Compendium DES

July 2016, CARN

Contents

1. Energy balance

Using Energy Balance for analyzing energy systems

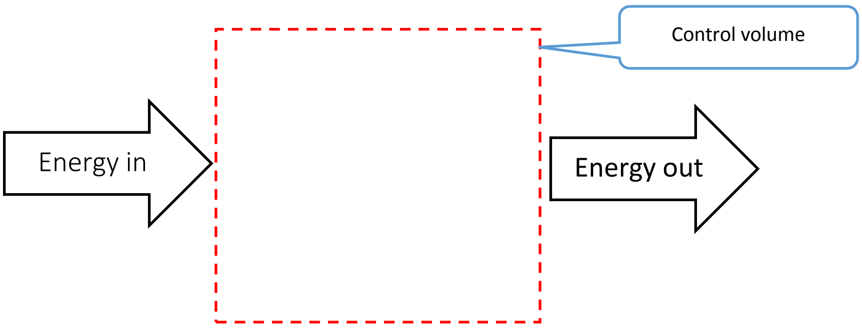
Energy Balance definition

Energy cannot be created or destroyed, only modified in form (First Law of Thermodynamics).

For a defined control volume there is a balance between supplied and extracted energy if the system is steady state.

If there is no balance energy will be added to (stored) or removed from the control volume, and the system is non-steady (dynamic).

A control volume can be defined for a total system (eg. a cold storage room, a refrigeration system/heat pump) or for a component (eg. a heat exchanger, a compressor, an expansion valve, a tank).



Figur 1

Steady state, steady flow

Mass balance: $\Sigma q_{m\_in} = \Sigma q_{m\_out}$ [kg/s]

Energy balance: $\Sigma \dot{E}_{in} = \Sigma \dot{E}_{out}$ [kW]

$$\overbrace{\Sigma\Phi_{in}}^{\text{Heat}} + \overbrace{\Sigma P_{in}}^{\text{Work or electricity}} = \Sigma\Phi_{out} + \Sigma P_{out} \quad \text{[kW]}$$

Heat:

$$\Sigma\Phi = \overbrace{\Sigma\Phi}^{\text{Heat to or from environment}} + \overbrace{\Sigma(q_m\, h)}^{\text{Mass flow with latent heat change}} + \overbrace{\Sigma\left(q_m\, c_p\, \Delta t\right)}^{\text{Mass flow with sensible heat change}} \quad \text{[kW]}$$

Methodology

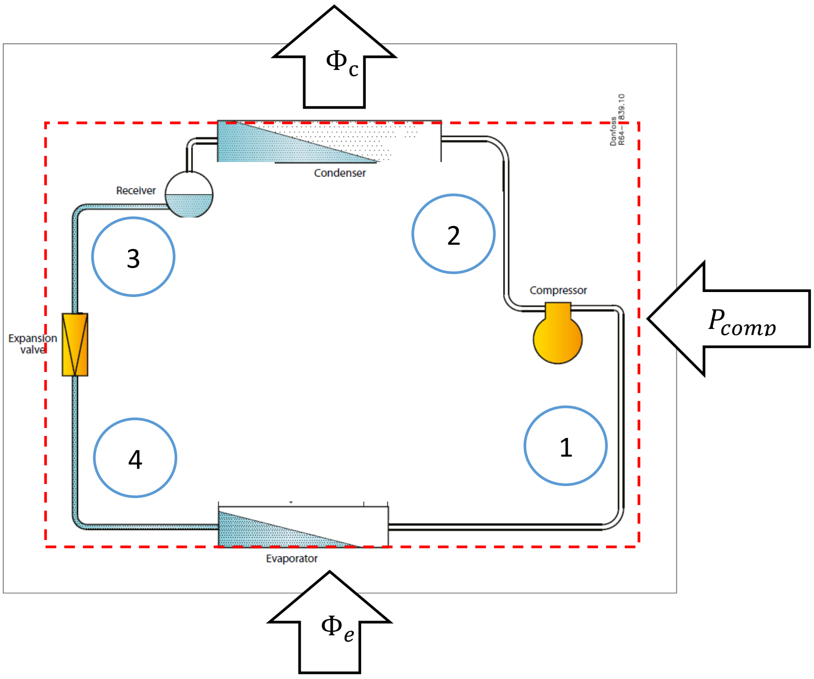
- Draw a simple model of the system using standard symbols

- Add numbers to identify the different parts of the process
- Add know information on eg. massflow, temperatures, pressure to the drawing
- Draw Control Volumes for the total system and parts of the system to be analyzed.
- Find enthalpies for latent processes: Refrigerants (log P h-diagram), air with condensation or humidification (hx-diagram). Find c_p values for sensible processes: Water, dry air and other substances (tables)
- Calculate energy flow in and out of each control volume using energy balance equations.

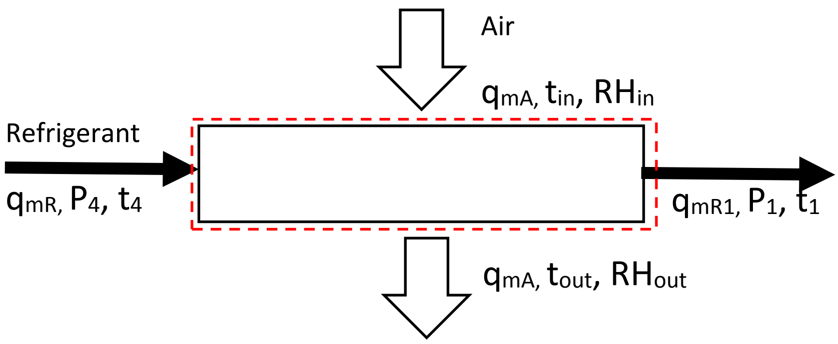
Energy balance examples

Overall system, refrigeration or heat pump

$\Phi_{evaporator} + P_{compressor} = \Phi_{condenser}$ [kW]



Evaporator, air cooled



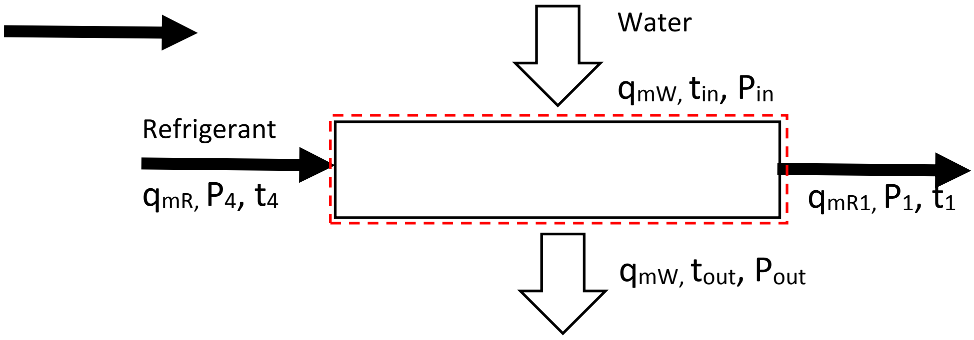
$q_{mR}\, h_4 + q_{mA}\, h_{in} = q_{mR}\, h_2 + q_{mA}\, h_{out}$ [kW]

or

$q_{mR}\, (h_{R1} - h_{R2}) + q_{mA}\, (h_{A1} - h_{A2}) = 0$ [kW]

Specific enthalpy is found in tables

Evaporator, water cooled

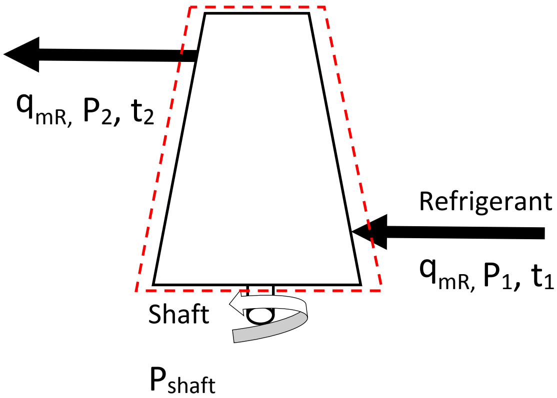


$$q_{mR}\, h_4 + q_{mW}\, c_{p\_in}\, t_{in} = q_{mR}\, h_1 + q_{mR}\, c_{p\_out}\, t_{out} \quad \text{[kW]}$$

or

$$q_{mR}\,(h_4 - h_1) + q_{mW}\,(c_{pWin}\, t_{Win} - c_{pWin}\, t_{Wout}) = 0 \quad \text{[kW]}$$

Compressor



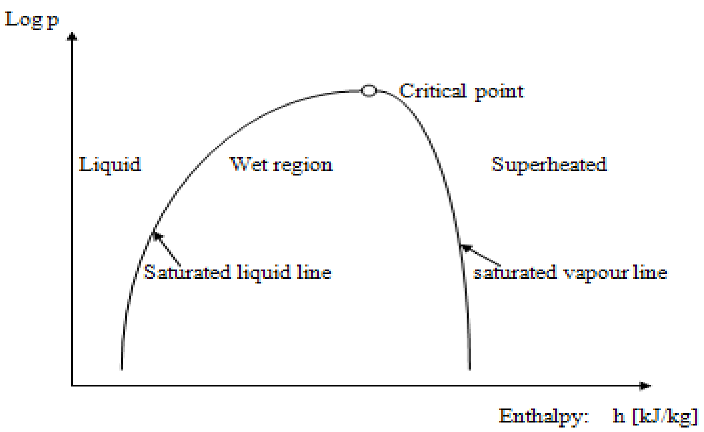
$$q_{mR}\, h_1 + P_{shaft} = q_{mR}\, h_2 \text{ [kW]}$$

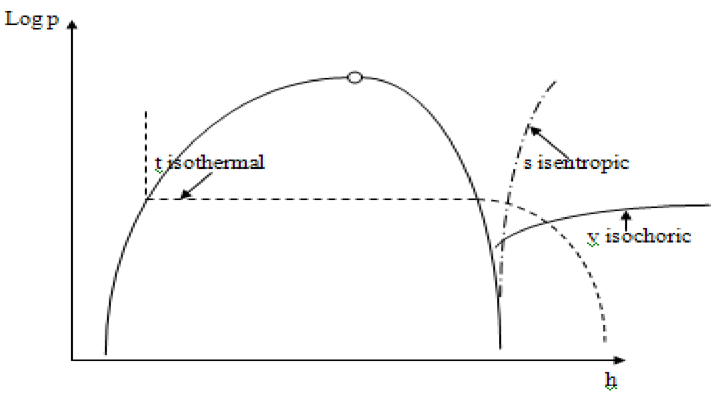
2. Refrigeration cycle

Refrigeration cycle and log p,h diagram

The diagram shows different phases: Liquid, saturated liquid (transition from liquid to wet region), wet region (mixture between vapour and liquid), saturated vapour (transition between wet region and superheated) and superheated phase. Furthermore, the following curves are depicted in the diagram: Isothermal, isochoric and isentropic curves.

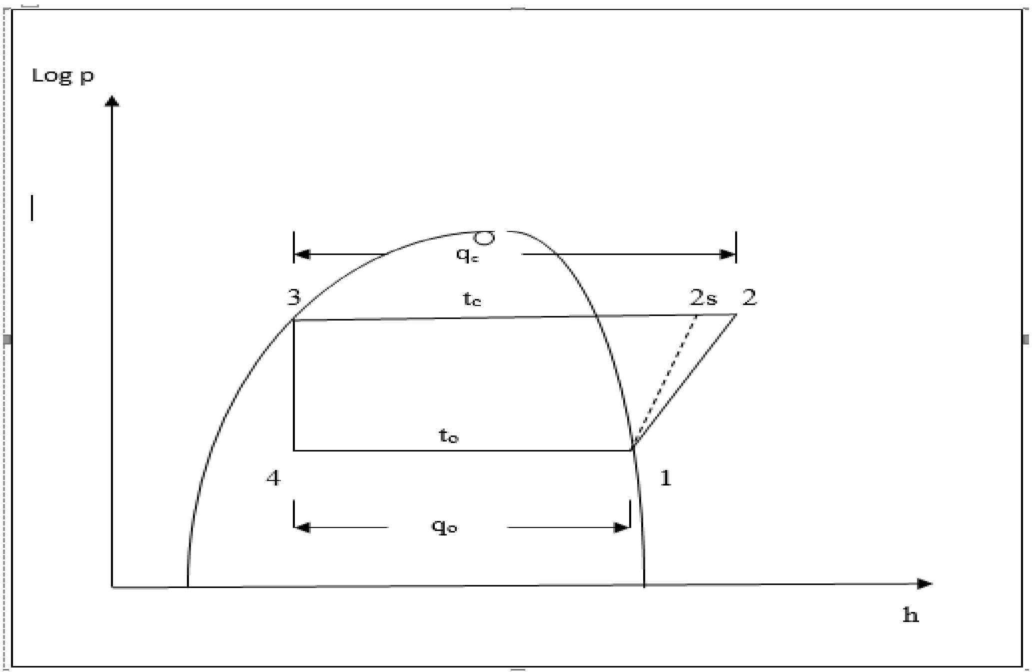
Phases in log p, h diagram:





A log p, h diagram for R 404a is shown on next page.

Single stage refrigeration cycle in log P,h diagram:



To draw the refrigeration process in a log P h diagram the following steps can be used.

Evaporaration process (4-1)

The evaporation pressure is determined by the evaporation temperature and the chosen refrigerant. Because of the evaporation pressure and temperature will be constant during the process.

The evaporation temperature t_o depends on the purpose of use. If the evaporator is to cool the air in a cold store to a given storage temperature t_s, the evaporation temperature t_o must be lower than t_s in order for the heat from the cold store to flow into the evaporator. The evaporation temperature t_o is normally 5 - 10°C lower than t_s.

The quality x is connected to the wet area and expresses the ratio between the vapour and liquid fractions. If the suction state 1 is in the wet region then x_1 is less than 1. This will not be acceptable since the compression of wet vapour will damage the compressor. Fluid droplets that are sucked into the compressor will evaporate promptly and be highly damaging to the compressor.

The temperature t_1 must hence be equal to or larger than (=superheated) the temperature in saturated vapour state.

Condensation process (1-2)

The condensing temperature t_c is determined by the fluid to which the heat from the condenser is transferred. This heat is often given off to outside air or cooling water e.g. seawater. The condensing temperature t_c is normally 10 - 15°C higher than the outside air/cooling water temperature.

The quality x_3 must be equal to 0 and the temperature t_3 must be equal to or smaller than (=subcooled) the temperature in saturated liquid state.

The temperature t3 after the condenser should be the same or lower than the condensation temperature, meaning that the refrigerant is liquid to avoid flash gas in the expansion valve. If state 3 is on the liquid side of the saturation curve (=subcooled).

Expansion process (3-4)

During the expansion process there will be no transfer of energy to or from the environment so h4=h3.

Compression process (1-2)

The temperature t_2 after the compressor is determined by the efficiency of the compressor and cooling of this. The ideal process is isentropic, which is defined as adiabatic (Q_{12}= 0) and frictionless (W_{diss12} = 0). The actual temperature t_2 will be higher than the theoretical t_{2s} since frictional heat cannot be avoided. Immediately after the compressor, the gas is strongly superheated. The superheated gas is cooled down in the condenser from t_2 to the condensing temperature $t_{c.}$

The calculation of the actual enthalpy h_2 after the compressor depends on its isentropic efficiency $\eta_{s.}$

The ideal compression process will have constant entropy, meaning that the process vill follow the isentropic curve and the intersection with the condensation pressure will be stage 2s. h_{2s} is enthalpy at discharge valve for the <u>ideal</u> compression process.

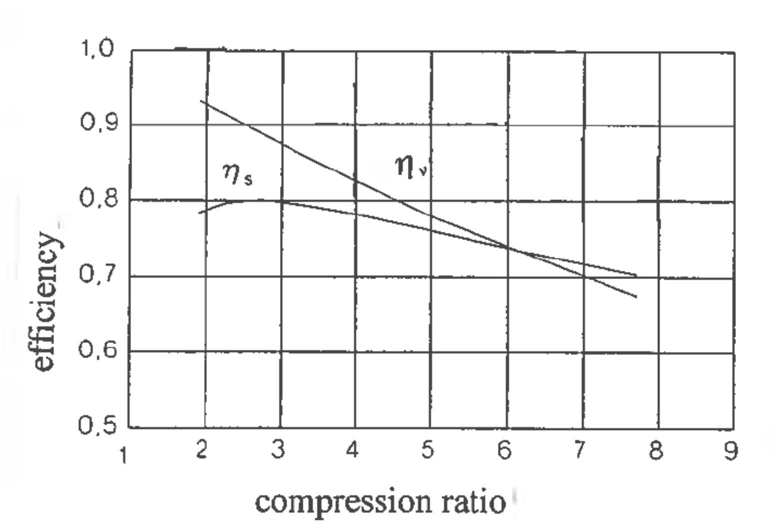
For a real process and if the compressor is <u>uncooled</u>, the friction losses are transferred to the refrigerant gas. In this case η_s is defined only by the enthalpies:

$$h_2 = \frac{h_{2s} - h_1}{\eta_s} + h_1 \quad \text{[kJ/kg]}$$

h_2 is enthalpy at discharge valve for the <u>real</u> compression process. h_1 is enthalpy at compressor inlet suction valve.

η_s is the compressors isentropic efficiency. It is depending on compressor type and pressure ratio between condensing pressure and evaporation pressure.

The diagram below shows typical values for $\eta_{.}$ (η_v is volumetric efficiency).



3. Coefficient of Performance - COP

COP definition

As a measure of the efficiency of a refrigeration system or a heat pump, Coefficient Of Performance = COP is used. This factor cannot be compared to an efficiency as COP is normally greater than 1. Of course, this relationship does not mean that a refrigeration system or a heat pump work as perpetual motion machines. The COP can be formulated as useful energy divided by supplied energy.

$$COP = \frac{Useful\ energy}{Paid\ energy}$$

COP calculation

Refrigeration system

$$COP_{cool} = \frac{q_o}{w_i}$$

If the efficiency of the electrical motor, fans and pump is included, we get:

$$COP_{cool} = \frac{\Phi_o}{P_{el}}$$

Heat pump

$$COP_{HP} = \frac{q_c}{w_i}$$

If the efficiency of the electrical motor, fans and pumps is included, we get:

$$COP_{HP} = \frac{\Phi_c}{P_{el}}$$

COP factors provides a snapshot of the operating conditions at certain temperatures and loads. It is evident that a high COP value yields the lowest operational expenses.

Seasonal performance factor -SPF

The COP factor is measured by using specific test data and often describe performance for worst case based on design data.

Since the outdoor temperatures change during the year, the COP for the plant will also vary according to the season. To achieve an exact calculation of the operational expenses, a seasonal performance factor SPF must be used instead.

To get an accurate picture of how efficient the heat pump functions over longer time, we can calculate the seasonal COP of the heat pump, where the whole year, summer and winter is taken into account. Seasonal performance factor COP_{season} is an average for the whole year. The factor is called SPF and is defined by:

$$SPF = COP_{season} = \frac{Annual\ energy\ production\ (J\ or\ kWh)}{Annual\ electricity\ consumption\ (J\ or\ kWh)}$$

Often annual SFP is found using monthly values for energy demand and COP.

4. Mass flow and capacities

Refrigerant mass flow

Refrigerant mass flow rate q_{mR} is calculated from evaporator or condenser capacity.

For a cooling plant a given cooling capacity Φ_o is needed:

$$q_{mR} = \frac{\Phi_o}{h_1 - h_4} \quad \text{[kg/s]}$$

For a heat pump a given condenser capacity Φ_c is needed:

$$q_{mR} = \frac{\Phi_c}{h_2 - h_3} \quad \text{[kg/s]}$$

Cooling capacity

The cooling capacity is describing the cooling performance of the refrigeration systems. The cooling capacity is the same as the evaporator capacity.

$\Phi_o = q_{mR} (h_1 - h_4) = q_{mR}\, q_o$ [W]

Condenser capacity

The condenser capacity is describing the heat from the condenser to the environment.

$\Phi_c = q_{mR} (h_2 - h_3) = q_{mR}\, q_c$ [W]

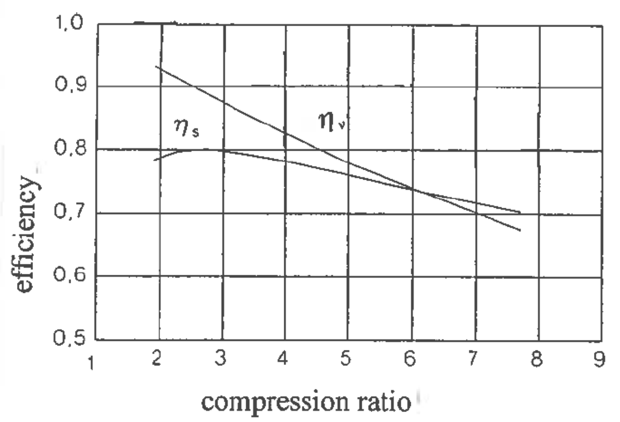
Compressor capacity and volumetric efficiency

The compressors capacity is characterized by the real volume flow rate q_{v1} in the suction inlet valve. The volume flow rate in the suction pipe depends on the compressors teortical swept volume $q_{v,s}$ and the pressure ratio between evaporation pressure and condenser pressure. The real volume flow rate q_{v1} will always be lower than the theoretical q_v because of e.g. recompression of refrigerant vapour from cylinder "top space", pressure drop in valves, leak from high pressure side to low pressure side, heating of refrigerant in suction inlet.

The volumetric efficiency η_v is defined as the ratio between actual volume flow rate and the theoretically swept volume:

$$\eta_v = \frac{q_{v1}}{q_{v,s}} = \frac{q_{mR}\, v_1}{q_{v,s}}$$

The diagram below shows typical values for η_v.



For a compressor with a given $q_{v,s}$ and η_v, the capacity is given by:

$$\Phi_o = q_{mR}\, q_o = \frac{q_{v,s}\eta_v}{v_1} q_o = q_{v,s}\eta_v q_{o,v} \quad \text{[W]}$$

where $q_{o,v}$ is the specific volumetric refrigerating capacity

$$q_{o,v} = \frac{q_o}{v_1} \quad \text{[kJ/m}^3\text{]}$$

5. Details in the refrigeration cycle

Superheating

The state 1 at the compressor suction inlet can be saturated vapour state or superheated. The actual state is linked to the regulation of the evaporator. If $t_1 = t_o$ the vapour at the compressor inlet is saturated. If $t_1 > t_o$ the vapour to the compressor is superheated. The superheating is controlled by the expansion valve.

We are often dealing with a superheating in the range of Δt_{sh} = 2 - 8ºC.

The main purpose of superheating is to avoid liquid in the compressor.

Subcooling

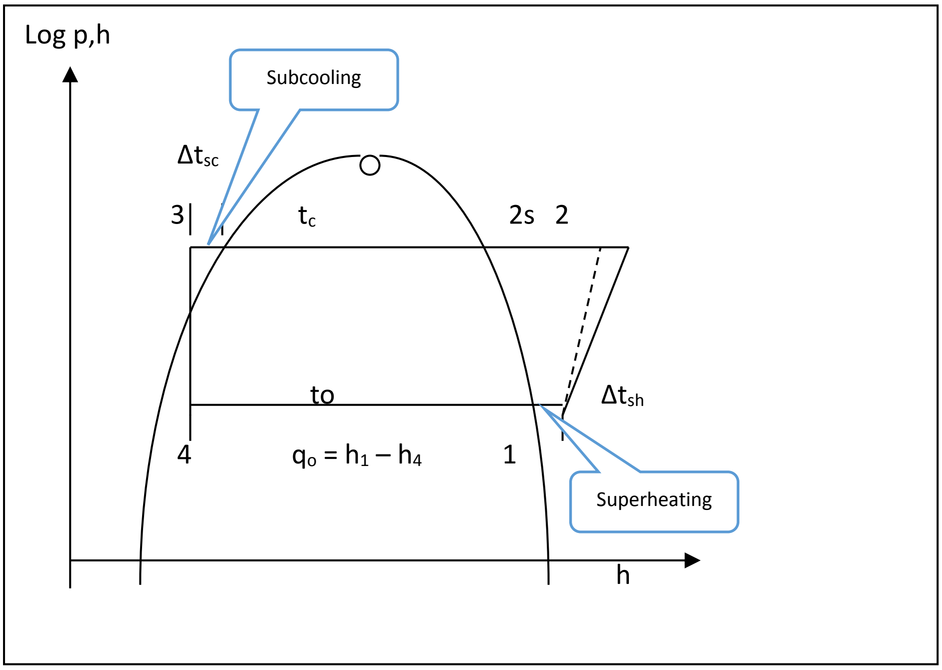
The state 3 right after the condenser must be saturated liquid or subcooled. A liquid is said to be subcooled, when its temperature t_3 is lower than the condensing temperature t_c that corresponds to the condensing pressure.

The subcooling $\Delta t_{sc} = t_c - t_3$ occurs by cooling in the bottom part of the condenser where there is liquid. Furthermore, the subcooling can also occur in pipes- and the receiver after the condenser- it is normal practice that these pipes are not insulated.

Subcooling avoids flash gas in the expansion valve. If the refrigerant just before the expansion valve enters the wet region state, gas bubbles in the liquid are formed- this is called flash gas. Flash gas can lead to disruptions in the operation, as the expansion valve gets a lower capacity.

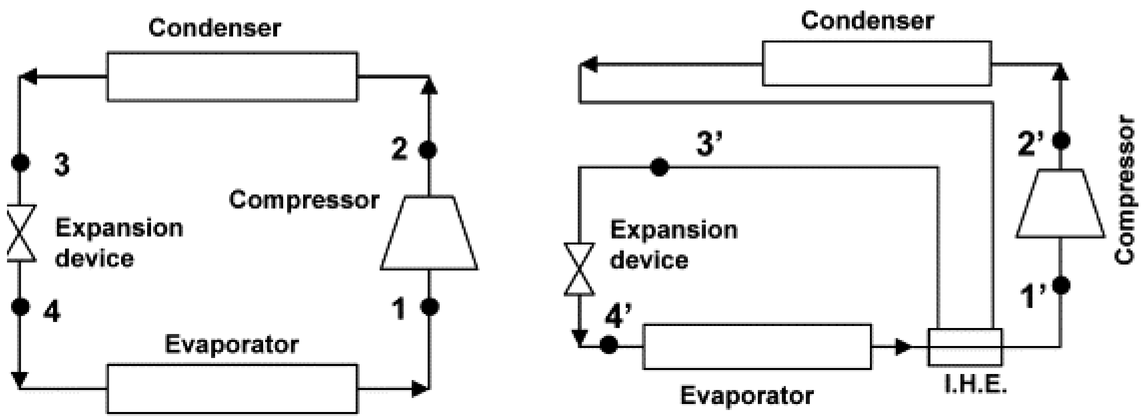
We are often dealing with a subcooling of Δt_{sc} = 2 - 10ºC. Another benefit of subcooling is increased cooling capacity Φ_o.

Superheating and subcooling in the log p,h diagram is shown below:



Superheating and subcooling with internal heat exchanger

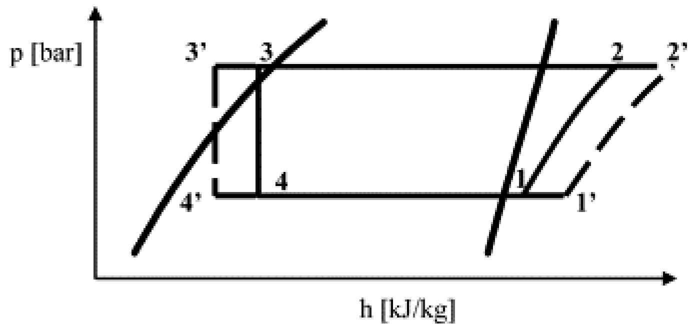
Super heating and subcooling can be obtained using an internal heat exchanger.

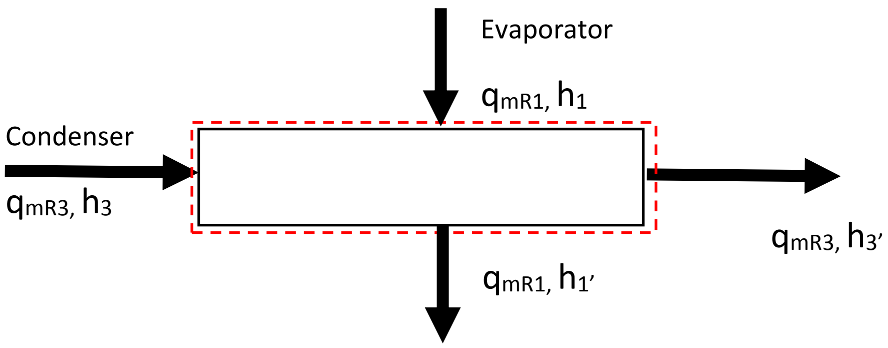


(a) Configuration without IHE. (b) Configuration with IHE adoption.

Fig. 1. Single-stage vapour compression plant with (a) and without (b) IHE.

The figure shows a system with an internal heat exchanger. In this, heat is exchanged between the liquid flow from the condenser and the suction gas to the compressor. The process is drawn in the log p, h diagram:





Since the mass flow of liquid from the condenser and the suction gas to the compressor are the same, the energy balance will yield the following equation:

$\Phi_{33'} = \Phi_{11'} = q_{mR3}\,(h_3 - h_{3'}) = q_{mR1}\,(h_{1'} - h_1)$ [W]

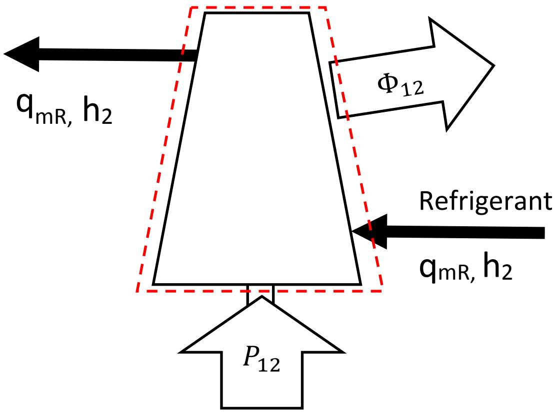
If there is only one evaporator ($q_{mR1} = q_{mR3}$) then $q_{ih} = h_3 - h_{3'} = h_{1'} - h_1$ [kJ/kg]

It is seen that the internal heat exchanger gives both increased subcooling and increased superheating. Therefore, for certain refrigerants, a higher COP is achieved while the opposite will be true for other refrigerants, cf. the paragraph on superheating.

Internal heat exchange decreases the risk of flash gas in the liquid pipe and in certain cases it can reduce the risk of intake of wet suction gas or liquid chock in the compressor. Please also note the increased temperature t_2 after the compressor.

A Cooled Compressor

Cooling a compressor means that heat is removed from during the compression process. Φ_{12} is the heat flow from the compressor to the surroundings (compressor cooling).



Using the energy balance for the compressor we get:

$P_{i12} + \Phi_{12} = q_{m,R}\,(h_2 - h_1)$ or $P_{i12} = q_{m,R}\,(h_2 - h_1) - \Phi_{12}$ [W]

If we want to determine the enthalpy h_2 in the compressor outlet valve we will find:

$$h_2 = h_1 + \frac{P_{i12} + \Phi_{12}}{q_{m,R}} = h_1 + \frac{\dfrac{P_{i12s}}{\eta_s} + \Phi_{12}}{q_{m,R}} \qquad \text{[KJ/kg]} \qquad (3.8)$$

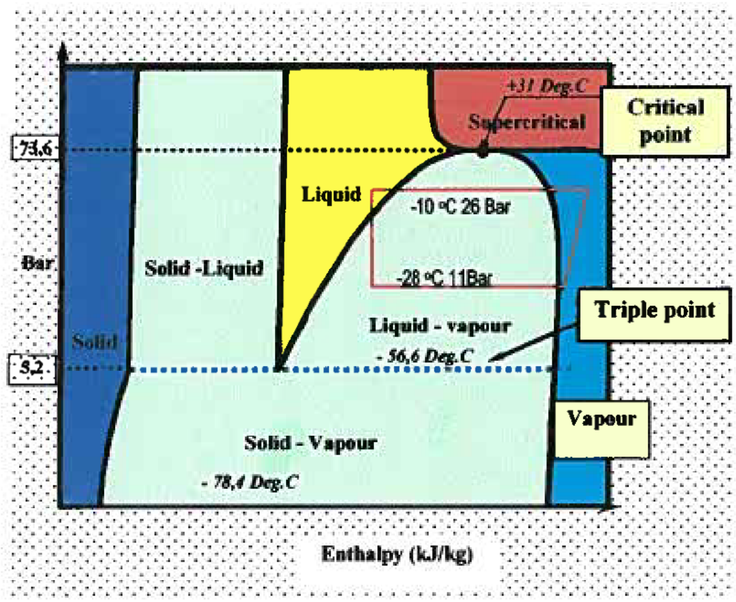
The enthalpy h_2 can then be plotted into the log p,h diagram.

6. Refrigerating plants with CO_2 and HC

CO_2 refrigerant

The critical point is reached at 31°C and 74 bar Generally, plants operate in trans-critical mode a part of the year. In this case there is no condensation after the compressor; the refrigerant is cooled in a gas-cooler. It is possible to have traditional operation (sub-critical) in winter time when outdoor temperature is low.

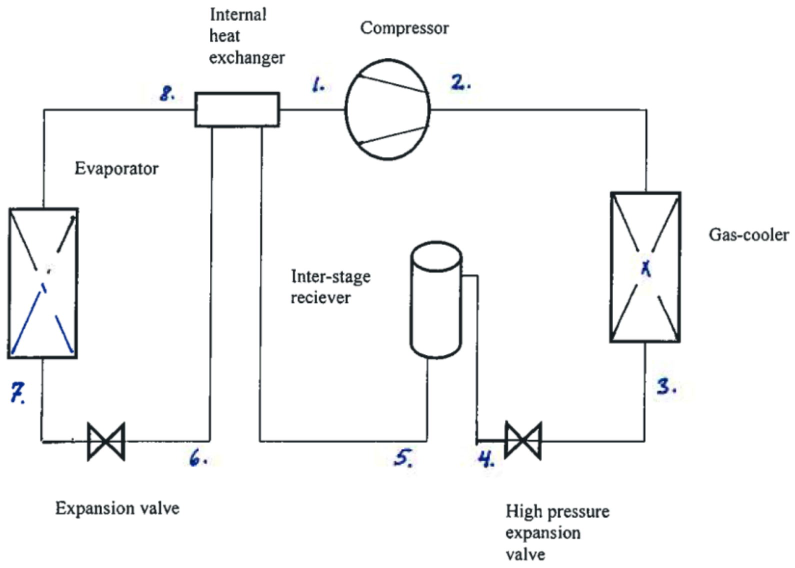
The different phases of CO_2 is shown on the log p,h diagram below:



Simplified P-H Diagram [1].

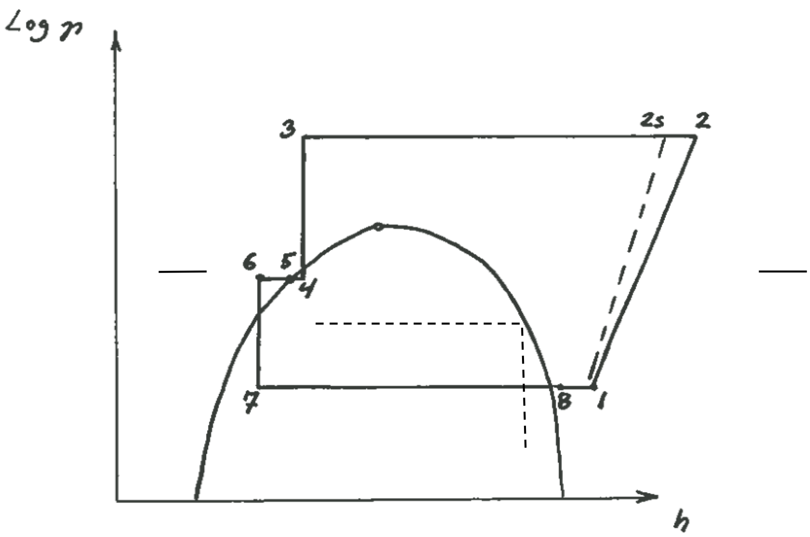
One stage refrigeration plant with direct expansion

There are several possible plant designs. A typical circuit with direct expansion has expansion in two steps and a build-in inter-stage receiver. To ensure sub-cooling before the second expansion valve there is mounted an internal heat exchanger (see figure below):



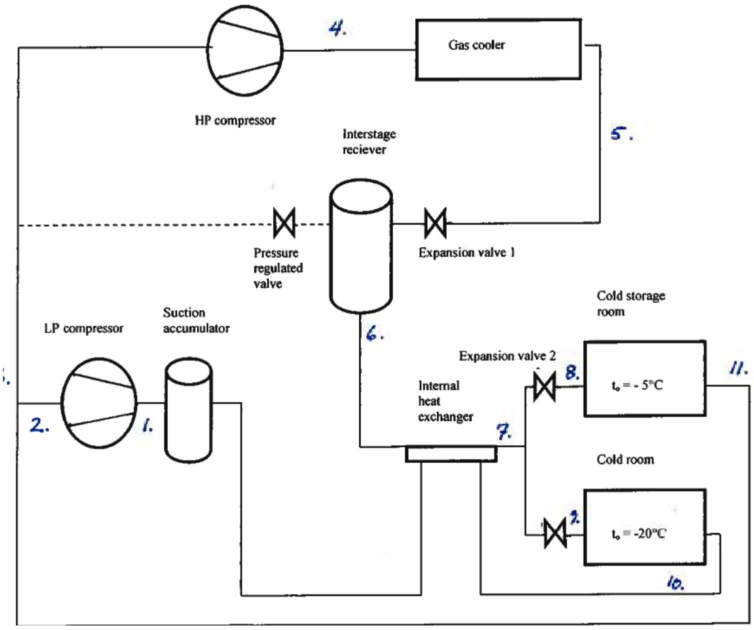
Traditional sub-critical operation: The refrigerant is condensed in “gas-cooler” and high pressure expansion valve is fully open. The process is as usual.

Trans-critical operation: The high pressure gas from compressor flows to the gas-cooler where the gas is cooled. The outlet temperature is regulated by the high pressure expansion valve. After the first throttling the refrigerant is in liquid / vapour region and stored in the receiver. The liquid flows through the internal heat exchanger which ensures sub-cooling. The second expansion valve is overheat regulated and works as usual thermostatic expansion valve. See the principal process in log p,h diagram below:

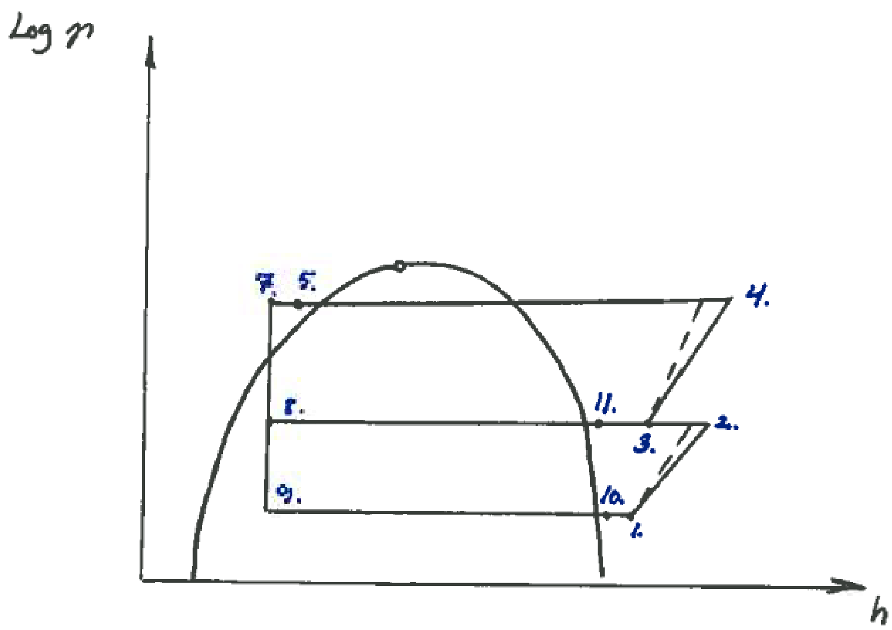


Two stage refrigeration plant for freezing and cold storage

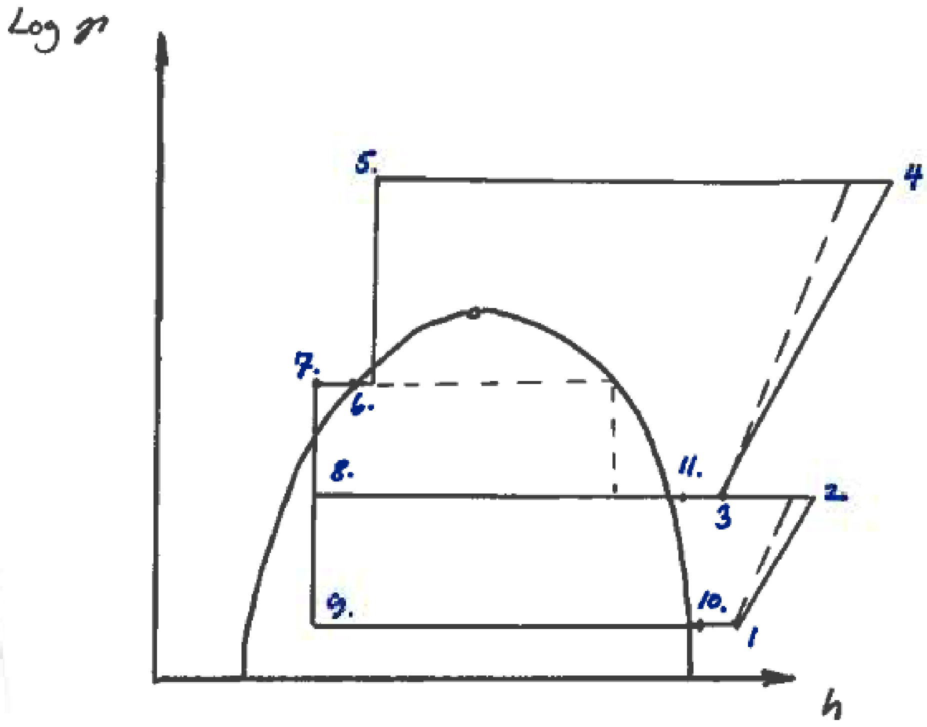
In most cases there will be more than one room to cool down and each room will have different temperatures. A cold store often contains a cold store room with minus temperature and cold storage rooms with plus temperatures. In this case a possible design could be with a two-stage compression and an interstage receiver as shown below:



Traditional sub-critical operation: Expansion valve 1 is fully open and the pressure regulated valve is closed. The principal process in log p,h diagram can be seen below:



Trans-critical operation: There is a throttling from gas-cooler to interstage pressure in expansion valve 1. The interstage pressure is regulated by the pressure regulated valve. The liquid flows from the receiver through the internal heat exchanger, which ensures sub-cooling. The second expansion valve is overheat regulated and works as usual thermostatic expansion valve. See principal process in log p,h diagram below:



Alternatively of two-stage compression cascade cooling plants could be used

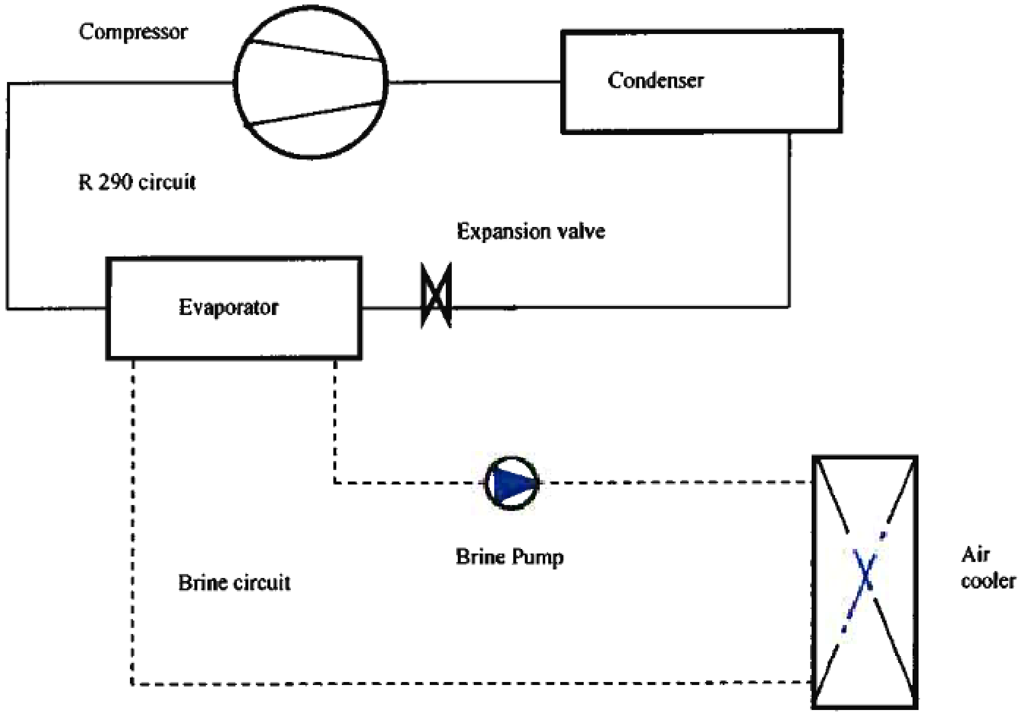
Refrigeration plants with HC refrigerants

R 600a (Isobutane) for small plants/fridges

R 290 (Propane) for large plants

The substance has good thermal qualities and is an effective refrigerant (high COP). You can easily buy all main components and regulators. The problem is that the substance is highly flammable. Therefore, it is difficult to use it in extensive plants, for example plants with direct expansion in cold storage rooms. A possibility is plants with indirect expansion and with circulating brine to cool air coolers in cold storage rooms. In this case the circuit with HC refrigerant is very compact and easier to protect against fire.

Diagram for indirect expansion and brine:



7. Pipe dimensioning

The choice of pipes which connect the components is a compromise between:

- Minimum velocity so that the oil is transported back to the compressor
- Not too high pressure loss due to operational expenses
- Not too large pipes due to investment

The piping system must hence be designed so that the oil is led through the pipes and back to the compressor.

The choice of refrigerant velocity, c, in the pipes and hereby choice of dimension is a complex area where one should consult special literature or a refrigeration consultant. The following velocity indications are guidelines:

Suction pipe: 5 – 20 m/s

Discharge pipe: 8 – 18 m/s

Liquid pipe: 0.4 – 1 m/s

The internal pipe dimension d_i is calculated based on the volume flow q_v and the velocity c:

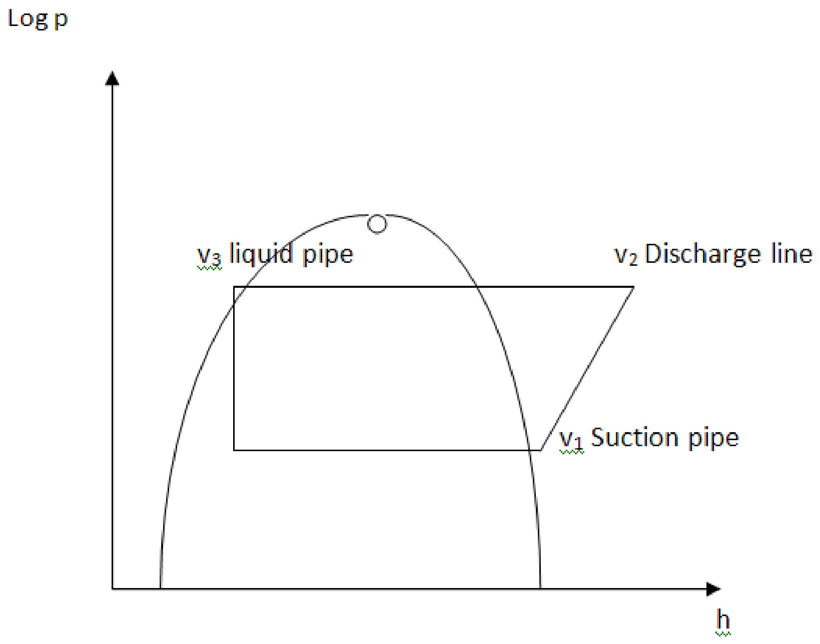
$$q_v = q_{mR}\, v = c\, \frac{\pi}{4}\, d_i^2$$

$$d_i = \sqrt{q_{mR}\, v\, \frac{4}{c\,\pi}}$$

v Specific volume, m^3/kg

c Velocity, m/s

The specific volume, v, can be determined from the log p,h diagram and the type of pipe:



Specific volume and type of pipe.

The pressure drop Δp [Pa] lost to friction is calculated from:

Straight pipes:

$$\Delta p = \frac{\lambda\, L}{d_i}\, 0{,}5\, \varrho\, c^2$$

Minor losses:

$$\Delta p = \zeta\, 0{,}5\, \varrho\, c^2$$

Where

$\varrho = \frac{1}{v}$ Density, kg/m^3

λ Pipe friction factor

L Pipe length, m