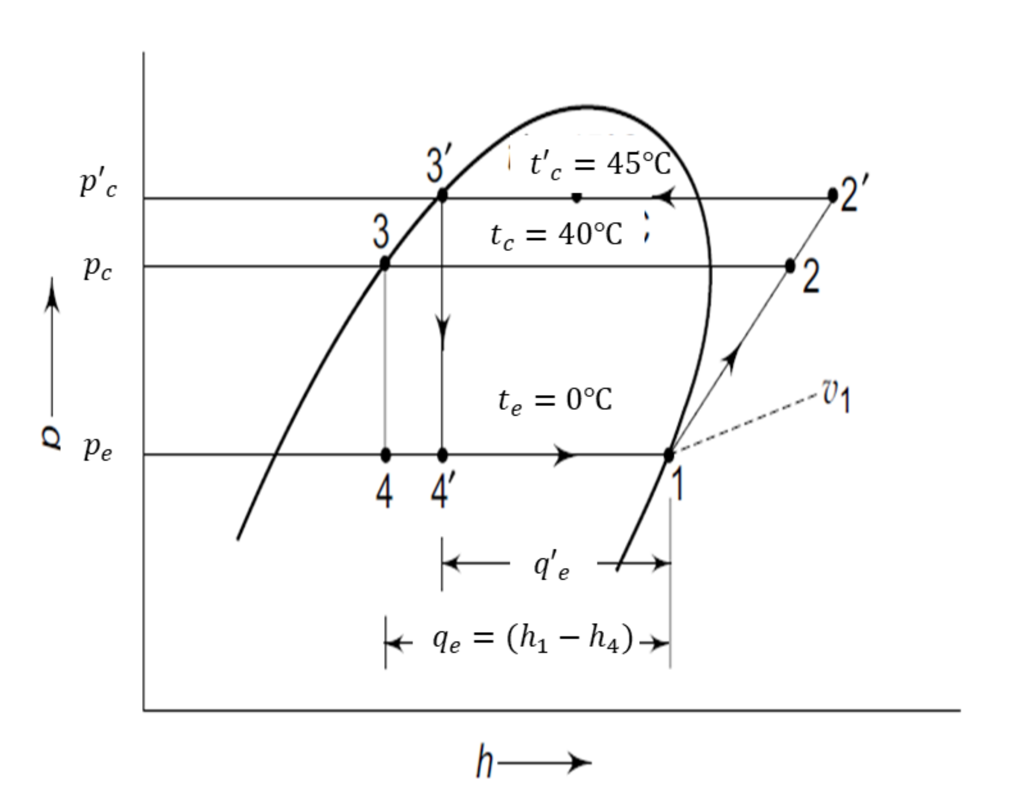
Effect of Condenser Pressure

- Increase in condenser pressure results in :

- Decrease in the refrigerating capacity

- Increase in power consumption.

- Decrease in Volumetric efficiency



2- A VCRS working with Freon 22 as working fluid is specified to give 40 TR capacity for air conditioning under standard operating conditions of 40^{o}C condensing and 5^{o}C evaporating temperature. What would be COP, refrigerating capacity and power consumption if it is operating under 3 different sets of conditions tabulated below:

S.No.	Evaporating temperature(^{o}C)	Condensing temperature(^{o}C)	Compressor suction temperature(^{o}C)
1	0	40	0
	-5	40	-5
	-10	40	-10
2	5	35	5
	5	30	5
	5	25	5
3	5	40	10
	5	40	15
	5	40	20

Comment on the relative effect of each and explain results obtained from plot as well.

$T_o = 5°C \qquad T_k = 40^{\circ} C$

$Q = 40TR = 40 * 3.5167 = 140.67kW$

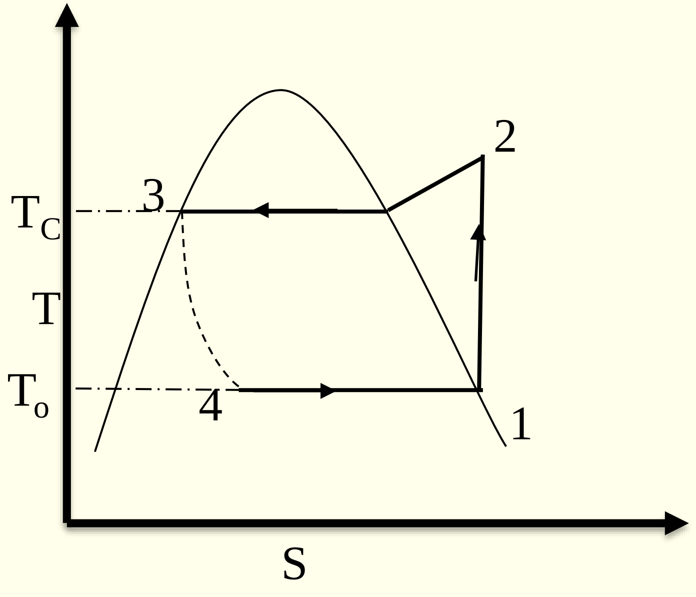
$h_1 = 407.15kJ / kg \qquad s_1 = 1.7446kJ / kgK = s_2$

$h_2 = 429.9417kJ / kg$ (from superheated table)

$h_3 = 249.08kJ / kg = h_4$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = 6.935$$

$$\dot{m} = \frac{Q}{q} = \frac{140.67}{h_1 - h_4} = 0.8899kg / s$$



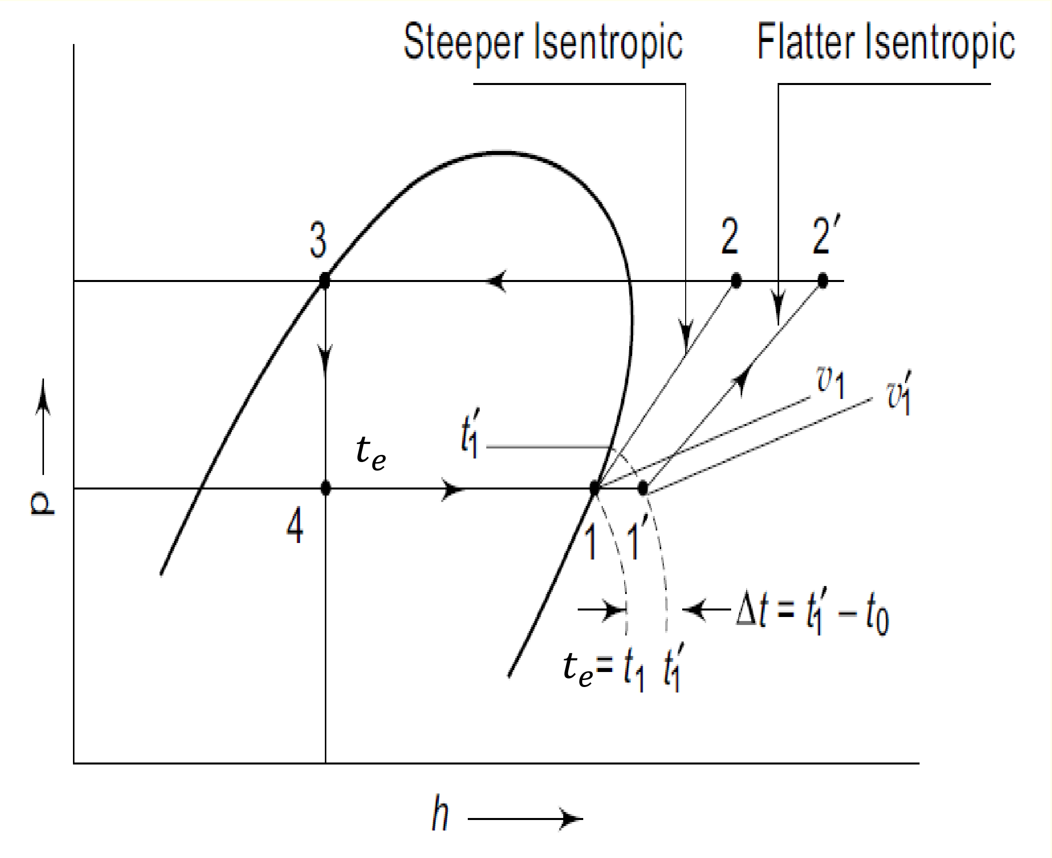
Volume of suction vapor in compressor $V = \dot{m}\upsilon_{g1} = 0.03586m^3 / s$

Power consumption $W = \dot{m}(h_2 - h_1) = 20.282kW$

S.No.	Evaporating temperature(°C)	Condensing temperature(°C)	Compressor suction temperature(°C)	COP	Refrigeration capacity(kW)	Power consumption (kW)
1	5	40	5	6.935	140.67	20.282
	0	40	0	5.811	118.9837	20.474
	-5	40	-5			
	-10	40	-10			
2	5	35	5	8.0695	146.1905	18.116
	5	30	5			
	5	25	5			
3	5	40	10	6.9084	139.7406	20.2274
	5	40	15			
	5	40	20			

Effect of Suction Vapour Superheat

- Superheating of suction vapour ensures complete vaporization of the liquid in the evaporator before it enters compressor.
- It also serves to actuate and modulate the capacity of the expansion valve.
- For R 134a, Isobutane, etc., maximum COP is obtained with superheating of the suction vapour.



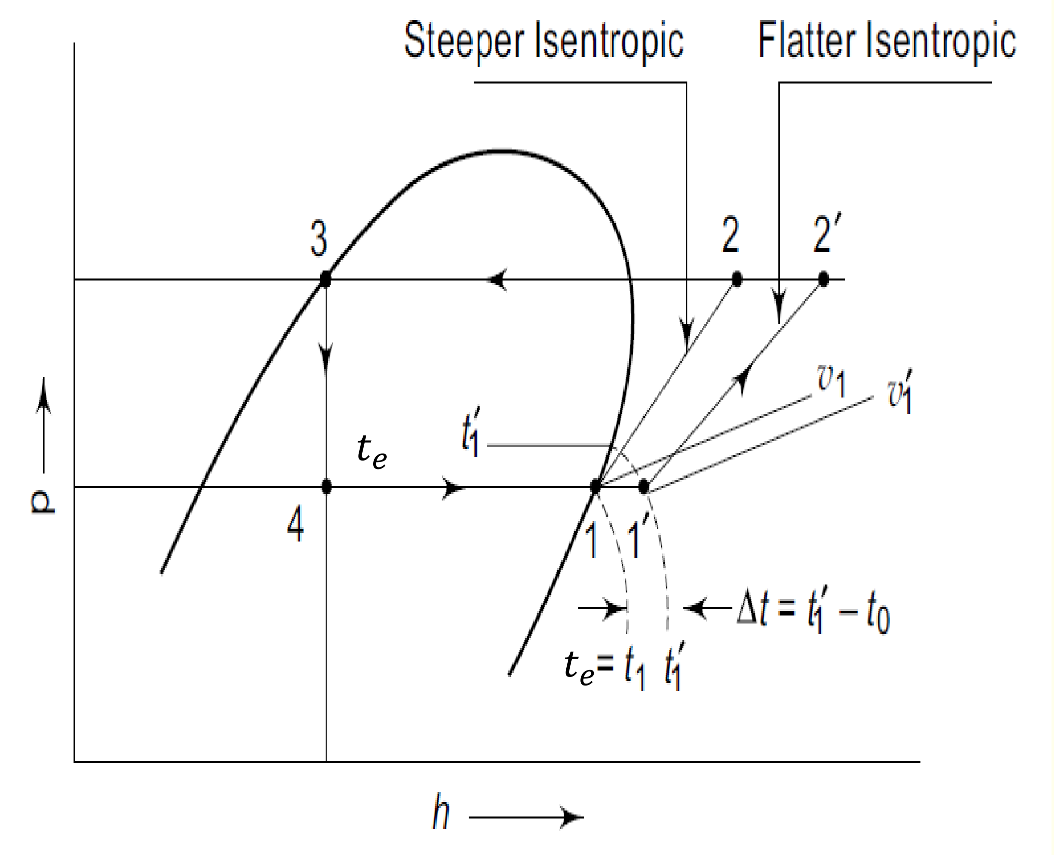
Effect of Suction Vapour Superheat

- Effect of superheating of the vapour from $t_1 = t_e$ to t_1' is as follows:

i. Increase in specific volume of suction vapour from 1 to 1'

ii. Increase in refrigerating effect from $(h_1 - h_4)$ to $(h_1' - h_4)$

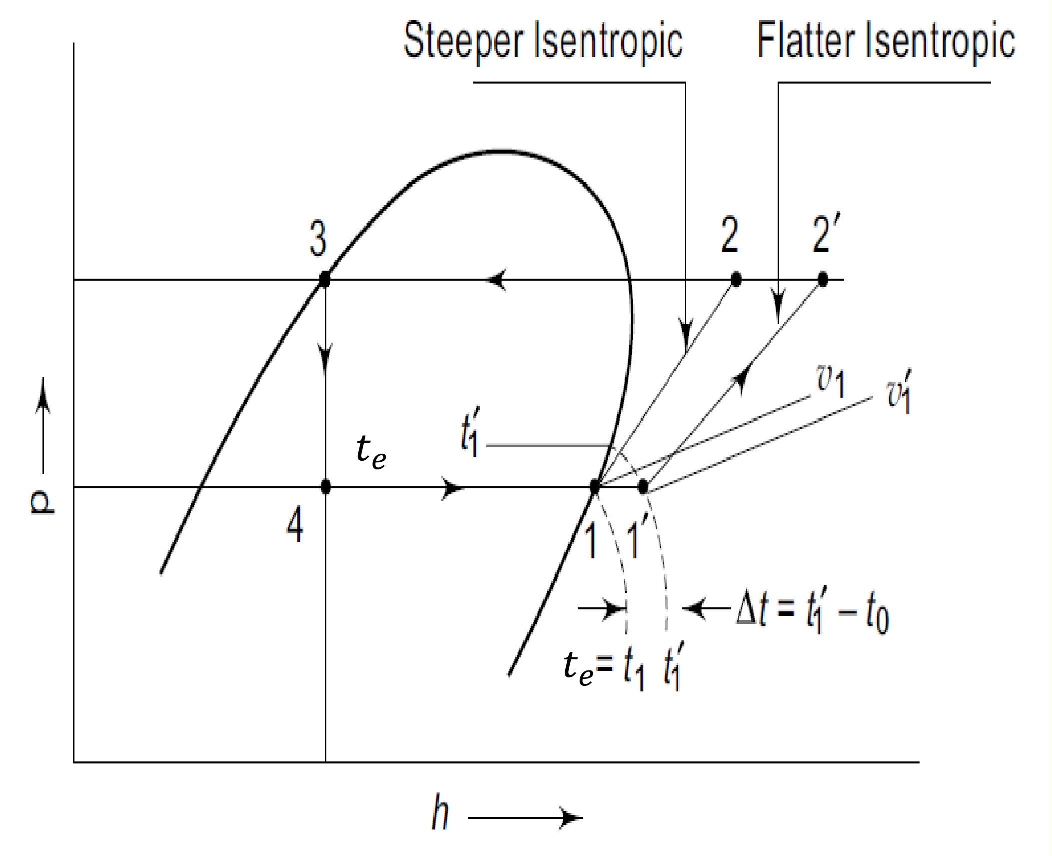
iii. Increase in specific work from $(h_2 - h_1)$ to $(h_2' - h_1')$.



Effect of Suction Vapour Superheat

- $(h_2' - h_1')$ is greater than $(h_2 - h_1)$ as the initial temperature t_1' is greater than t_1

- $w = \frac{\gamma R T_1}{\gamma - 1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = f\left(T_1, \frac{p_2}{p_1}, \gamma\right)$

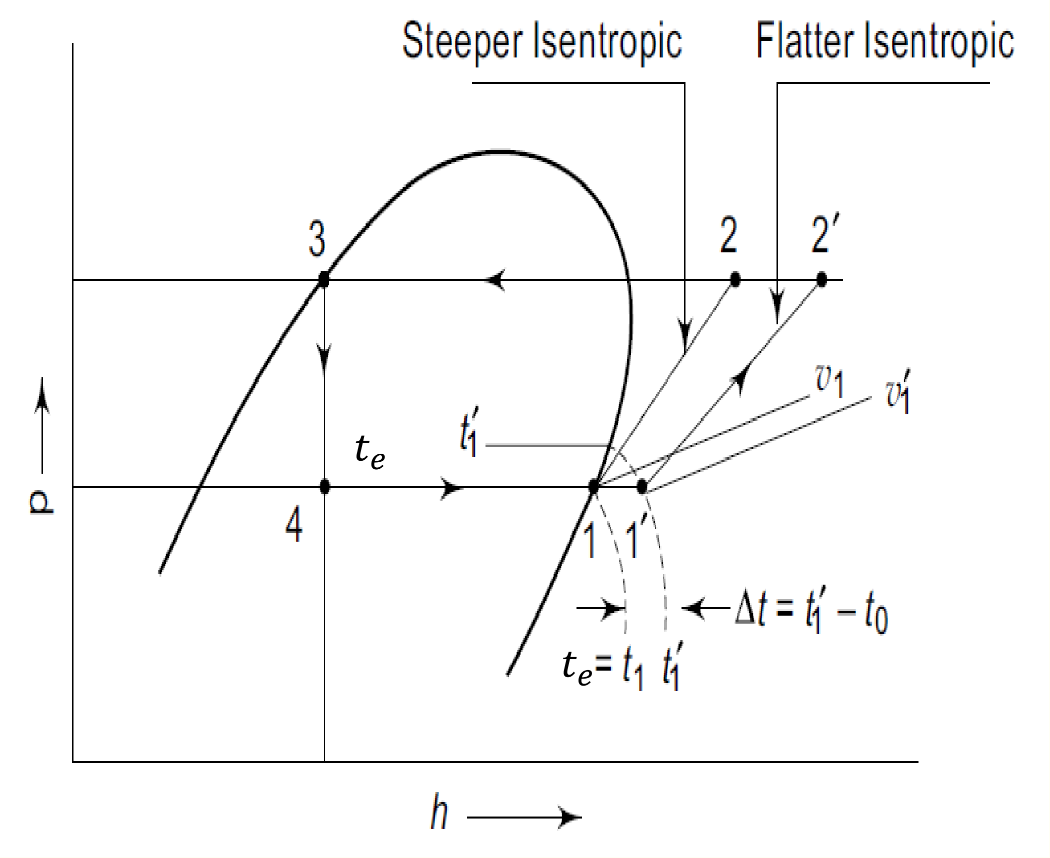


Effect of Suction Vapour Superheat

- Two contradictory requirements for work done per ton of refrigeration :

- Increase in RE-need to decrease mass flow rate requirement and hence work. And increase in sp work due to increase in suction temperature.

- The resulting work per unit refrigeration may, therefore, increase or decrease depending on the refrigerant and operating temperatures.



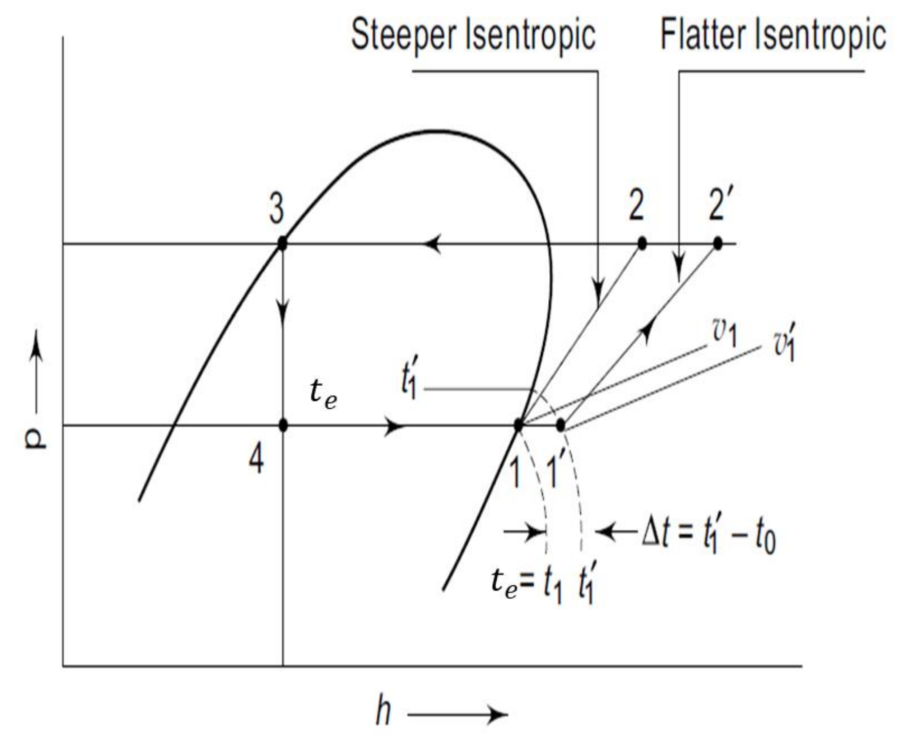
Effect of Suction Vapour Superheat

- Work per unit refrigeration therefore increases or decreases depending on refrigerant and operating temperatures.

$$\frac{Q'_e}{Q_e} = \frac{h'_1 - h_4}{h_1 - h_4}$$

$$E'_c = \frac{h'_1 - h_4}{h'_2 - h'_1}$$

$$= \frac{(h_1 - h_4) + (h'_1 - h_1)}{(h_2 - h_1) + [(h'_2 - h'_1) - (h_2 - h_1)]}$$



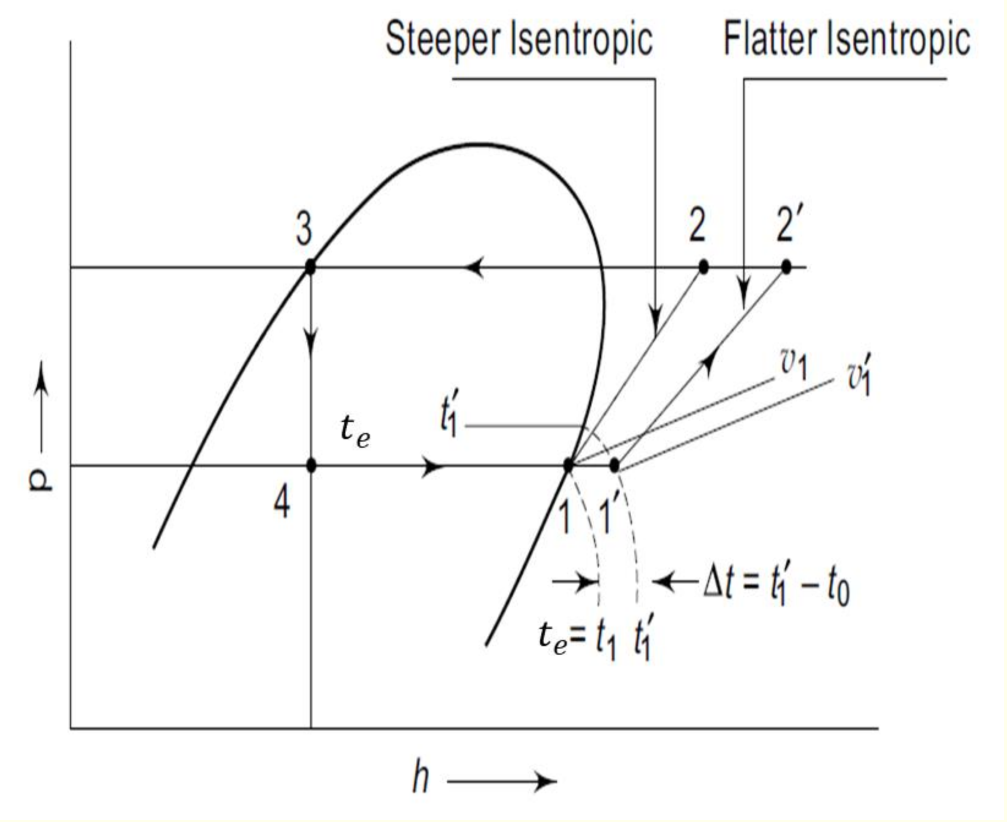
Effect of Suction Vapour Superheat

- $= \dfrac{(h_1-h_4)+(h_1'-h_1)}{(h_2-h_1)+[(h_2'-h_1')-(h_2-h_1)]}$

- Both Nr and Dr increase – COP may increase or decrease or may remain same.

- For R12, superheating increases the COP while for R22 and NH3 it decreases.

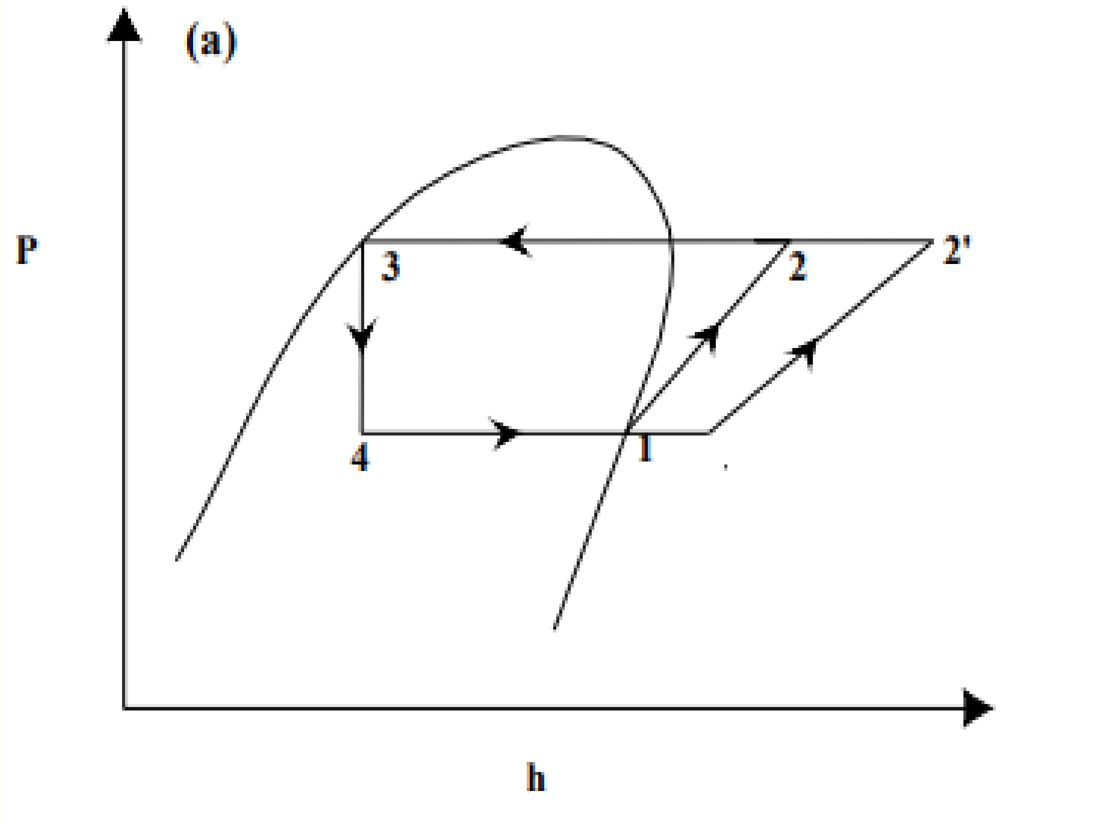
- Slight superheat increases η_v and COP. Superheat outside the evaporator or cold space results in loss.

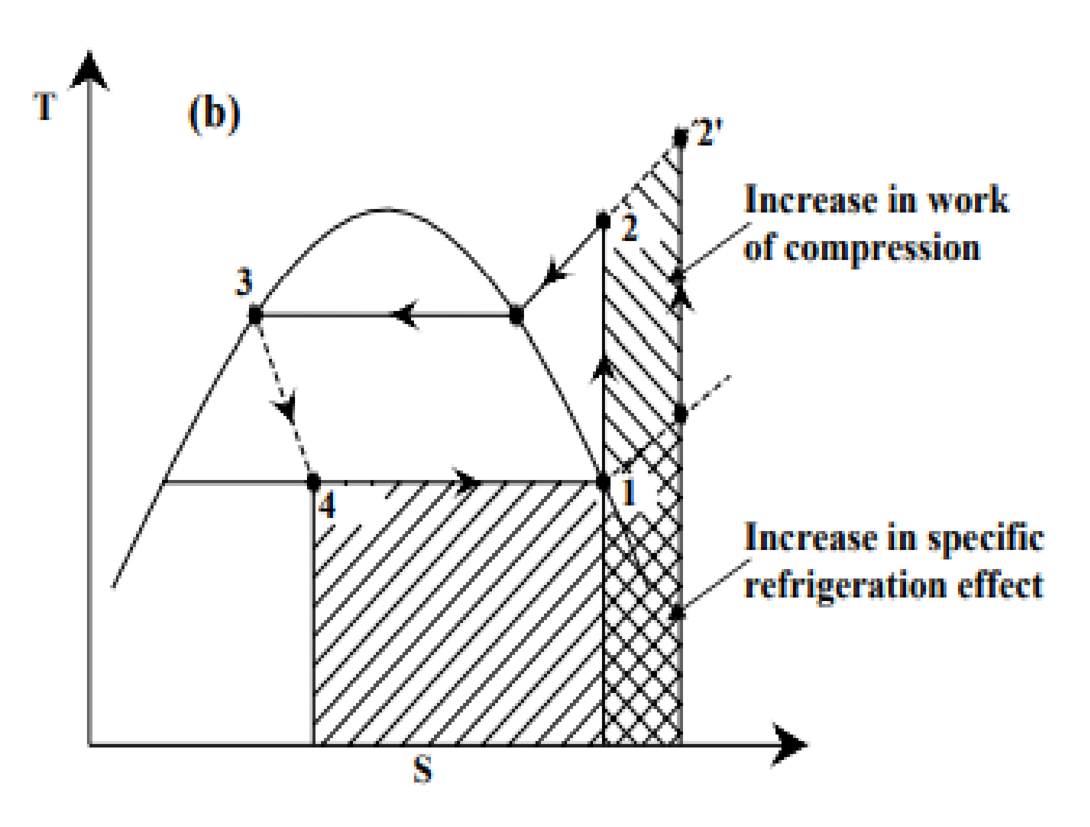


Modifications to VCR cycle - Superheating

- Temperature of heat source will be a few degrees higher than the evaporator temperature, hence the vapour at the exit of the evaporator can be superheated
- If the superheating of refrigerant takes place due to heat transfer with the refrigerated space then it is called as **useful** superheating
- On the other hand if refrigerant vapour becomes superheated by exchanging heat with the surroundings it is called as **useless** superheating

Effect of superheat





Effect of useful superheating

- Increases both refrigeration effect and work of compression
- Increases compressor temperature and specific volume of refrigerant at compressor inlet discharge
- Whether COP and volumetric refrigeration effect increase or not depend on the nature of the refrigerant and operating conditions
- Some amount of superheat is always used to prevent entry of liquid into compressor
- Useless superheat is detrimental suction lines should be insulated

Effect of superheating

- Whether superheating improves COP or not, in all practical systems a minimum amount of superheating is provided to:

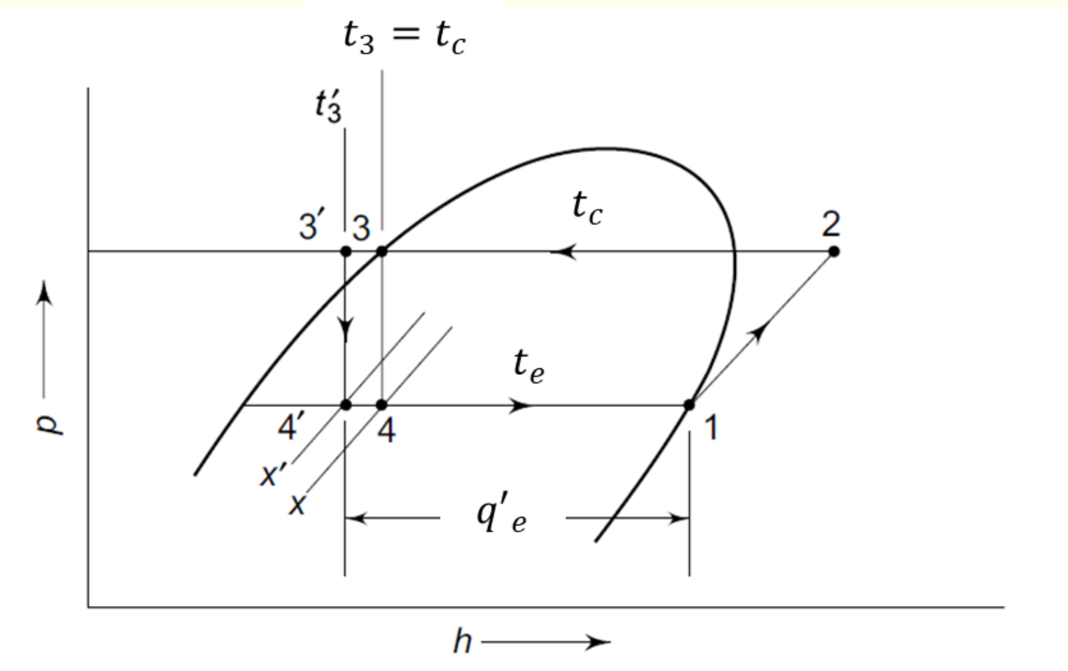
- Ensure only refrigerant vapour entry into compressor

- Improve volumetric efficiency of compressor, and

- To prevent moisture condensation on suction lines in domestic refrigerators

Effect of Liquid Subcooling

- A Sub cooler between the condenser and the expansion valve.
- The temperature of the refrigerant can be reduced below its saturation liquid temperature.
- Sub cooling reduces flashing of the liquid during expansion and increases the RE.
- This reduces the power per ton.



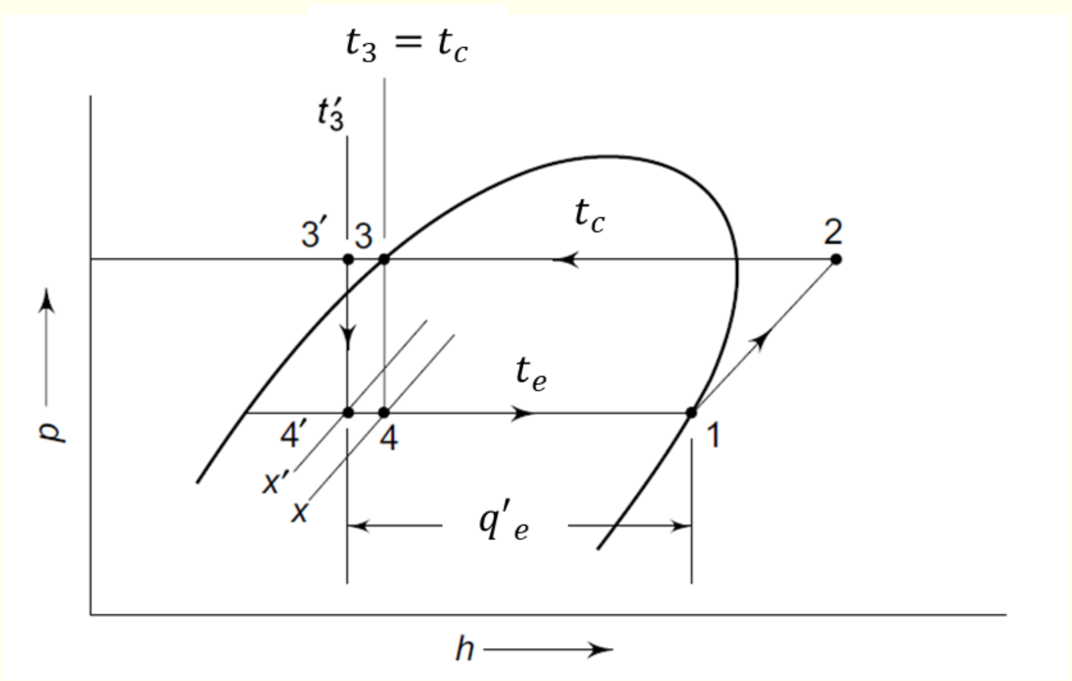
Effect of Liquid Subcooling

- Normally cooling water first passes through sub-cooler and then through condenser. This results in a warmer water entering the condenser and therefore higher condensing temperature and pressure.

- The advantage of sub-cooling is offset by the increased work of compression.

- Parallel cooling water inlets to the sub-cooler and condenser may be installed.

- But degree of sub cooling can be small – economics.



The functions of the condenser as well as the sub cooler can be combined in the condenser itself by slightly oversizing the condenser.

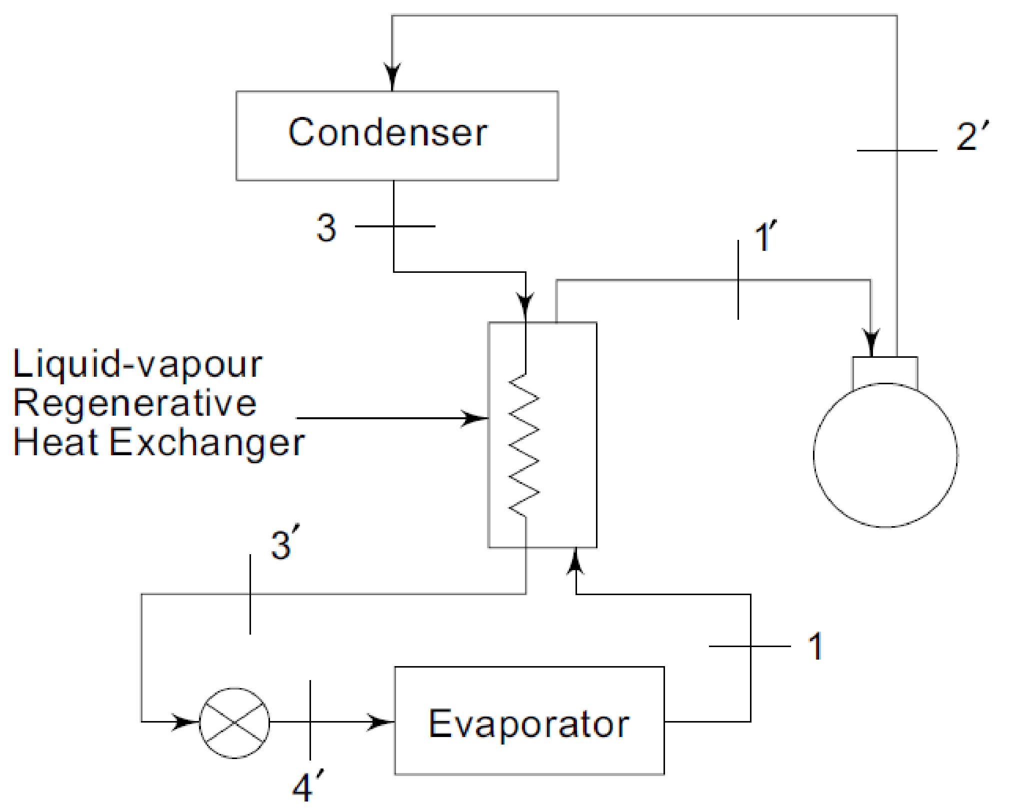
Liquid–Vapour Regenerative Heat Exchanger

- Combining vapor superheat and liquid sub-cooling – LV HX

- Mass flow rates are same.

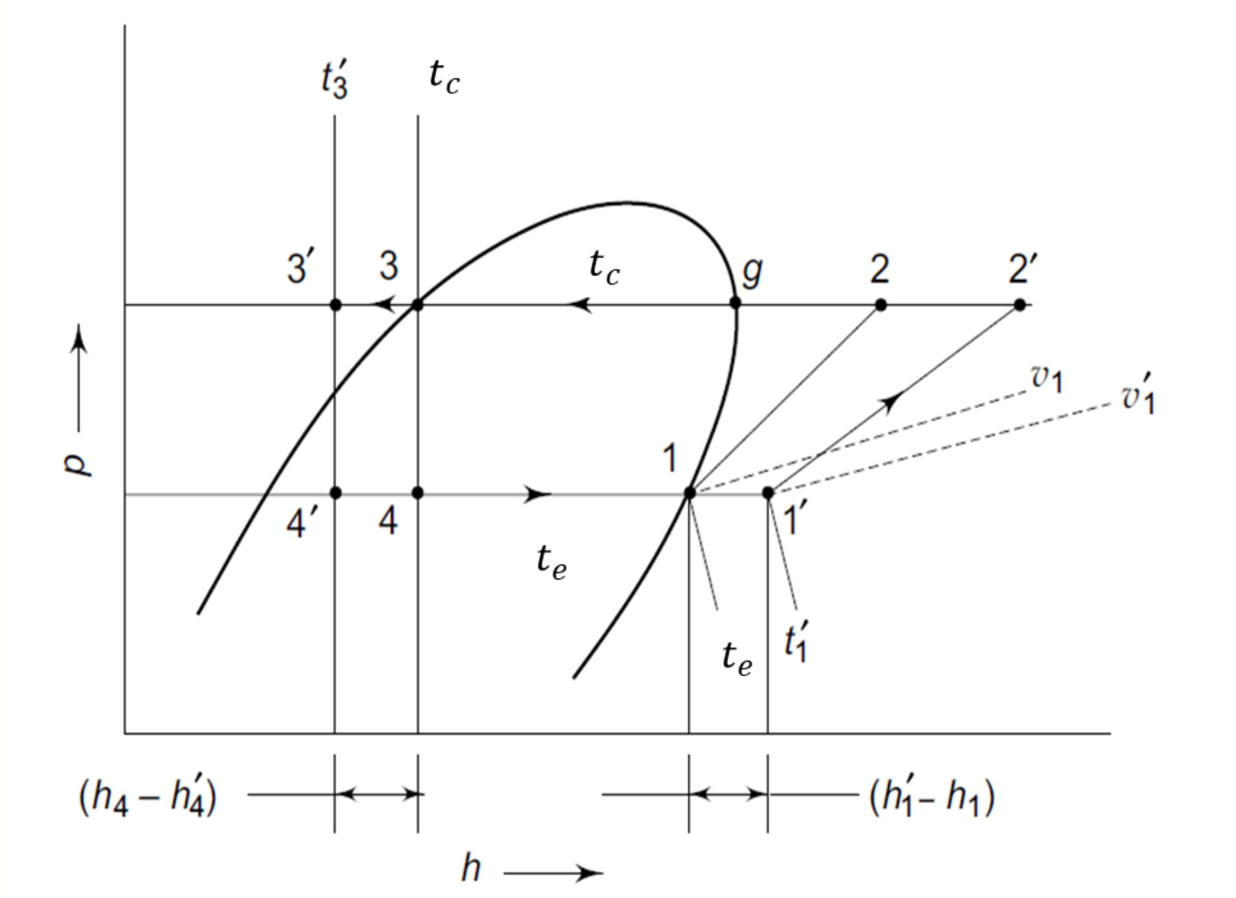
- $q_n = h_1' - h_1 = h_3 - h_3'$

- Degree of superheat ($t_1' - t_e$) and the degree of subcooling ($t_c - t_3'$) need not be same as the specific heats of the vapour and liquid phases are different.



Liquid–Vapour Regenerative Heat Exchanger

- $q_n = h_1' - h_1 = h_3 - h_3'$
- $\frac{\dot{Q}_e{}'}{\dot{Q}_e} = \frac{h_1 - h_{4'}}{h_1 - h_4} \cdot \frac{v_1}{v_{1'}}$
- $\frac{W^{*\prime}}{W^*} = \frac{h_1 - h_4}{h_1 - h_4'} \cdot \frac{h_2' - h_1'}{h_2 - h_1}$
- $E_c' = \frac{(h_1 - h_4) + (h_1' - h_1)}{(h_2 - h_1) + [(h_2' - h_1') - (h_2 - h_1)]}$
- both numerators and denominators increase. The net effect depends on the refrigerant used and the operating conditions.



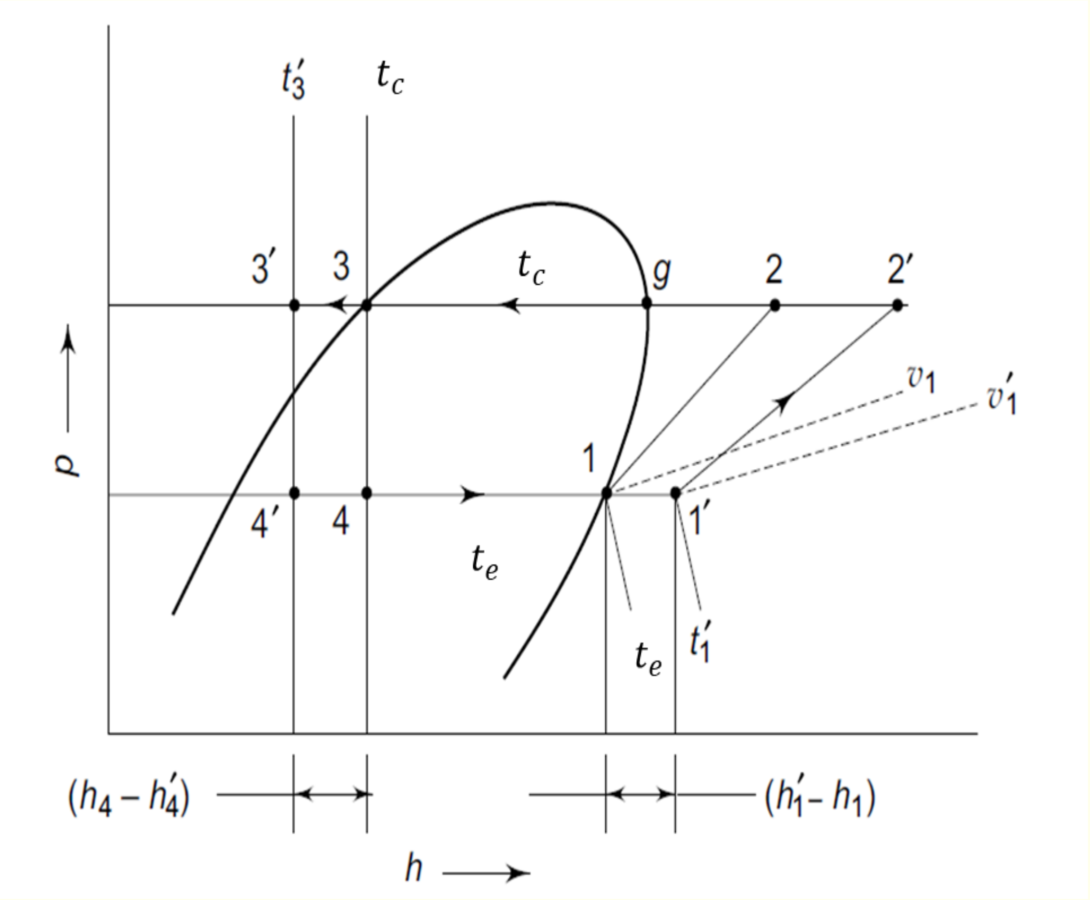
Liquid–Vapour Regenerative Heat Exchanger

- Suction volume per ton and HP per ton reduce for R12 and R134a but for R22 and NH_3

- Volumetric efficiency of most reciprocating compressors improves with superheat, Super heat is preferable in evaporator itself.

- Increased refrigerating effect ($h_1' - h_1$) due to superheating increasing temperature t_e to $t_{1'}$ is transferred as ($h_4 - h_4'$) at temperature t_e

- This increases mean refrigeration temperature.



Actual VCRS systems

- They differ from theoretical cycles due to:

  a) Pressure drops in evaporator, condenser and LSHX

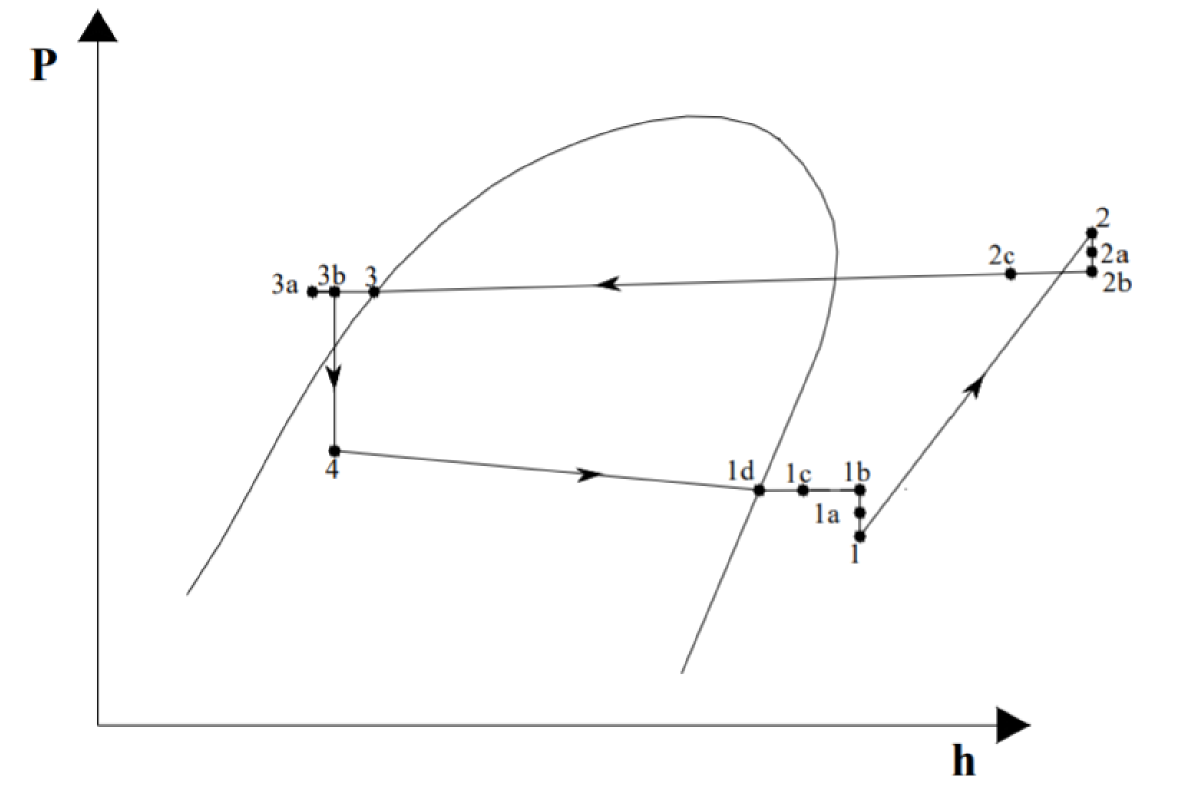
  b) Pressure drop across Suction and discharge valves of the compressor

  c) Friction and heat transfer in compressor

  d) Pressure drop and heat transfer in connecting pipe lines, and Presence of foreign matter

Actual vapour compression cycle

- Pressure drop in the piping
- Heat losses or gains depending on temperature differences
- Polytropic compression with friction and heat transfer – not isentropic.
- Actual VCR will have all these losses

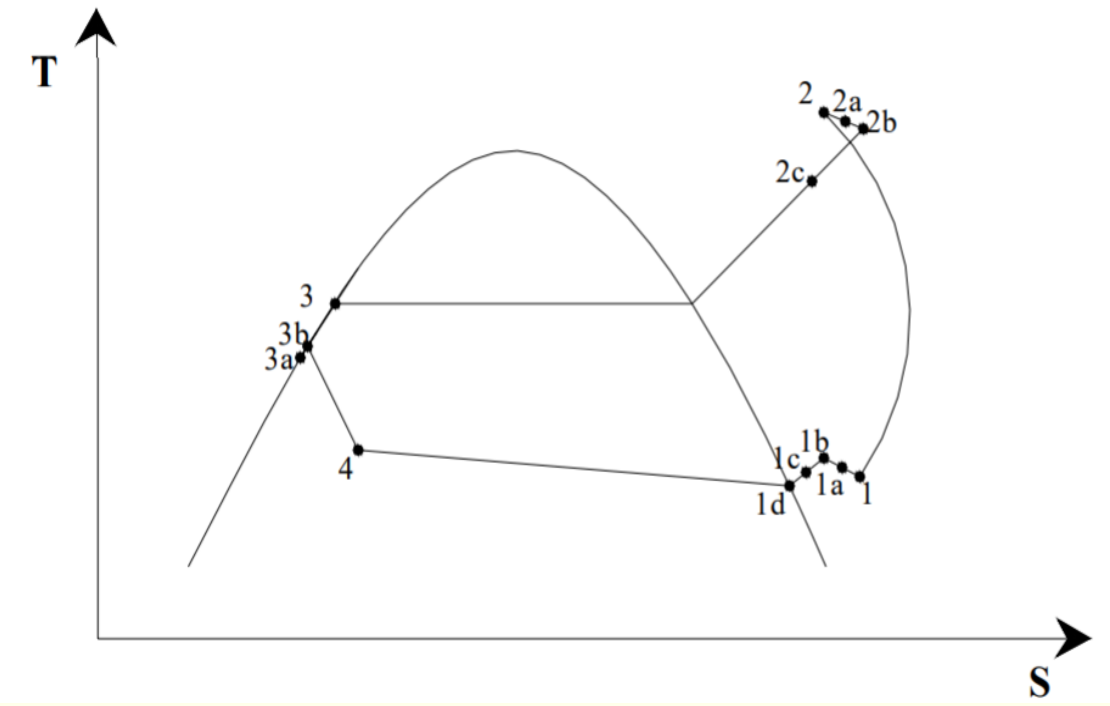


Actual vapour compression cycle

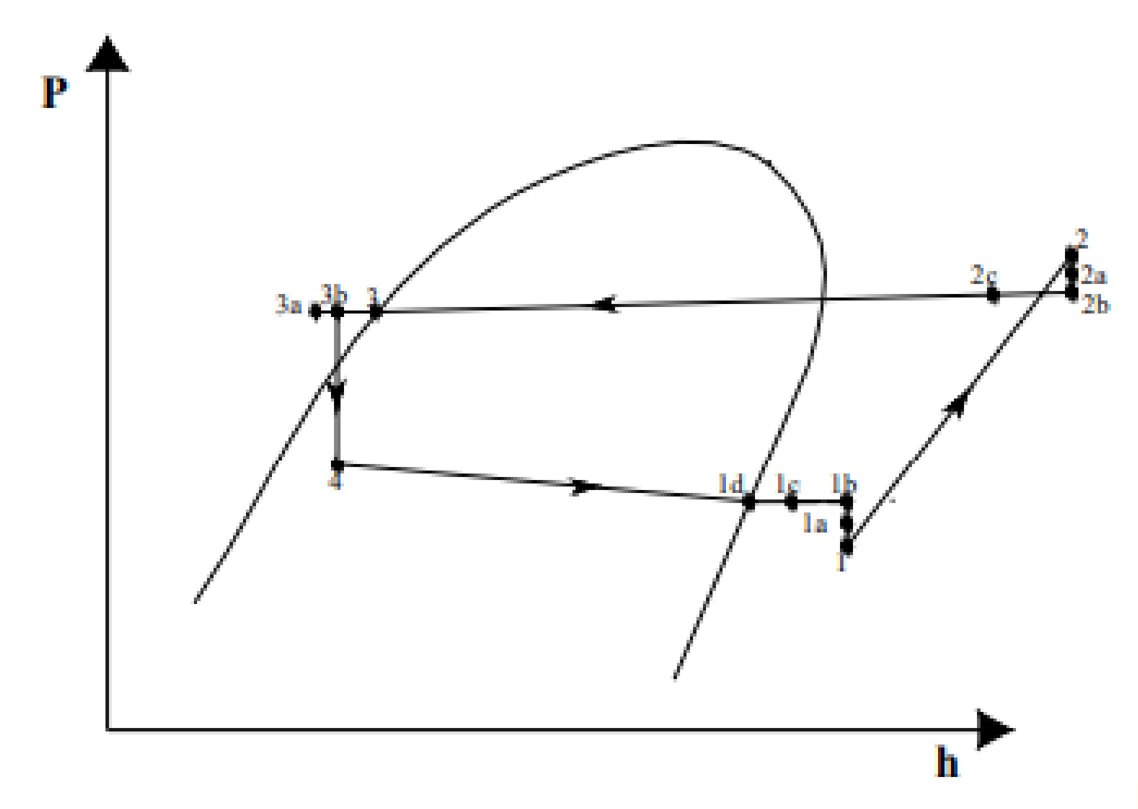
- Pressure drop in evaporator is large due to friction and velocity increase (momentum pressure drop) This would increase power consumption greatly or alternatively reduce RE. – Temp is not constant in evaporator.

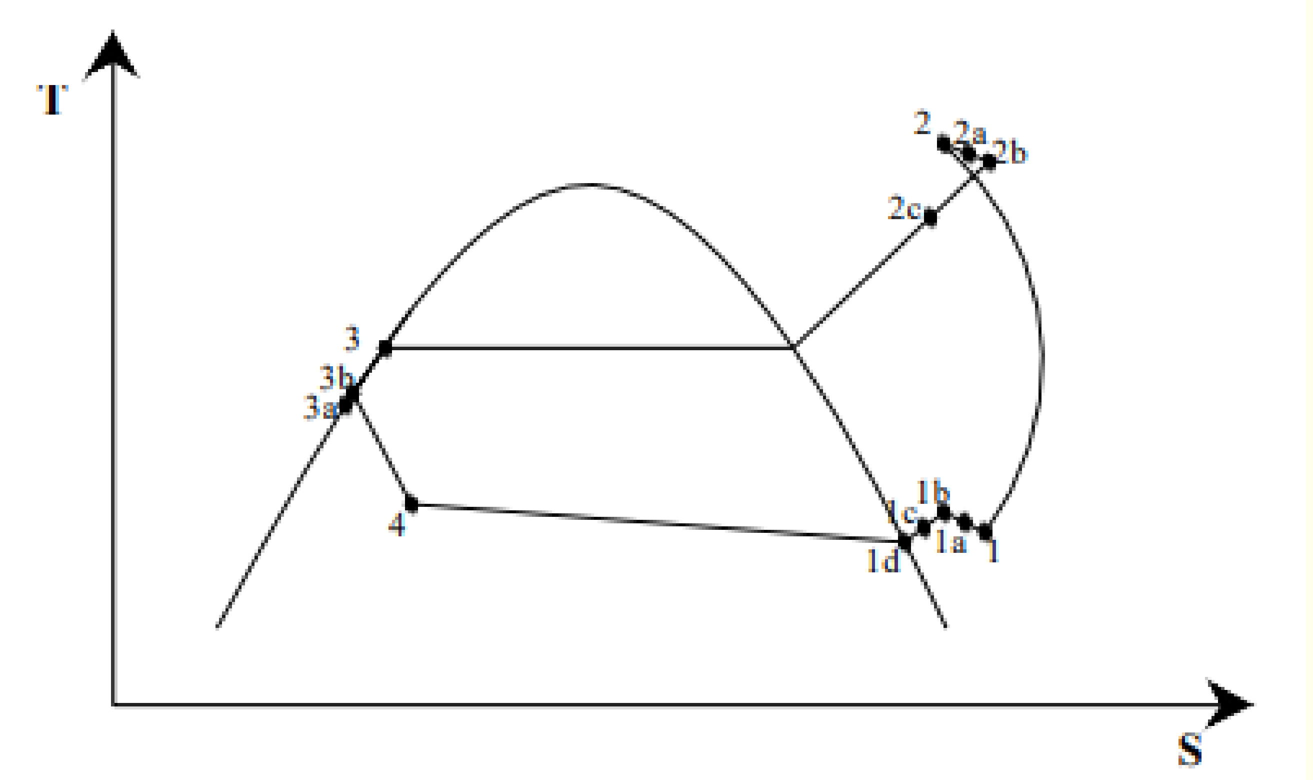
- Pressure drop in condenser is not critical.

- The capacity of the plant is decreased and the unit power consumption (per unit of refrigeration) increases, COP decreases.



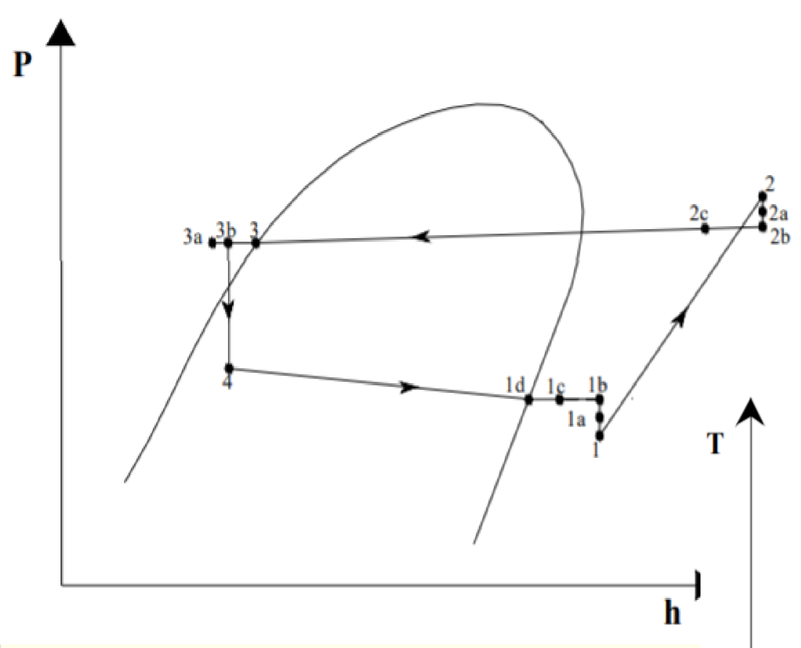
Actual VCRS cycle on P-h and T-s diagrams

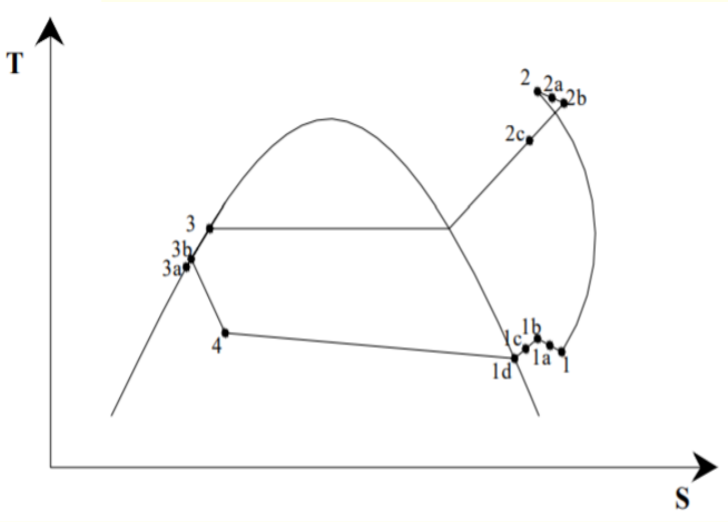




Actual VCRS cycle on P-h and T-s diagrams

Process	State
Pressure drop in evaporator	4-1d
Superheat of vapour in evaporator	1d-1c
Useless superheat in suction line	1c-1b
Suction line pressure drop	1b-la
Pressure drop across suction valve	1a-1
Non-isentropic compression	1-2
Pressure drop across discharge valve	2-2a
Pressure drop in the delivery line	2a-2b
Desuperheating of vapour in delivery pipe	2b-2c
Pressure drop in the condenser	2b-3
Subcooling of liquid refrigerant	3-3a
Heat gain in liquid line	3a-3b





Effect of pressure drop and heat transfer

- Pressure drop and heat transfer in the vapour line affects the performance significantly by reducing
- System capacity, and COP
- Normally a pressure drop leading to a saturation temperature of 1-2 K in evaporator and 1 K in suction line is considered to be acceptable

Effect of pressure drop and heat transfer

- Pressure drop in evaporator and suction line depends on:

- Layout and type of the refrigerant tubing
- Velocity of refrigerant, and
- Type of refrigerant

- Pressure drop can be reduced by reducing refrigerant velocity, however, a minimum velocity of 4-6 m/s is required in evaporator and suction lines for proper carry-over of lubricating oil to the compressor

Effect of pressure drop and heat transfer

- Pressure drop and heat transfer in liquid lines is not very critical
- Pressure drops across the valves of the compressor can be quite considerable and may affect the performance adversely
- Heat transfer from the compressor is deliberately provided in most of the cases so as to operate the compressor within safe temperature limits
- Hence, compression in actual systems is polytropic. Isentropic efficiency is an indication of the deviation

Use of pressure-enthalpy P-h charts

- Since various performance parameters are expressed in terms of enthalpies, it is very convenient to use a pressure - enthalpy chart for property evaluation and performance analysis

- Using P-h charts one can easily find system performance from known values of evaporato and condenser temperatures

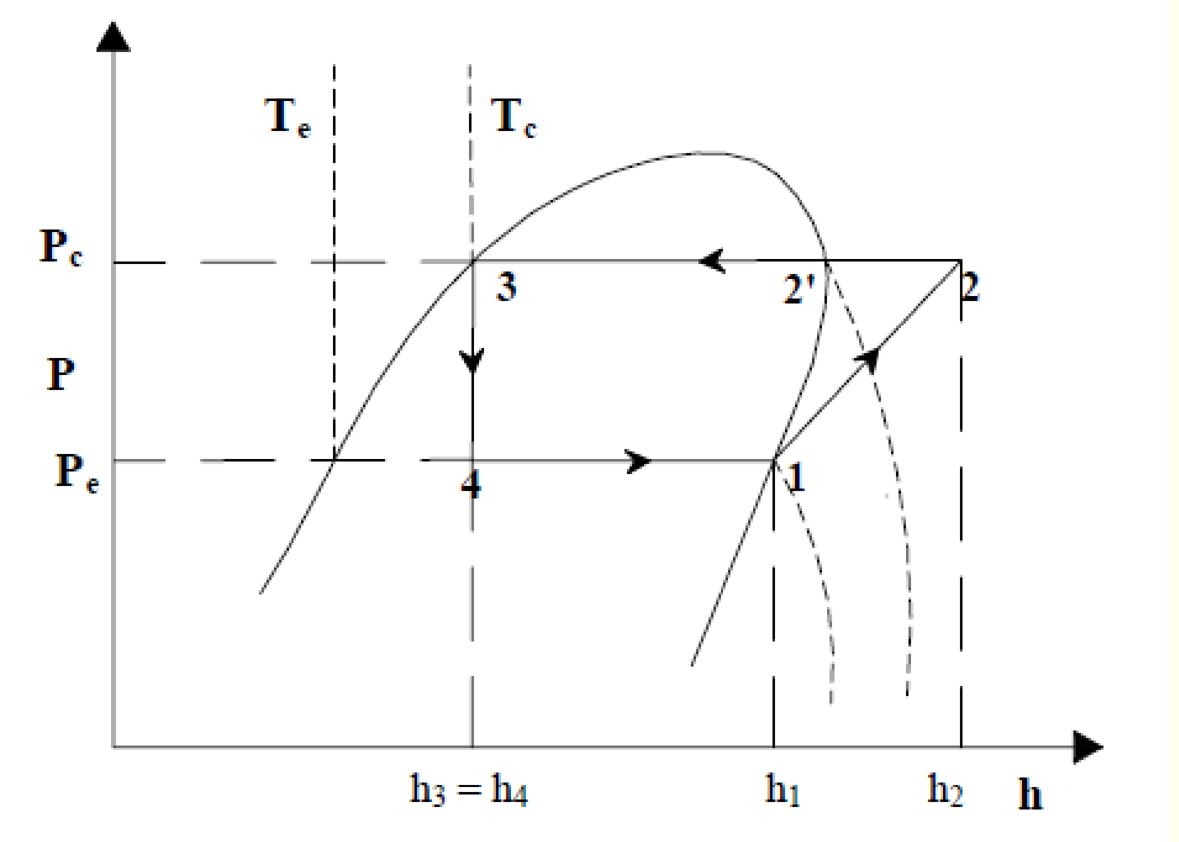


Image credit: NPTEL

Performance of a VCR cycle

- For good system performance the evaporator temperature should be high and condensing temperature should be low
- At low evaporator temperature effect of condenser temperature is marginal
- At very low evaporator temperatures, SSS cycle is not viable, in such cases a multistage or cascade system should be used

Summary

- Carnot refrigeration cycle and its practical limitations are discussed
- The standard vapour compression refrigeration system (VCRS) is introduced
- VCRS is compared with Carnot system
- Performance analysis of VCRS is presented
- Effect of change in parameters is discussed.
- Actual VCR system is also discussed.

Thank you !