Heat Pump Sytems

Summary W4

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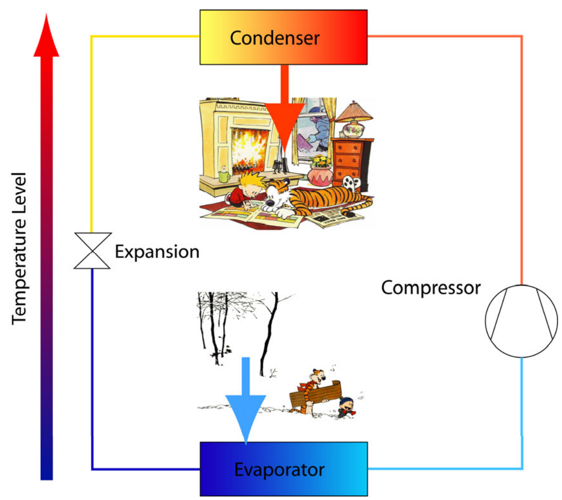
What is a Heat Pump Thermodynamically?

- Bithermal thermodynamic cycle working in anti-clockwise direction
- Work is invested to drive the cycle, which absorbs and supplies heat at different temperatures



Heating heat pump cycle

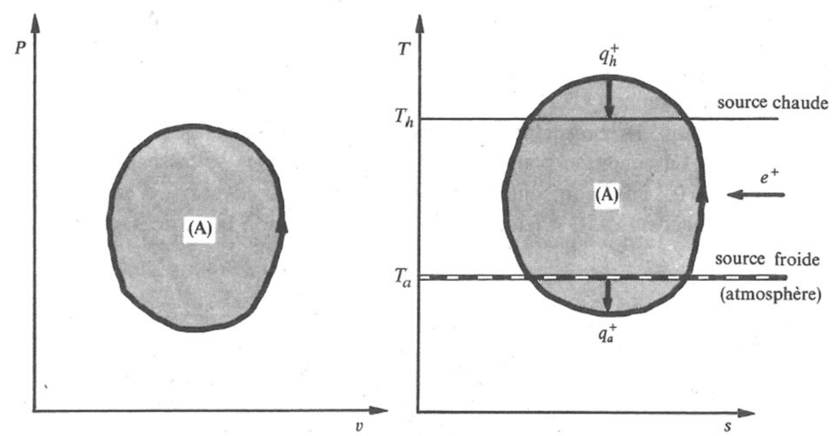


Fig 13.18 Favrat

Heating Heat Pump Effectiveness

- Heating effectiveness definition

$$\epsilon_h = \frac{q_h^-}{e^+} = 1 + \frac{q_a^+}{e^+} = \frac{1}{\Theta_h}\underbrace{\left(1 - \frac{l}{e^+}\right)}_{\eta} = COP_h$$

Specific exergy losses: l

Coefficient of performance: COP_h

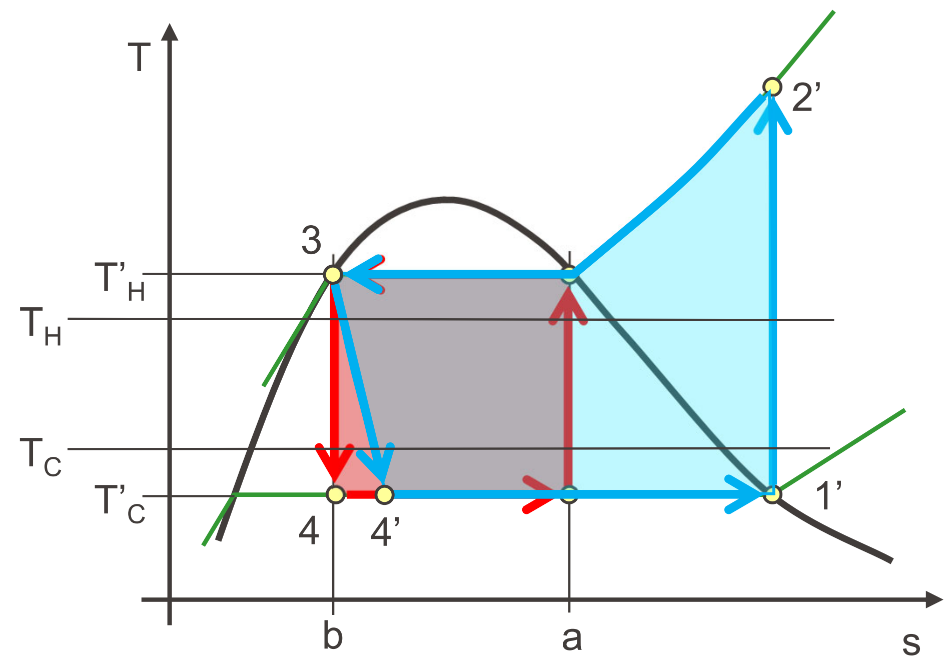
Exergy efficiency: η

Carnot factor $\Theta_h = 1 - \frac{T_a}{T_h}$

- Link between Carnot factor and exergy efficiency in heating mode

Technical Challenges of Reversed Carnot Cycle

- Dry compression is preferred to protect compression machine from destruction
- Use of two phase expansion valve preferred solution
  → isenthalpic expansion
- Heat transfer in condenser and evaporator requires finite temperature difference



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Heat Pump Systems

Analysis of Performance Metrics

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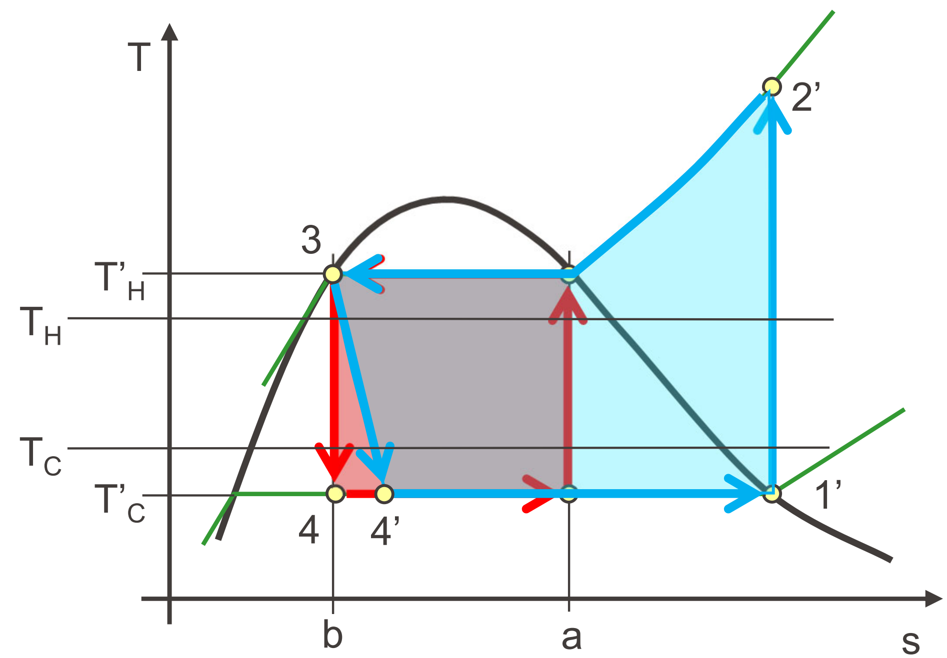
EPFL

Technical Challenges

- Implementation of Carnot cycle challenging due to technical limitations
  - Heat transfer to thermal sources requires temperature difference
  - Dry compression preferred
  - Use of two phase expansion valve preferred solution
- Technical solutions imply deviations from Carnot!



Analysis of Heating and Refrigeration HP Cycle

- Consider theoretical cycle with two isothermal, one isentropic and one polytropic process, including devaluation at low and high temperature and dissipation

- Energy balance

$$q_h^- - q_f^+ = e^+ = r - \oint T ds$$

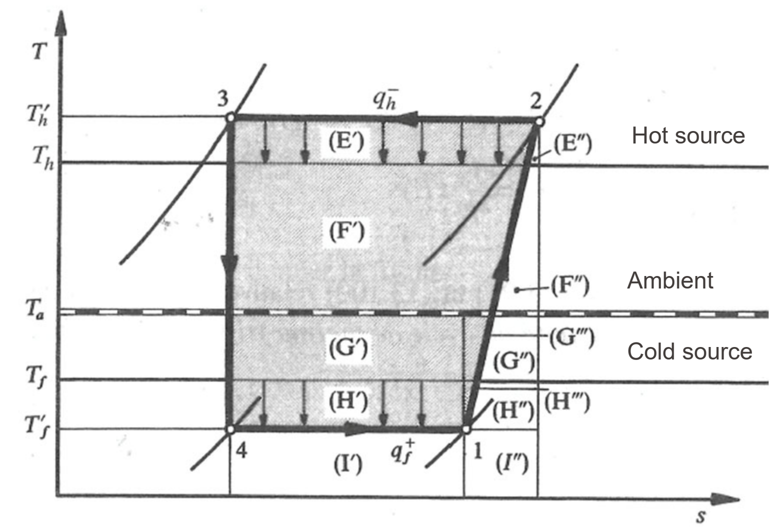


Fig 13.26 Favrat

Analysis of Heating and Refrigeration HP Cycle
Energy Balance

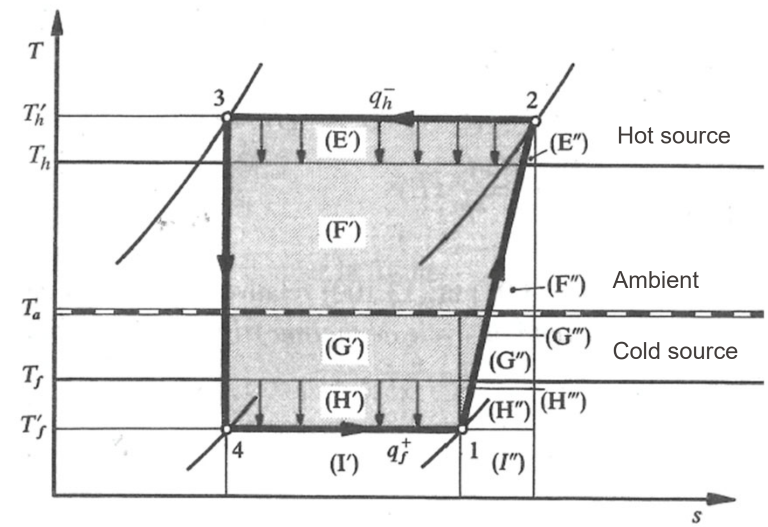


Fig 13.26 Favrat

Analysis of Heating and Refrigeration HP Cycle
Energy Balance

$$q_f^+ = T_f'(s_1 - s_4) = i'$$

$$q_h^- = T_h'(s_2 - s_3) = e + f + g + h + i$$

$$e^+ = \left(T_h' - T_f'\right)(s_2 - s_3) + T_f'(s_2 - s_1)$$

$$e^+ = e + f + g + h + i''$$

$$-\oint T ds = e' + f' + g' + g''' + h' + h'''$$

$$r = \int_1^2 T ds = e'' + f'' + g'' + h'' + i''$$

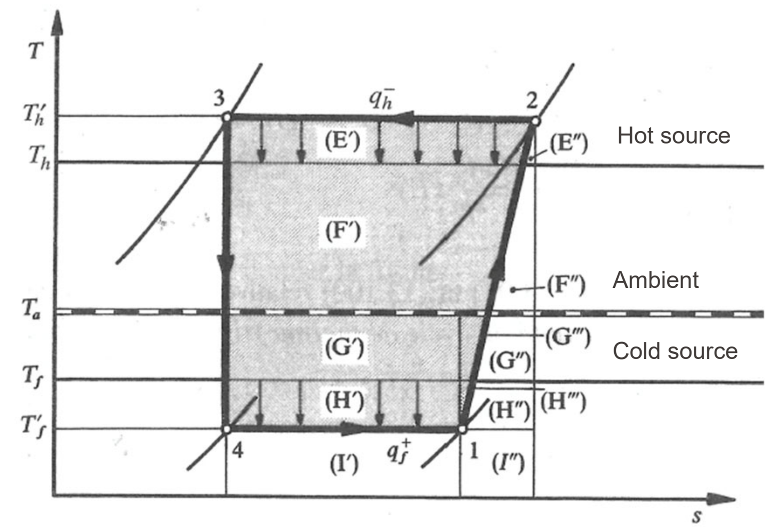


Fig 13.26 Favrat

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Analysis of Heating and Refrigeration HP Cycle
Energy Balance

- Effectiveness of heating and refrigeration cycle

$$\epsilon_{hf} = \frac{q_h^- + q_f^+}{e^+}$$

- Inadequate adding received and delivered heat without care
- Definition disconcerting and meaningless without temperature levels
- Effectiveness cannot express real thermodynamic efficiency

Analysis of Heating and Refrigeration HP Cycle

- Consider theoretical cycle with two isothermal, one isentropic and one polytropic process, including devaluation at low and high temperature and dissipation

- Exergy balance

$$e_{qh}^{-} + e_{qf}^{-} = e^{+} - (l_{qh} + l_{qf} + l_c + l_e)$$

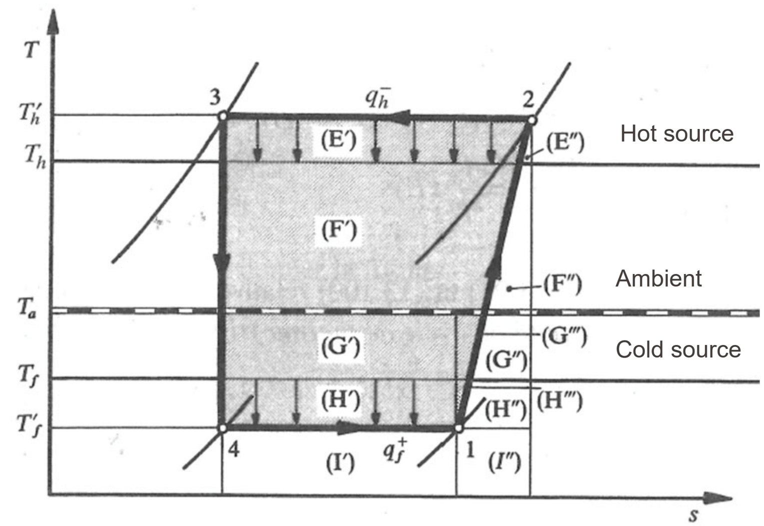


Fig 13.26 Favrat

Analysis of Heating and Refrigeration HP Cycle

- Exergy balance: heat exchange at condenser

$$e^{+}_{qh'} - e^{-}_{qh} = l_{qh}$$ Exergy balance

$$e^{+}_{qh'} = \int \left(1 - \frac{T_a}{T'_h}\right) \delta q_h = \left(1 - \frac{T_a}{T'_h}\right) q^{-}_h$$ Received heat exergy

$$e^{-}_{qh} = \int \left(1 - \frac{T_a}{T_h}\right) \delta q_h = \left(1 - \frac{T_a}{T_h}\right) q^{-}_h$$ Delivered heat exergy

$$l_{qh} = q^{-}_h \left(\frac{T'_h - T_h}{T'_h T_h}\right) T_a = \frac{T_a}{T_h}\left(T'_h - T_h\right)(s_2 - s_3)$$ Exergy losses

Analysis of Heating and Refrigeration HP Cycle

- Exergy balance: heat exchange at evaporator

$$e_{qf}^{+} - e_{qf'}^{-} = l_{qf}$$ Exergy balance

$$e_{qf}^{+} = \int \left(1 - \frac{T_a}{T_f}\right) \delta q_h = \left(1 - \frac{T_a}{T_f}\right) q_f^{+}$$ Received heat exergy

$$e_{qf'}^{-} = \int \left(1 - \frac{T_a}{T_f'}\right) \delta q_f = \left(1 - \frac{T_a}{T_f'}\right) q_f^{+}$$ Delivered heat exergy

$$l_{qf} = q_f^{+} \left(\frac{T_f - T_f'}{T_f' T_f}\right) T_a = \frac{T_a}{T_f} \left(T_f - T_f'\right) (s_1 - s_4)$$ Exergy losses

Analysis of Heating and Refrigeration HP Cycle

- Exergy balance: adiabatic compression

$$e_y^+ + e^+ - e_y^- = l_c$$ Exergy balance

$$e_y^+ = k_1 = h_1 - T_a s_1$$ Received transformation exergy

$$e_y^- = k_2 = h_2 - T_a s_2$$ Delivered transformation exergy

$$\underbrace{h_1 - h_2}_{-e^+} + e^+ + T_a\left(s_2 - s_1\right) = l_c$$

$$T_a\left(s_2 - s_1\right) = l_c$$ Exergy losses

Analysis of Heating and Refrigeration HP Cycle

- Exergy balance: expansion

$$e_y^+ - e^- - e_y^+ = l_e$$ Exergy balance

$$e_y^+ = k_3 = h_3 - T_a s_3$$ Received transformation exergy

$$e_y^- = k_4 = h_4 - T_a s_4$$ Delivered transformation exergy

$$\underbrace{h_3 - h_4}_{+e^-} - e^- + T_a \left(s_4 - s_3 \right) = l_e$$

$$T_a \left(s_4 - s_3 \right) = l_e$$ Exergy losses

Analysis of Heating and Refrigeration HP Cycle

- Graphical representation of exergy losses

$$l_{qh} = \frac{T_a}{T_h}\left(T_h^{'} - T_h\right)(s_2 - s_3)$$

$$T_a\left(s_4 - s_3\right) = l_e$$

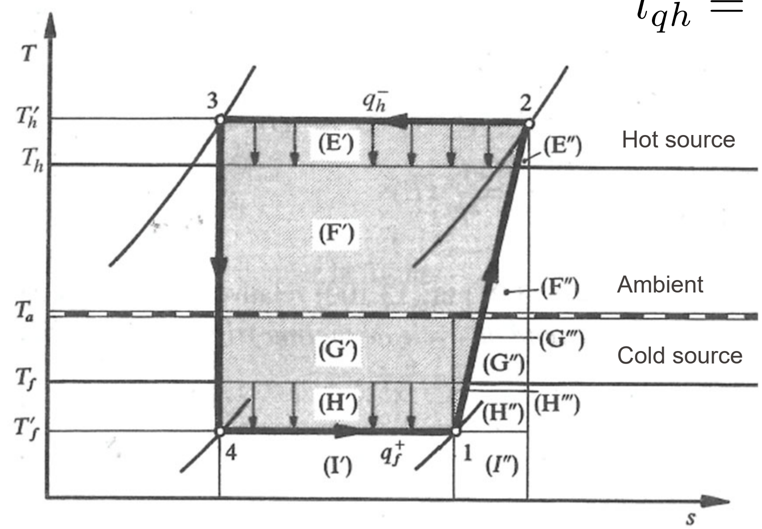


Fig 13.26 Favrat

$$l_{qf} = \frac{T_a}{T_f}\left(T_f - T_f^{'}\right)(s_1 - s_4)$$

$$T_a\left(s_2 - s_1\right) = l_c$$

Analysis of Heating and Refrigeration HP Cycle

- Exergy efficiency of heating and refrigeration cycle

$$\eta = \frac{e_{qh}^{-} + e_{qf}^{-}}{e^{+}} = 1 - \frac{T_a}{T_h}\left[1 - \frac{\frac{T_h}{T_f} - 1}{\frac{T_h + \Delta T_h}{T_f - \Delta T_f}\left(1 + \frac{\Delta s}{s_1 - s_4}\right) - 1}\right]$$

with

$$\Delta T_f = T_h^{'} - T_h$$

$$\Delta T_f = T_f - T_f^{'}$$

$$\Delta s = s_2 - s_1$$

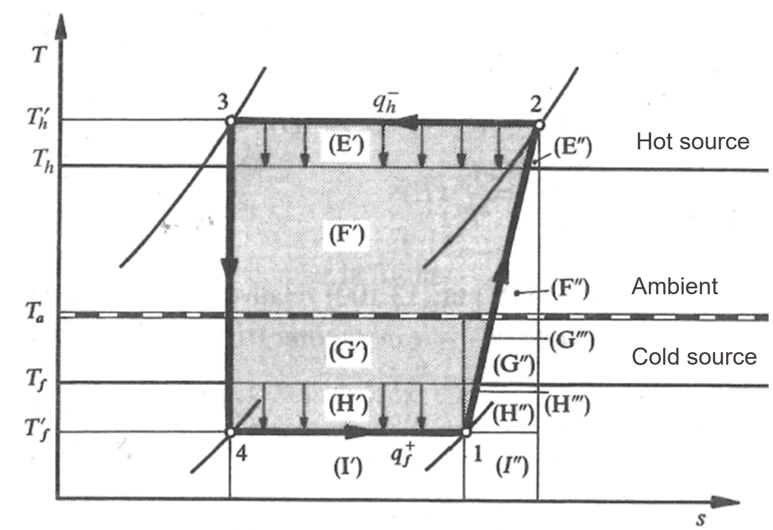


Fig 13.26 Favrat

Analysis of Heating and Refrigeration HP Cycle

- Assume example case without dissipation ($\Delta s = 0$) but with thermal devaluation and with $T_f = T_a$ → heat pump mode

- Exergy efficiency

$$\eta = \frac{(T_h + \Delta T_h)(T_h - T_a)}{T_h (T_h - T_a + \Delta T_h + \Delta T_f)}$$

- Effectiveness

$$\epsilon = \frac{T_h + \Delta T_h}{T_h - T_a + \Delta T_h + \Delta T_f}$$

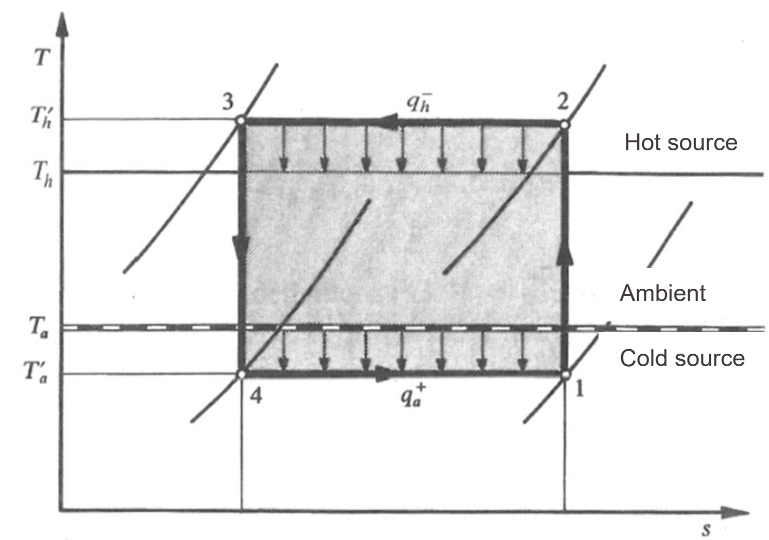
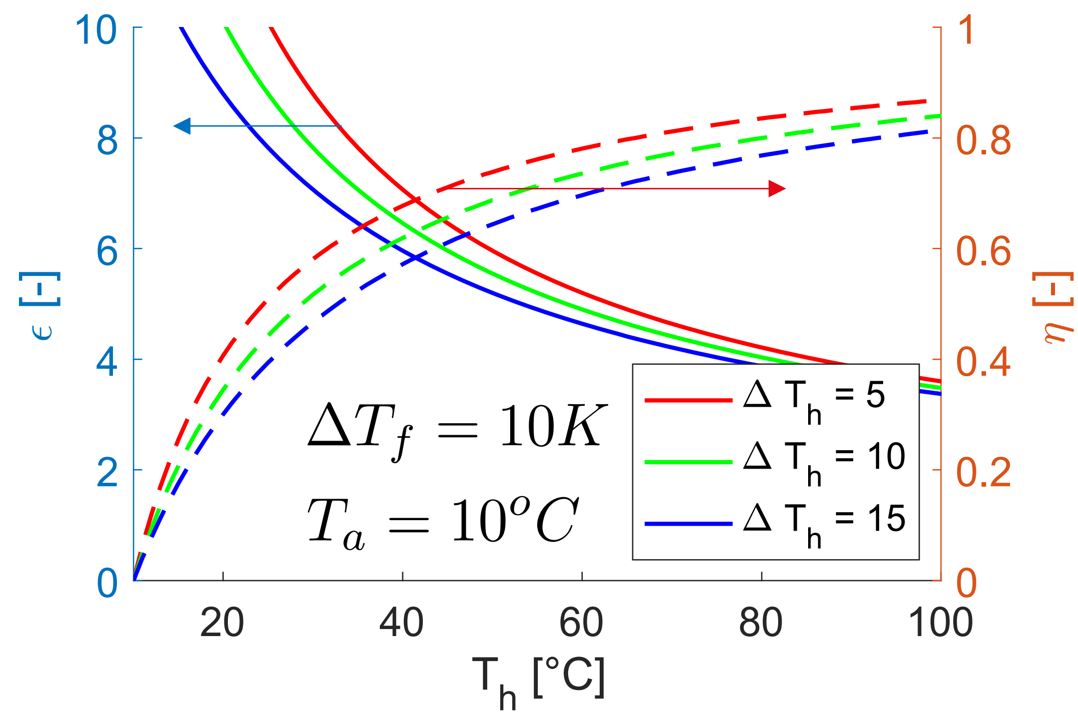


Fig. 13.27 Favrat

Analysis of Heating and Refrigeration HP Cycle

- Plot clearly demonstrates advantage of exergy analysis
- Devaluation yields strong impact at low temperature lifts



Heat Pumps Systems

Analysis of real heat pumps

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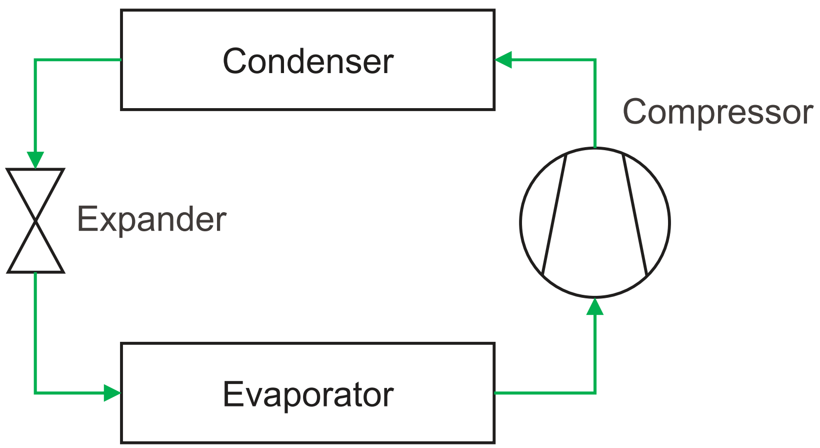
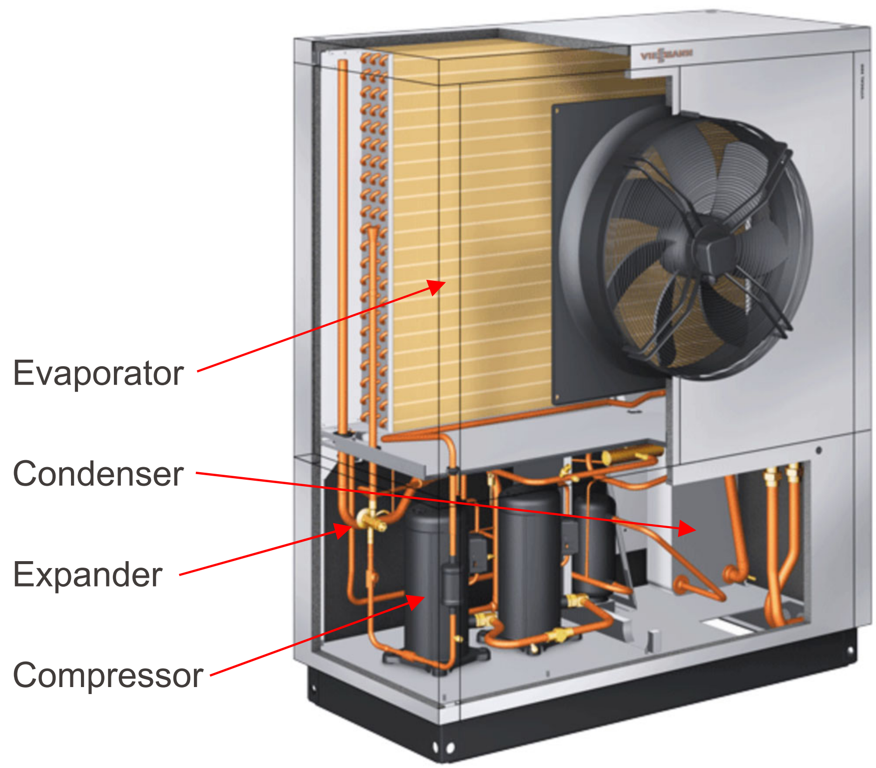
Real HP Cycle

- Typical layout of single stage heat pump cycle





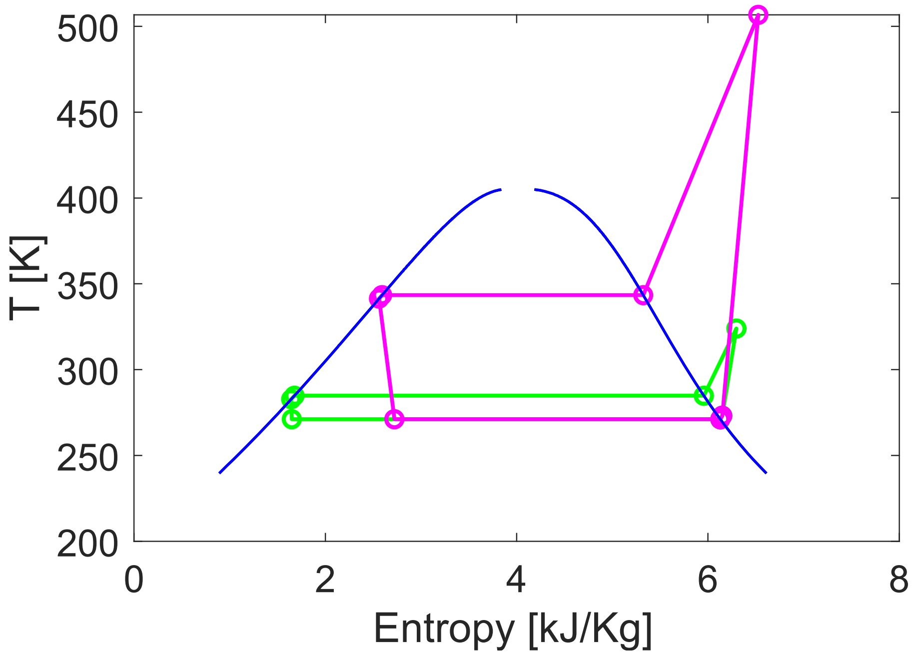
Conditions on Real HP Cycle

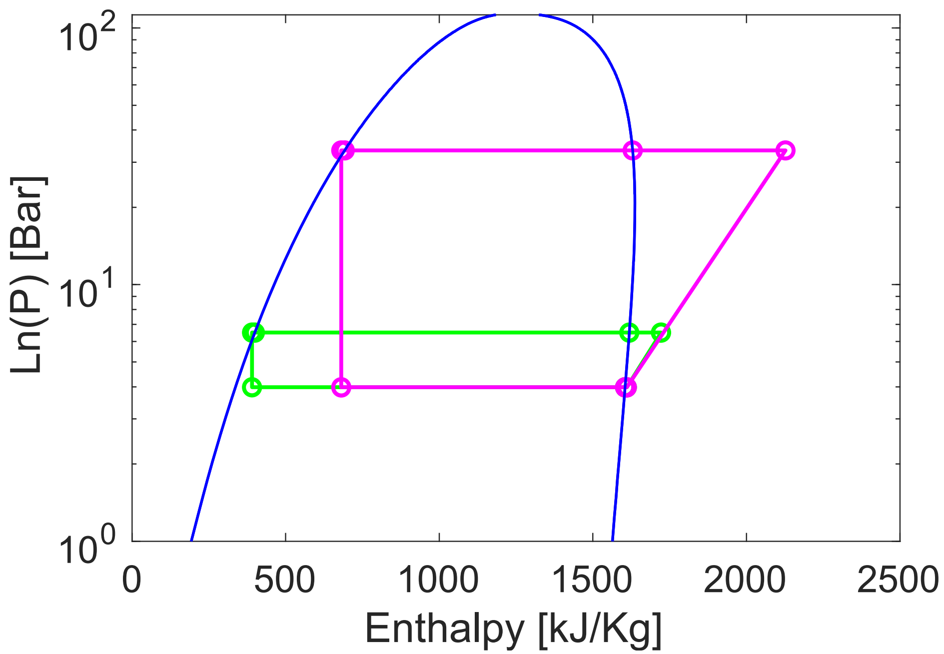
- Single stage vapor compression heat pump
  - Adiabatic compression with isentropic efficiency function of pressure ratio
  - Condensation with 2K subcooling
  - Isenthalpic expansion in valve
  - Evaporation with 2K superheat
  - Cold source is air at 5°C
  - Hot source is water provided at 15 – 90°C (10kW)
  - Temperature difference on water and air 5K
  - Pinch in condenser and evaporator 2K
  - Working fluid ammonia

- Assumptions
  - Perfect thermal insulation, negligible dissipation in condenser, evaporator and ducts, steady-state operation

Real HP Cycle Representation

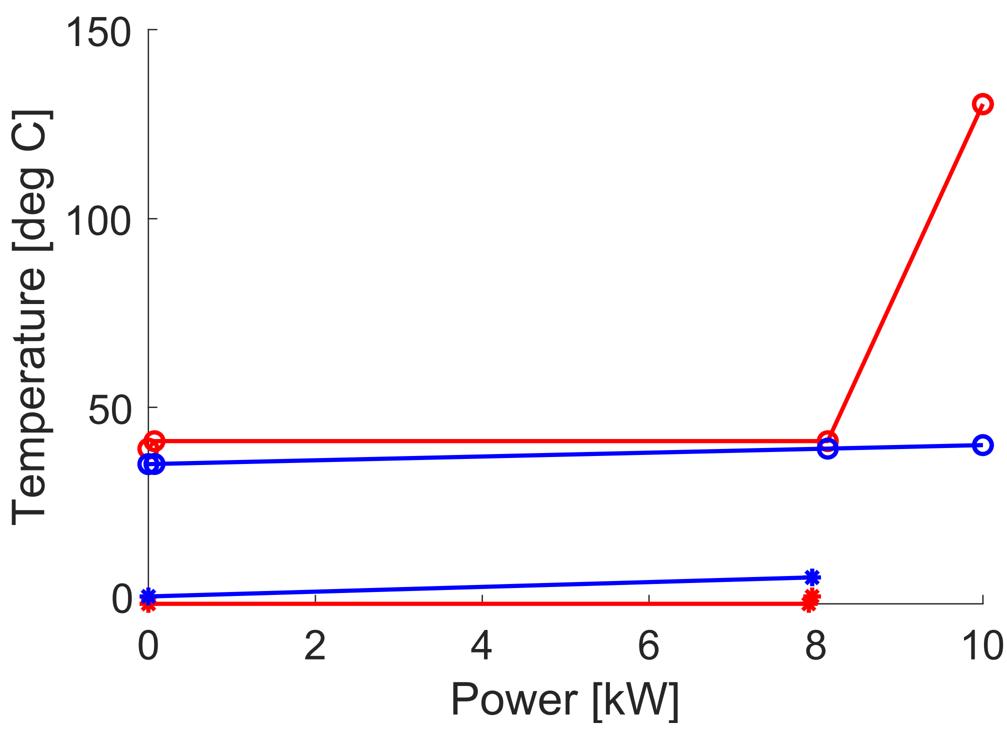
- Vapor compression heat pump, T-s and P-h diagrams





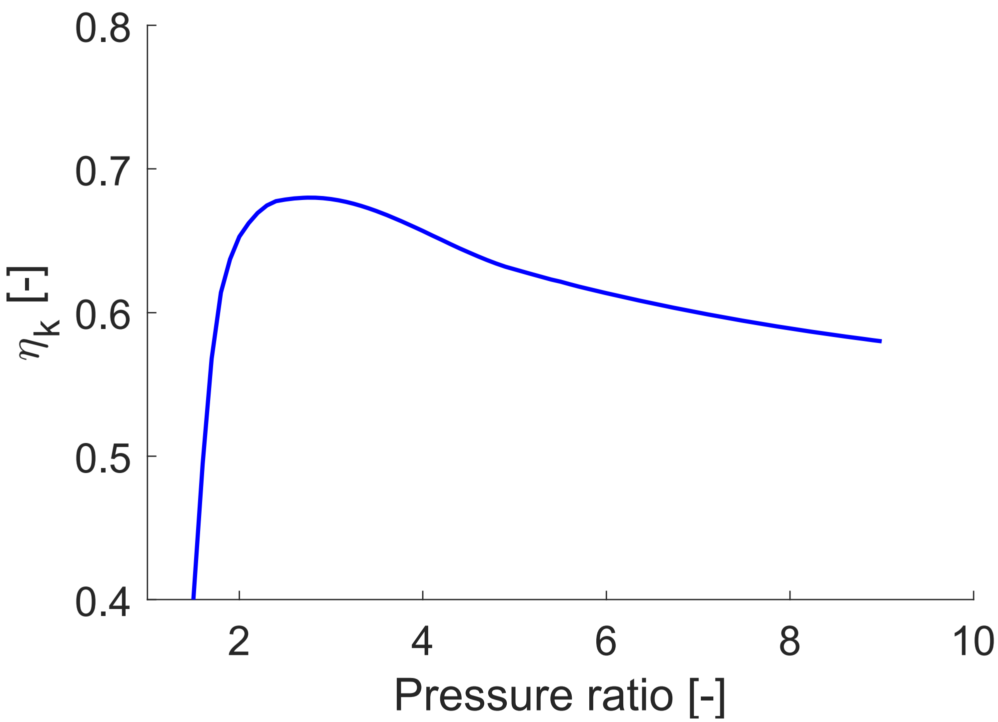
Real HP Cycle Composites

- Condenser and evaporator composites
  - 2K superheat & subcooling
  - Cold source is air at 5°C
  - Temperature difference on water and air 5K
  - Pinch in condenser and evaporator 2K



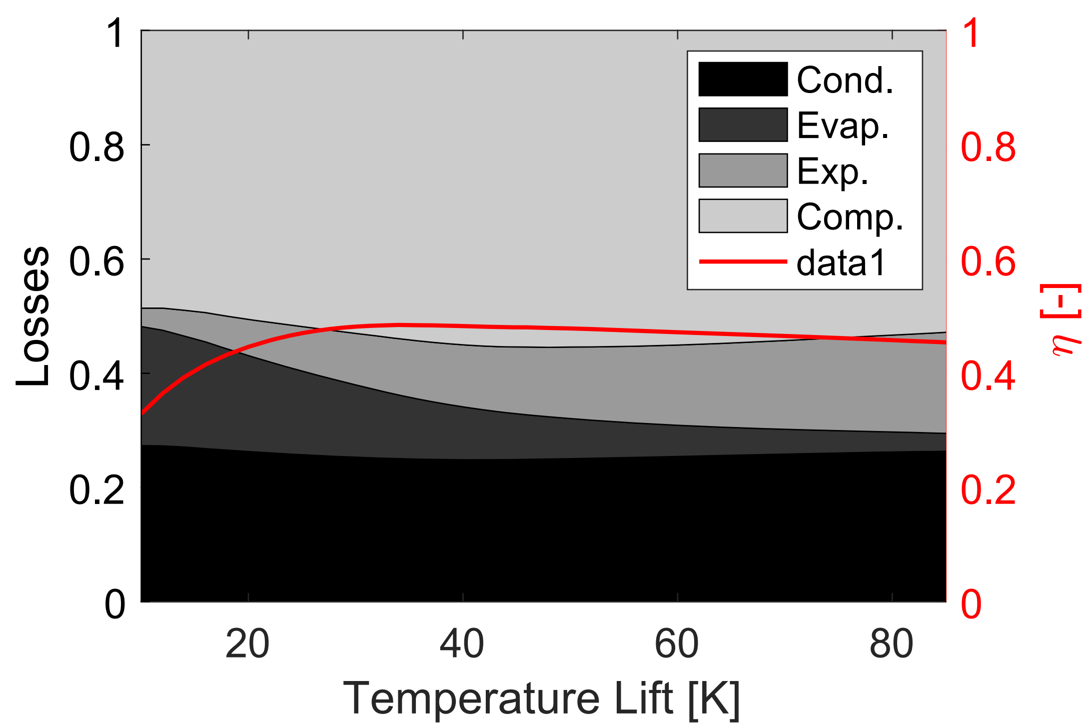
Real HP Cycle Compressor (Scroll)

- Isentropic efficiency of typical heat pump scroll compressor
  - Max efficiency governed by internal geometry
  - Efficiency rises rapidly due to reduction of over-compression
  - At high pressure ratios efficiency drops due to leakage and to under-compression



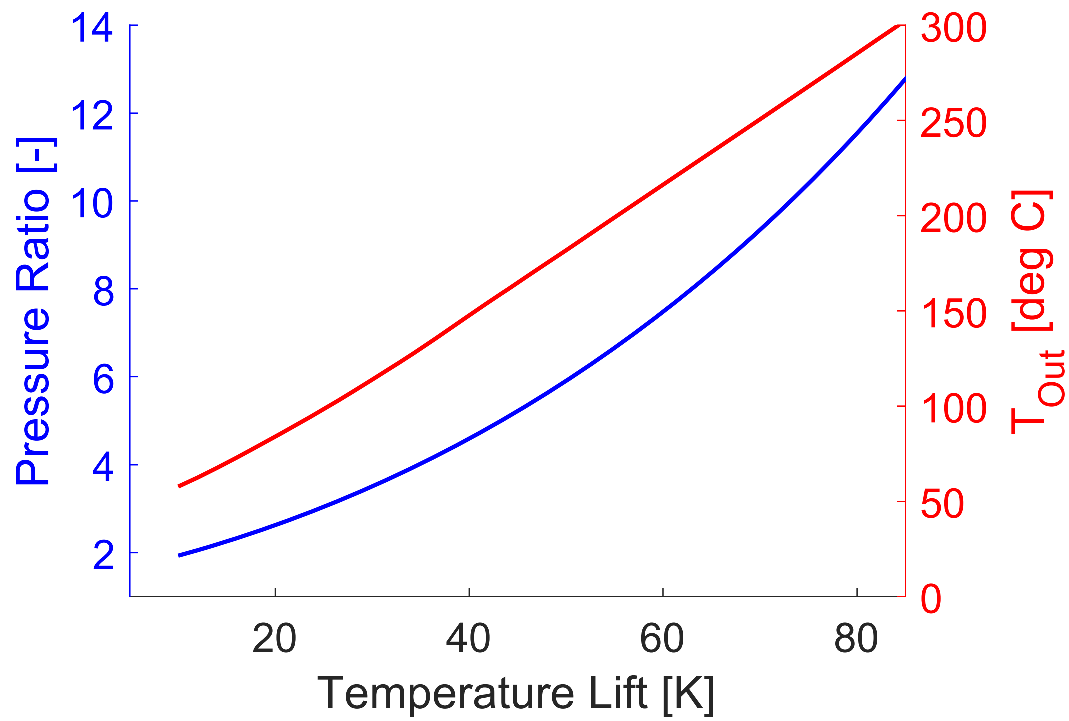
Exergy Analysis of Real HP Cycle

- Exergy losses evolution
  - Heat exchanger loss dominate at low temperature lifts
  - Compressor accounts for 50% of the losses
  - Compressor and expansion losses dominate at high temperature lifts



Compressor in Real HP Cycle

- Pressure ratio and exhaust temperature increase with lift
- High temperatures may damage oil and deteriorate working fluid molecules
- High superheat increases exergy losses in condenser
- Temperature lift with single stage heat pump cycle limited



Limitations of Real HP Cycle

- Single stage cycle (compression & expansion) well suited for low temperature lifts
- Heat exchangers are key for decreasing losses at low lifts
- At high temperature lifts compressor and expansion dominate losses
- High compressor outlet temperatures limit feasible temperature lifts
- Exploitation of latent heat in evaporator decreases with temperature lift

Heat Pump Systems

Heat pump cycle improvements I

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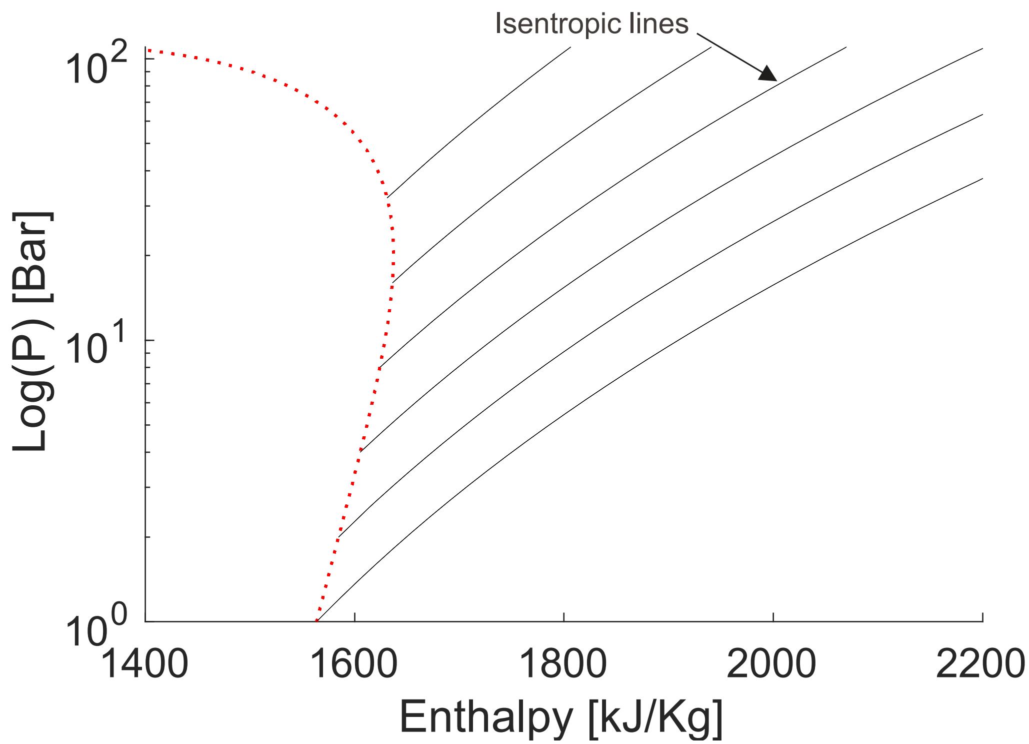
Compression Process

- Isentropic lines in Ph-diagram decrease in slope with increasing distance from saturation line

- Compression process close to saturation line promising means to decrease compression power

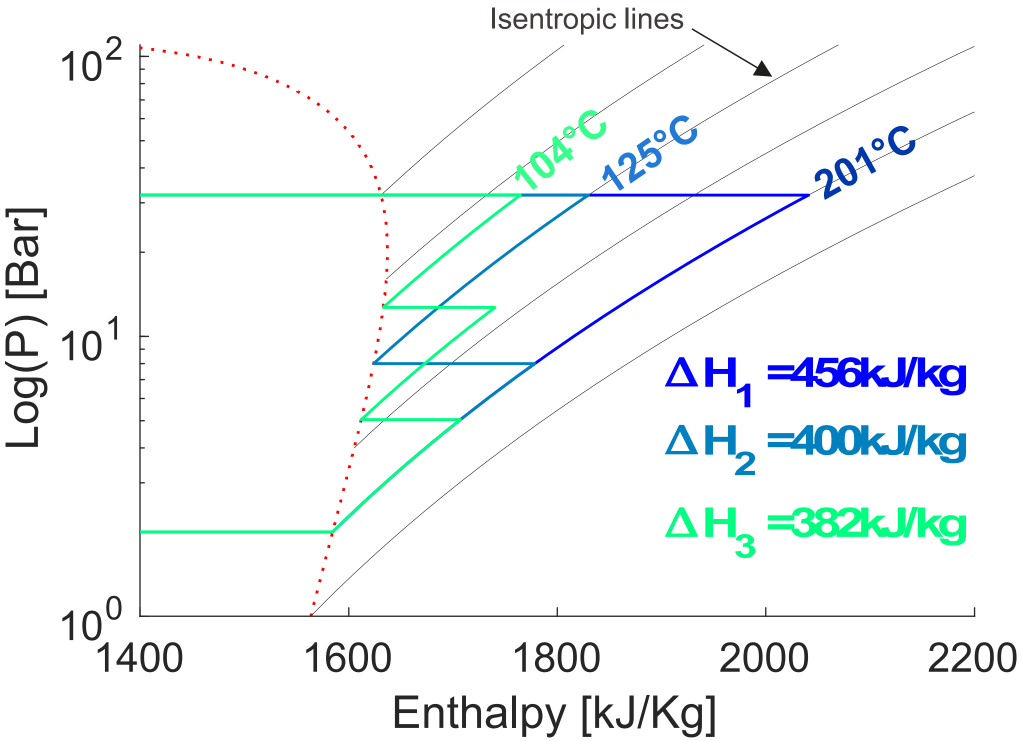


Compression Process

- Assume ideal compression with intercooling

- Splitting compression process with intercooling reduces power to achieve same pressure ratio

- Splitting and intercooling reduces exhaust temperature

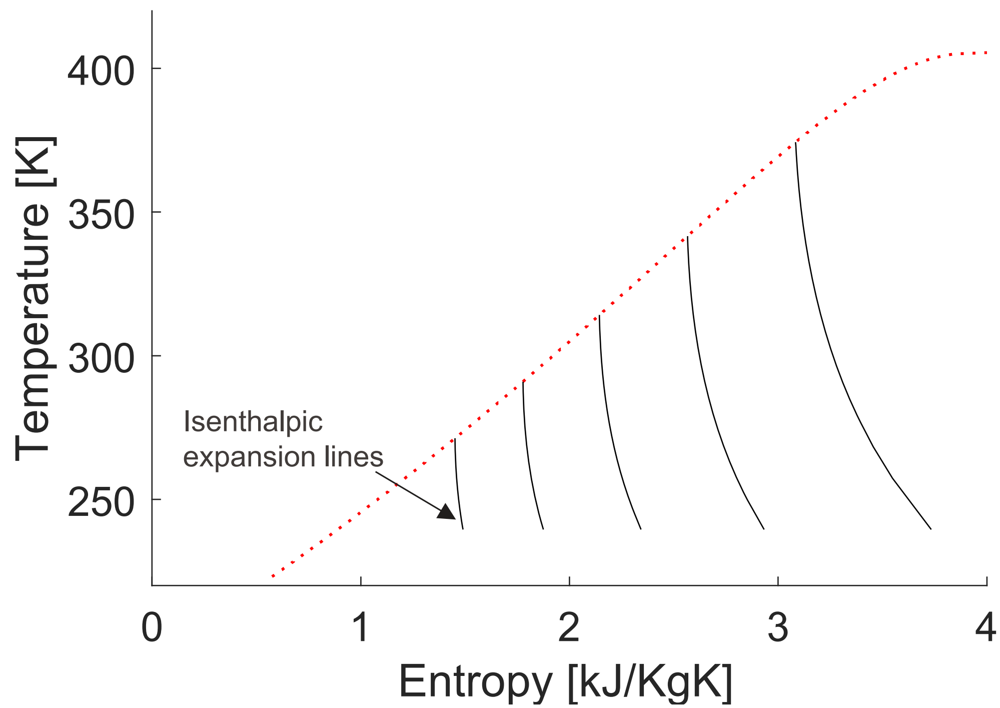


Expansion Process

- Isenthalpic lines in Ts-diagram decrease in slope with increasing expansion from saturation

- Limit expansion process and combining with intercooling is promising means to reduce expansion losses

- Valorize subcooling region from saturation to room temperature



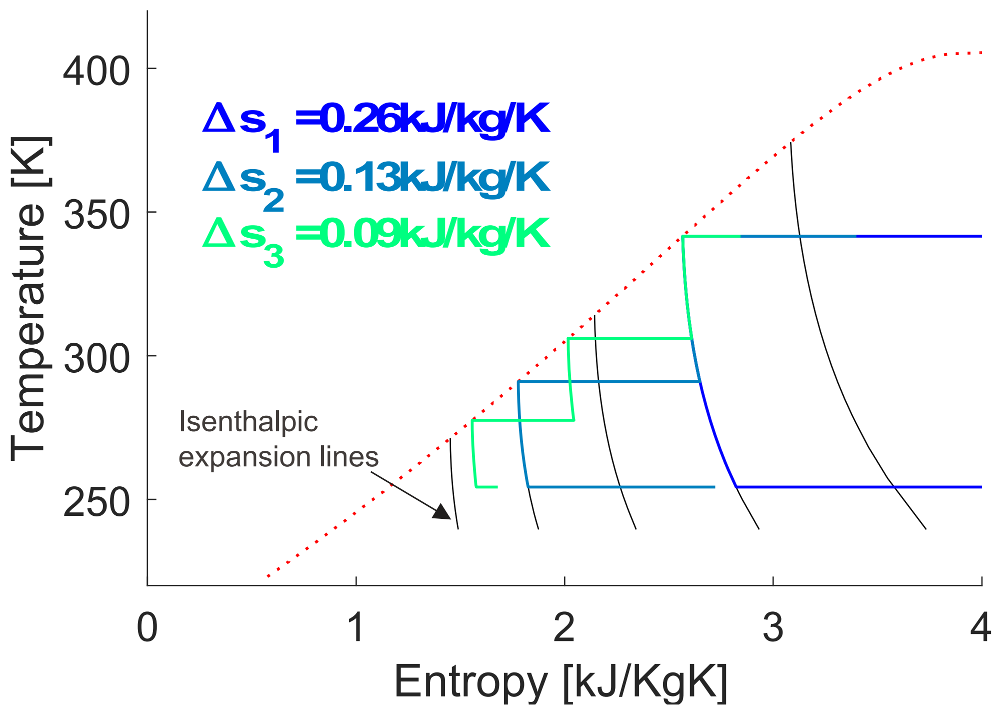
Expansion Process

- Assume isenthalpic expansion with intercooling starting from saturation

- Splitting expansion process reduces losses and increases latent heat pickup in evaporator

- Considering exergy losses savings can be significant in particular in high temperature cycles

$$T_a\left(s_4 - s_3\right) = l_e$$



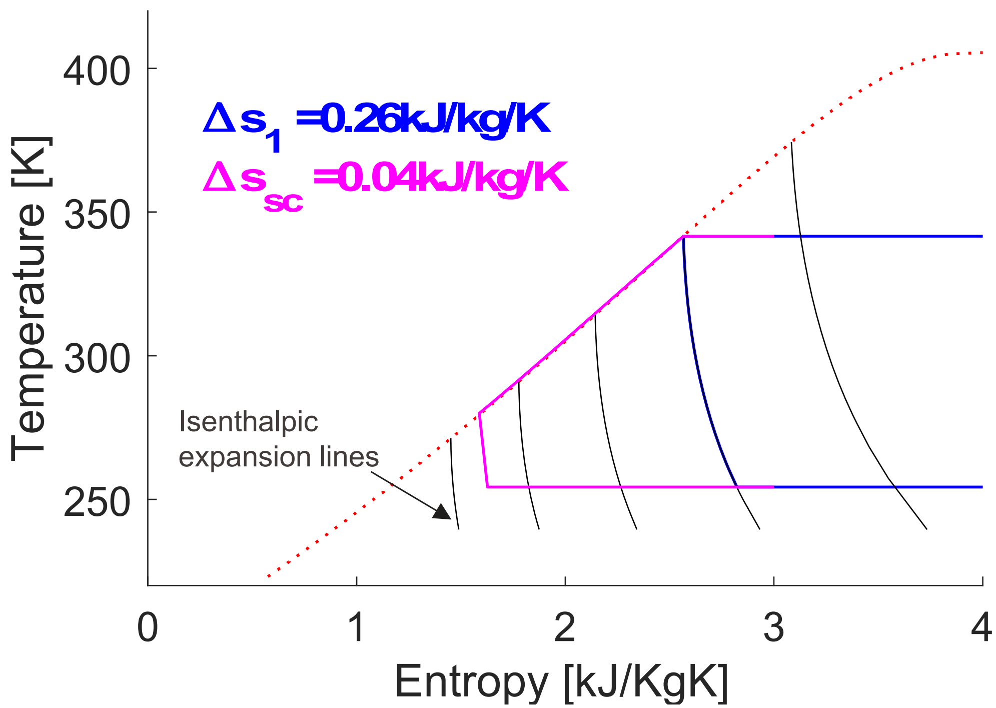
Expansion Process

- Assume isenthalpic expansion with initial subcooling

- Valorization of subcooling region decreases expansion losses

- Subcooling heat difficult to distribute to ambient

- Subcooling used in cycle for compression superheat

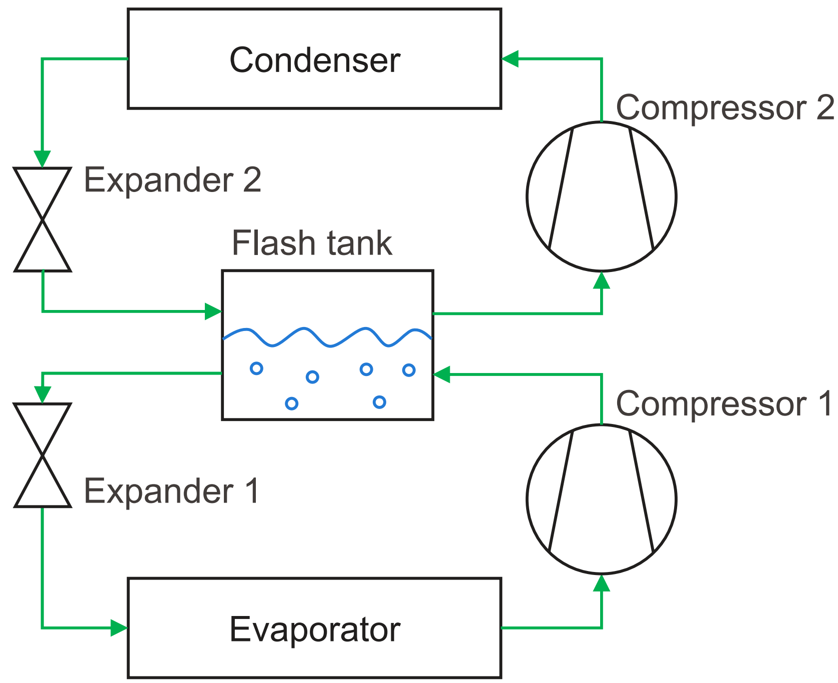


How to Improve HP Cycle?

- Splitting compression process coupled to intercooling
- Use subcooling heat for heating purpose (external) or directly in cycle
- Splitting expansion with intercooling
- Careful heat exchanger design to decrease pinch
- Appropriate selection of working fluid
  - Effect on compressor exhaust temperature
  - Temperature glide may decrease losses in heat exchangers

Implementing Improvements in HP Cycle

- Two stage compression heat pump cycle
  - Two compression stages in series
  - Economizer acts as phase separator (economizer-flash-tank)
  - Requires two expanders
  - Adds 3 components compared to single stage cycle
    → compressor is expensive!



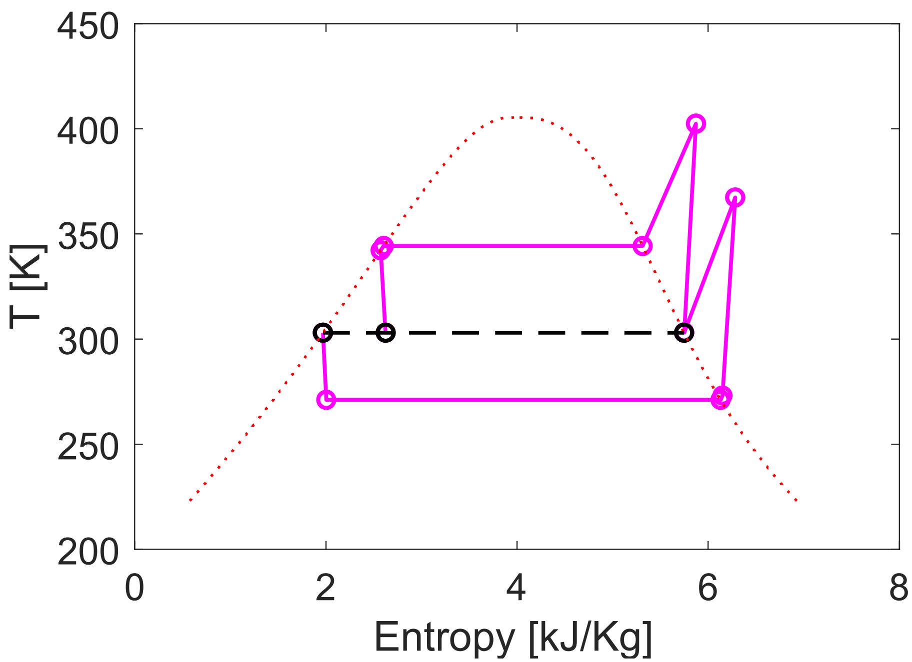
Conditions on Two Stage HP Cycle

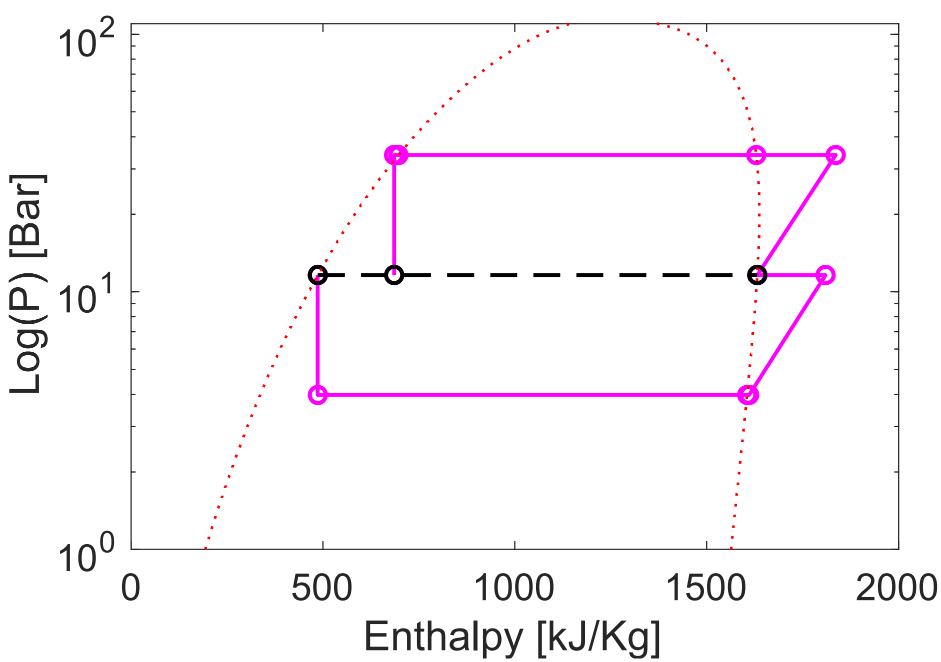
- Two stage vapor compression heat pump
  - Adiabatic compression with isentropic efficiency function of pressure ratio
  - Condensation with 2K subcooling
  - Isenthalpic expansion in valve
  - Evaporation with 2K superheat
  - Cold source is air at 5°C
  - Hot source is water provided at 15 – 90°C, heat rate 10kW
  - Temperature difference on water and air 5K
  - Pinch in condenser and evaporator 2K
  - Open flash tank economizer at intermediate pressure ($PR_1 = PR_2$)
  - Working fluid ammonia

- Assumptions
  - Perfect thermal insulation, negligible dissipation in condenser, evaporator and ducts, steady-state operation

Two Stage Compression HP Cycle

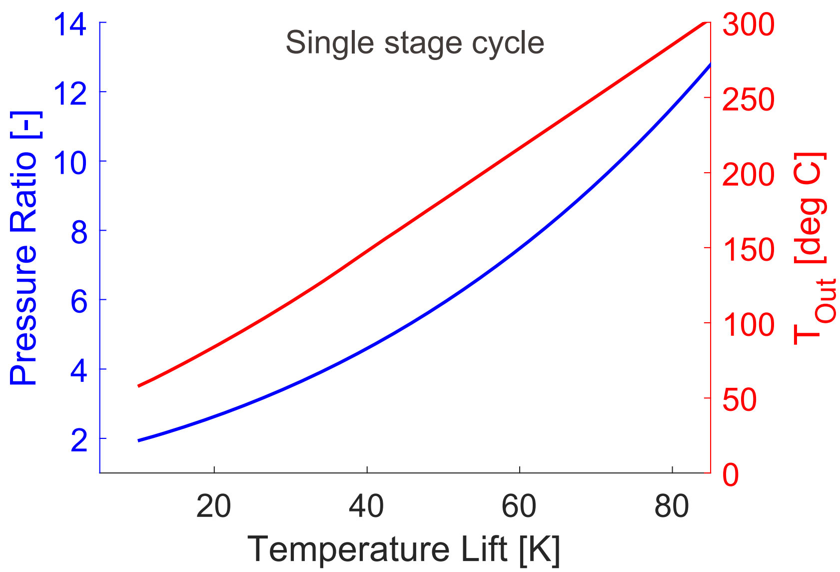
- Two stage vapor compression heat pump, T-s and P-h diagrams

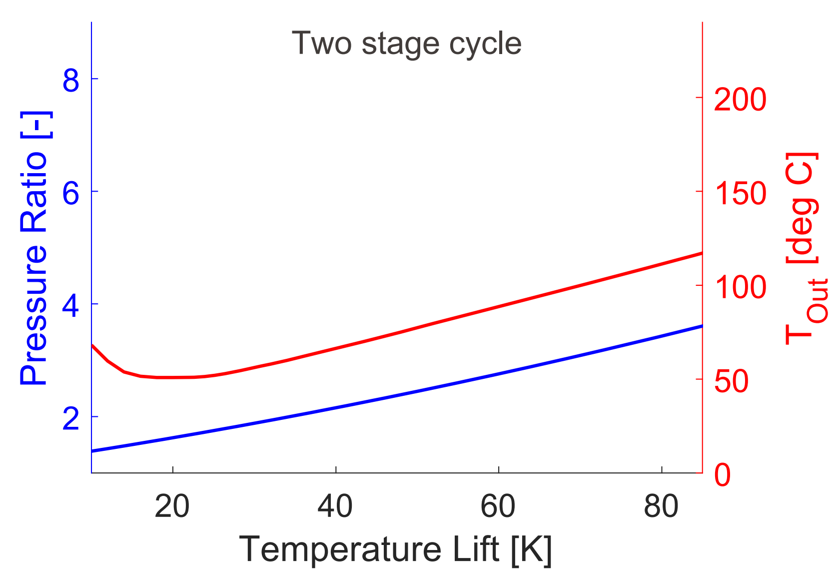




Benefits of Two Stage Compression HP Cycle

- Lower exhaust temperature protects compressor, working fluid and lubricant
- Splitting into two stage reduces pressure ratio across compressors → works at higher efficiency

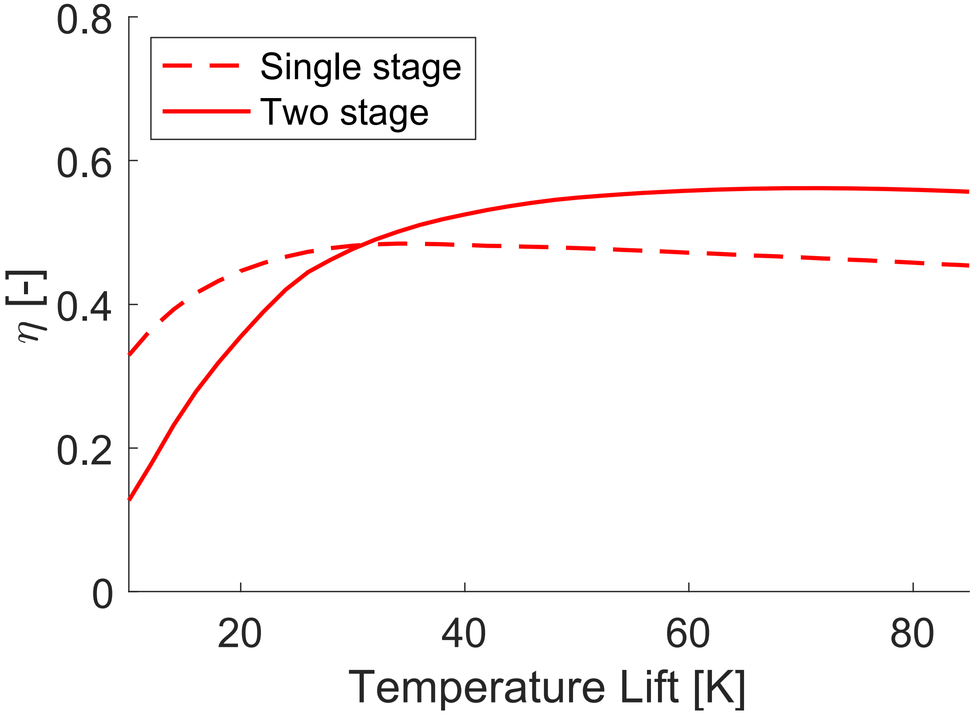




Benefits of Two Stage Compression HP Cycle

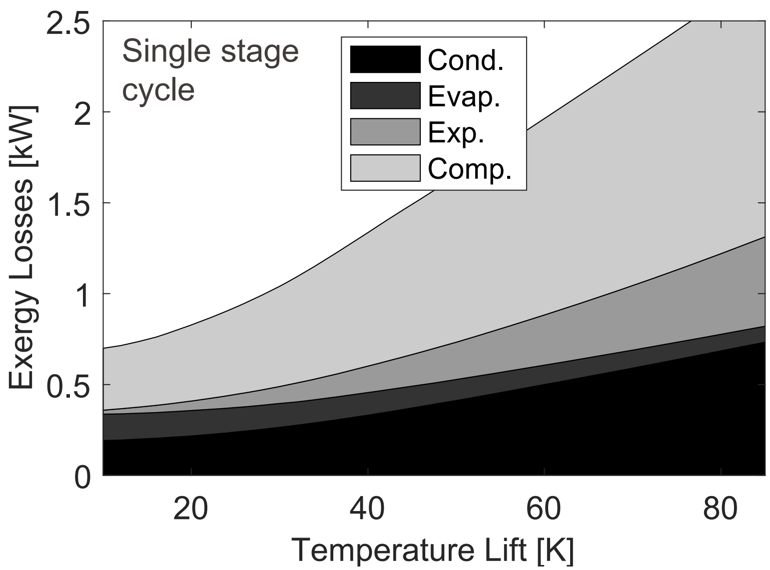
- Efficiency improves by ~20% compared to single stage HP cycle at high temperature lifts

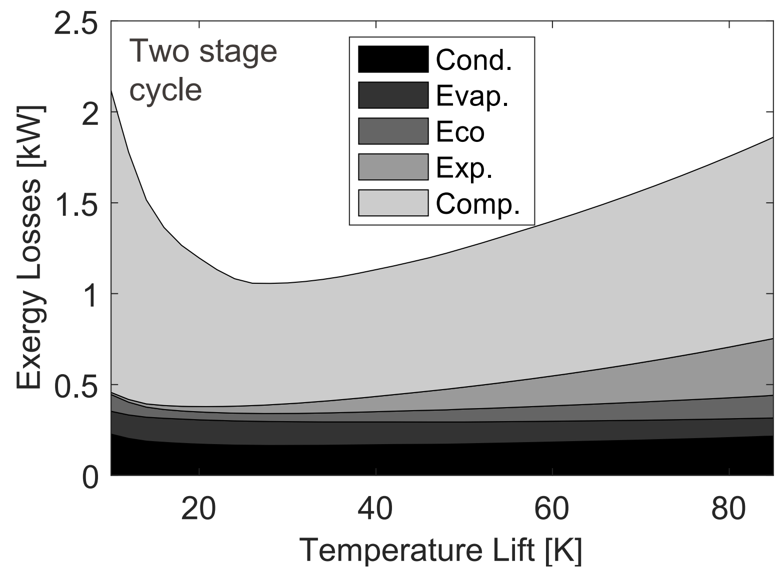
- Two stage cycle makes no sense at low lifts due to low compressor efficiency (over-compression)



Benefits of Two Stage Compression HP Cycle

- Condenser losses decreased with twin stage cycle due to lower temperatures at compressor exhaust
- Expansion losses reduced with two stage cycle
- Compression losses decrease only at high lifts





Limitations of Two Stage HP Cycle

- Suited for high temperature lifts
- Low pressure ratio at low temperature lifts penalize cycle due to low compressor efficiency
- Ratio between lower and upper stage pressure ratio is additional variable to optimize cycle → depends on working fluid
- Higher investment cost due to additional compressor, expander and economizer
- Makes better use of latent heat in evaporator
- Yields lower compressor exhaust temperatures

Outlook for W6

- Practical cycle improvements

Exercises W5

- Theory questions
- A compressor driven heat pump installation