Refrigeration Cycle

Heat flows in direction of decreasing temperature, i.e., from high-temperature to low temperature regions. The transfer of heat from a low-temperature to high-temperature requires a *refrigerator* and/or *heat pump*.

Refrigerators and heat pumps are essentially the same device; they only differ in their objectives.

The performance of refrigerators and heat pumps is expressed in terms of coefficient of performance (COP):

$$COP_R = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{Q_H}{W_{net,in}}$$

The Reversed Carnot Cycle

Reversing the Carnot cycle does reverse the directions of heat and work interactions. A refrigerator or heat pump that operates on the reversed Carnot cycle is called a *Carnot refrigerator* or a *Carnot heat pump*.
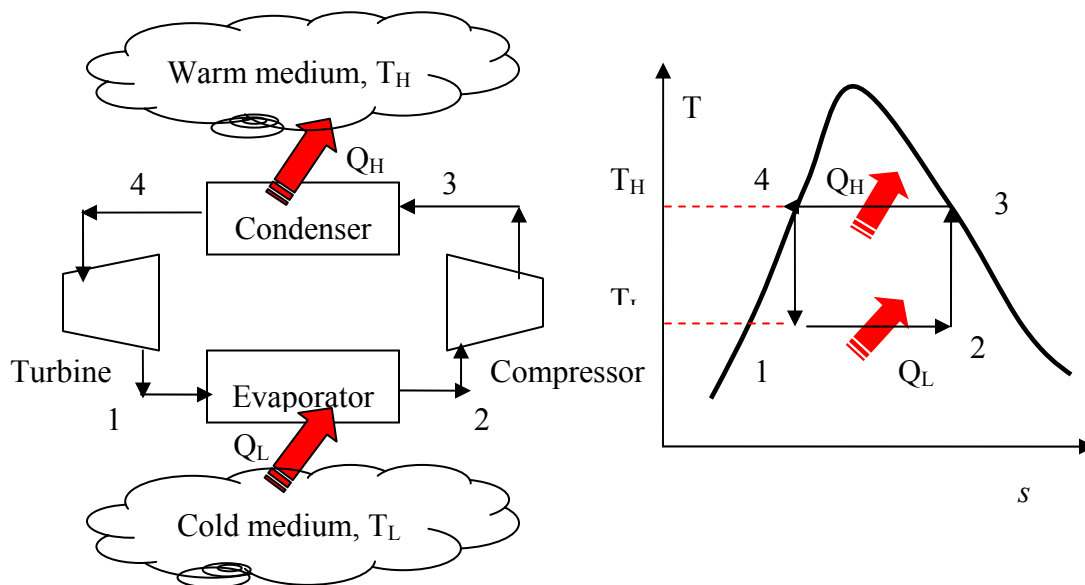

Fig. 5-1: T-s diagram and major components for Carnot refrigerator.

The reversed Carnot cycle is the most efficient refrigeration cycle operating between two specified temperature levels. It sets the highest theoretical COP. The coefficient of performance for Carnot refrigerators and heat pumps are:

$$COP_{Ref,Carnot} = \frac{1}{T_H/T_L - 1} \qquad COP_{HP,Carnot} = \frac{1}{1 - T_L/T_H}$$

The Carnot cycle cannot be approximated in an actual cycle, because:

1- executing Carnot cycle requires a compressor that can handle two-phases

2- also process 4-1 involves expansion of two-phase flow in a turbine.

The Ideal Vapor-Compression Refrigeration Cycle

The vapor-compression refrigeration is the most widely used cycle for refrigerators, air-conditioners, and heat pumps.
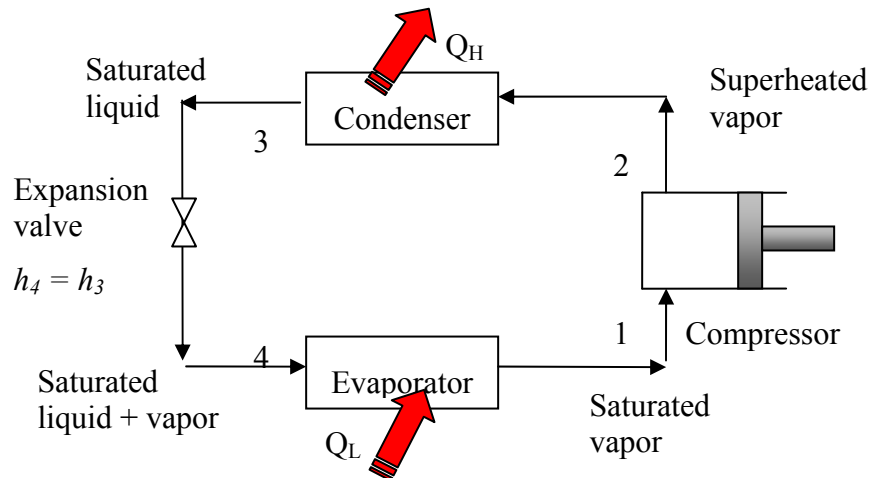
Fig. 5-2: Schematic for ideal vapor-compression refrigeration cycle.

Assumptions for ideal vapor-compression cycle:

- irreversibilities within the evaporator, condenser and compressor are ignored

- no frictional pressure drops

- refrigerant flows at constant pressure through the two heat exchangers (evaporator and condenser)

- heat losses to the surroundings are ignored
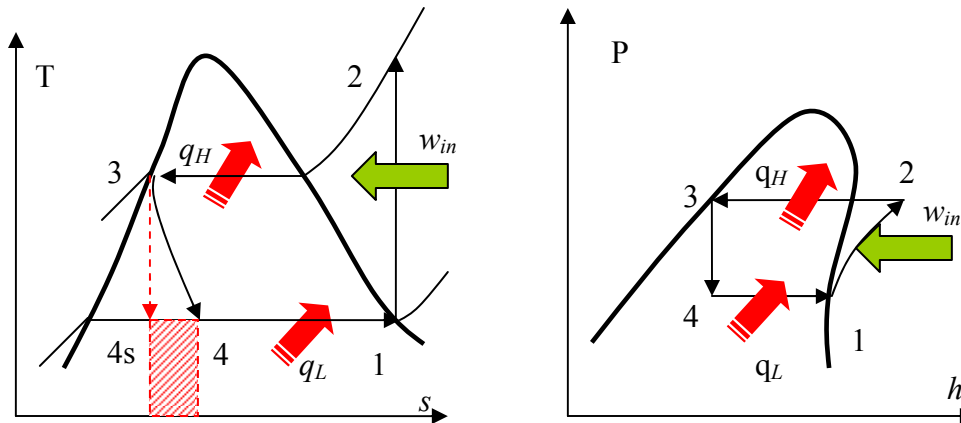
- compression process is isentropic

Fig. 5-3: *T-s* and *P-h* diagrams for an ideal vapor-compression refrigeration cycle.

1-2:   A reversible, adiabatic (isentropic) compression of the refrigerant. The saturated vapor at state 1 is superheated to state 2.

$$w_c = h_2 - h_1$$

2-3:   An internally, reversible, constant pressure heat rejection in which the working substance is de-superheated and then condensed to a saturated liquid at 3. During this process, the working substance rejects most of its energy to the condenser cooling water.

$$q_H = h_2 - h_3$$

3-4:   An irreversible throttling process in which the temperature and pressure decrease at constant enthalpy. The refrigerant enters the evaporator at state 4 as a low-quality saturated mixture.

$$h_3 = h_4$$

4-1:   An internally, reversible, constant pressure heat interaction in which the refrigerant (two-phase mixture) is evaporated to a saturated vapor at state point 1. The latent enthalpy necessary for evaporation is supplied by the refrigerated space surrounding the evaporator. The amount of heat transferred to the working fluid in the evaporator is called the refrigeration load.

$$q_L = h_1 - h_4$$

Notes:

The ideal compression refrigeration cycle is **not an internally reversible** cycle, since it involves throttling which is an irreversible process.

If the expansion valve (throttling device) were replaced by an isentropic turbine, the refrigerant would enter the evaporator at state 4s. As a result the refrigeration capacity would increase (area under 4-4s) and the net work input would decrease (turbine will

produce some work). However; replacing the expansion valve by a turbine is not practical due to the added cost and complexity.

The COP improves by 2 to 4% for each °C the evaporating temperature is raised or the condensing temperature is lowered.

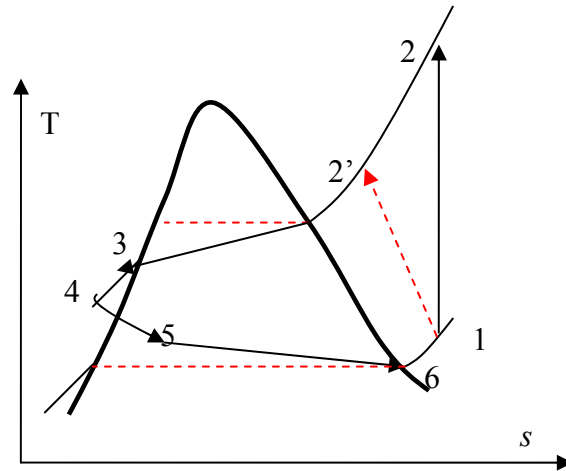## Actual Vapor-Compression Refrigeration Cycle

Fig. 5-4: T-s diagram for actual vapor-compression cycle.

Most of the differences between the ideal and the actual cycles are because of the irreversibilities in various components which are:

1-In practice, the refrigerant enters the compressor at state 1, slightly superheated vapor, instead of saturated vapor in the ideal cycle.

2- The suction line (the line connecting the evaporator to the compressor) is very long. Thus pressure drop and heat transfer to the surroundings can be significant, process 6-1.

3- The compressor is not internally reversible in practice, which increase entropy. However, using a multi-stage compressor with intercooler, or cooling the refrigerant during the compression process, will result in lower entropy, state 2'.

4- In reality, the refrigerant leaves condenser as sub-cooled liquid. The sub-cooling process is shown by 3-4 in Fig. 5-4. Sub-cooling increases the cooling capacity and will prevent entering any vapor (bubbles) to the expansion valve.

5- Heat rejection and addition in the condenser and evaporator do not occur in constant pressure (and temperature) as a result of pressure drop in the refrigerant.

Selecting the Right Refrigerant

When designing a refrigeration system, there are several refrigerants from which to choose. The right choice of refrigerant depends on the situation at hand. The most common refrigerants are: R-11, R-12, R-22, R-134a, and R-502.

R12: CCl_2F_2 dichlorofluoromethane, used for refrigeration systems at higher temperature levels- typically, water chillers and air conditioning (banned due to ozone layer effects)

R22: $CHClF_2$ has less chlorine, a little better for the environment than R12 - used for lower temperature applications

R134a: CFH_2CF_3 tetrafluorethane - no chlorine- went into production in 1991- replacement for R12

Ammonia NH_3: corrosive and toxic- used in absorption systems-cheap- high COP

R744: CO_2 behaves in the supercritical region- low efficiency

R290: propane combustible.

Many factors need to be considered when choosing a refrigerant:

ozone depletion potential ,global warming potential:	CFC (chlorofluorocarbons) refrigerants allow more ultraviolet radiation into the earth's atmosphere by destroying the protective ozone layer and thus contributing to the greenhouse effect that causes the global warming. As a result the use of some CFCs (e.g. R-11, R-12, and R-115) are banned by international treaties.
combustibility:	all hydro-carbon fuels, such as propane
leak detectability:	the refrigerant saturated pressure at the evaporator should be above P_{atm}.
thermal factors:	1- the heat of vaporization of the refrigerant should be high. The higher h_{fg}, the greater the refrigerating effect per *kg* of fluid circulated.
thermal factors cont'd.:	2- the specific heat of the refrigerant should be low. The lower the specific heat, the less heat it will pick up for a given change in temperature during the throttling or in flow through the piping, and consequently the greater the refrigerating effect per *kg* of refrigerant
	3- the specific volume of the refrigerant should be low to minimize the work required per *kg* of refrigerant circulated since evaporation and condenser temperatures are fixed by the temperatures of the surroundings
	4- selection is based on operating pressures in the evaporator and the condenser.

Other desirable characteristics of refrigerant are: non-flammable, being chemically stable, and be available at low cost.

Cascade Refrigeration Cycle

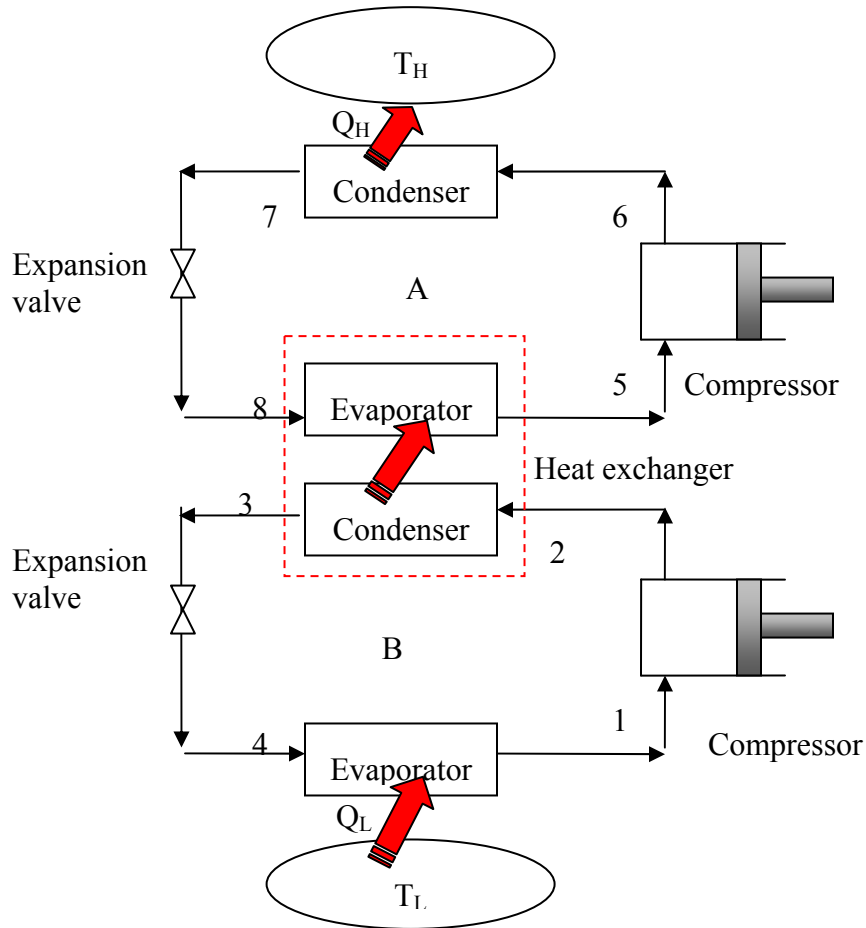Systems that have 2 (or more) refrigeration cycles operating in series.

Fig.5-5: A 2-stage cascade refrigeration cycle.

Cascade cycle is used where a very wide range of temperature between T_L and T_H is required. As shown in Fig. 5-5, the condenser for the low temperature refrigerator is used as the evaporator for the high temperature refrigerator.

Cascading improves the COP of a refrigeration cycle. Moreover, the refrigerants can be selected to have reasonable evaporator and condenser pressures in the two or more temperature ranges.
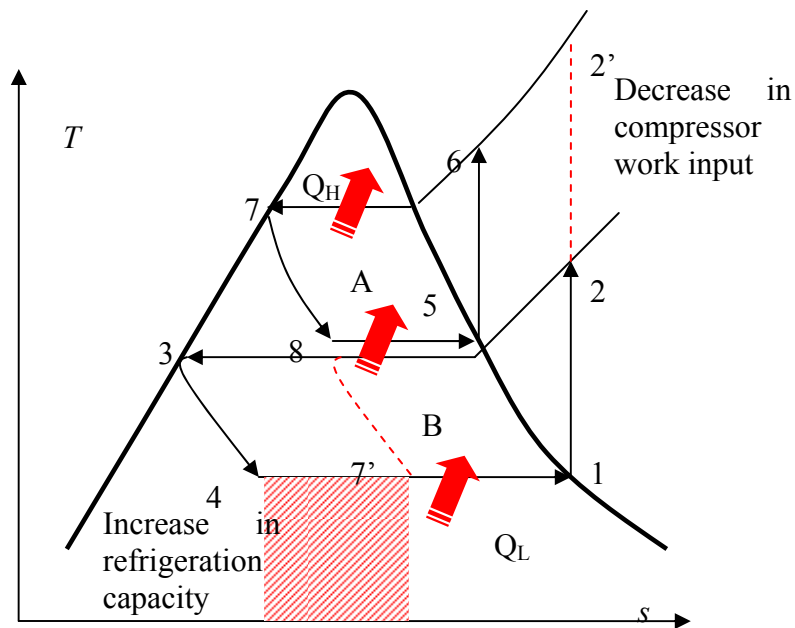
Fig. 5-6: T-s diagram for 2-stage cascade system.

The two cycles are connected through the heat exchanger in the middle, which serves as evaporator (cycle A) and condenser (cycle B). One can write:

$$\dot{m}_A\left(h_5 - h_8\right) = \dot{m}_B\left(h_2 - h_3\right)$$

$$COP_{R,cascade} = \frac{\dot{m}_B\left(h_1 - h_4\right)}{\dot{m}_A\left(h_6 - h_5\right) + \dot{m}_B\left(h_2 - h_1\right)}$$

Figure 5-6 shows the increase in refrigeration capacity (area under 4-7') and decrease in compressor work (2-2'-6-5).

Example 5-1: A two-stage Refrigeration cycle

Consider a two-stage cascade refrigeration system operating between pressure limits of 0.8 and 0.14 MPa. Each stage operates on an ideal vapor-compression refrigeration cycle with refrigerant R-134a as working fluid. Heat rejection from the lower cycle to the upper cycle takes place in an adiabatic counter flow heat exchanger where both streams enter at about 0.32 MPa. If the mass flow rate of the refrigerant through the upper cycle is 0.05 kg/s, determine:

  a) the mass flow rate of the refrigerant through the lower cycle

  b) the rate of heat removal from the refrigerated space and the power input to the compressor

  c) COP.

Assumptions:

1) Steady operation

2) ΔKE=ΔPE=0

3) Adiabatic heat exchanger.

Figure 5-6 shows the *T-s* diagram for the cascade cycle. Enthalpies for all 8 states of refrigerant R-134a can be read off Tables A-12 and A-13.

Writing energy balance for the heat exchanger, the mass flow rate of the refrigerant through the lower cycle can be found:

$$\dot{m}_A h_5 + \dot{m}_B h_3 = \dot{m}_A h_8 + \dot{m}_B h_2$$

$$\dot{m}_A(h_5 - h_8) = \dot{m}_B(h_2 - h_3)$$

$$0.05 kg/s (251.88 - 95.47)kJ/kg = \dot{m}_B (255.93 - 55.16)kJ/kg$$

$$\dot{m}_B = 0.039 kg/s$$

b) the heat removal by the cascade cycle can be determined from:

$$\dot{Q}_L = \dot{m}_B(h_1 - h_4) = 0.0390 kg/s (239.16 - 55.16)kJ/kg = 7.18 kW$$

$$\dot{W}_{in} = \dot{W}_{Comp1,in} + \dot{W}_{Comp2,in} = \dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1) = 1.61 kW$$

c) the COP of the cycle will be:

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{7.18 kW}{1.61 kW} = 4.46$$