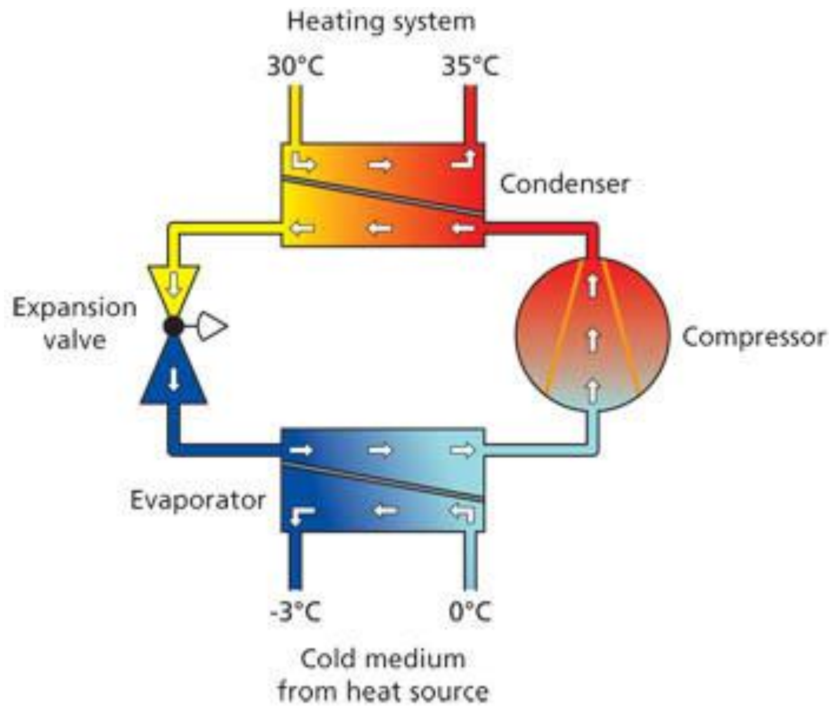
Heat pumps - principles

- heat pumping
- basic cycles
- main components of HPs

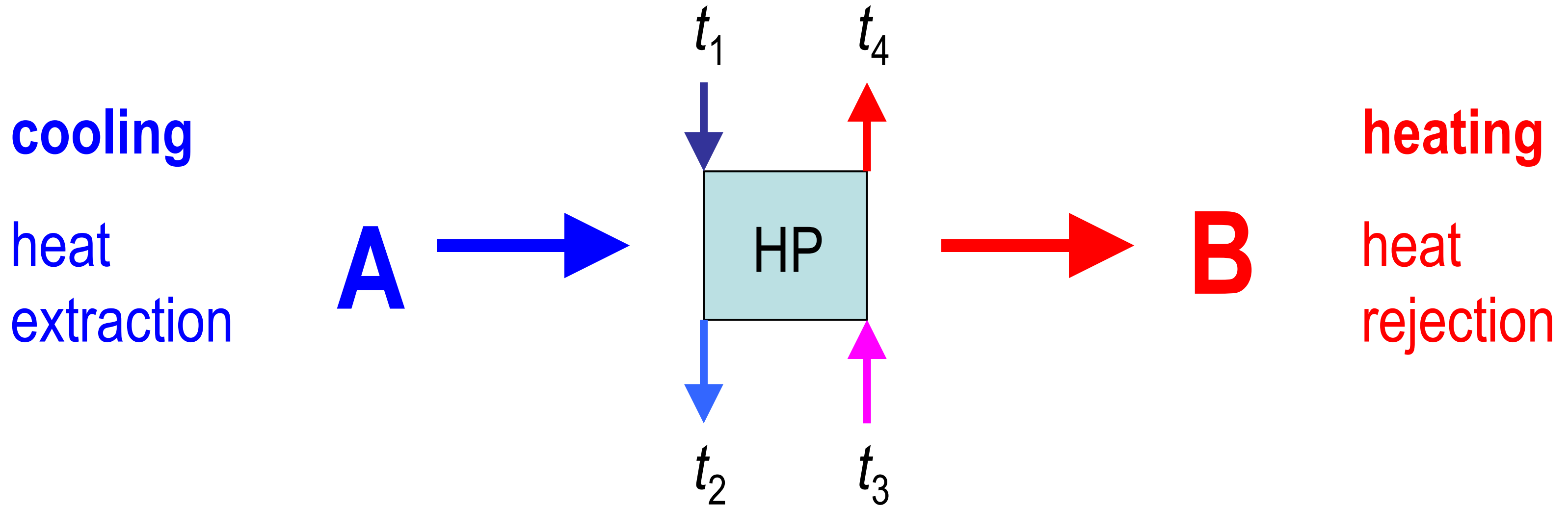


Heat pumps HP

… generally devices for:

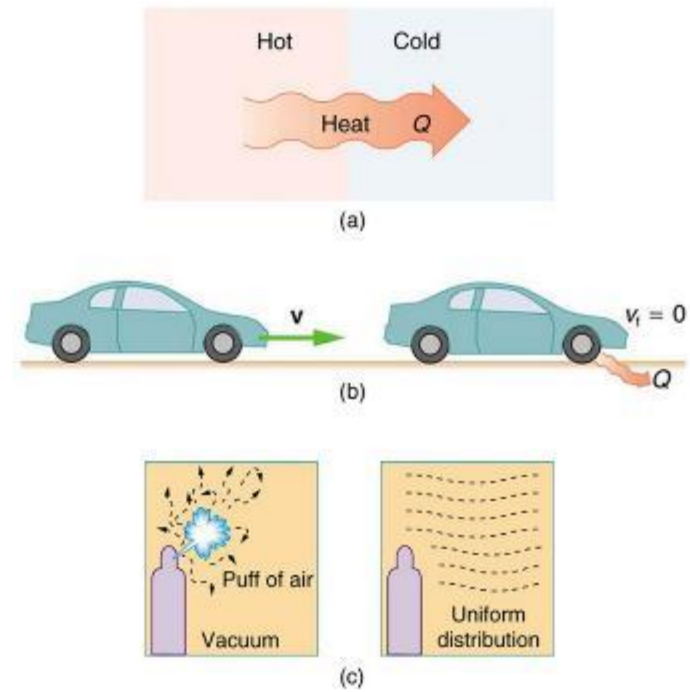
pumping the thermal energy from environment **A** at **low (= nonutilisable)** temperature

transferring it to environment **B** at **higher (=utilisable)** temperature



Heat pumps – basic principles

- **2nd law of thermodynamics**
  (increase of entropy in isolated systems,
  irreversibility of heat processes):
  - „thermal energy **cannot** be **freely** transferred from environment at lower temperature to environment at higher temperature “
  - the process can be realised only if external energy at **higher quality** (potential, temperature) enters the system



(entropy …the rate of energy degradation)

Heat pumps – basic principles

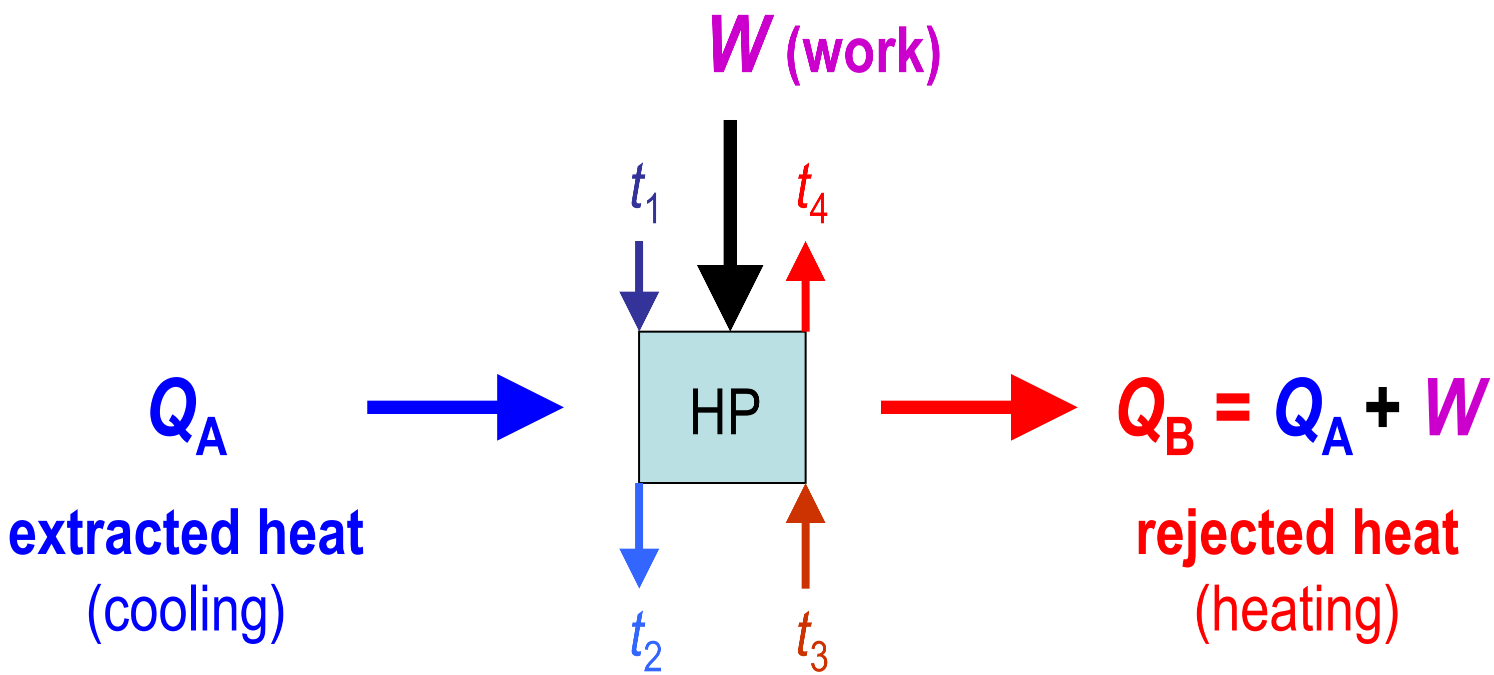
- high-potential energy
  1. **electric** (electric engine)
  2. **mechanical** (shaft, gearing)

  3. heat at **higher temperature** than temperature, to which the heat is pumped (gas burner)

Heat pumps – basic principles

- heat **pumping**:

  driving high-potential energy (work) *W* degrades and is transferred to environment **B** with the extracted (pumped) energy



Devices

cooling machine

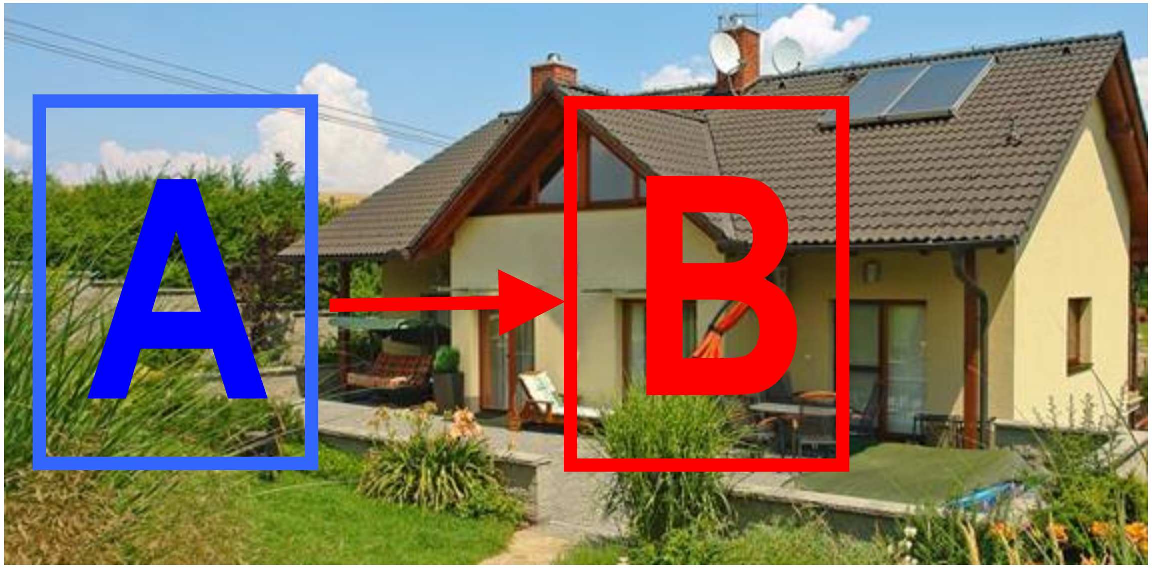
- uses primarily the cooling effect
- usable heat is extracted heat from environment A (lowering the temperature)

»

heat rejected to environment B is not used (waste heat)

Devices

- **heat pump**
  - usable heat is the rejected heat to environment **B**



- difference is not in the principle, but in character of heat management
- both devices can’t be simply mixed – differences in practical construction

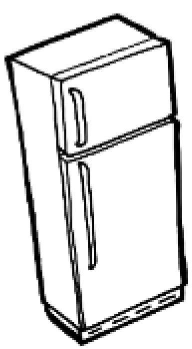
Energy performance

$$Coef.of \cdot performance = \frac{what \cdot we \cdot want}{what \cdot we \cdot pay}$$

coefficient of performance

COP

$$COP = \frac{Q_B}{W}$$



energy efficiency ratio

EER

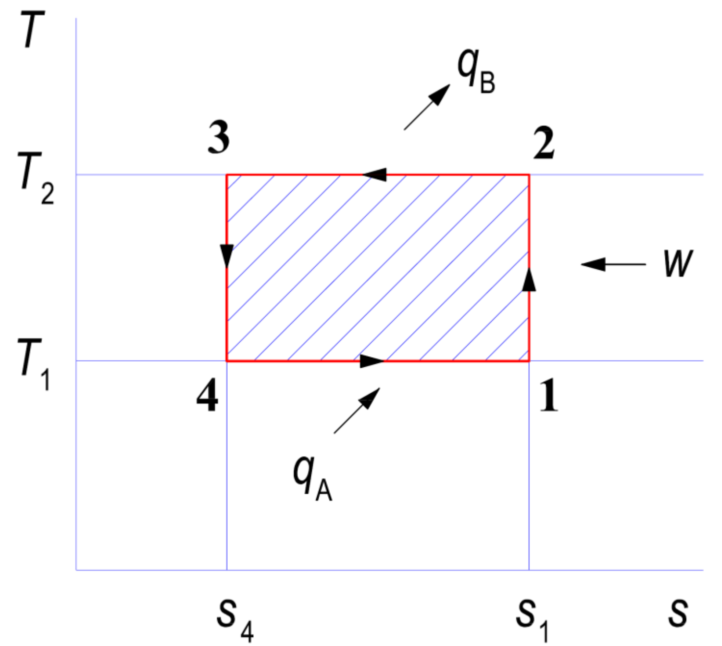
$$EER = \frac{Q_A}{W}$$

Carnot cycle

- **theoretical cycle**
  - reversible (ideal)
  - the most efficient thermal cycle
  - can't be realised in reality
- **isoentropic changes (*s* = const.)**
  - Expansion, compression I I
- **isothermal changes (*T* = const.)**

heat output q_B ____

heat input q_A ____



Carnot cycle

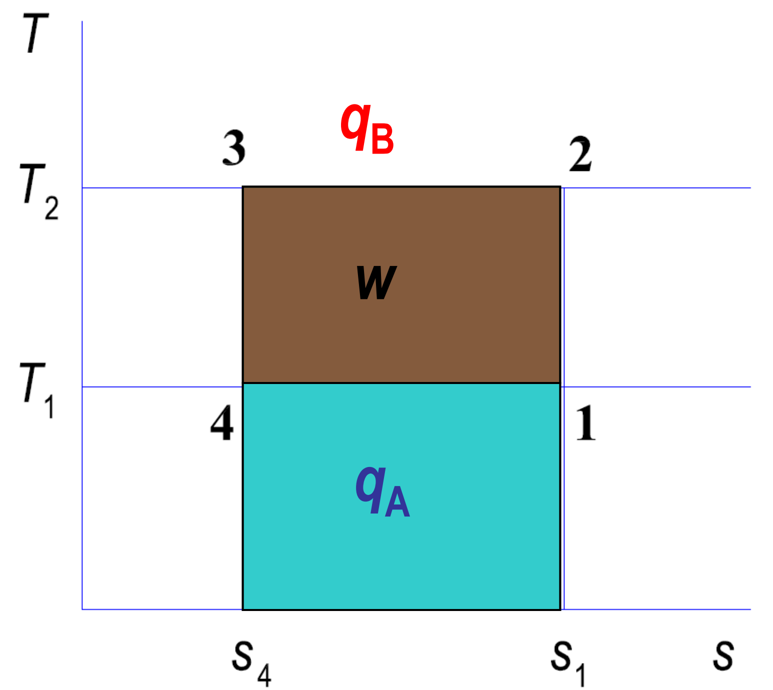
specific energy

$$q_B = q_{23} = T_2 \cdot (s_1 - s_4)$$

$$q_A = q_{41} = T_1 \cdot (s_1 - s_4)$$

[J/kg]

$$w = q_B - q_A = (T_2 - T_1) \cdot (s_1 - s_4)$$



$$COP_C = \frac{q_B}{w} = \frac{T_2}{T_2 - T_1}$$

$$EER_C = \frac{q_A}{w} = \frac{T_1}{T_2 - T_1} = COP_C - 1$$

Carnot cycle

- **unrealistic cycle – not considering:**
  - finite surface area of heat exchangers
  - thermophysical properties of working fluids (refrigerants)
  - real efficiency of driving energy source
  - heat losses
  - auxilliary energy (pumps to overcome hydraulics losses)

- **real coefficient of performance – comparison with Carnot**

$$COP_{HP} = \eta_{HP} \frac{T_2}{T_2 - T_1}$$

comparative efficiency η_{HP} = 0,4 to 0,6

0,4: small HP capacity

0,6: large HP capacity

Example

- environment A 0 °C
- environment B 40 °C 60 °C
- calculate Carnot $COP_{HP} = \eta_{HP} \frac{T_2}{T_2 - T_1}$

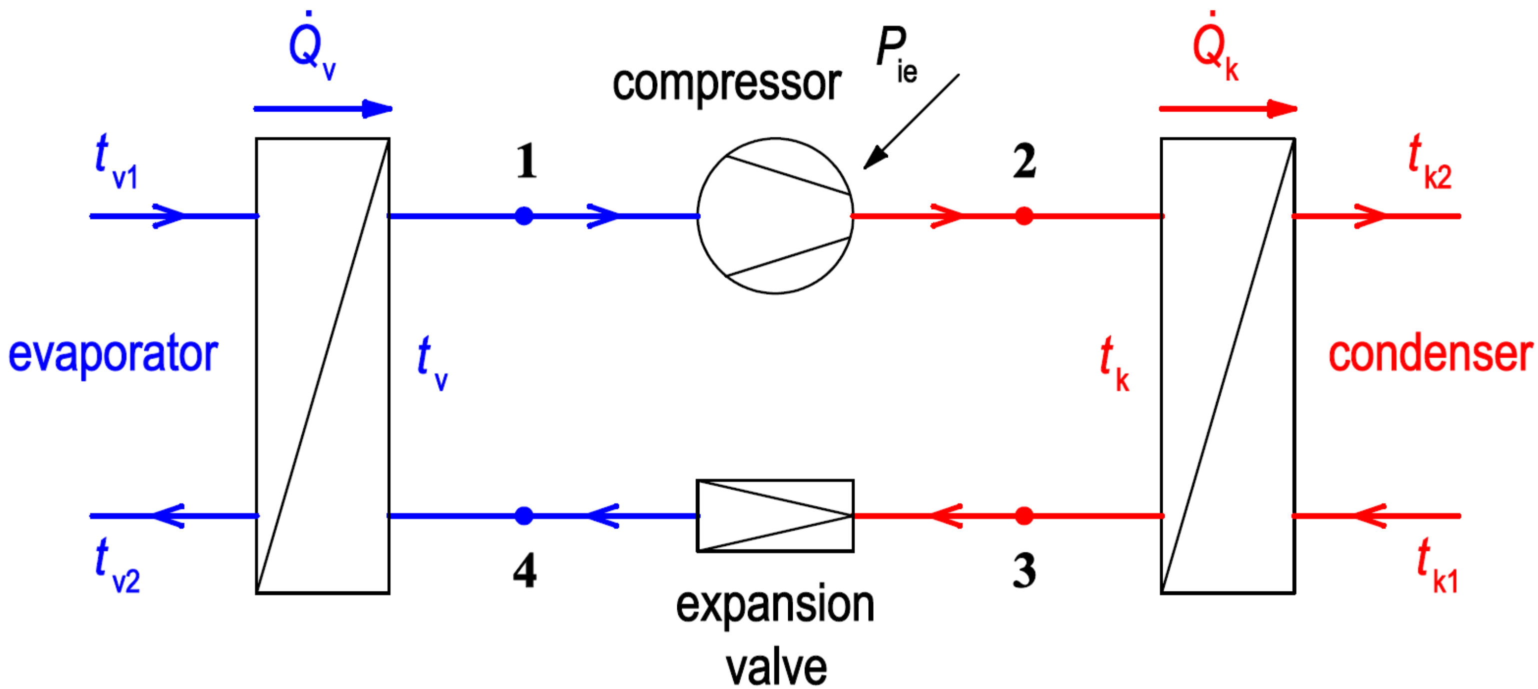
Carnot COP … 6,8 4,6

Real COP … 2,7 – 4,1 1,8 – 2,7

2,7 → small or/and unefficient HP

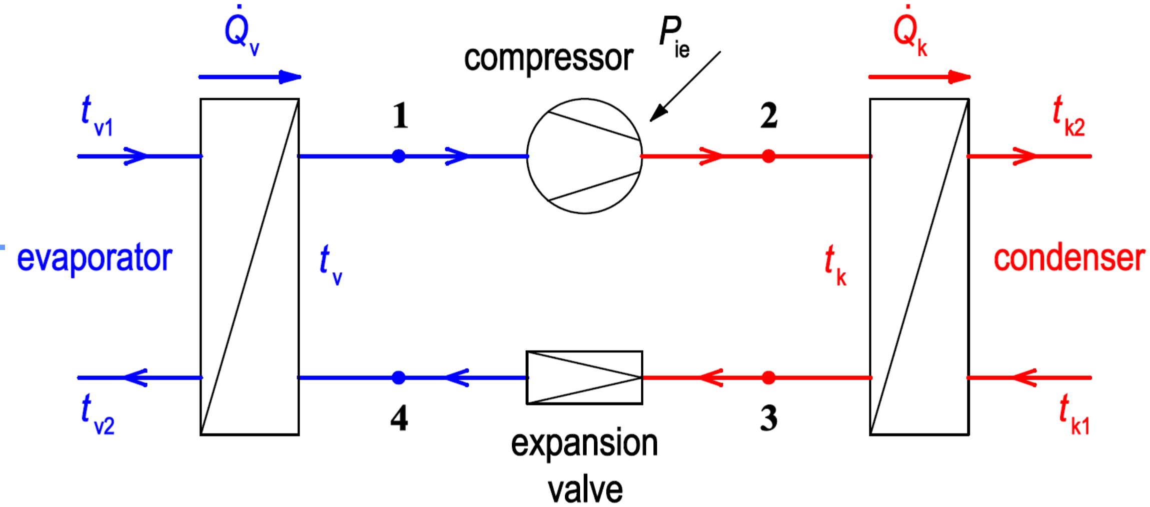
4,1 → big or/and efficient HP

vapour cycle (ideal)



$$Q_k = Q_v + P_{ie}$$

$$COP = Q_k / P_{ie}$$



- most widespread cycle

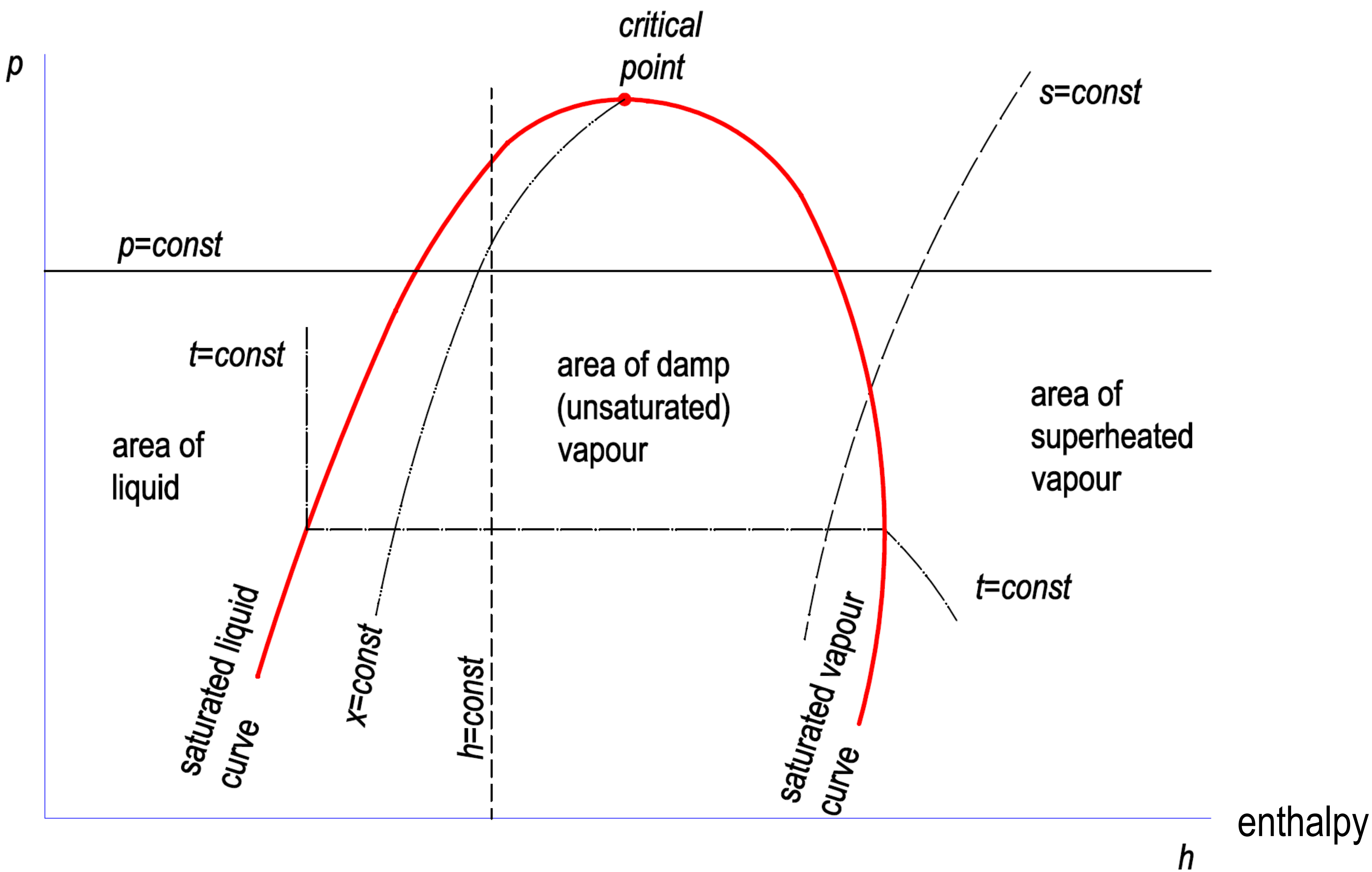
1) heat extraction at **low** temperature and **low** constant pressure with phase change (evaporation) of working fluid in **evaporator**

2) vapour suction and compression by a **compressor**
increase of pressure = increase of boiling point of the fluid

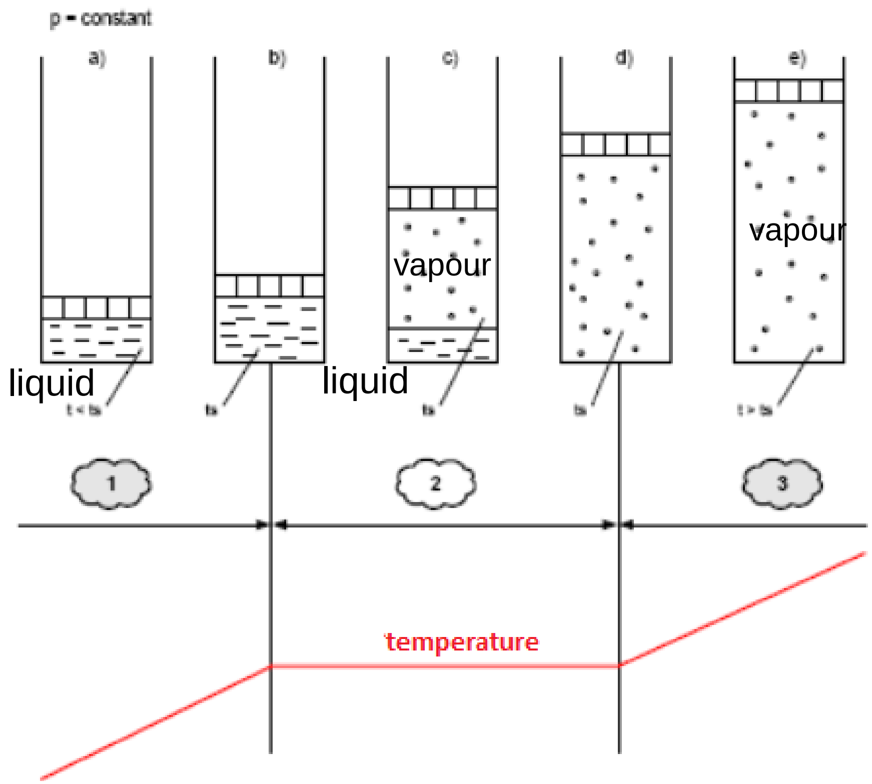
3) heat rejection at **high** temperature and **high** constant pressure with phase change (condensation) of working fluid in **condenser**

4) decrease of pressure (expansion) in **expansion valve**
decrease of pressure = decrease of the boiling point of the fluid

Working fluid – refrigerant (diagram)



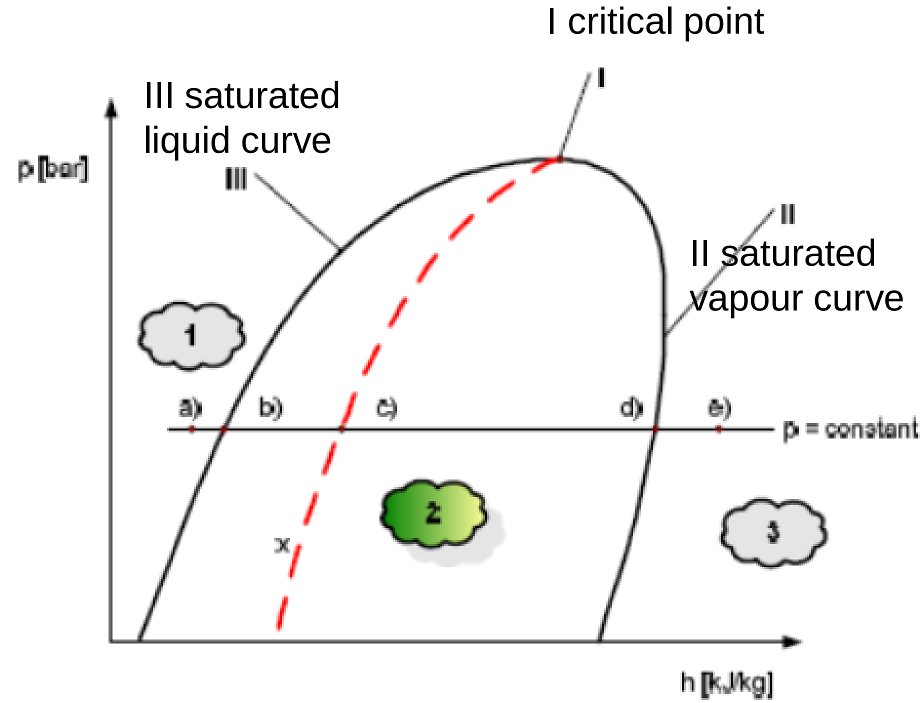
Evaporation of water in the pot



1 liquid heating

2 evaporation

3 superheating

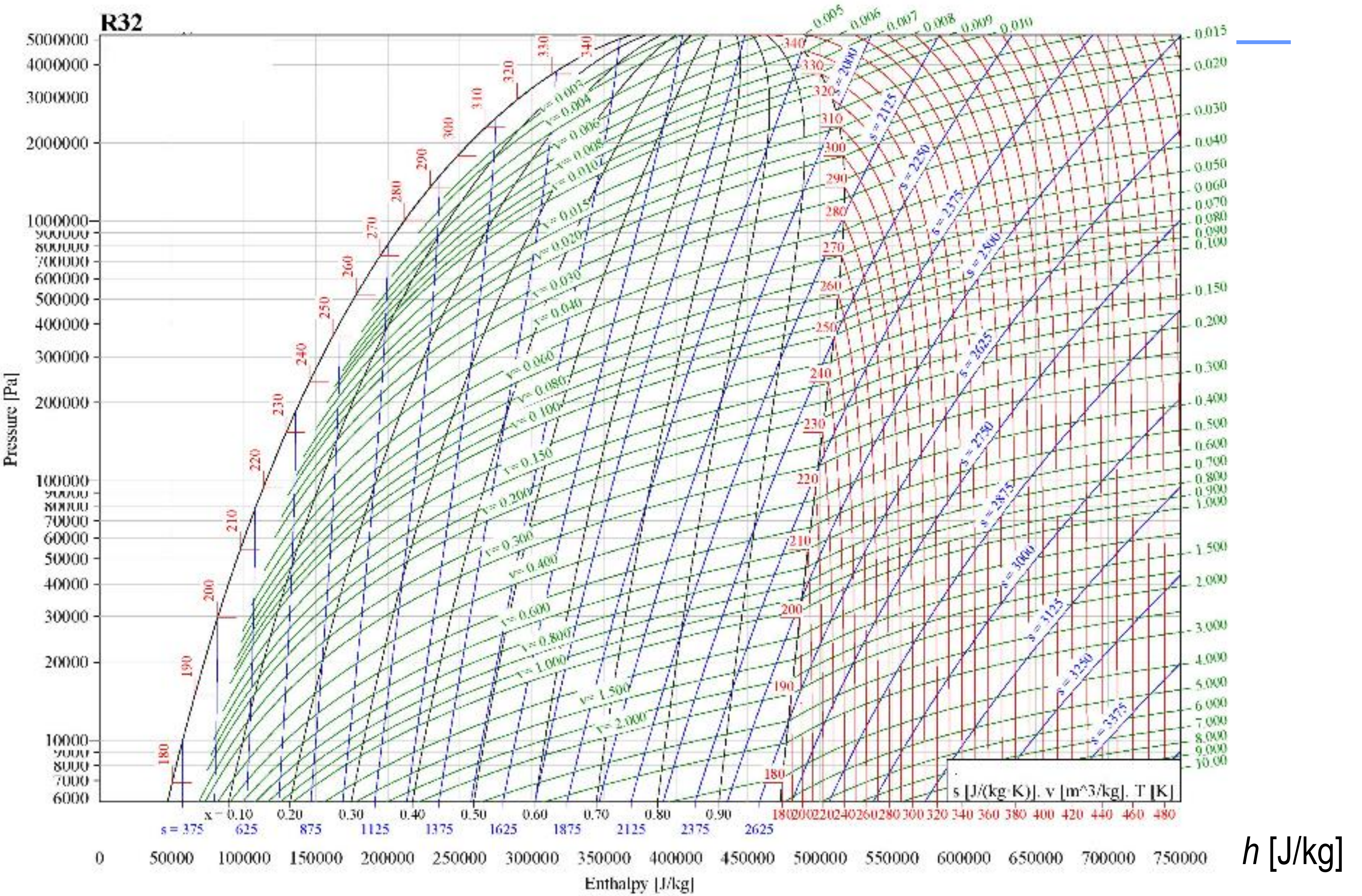


1 liquid

2 damp vapour

3 superheated vapour

Working fluid – refrigerant (diagram)



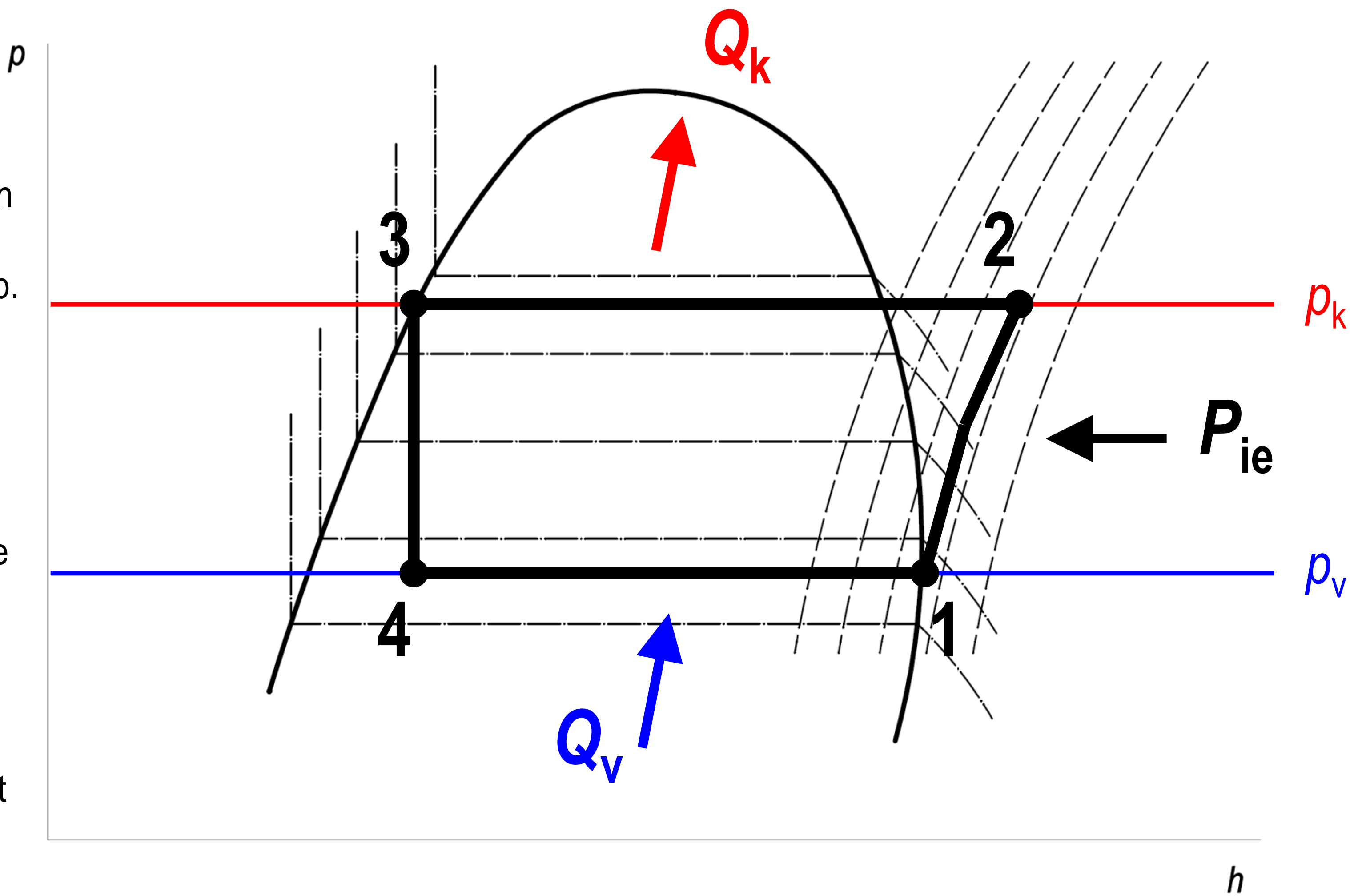
Rankin vapour cycle (ideal)

1-2 vapour suction+compres in **compressor,** increasing boiling p.

2-3 h.rejection at **high** temp. in **condenser**

3-4 decr. pressure in **expansion valve** decrease boiling point

4-1 h.extraction at **low temp** in **evaporator**



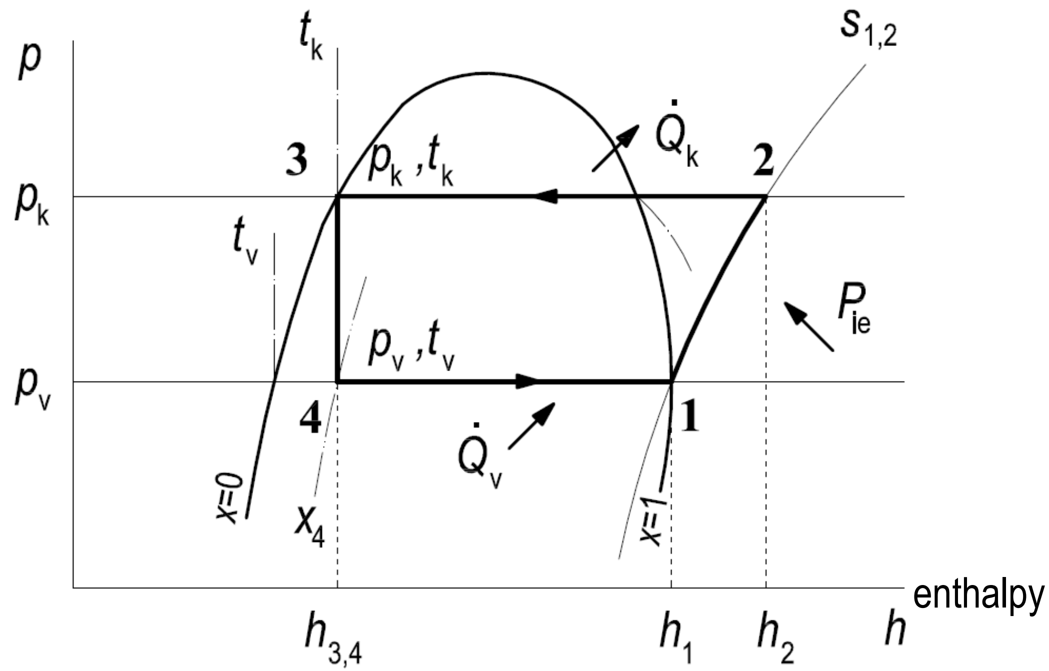
Balance of Rankin vapour cycle

$$\dot{Q}_v = \dot{M}_r \cdot (h_1 - h_4)$$

Mass flow rate

$$\dot{Q}_k = \dot{M}_r \cdot (h_2 - h_4)$$

$$P_{ie} = \dot{M}_r \cdot (h_2 - h_1)$$



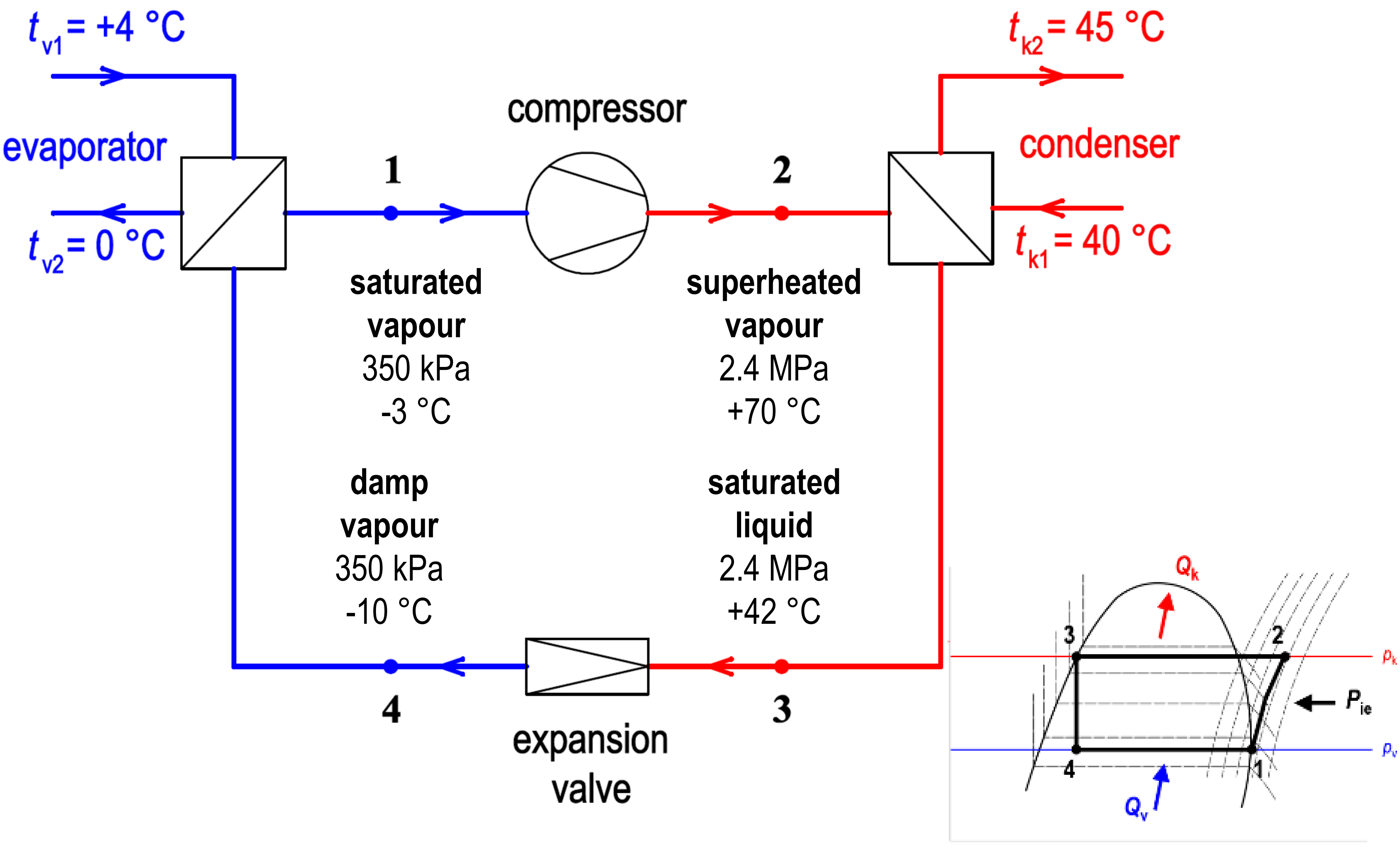
Coefficient of Performance

$$COP_R = \frac{\dot{Q}_k}{P_{ie}} = \frac{h_2 - h_4}{h_2 - h_1}$$

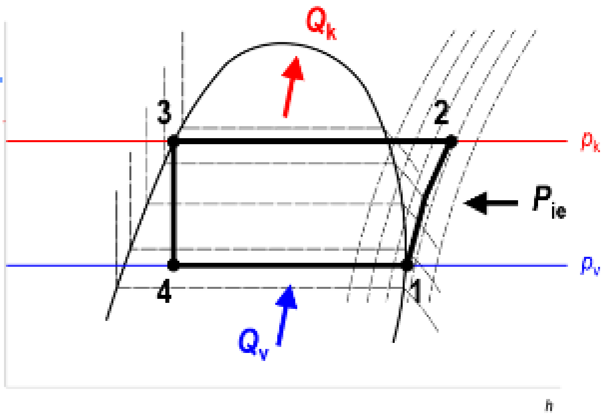
Energy Efficiency Ratio

$$EER_R = \frac{\dot{Q}_v}{P_{ie}} = \frac{h_1 - h_4}{h_2 - h_1}$$

(ideal) Rankin vapour cycle - example



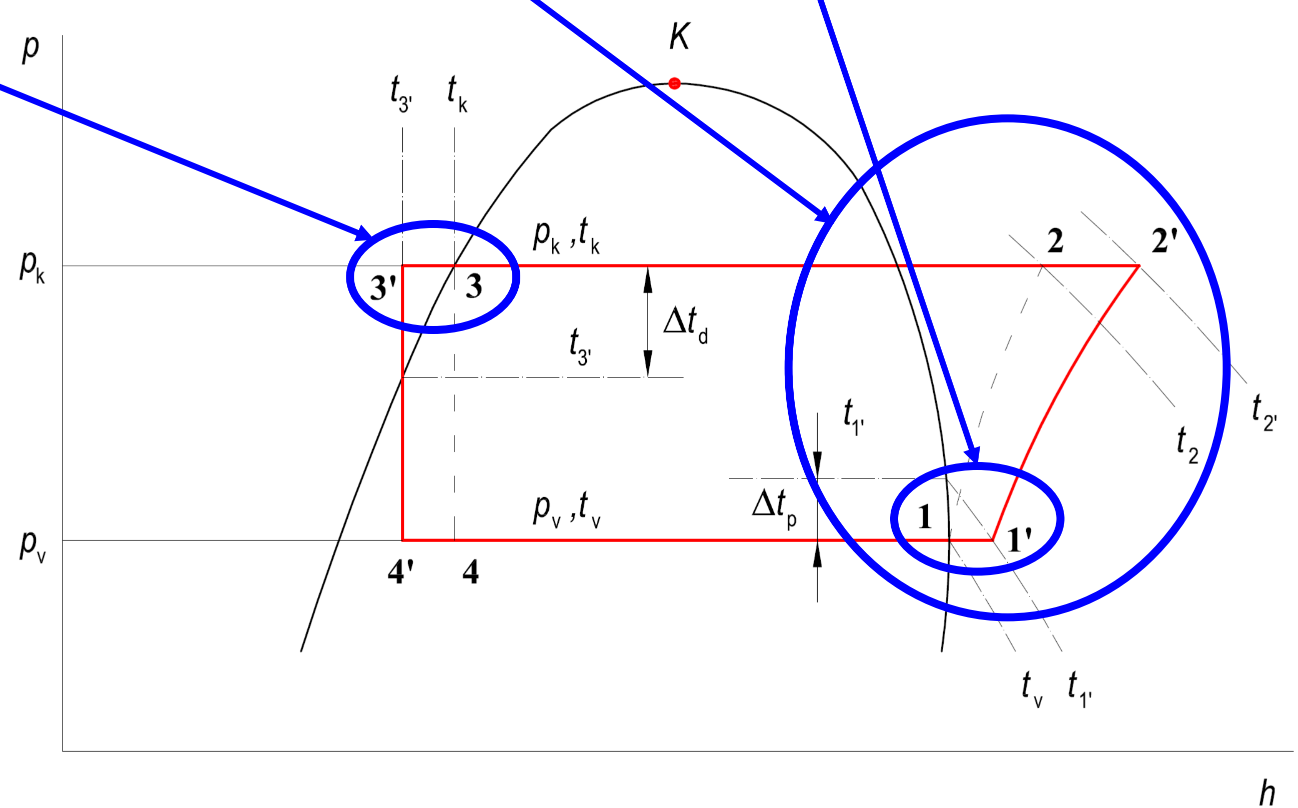
Rankin vapour cycle

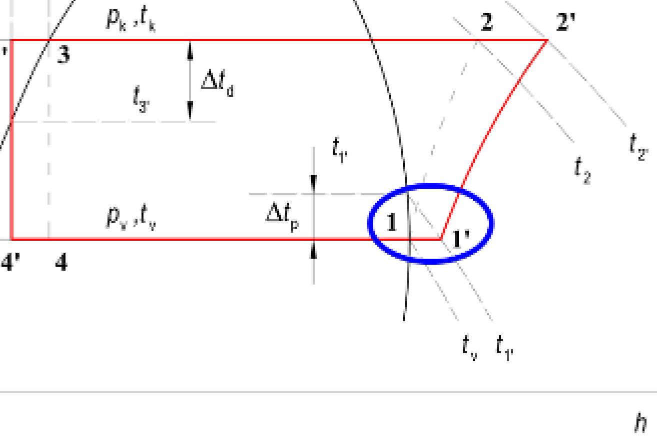


- **ideal Rankin cycle** assumes:
  - no subcooling at condenser, no superheating at evaporator, refrigerant states at saturated curves
  - no pressure losses (pipes, heat exchangers)
  - no heat loss of heat pump
  - isoentropic (ideal) compression
- Rankin cannot be realised, but differences from real cycle are quite small

Real vapour cycle

- **differences from Rankin cycle in:**
  - **superheating of refrigerant at evaporator 1 – 1´**
  - **polytropic compression 1´ - 2´**
  - **subccoling of refrigerant liquid at condenser 3 – 3´**



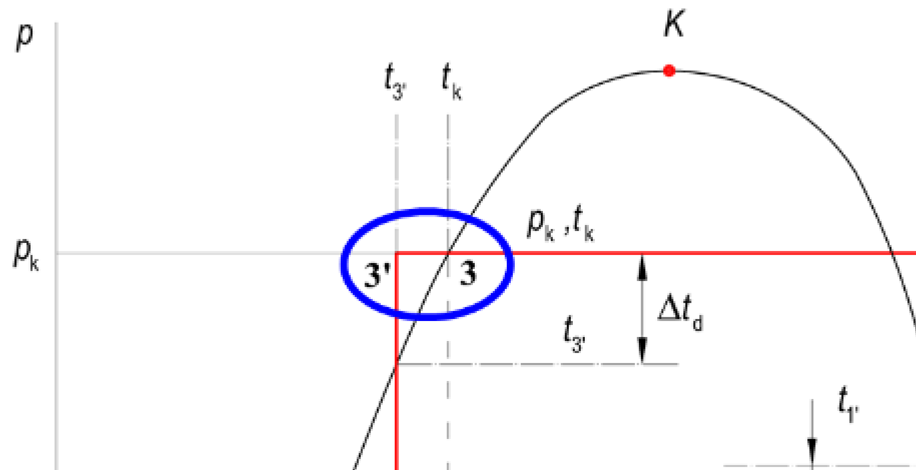


Superheating at evaporator

- compressor sucks the superheated vapour (1´)
- superheating has **an advantage** (contrary to cooling devices) – higher specific heat output
- superheated vapour at compressor intake = **longer durability**
- superheating due to:
  - function of controlled expansion valve
  - heat input from ambient = heat gains to pipe between evaporator and compressor
  - heat input from electric motor in hermetic compressor

Subcooling at condenser

- subcooling of liquid refrigerant under saturated state
- subcooling has benefits:
  - **proper function of expansion valve** – subcooling provides liquid refrigerant input = stabilisation, elimination of cavitation effects, higher durability
  - **increase of effectiveness** – increase of specific heat output without additional electric power demand



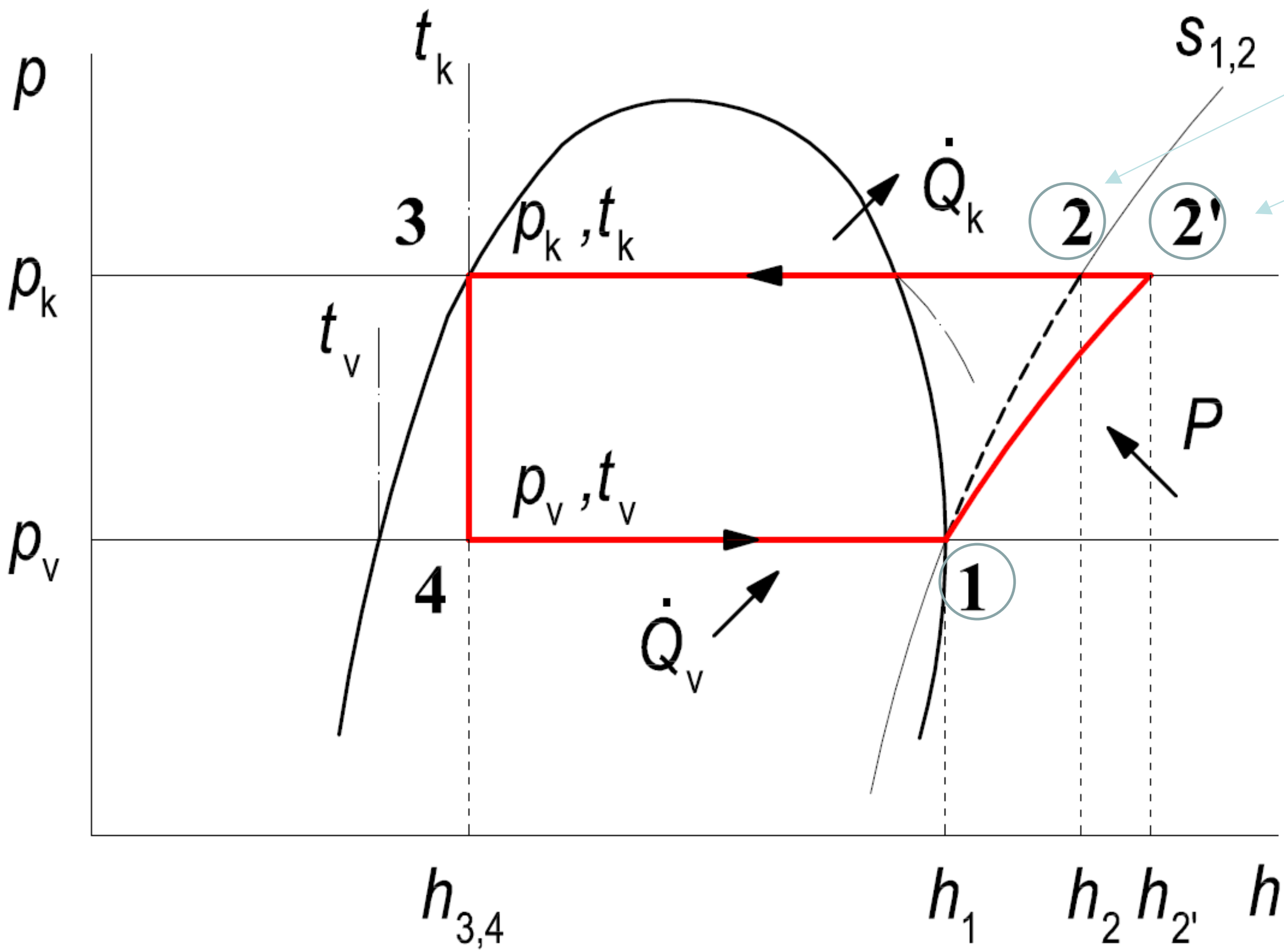
Real compression

- compression of vapour is not isoentropic (without losses)
- **polytropic compression**: increase of energy demand by real processes in compressor
- isoentropic efficiency

$$\eta_{ie} = \frac{h_2 - h_1}{h_{2'} - h_1} = \frac{\text{isoentropic work}}{\text{actual work}} = \frac{P_{ie}}{P_e} = \frac{\text{theoretical isoentropic input}}{\text{electric (indicated) input}}$$

Real compression

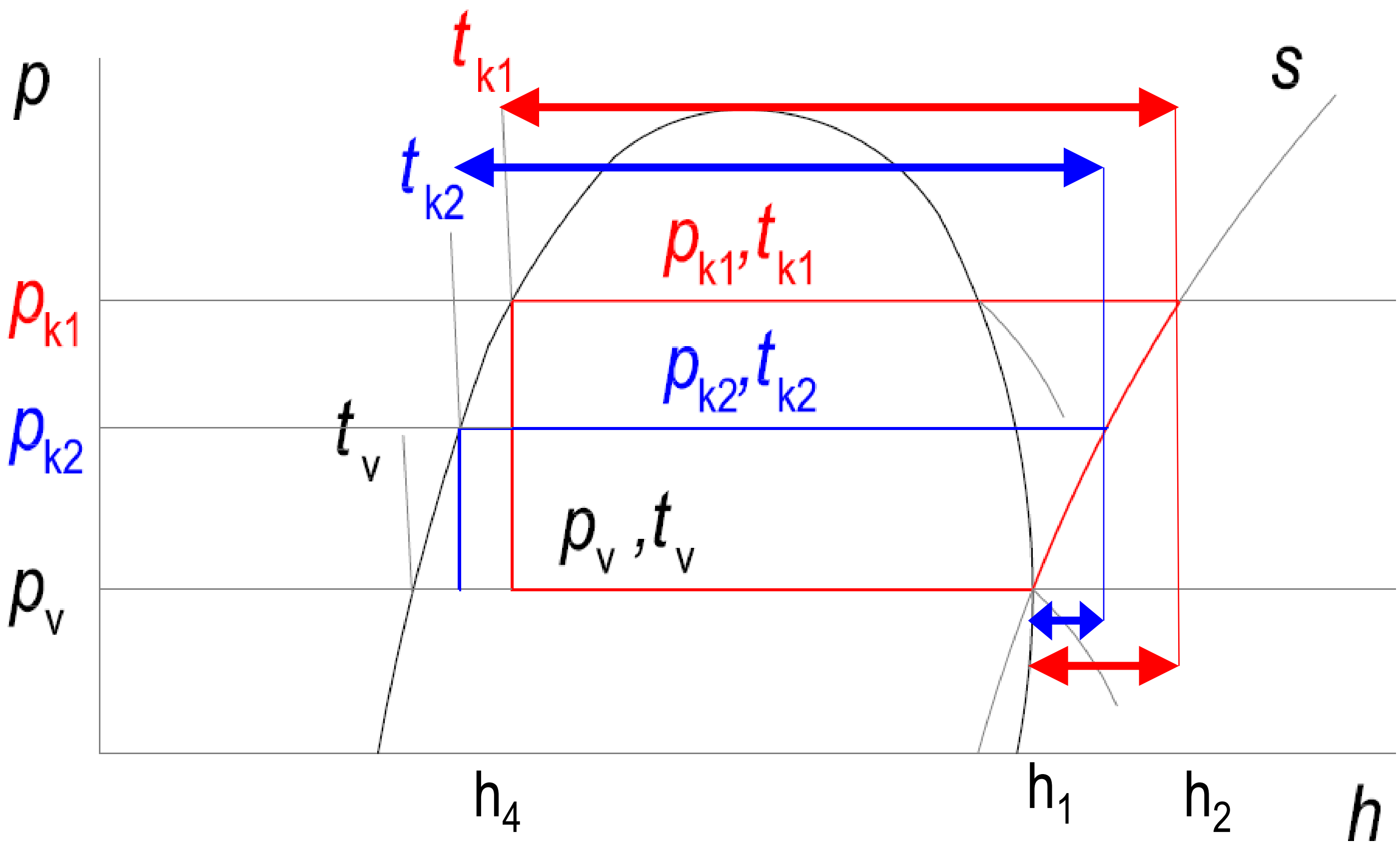
$$\eta_{ie} = \frac{h_2 - h_1}{h_{2'} - h_1}$$



COP dependent on temperatures

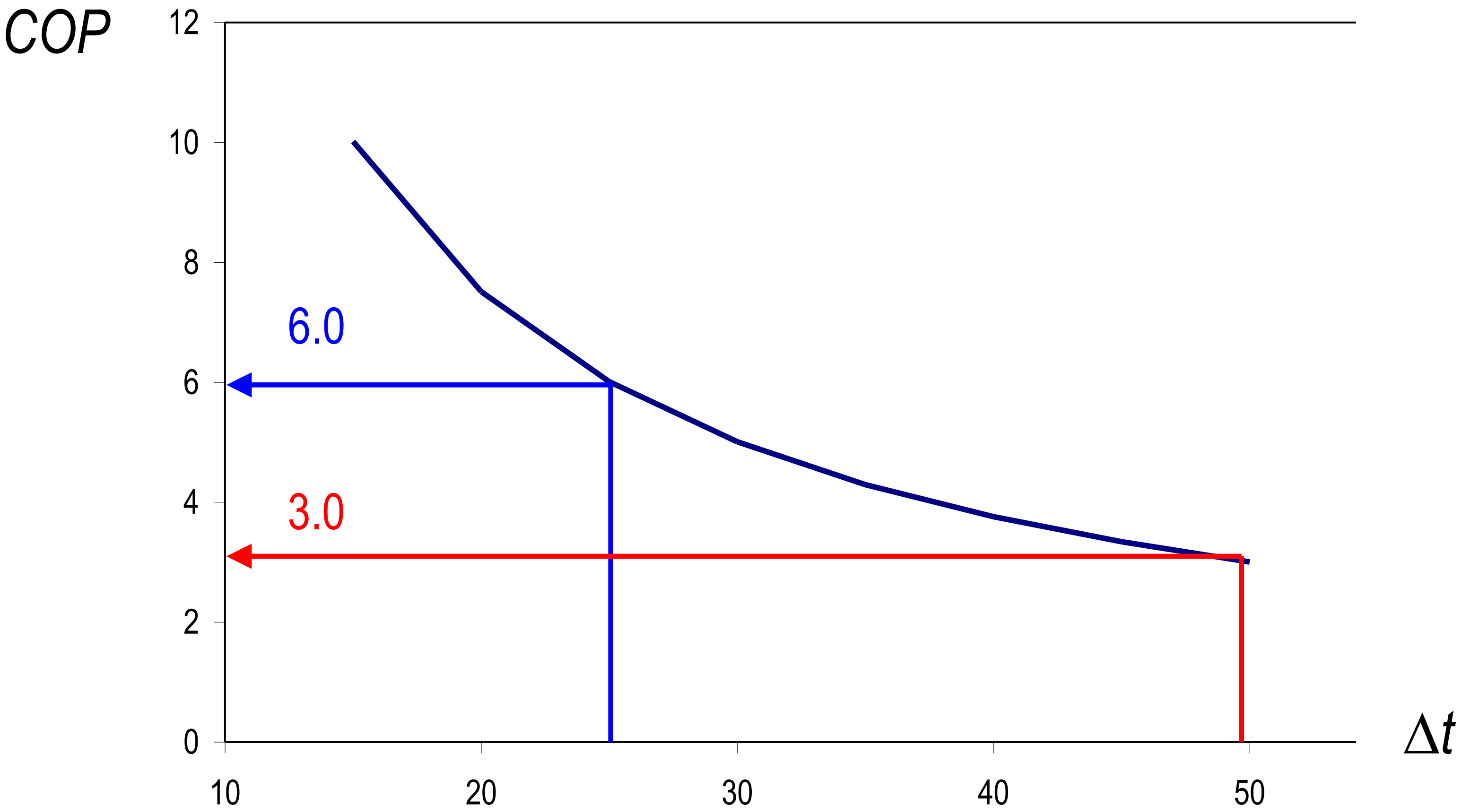
- **condensation temperature t_k** – given by the rejection system
  - space heating systems
  - hot water preparation
- **evaporation temperature t_v** – given by temperature of heat source (cooled environment)
  - ground, air, water, waste heat

COP dependent on temperatures



$$COP_R = \frac{\dot{Q}_k}{P_{ie}} = \frac{h_2 - h_4}{h_2 - h_1}$$

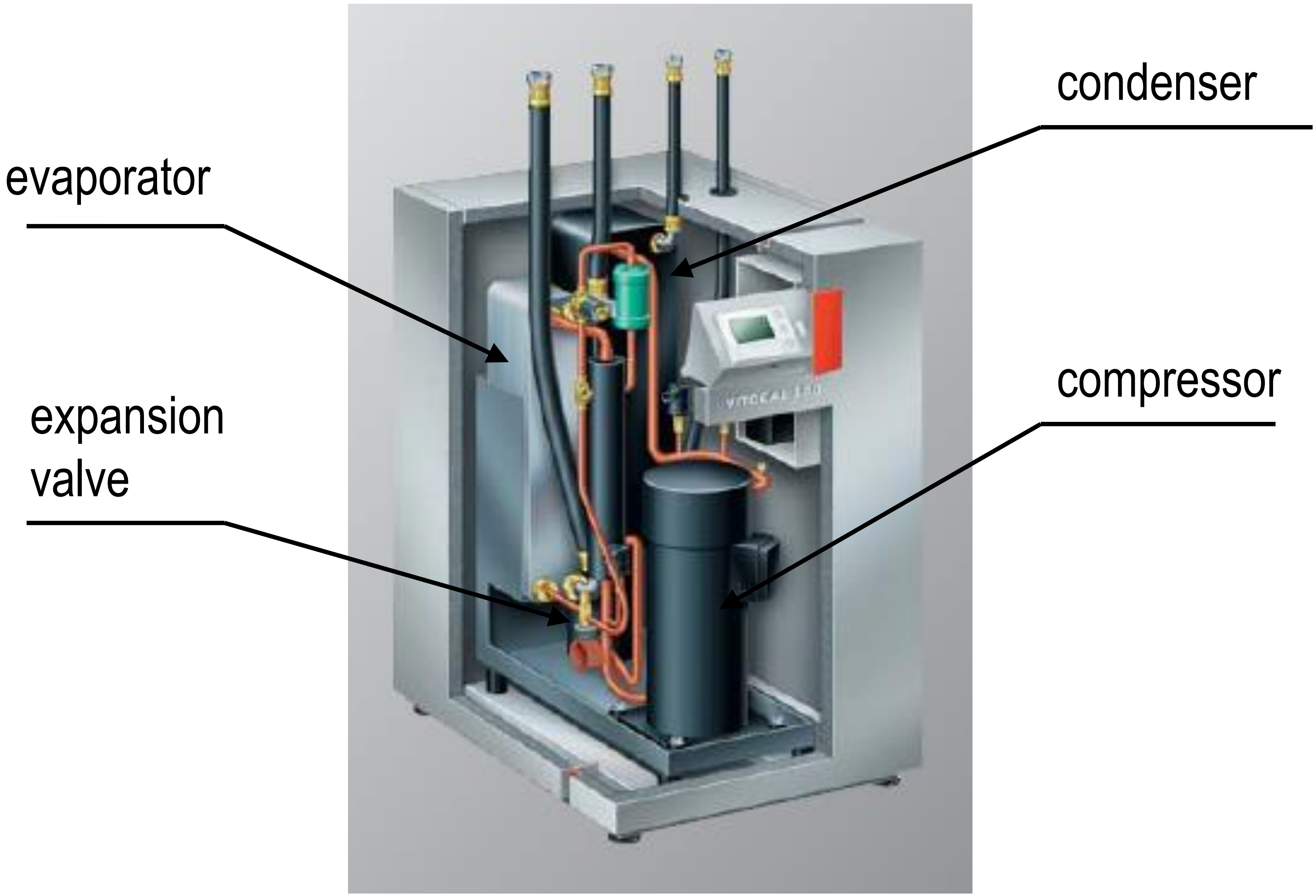
COP dependent on temperatures



COP dependent on

- type of **refrigerant**
- type of **compressor**
- sizing of **heat exchangers**

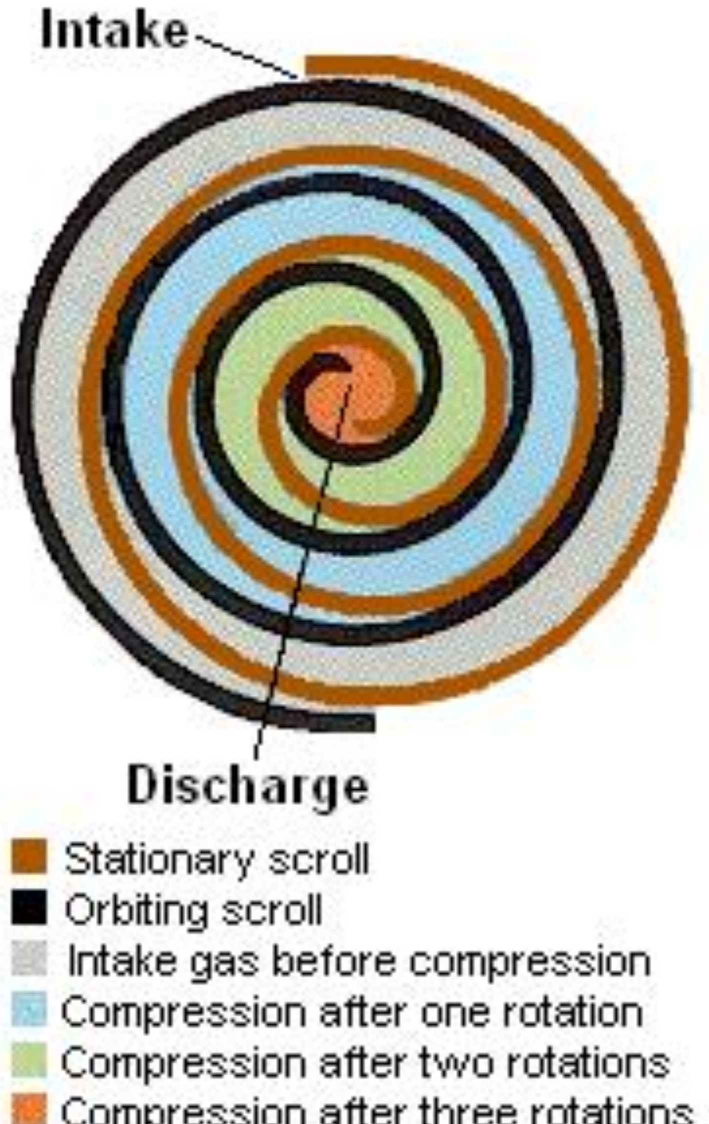
Heat pump components



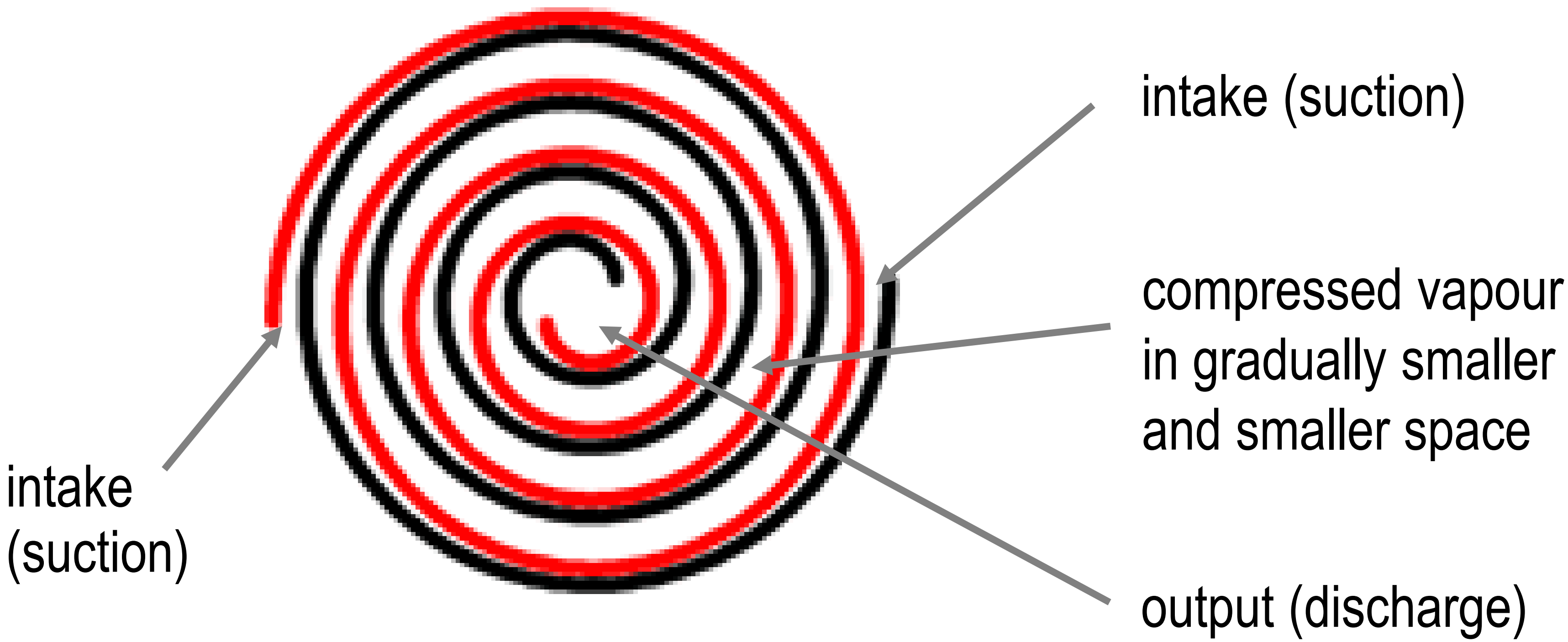
Compressor

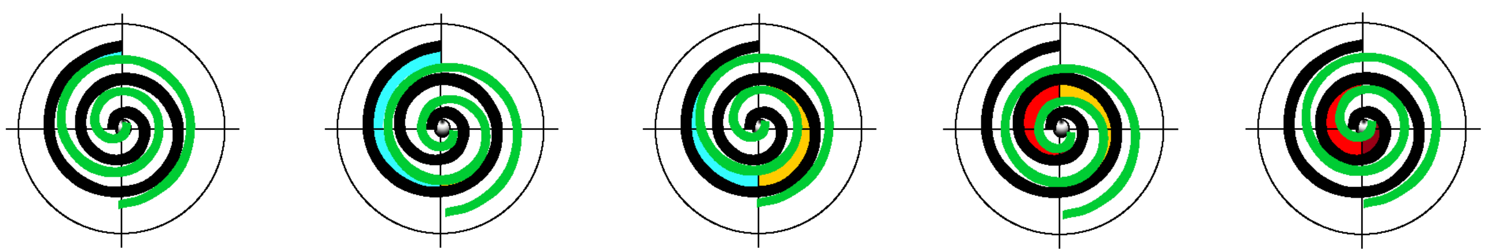
- **rotating spiral compressors (scroll)**
  - working cycle: suction-compression-discharge
  - motion of rotor spiral on stator spiral
  - continuos change of compression volume
  - suction at perimeter, discharge in center
  - durability, longlife, low vibration, low noise level

Spiral compressor - function



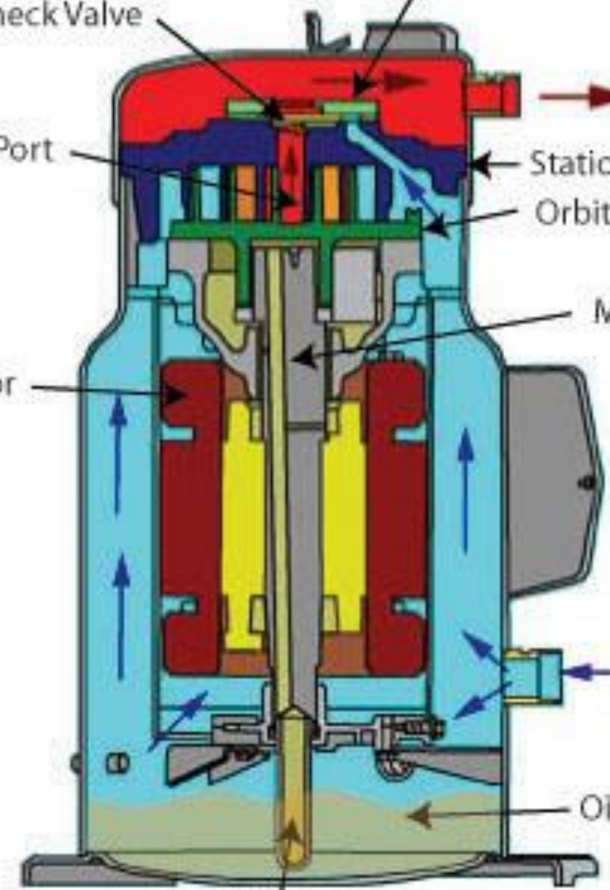
Spiral compressor - function





Compressor construction

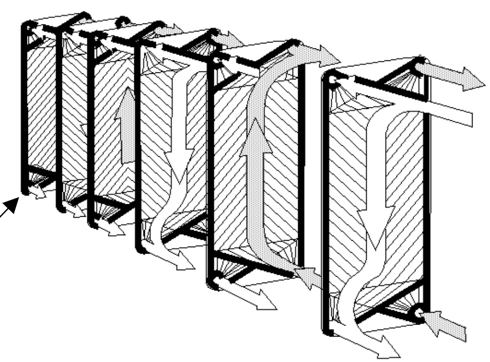
rotating spiral compressor

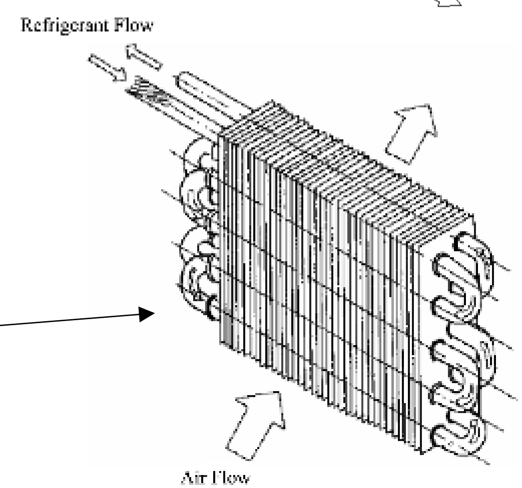


Evaporator

- extracts the heat from low temperature heat source (cooled environment) by evaporation of refrigerant at low pressure at temperature **lower** than output temperature of cooled fluid

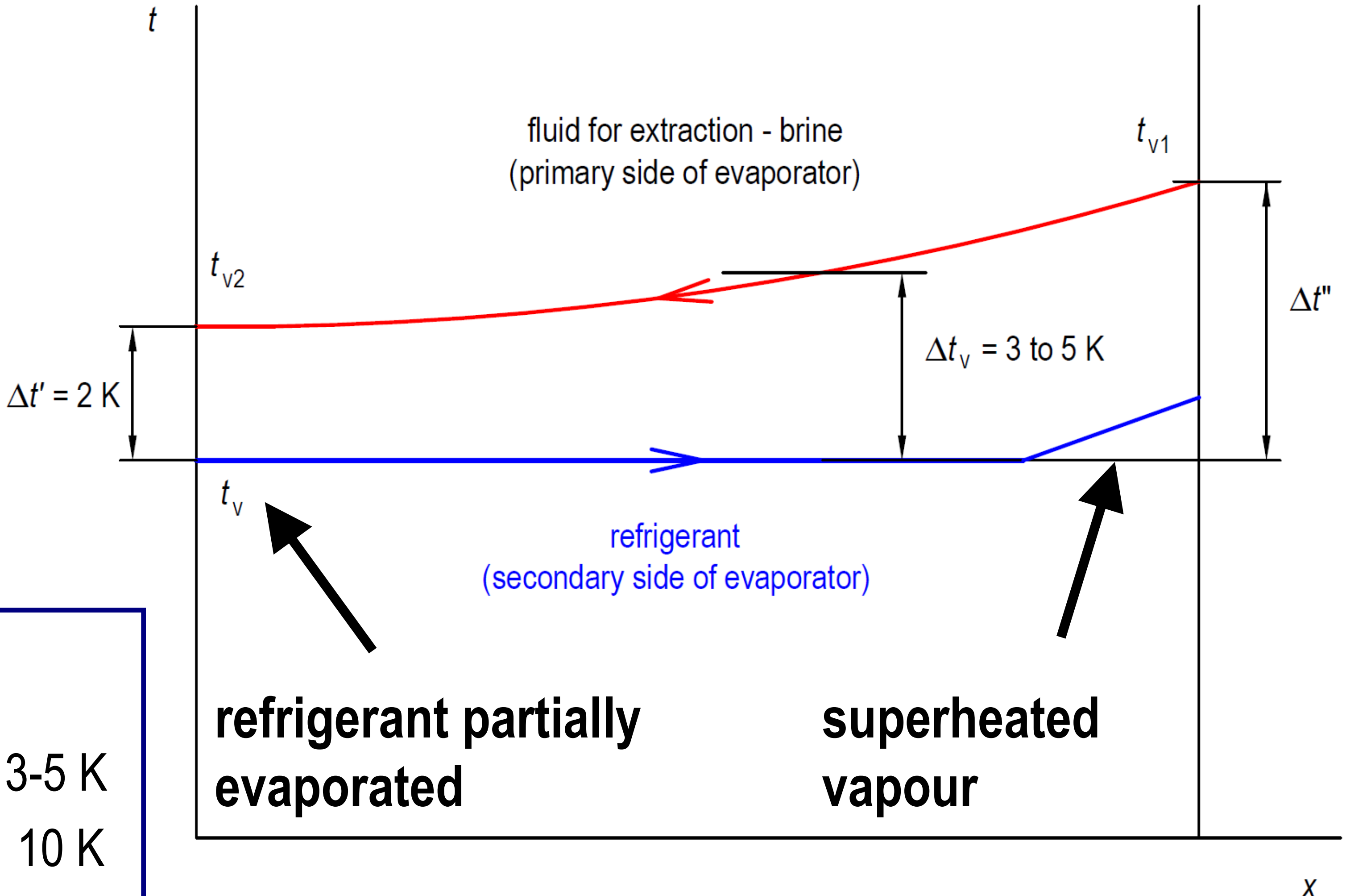
- **cooling of heat transfer fluid** :
  - brine (ground source)
  - water (water source)
  - air (air source)





- **heat exchangers**:
  - liquids: plate heat exchanger
  - air: pipe with fins heat exchanger

Evaporator



refrigerant partially evaporated

superheated vapour

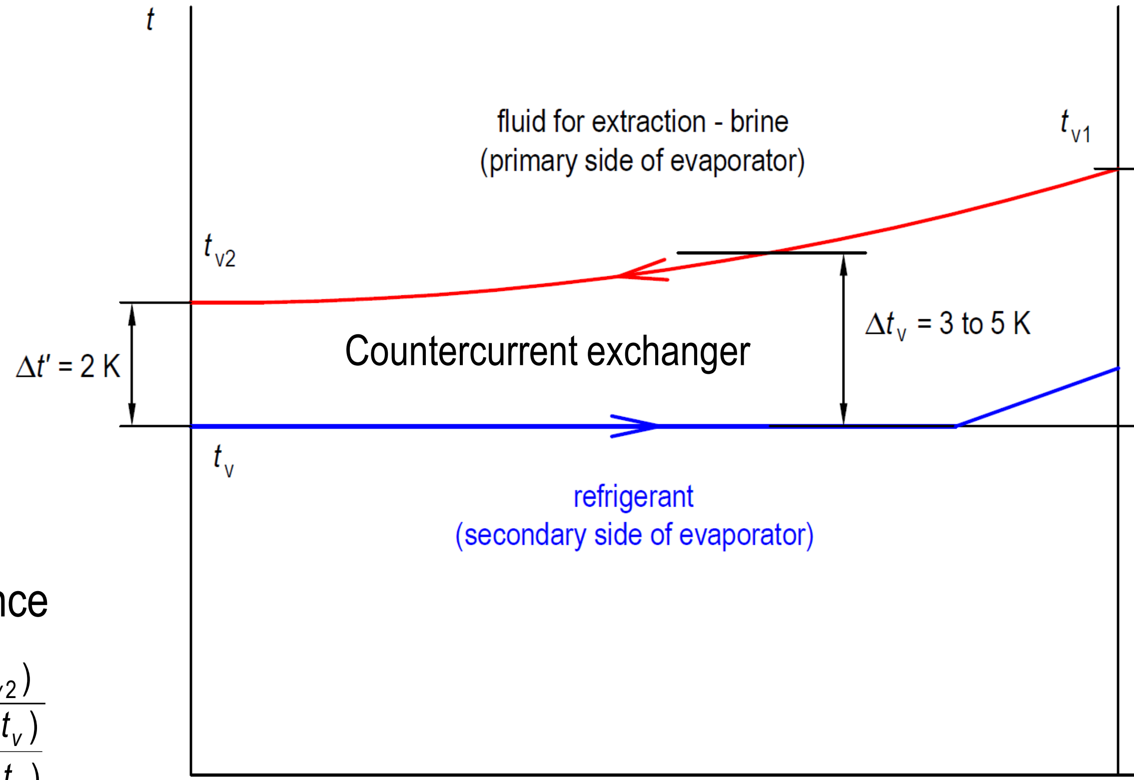
$t_{v1} - t_{v2}$	
liquids	3-5 K
air	10 K

Heat capacity Q_v of evaporator

$$\dot{Q}_v = U_v \cdot A \cdot \Delta t_v$$

Heat transfer coef. W/(m²K)

Area (m²)



(logarithmic) mean temperature difference

$$\Delta t_v = \frac{\Delta t'' - \Delta t'}{\ln \frac{\Delta t''}{\Delta t'}} = \frac{(t_{v1} - t_v) - (t_{v2} - t_v)}{\ln \frac{(t_{v1} - t_v)}{(t_{v2} - t_v)}} = \frac{(t_{v1} - t_{v2})}{\ln \frac{(t_{v1} - t_v)}{(t_{v2} - t_v)}}$$

After linearization

$$\Delta t_v = \frac{t_{v1} + t_{v2}}{2} - t_v$$

Condenser

- rejects the heat into heat transfer fluid (heated environment) by condensation of refrigerant at high pressure and temperature **higher** than output temperature of heated fluid

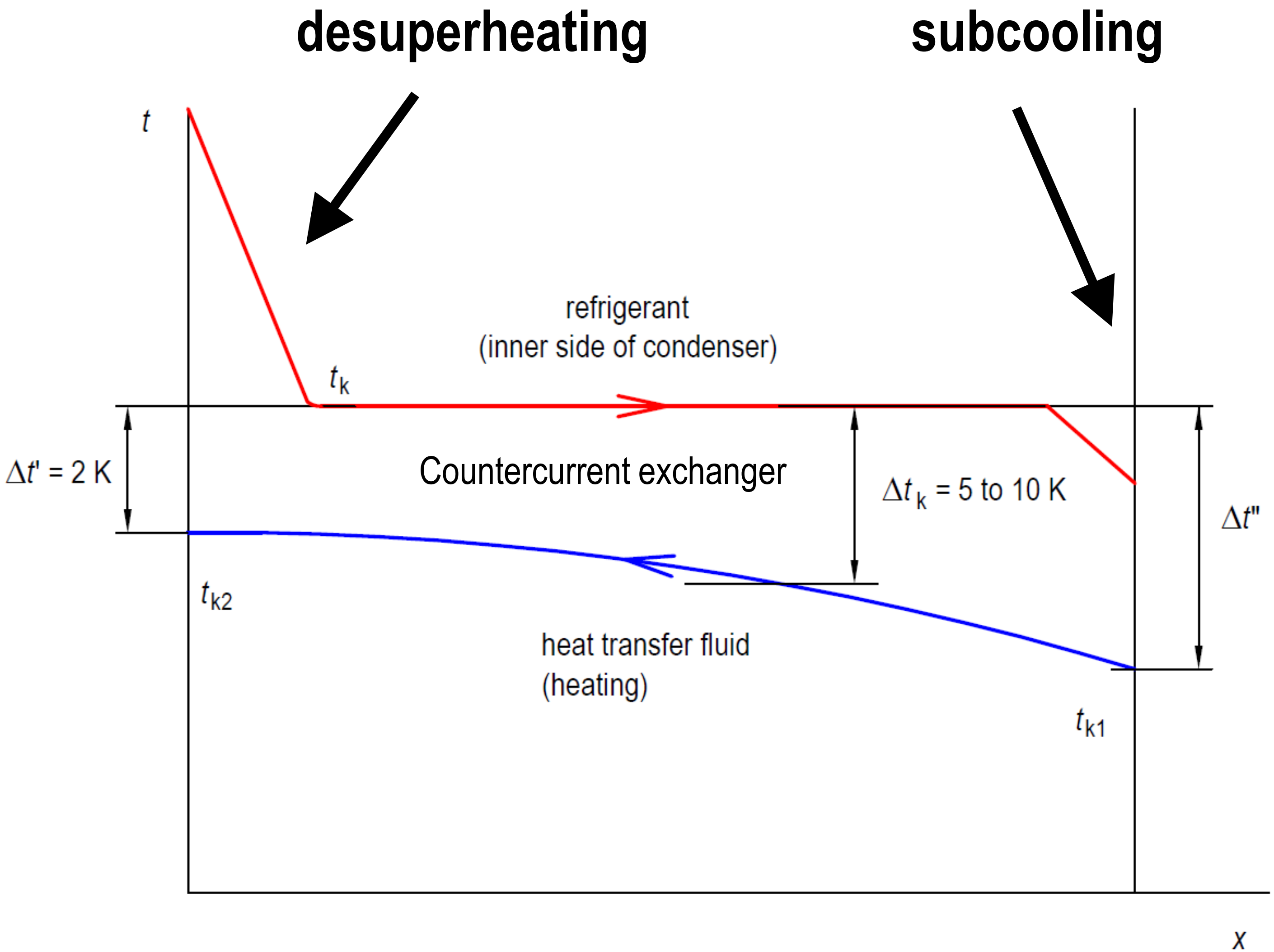
- **heating of heat transfer fluid**:
  - heating water (usual HP)
  - DHW (water heaters with HP)

- **heat exchangers**:
  - plate HX
  - pipe with fins (inside the tank)

Condenser

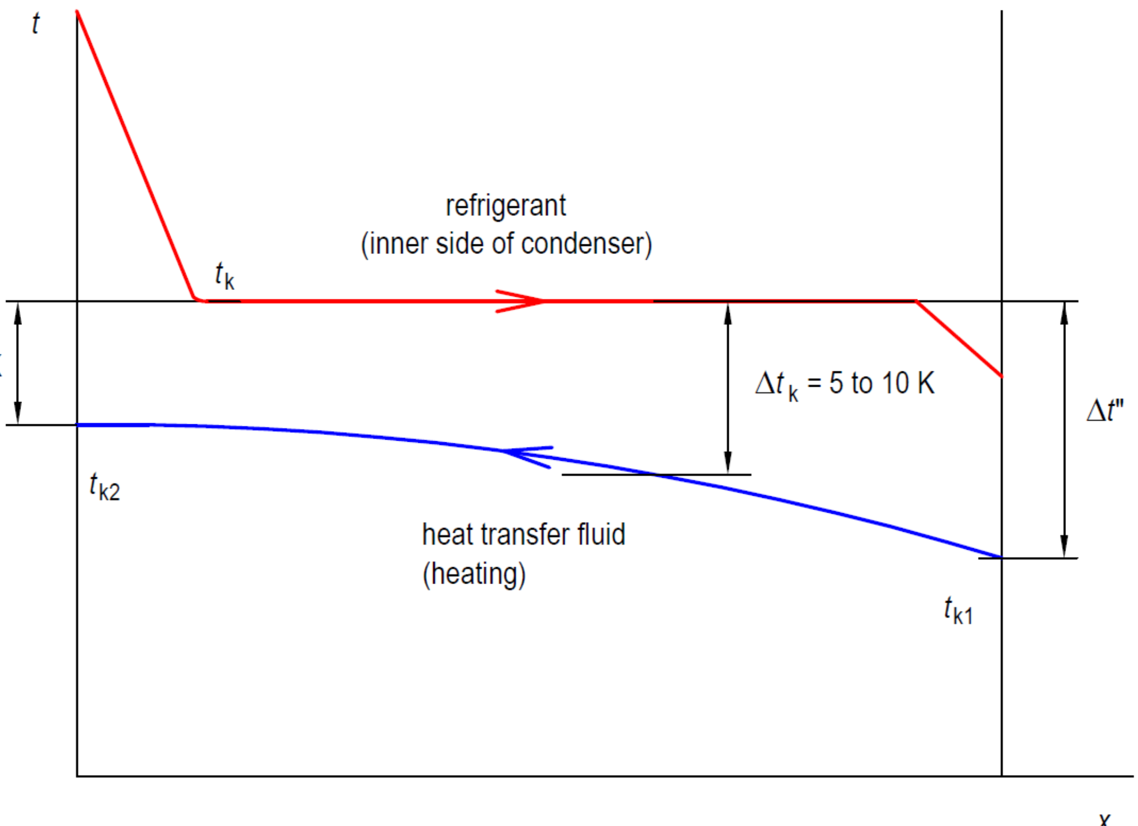
$t_{k1} - t_{k2}$ = 5-10 K

depends on heat capacity and flowrate



Heat capacity of condenser

$$\dot{Q}_k = U_k \cdot A \cdot \Delta t_k$$



$$\Delta t_k = \frac{\Delta t'' - \Delta t'}{\ln \frac{\Delta t''}{\Delta t'}} = \frac{(t_k - t_{k1}) - (t_k - t_{k2})}{\ln \frac{(t_k - t_{k1})}{(t_k - t_{k2})}} = \frac{(t_{k2} - t_{k1})}{\ln \frac{(t_k - t_{k1})}{(t_k - t_{k2})}}$$

linearization

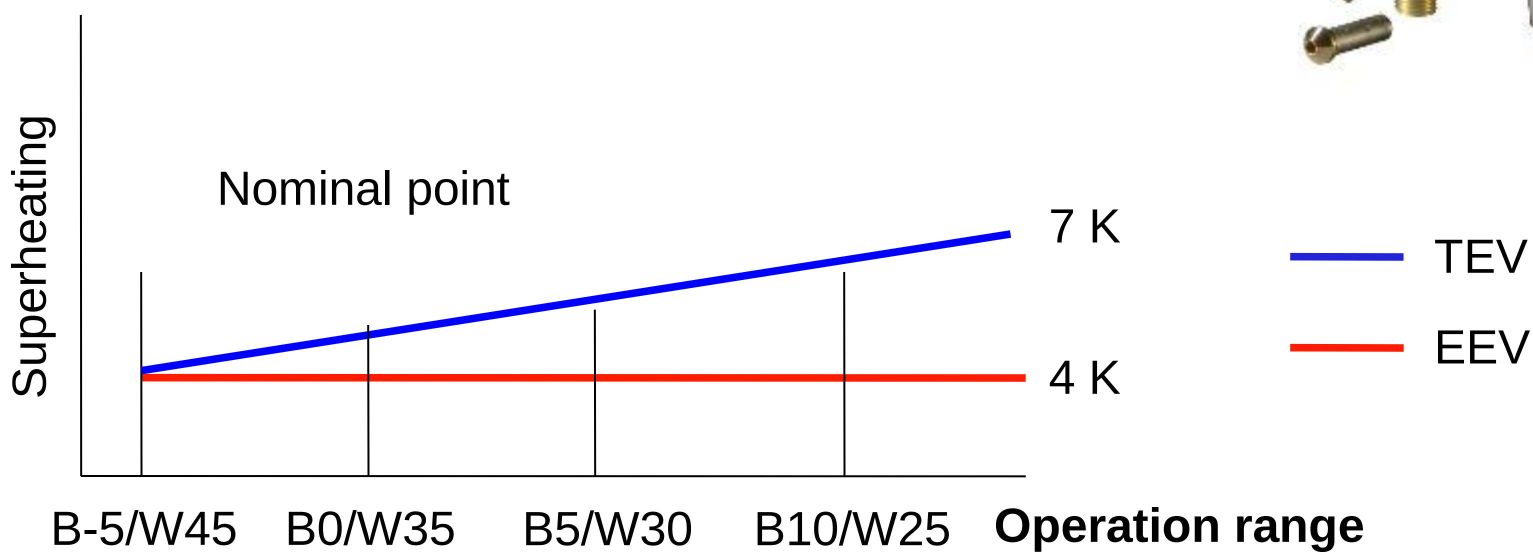
$$\Delta t_k = t_k - \frac{t_{k1} + t_{k2}}{2}$$

Expansion valve

- **keep pressure difference** between high-pressure and low-pressure side of the cycle
- **controls the refrigerant flowrate** from condenser to evaporator in dependence on output temperature from evaporator
- **keep refrigerant superheating** at evaporator output $\Delta t_s > 5$ K
- refrigerant passing through EV is partially evaporated by expansion and the damp vapour (mixture of vapour and liquid droplets) enters into the evaporator

Expansion valve

- expansion valve
  - capillary – for constant operation conditions (refrigerator)
  - termostatic expansion valve (TEV)
  - electronic expansion valve (EEV)
    accurate control of superheating



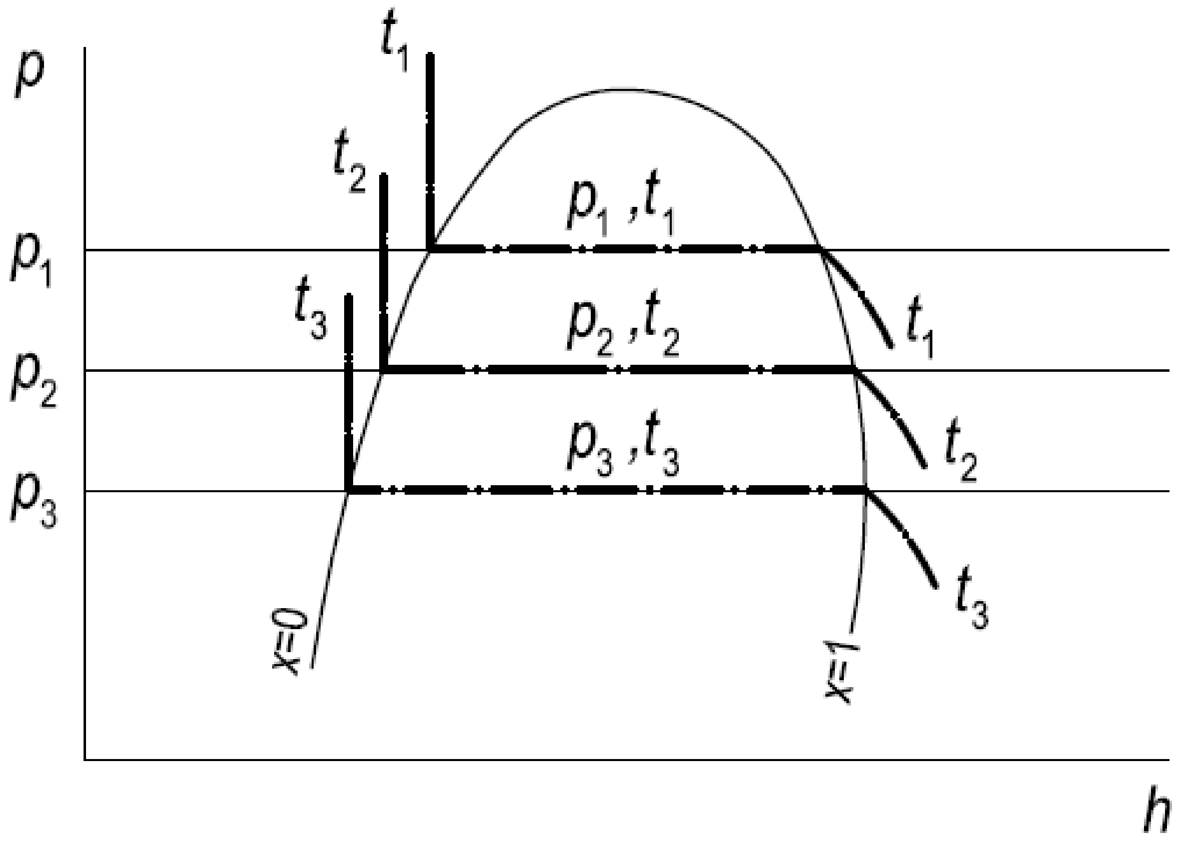
(B5/W30 = brine 5°C, water 30°C)

Refrigerants

- **azeotropic**
  - perform as pure liquids, vapour content is not changing at boiling point (phase change)
  - R22, R290, azeotropic mixture: R502 or R507
- **zeotropic**
  - mixtures usually from 2 to 4 refrigerants
  - **temperature glide** – nonuniform evaporation of refrigerant components, difference in evaporation temperatures of components at constant pressure. Evaporation: temperature moderately increases.
  - e.g. R407a

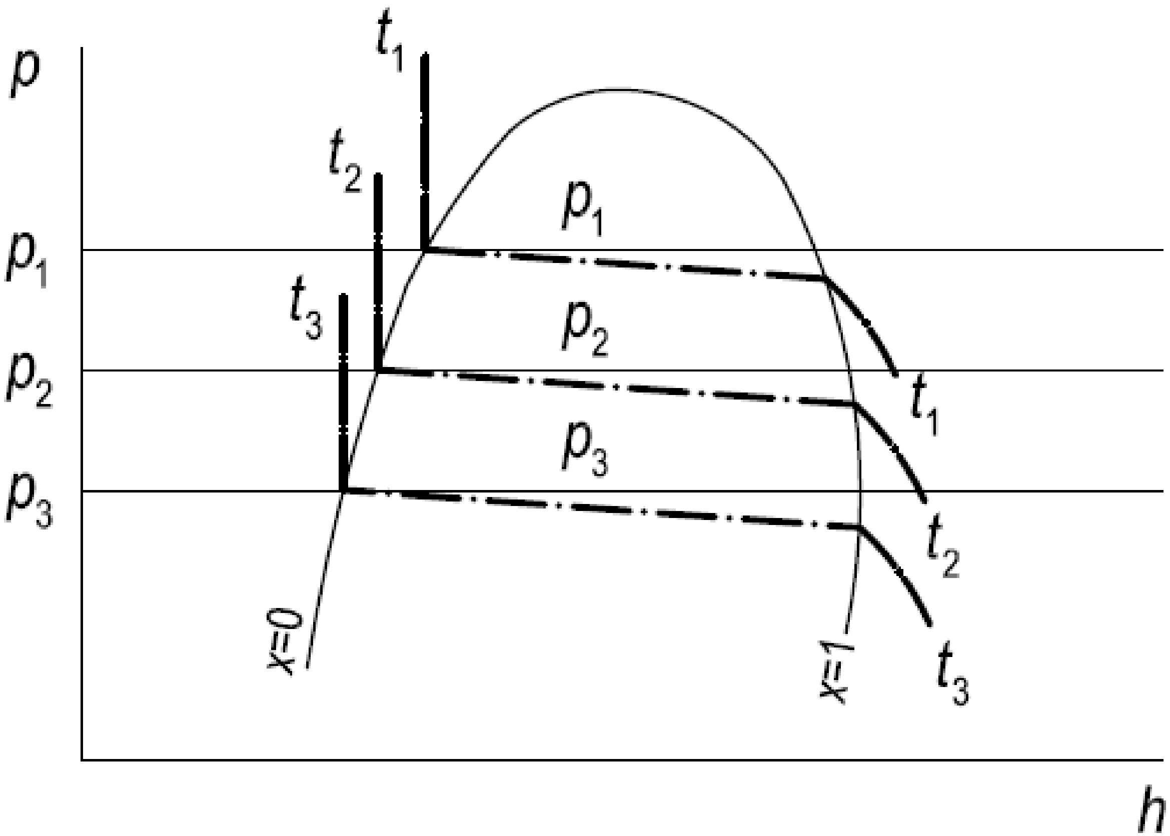
Refrigerants

azeotropic



Refrigerants

zeotropic (temperature glide)



Refrigerants

- **CFC**
  - fully halogenated hydrocarbons and mixtures, i.e. all atoms of **H** in molecule are replaced by halogenid atoms (**Cl**, **F**, **Br**)
  - „hard freons“
  - R11, R12, R13, R113, R114, R115, R502, R503 etc.

forbiden
no servis

- **HCFC**
  - chloro-fluorinated hydrocarbons, atoms of **H** in molecules
  - „soft freons“
  - R21, R22, R141b, R142b, R123, R124

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Refrigerants

- **HFC**
  - no chlor atoms in molecule, only fluor
  - R152a, R125, R407c, **R134a, R410c, R32**
- **HFO (hydro-fluor-olefin)**

  also composed of hydrogen, fluorine and carbon atoms, but contain at least one double bond between carbon atoms
  - **R1234yf**
- **HC** natural hydrocarbons and mixtures
  - amonnia, propan (R290)
  - no halogenids, flammable, toxic
- **CO_2 (R744)** **???? (R718)**

expensive, gradually replaced

longterm alternative

green refrigerants

preferred