

Advanced Thermodynamics

Homework Assignment #1

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1.

It is assumed the system is a simple compressible system, therefore there are two degrees of freedom and the rest of the intensive properties can be calculated using an equation of state, or found in a table of common fluids.

2.

- Closed system: (a)
- Open system: (c)
- Isolated system: (b)

3.

Answer:

- a. internal energy
- b. kinetic energy
- c. potential energy

Notes:

- f. Electrical energy could potentially be considered if the mass crossing a system boundary is electrically charged, but it would be a rare circumstance.
- e. Flow work is a summation of other components, not necessarily carried by the mass itself.
- c. $E = mc^2$, so an argument could be made but it is not relevant here.

4.

False. Only solids resist shear stress by static deflection. Fluids continuously deform under shear stress.

5.

(c), assuming V_n is the velocity normal to A_c , then the final units are kg/s.

Problem 6: Hydro Storage System Energy Analysis

Given:

- Volumetric flow rate: $\dot{V} = 500 \text{ m}^3/\text{s}$
- Elevation drop: $\Delta z = 85 \text{ m}$
- Water density: $\rho = 1000 \text{ kg/m}^3$ (assumed)

(a) Maximum possible power with upper reservoir as a lake ($V_1 = 0$):

- Apply the Energy Equation:

$$\frac{V_1^2}{2} + gz_1 = \frac{V_2^2}{2} + gz_2 + w_{\text{turbine}}$$

where:

V_1 = velocity at upper reservoir (lake) = 0

V_2 = velocity at lower lake (assume 0)

z_1 = elevation of upper reservoir

z_2 = elevation of lower lake

$g = 9.81 \text{ m/s}^2$

- Solve for work per unit mass:

$$w_{\text{turbine}} = g(z_1 - z_2)$$

- Calculate total power:

$$\dot{W}_{\text{turbine}} = \dot{m} \cdot w_{\text{turbine}}$$

$$\dot{m} = \rho Q = 1000 \text{ kg/m}^3 \times 500 \text{ m}^3/\text{s} = 500,000 \text{ kg/s}$$

$$w_{\text{turbine}} = 9.81 \text{ m/s}^2 \times 85 \text{ m} = 833.85 \text{ J/kg}$$

$$\dot{W}_{\text{turbine}} = 500,000 \text{ kg/s} \times 833.85 \text{ J/kg} = 416,925,000 \text{ W} = \boxed{416.9 \text{ MW}}$$

(b) Maximum possible power with upper reservoir as a river ($V_1 = 3.75 \text{ m/s}$):

- Include kinetic energy at inlet:

$$\begin{aligned} w_{\text{turbine}} &= \frac{V_1^2}{2} + g(z_1 - z_2) \\ &= \frac{(3.75 \text{ m/s})^2}{2} + 9.81 \text{ m/s}^2 \times 85 \text{ m} \\ &= 7.03 \text{ J/kg} + 833.85 \text{ J/kg} = 840.88 \text{ J/kg} \end{aligned}$$

$$\dot{W}_{\text{turbine}} = 500,000 \text{ kg/s} \times 840.88 \text{ J/kg} = 420,440,000 \text{ W} = \boxed{420.4 \text{ MW}}$$

- **Percent increase in power:**

$$\% \text{ increase} = \frac{420.4 - 416.9}{416.9} \times 100\% = \boxed{0.84\%}$$

- **Physical explanation:**

The increase in possible power output is due to the additional kinetic energy of the water entering the turbine when the upper reservoir is a river. This kinetic energy is converted to useful work by the turbine, resulting in a slightly higher power output compared to the case where the upper reservoir is a still lake.

Problem 7: Piston-Cylinder Energy Balance

Given:

- Mass of H₂ gas: $m = 3.8 \text{ kg}$
- Gas constant: $R = 4.124 \text{ kJ/kg}\cdot\text{K}$
- Initial state: $P_1 = 250 \text{ kPa}$, $T_1 = 900 \text{ K}$
- Final temperature: $T_2 = 400 \text{ K}$
- Process: Constant pressure (heat removal)

Part A: Find the change in volume (m³) of the system.

Using the ideal gas law to find the initial volume:

$$\begin{aligned} PV &= mRT \\ V_1 &= \frac{mRT_1}{P} \\ &= \frac{(3.8 \text{ kg})(4.124 \text{ kJ/kg}\cdot\text{K})(900 \text{ K})}{250 \text{ kPa}} \\ &= \boxed{56.416 \text{ m}^3} \end{aligned}$$

For a constant pressure process, $\frac{T_1}{V_1} = \frac{T_2}{V_2}$:

$$\begin{aligned} \frac{900}{56.416} &= \frac{400}{V_2} \\ V_2 &= \frac{400 \times 56.416}{900} = \boxed{25.073 \text{ m}^3} \end{aligned}$$

The change in volume is:

$$\Delta V = V_2 - V_1 = 25.073 - 56.416 = \boxed{-31.343 \text{ m}^3}$$

The negative sign indicates the volume decreased as the gas cooled.

Part B: Find the magnitude (kJ) and direction (in or out of system) of the boundary work during this process. Defend why your answer makes sense.

For a constant pressure process, the boundary work is:

$$\begin{aligned}
 W &= P\Delta V = P(V_2 - V_1) \\
 &= 250 \text{ kPa} \times (25.073 - 56.416) \text{ m}^3 \\
 &= 250 \text{ kPa} \times (-31.343 \text{ m}^3) \\
 &= \boxed{-7,836 \text{ kJ}}
 \end{aligned}$$

The negative sign indicates work is done **by** the system (the gas does work on the surroundings as it contracts). If you define work done **on** the system, $W = +7,836 \text{ kJ}$ into the system. This makes sense because as heat is removed at constant pressure, the gas contracts and the surroundings do work on the gas.

Part C: Calculate the change in specific enthalpy (kJ/kg) and internal energy (kJ/kg) of the gas during this process using the given property tables.

- **Change in specific enthalpy (Δh):**

$$\begin{aligned}
 \Delta h &= \int_{T_1}^{T_2} C_p(T) dT \\
 &\approx \bar{C}_p \times (T_2 - T_1)
 \end{aligned}$$

Where \bar{C}_p is the average of C_p at 900 K and 400 K (read from the graph):

$$\begin{aligned}
 \bar{C}_p &\approx \frac{14.35 + 14.93}{2} = 14.64 \text{ kJ/kg} \cdot \text{K} \\
 \Delta h &= 14.64 \text{ kJ/kg} \cdot \text{K} \times (400 - 900) \text{ K} \\
 &= 14.64 \times (-500) = \boxed{-7,320 \text{ kJ/kg}}
 \end{aligned}$$

- **Change in specific internal energy (Δu):**

$$\begin{aligned}
 \Delta u &= \int_{T_1}^{T_2} C_v(T) dT \\
 &\approx \bar{C}_v \times (T_2 - T_1)
 \end{aligned}$$

Where \bar{C}_v is the average of C_v at 900 K and 400 K (read from the graph, e.g. 10.2 and 10.9):

$$\begin{aligned}
 \bar{C}_v &\approx \frac{10.2 + 10.9}{2} = 10.55 \text{ kJ/kg} \cdot \text{K} \\
 \Delta u &= 10.55 \text{ kJ/kg} \cdot \text{K} \times (400 - 900) \text{ K} \\
 &= 10.55 \times (-500) = \boxed{-5,275 \text{ kJ/kg}}
 \end{aligned}$$

- Both Δh and Δu are negative, indicating the gas lost enthalpy and internal energy as

it cooled from 900 K to 400 K.

Problem 8: Centrifugal Pump Energy Balance

Given:

- Shaft work: $\dot{W}_s = 10 \text{ kW}$
- Mass flow rate: $\dot{m} = 0.05 \text{ kg/s}$
- Water properties: $\rho = 1000 \text{ kg/m}^3$, $C = 4.2 \text{ kJ/kg} \cdot \text{K}$
- Inlet temperature: $T_1 = 20^\circ\text{C}$
- Outlet temperature: $T_2 = 45^\circ\text{C}$
- Cross-sectional areas: $A_{inlet} = A_{outlet}$

Part A: Prove that the change in kinetic energy is negligible

• **Given:**

$$\dot{m} = 0.05 \text{ kg/s}$$

$$A_2 = A_3$$

$$\rho = 1000 \text{ kg/m}^3$$

• **Conservation of mass for steady flow:**

$$\begin{aligned} \dot{m} &= \rho A_2 v_2 = \rho A_3 v_3 \\ \Rightarrow v_2 &= v_3 \end{aligned}$$

• **Change in kinetic energy:**

$$\Delta KE = \frac{1}{2} \dot{m} (v_3^2 - v_2^2) = 0$$

• **Conclusion:** The change in kinetic energy is negligible (zero) for the water in this pump.

Part B: Estimate the heat loss (kW) from the pump

Applying the flow energy equation:

$$\dot{Q}_{\text{loss}} - \dot{W}_s = \dot{m} [\Delta h + \Delta KE + \Delta PE]$$

With the assumptions that $\Delta KE = 0$ (from Part A), $\Delta PE \approx 0$ (not significant), and for incompressible liquid $\Delta h = C\Delta T$, the energy balance becomes:

$$\begin{aligned}\dot{Q}_{\text{loss}} - \dot{W}_s &= \dot{m} \cdot C \cdot (T_2 - T_1) \\ \dot{m} \cdot C \cdot \Delta T &= 0.05 \times 4.2 \times (45 - 20) \\ &= 0.05 \times 4.2 \times 25 \\ &= 5.25 \text{ kW}\end{aligned}$$

Solving for the heat loss:

$$\begin{aligned}\dot{Q}_{\text{loss}} &= \dot{W}_s - \dot{m} \cdot C \cdot \Delta T \\ &= 10 - 5.25 = \boxed{4.75 \text{ kW}}\end{aligned}$$

Therefore, the heat loss from the pump is approximately 4.75 kW.

Problem 9: Isentropic Steam Turbine with Extraction

Given:

- Steam inlet: $\dot{m}_1 = 3 \text{ kg/s}$ at $P_1 = 3 \text{ MPa}$ (saturated vapor)
- Extraction: 12% of flow at $P_2 = 500 \text{ kPa}$
- Exhaust: 88% of flow at $P_3 = 50 \text{ kPa}$, $T_3 = 100^\circ\text{C}$
- Process: Isentropic (reversible adiabatic)

Solution:

The mass flow rates are $\dot{m}_2 = 0.12 \times \dot{m}_1 = 0.36 \text{ kg/s}$ and $\dot{m}_3 = \dot{m}_1 - \dot{m}_2 = 2.64 \text{ kg/s}$.
The steady flow energy equation for the turbine is:

$$\dot{W}_{\text{out}} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$

where h_1 is the specific enthalpy at inlet (3 MPa, saturated vapor), h_2 is at extraction (500 kPa), and h_3 is at exhaust (50 kPa, 100°C).

Steam Properties from CoolProp:

- **State 1 (Inlet):** $P_1 = 3 \text{ MPa}$, saturated vapor

$$T_1 = 507.0 \text{ K} = 234^\circ\text{C}$$

$$h_1 = 2803.2 \text{ kJ/kg}$$

$$s_1 = 6185.6 \text{ J/kg} \cdot \text{K} = 6.186 \text{ kJ/kg} \cdot \text{K}$$

- **State 2 (Extraction):** $P_2 = 500 \text{ kPa}$, $s_2 = s_1$ (isentropic)

$$s_2 = 6185.6 \text{ J/kg} \cdot \text{K}$$

$$h_2 = 2478.2 \text{ kJ/kg}$$

- **State 3 (Exhaust):** $P_3 = 50 \text{ kPa}$, $T_3 = 100^\circ\text{C}$

$$h_3 = 2682.4 \text{ kJ/kg}$$

The power output is:

$$\begin{aligned} \dot{W}_{\text{out}} &= \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3 \\ &= (3.0)(2803.2) - (0.36)(2478.2) - (2.64)(2682.4) \\ &= 8409.6 - 892.2 - 7081.5 \\ &= \boxed{435.9 \text{ kW}} \end{aligned}$$

Problem 10: Real Steam Turbine Entropy Analysis

For this real steam turbine analysis, we have a mass flow rate of $\dot{m} = 2.5$ kg/s with inlet conditions of $P_1 = 3$ MPa and $T_1 = 400^\circ\text{C}$ (superheated steam). The steam expands to an outlet pressure of $P_2 = 100$ kPa with an isentropic efficiency of $\eta_s = 0.85$. The process is adiabatic expansion ($\dot{Q} = 0$) with negligible kinetic and potential energy changes.

PART A: Isentropic Operation

- **Goal:** Find T_{2s} , h_{2s} , x_{2s} (quality), s_{2s}
- **Process:**
 1. Look up inlet state properties (superheated steam at $P_1 = 3$ MPa, $T_1 = 400^\circ\text{C}$): h_1 , s_1 (from steam tables)
 2. Apply isentropic condition: $s_{2s} = s_1$
 3. At $P_2 = 100$ kPa, $s_{2s} = s_1$: compare s_{2s} with s_f and s_g at 100 kPa
 4. If $s_f < s_{2s} < s_g$, two-phase mixture: $x_{2s} = (s_{2s} - s_f)/s_{fg}$
 5. Find enthalpy: $h_{2s} = h_f + x_{2s}h_{fg}$
 6. Temperature: $T_{2s} = T_{sat}$ at 100 kPa
- **Summary for Part A:**
 - $h_1 = 3231.7$ kJ/kg, $s_1 = 6.923$ kJ/kg-K (inlet properties)
 - $T_{2s} = 372.8$ K = 99.6°C
 - $h_{2s} = 2512.6$ kJ/kg
 - $x_{2s} = 0.928$ (92.8% vapor)
 - $s_{2s} = 6.923$ kJ/kg-K (isentropic)

PART B: Real Operation

- **Goal:** Find T_2 , h_2 , x_2 , s_2 (actual outlet conditions)
- **Process:**
 1. Use isentropic efficiency: $\eta_s = (h_1 - h_2)/(h_1 - h_{2s}) = 0.85$
 2. Solve for h_2 : $h_2 = h_1 - \eta_s(h_1 - h_{2s})$
 3. At $P_2 = 100$ kPa, h_2 known: compare h_2 with h_f and h_g at 100 kPa
 4. If $h_f < h_2 < h_g$, two-phase mixture: $x_2 = (h_2 - h_f)/h_{fg}$
 5. Find entropy: $s_2 = s_f + x_2s_{fg}$
 6. Temperature: $T_2 = T_{sat}$ at 100 kPa
- **Summary for Part B:**

- Isentropic work: $w_s = h_1 - h_{2s} = 3231.7 - 2512.6 = 719.1$ kJ/kg
- Actual work: $w_{actual} = \eta_s \times w_s = 0.85 \times 719.1 = 611.2$ kJ/kg
- Actual outlet enthalpy: $h_2 = h_1 - w_{actual} = 3231.7 - 611.2 = 2620.5$ kJ/kg
- $T_2 = 372.8$ K = 99.6°C
- $h_2 = 2620.5$ kJ/kg
- $x_2 = 0.976$ (97.6% vapor)
- $s_2 = 7.213$ kJ/kg-K

PART C: Work Comparison

To calculate and compare the turbine work output, we first find the isentropic work: $w_s = h_1 - h_{2s} = 3231.7 - 2512.6 = 719.1$ kJ/kg. The actual work is $w_{actual} = h_1 - h_2 = 3231.7 - 2620.5 = 611.2$ kJ/kg.

The corresponding power outputs are:

$$\begin{aligned}\dot{W}_s &= \dot{m}w_s = 2.5 \times 719.1 = \boxed{1797.8 \text{ kW}} \\ \dot{W}_{actual} &= \dot{m}w_{actual} = 2.5 \times 611.2 = \boxed{1528.0 \text{ kW}}\end{aligned}$$

The difference is $\Delta \dot{W} = \dot{W}_s - \dot{W}_{actual} = 1797.8 - 1528.0 = \boxed{269.8 \text{ kW}}$. This verifies the calculation since $w_{actual} = \eta_s \times w_s = 0.85 \times 719.1 = 611.2$ kJ/kg.

The real turbine produces 15% less work due to irreversibilities (friction, turbulence, heat transfer within fluid), with the energy lost to entropy generation.

PART D: Entropy Generation Rate

- **Goal:** Calculate \dot{S}_{gen} using entropy balance
- **Given entropy balance:**

$$\sum \frac{\dot{Q}}{T_{boundary}} + \sum \dot{m}s_{in} - \sum \dot{m}s_{out} + \dot{S}_{gen} = \frac{dS_{sys}}{dt}$$

- **For steady flow, well-insulated turbine:**

- $\frac{dS_{sys}}{dt} = 0$ (steady state)
- $\sum \frac{\dot{Q}}{T_{boundary}} = 0$ (adiabatic)
- Single inlet and outlet stream

- **Simplified:**

$$\begin{aligned}0 + \dot{m}s_1 - \dot{m}s_2 + \dot{S}_{gen} &= 0 \\ \Rightarrow \dot{S}_{gen} &= \dot{m}(s_2 - s_1)\end{aligned}$$

- **Calculation:**

$$\begin{aligned}\dot{S}_{gen} &= \dot{m}(s_2 - s_1) = 2.5 \times (7.213 - 6.923) \\ &= 2.5 \times 0.290 = \boxed{0.725 \text{ kW/K}}\end{aligned}$$

- **Check:** $\dot{S}_{gen} = 0.725 > 0$ (confirms 2nd law for real process)

PART E: Comparison and Benefit

Comparing the isentropic (A) versus real (B) turbine properties:

Property	Isentropic (A)	Real (B)	Difference
h_2 (kJ/kg)	2512.6	2620.5	+107.9
s_2 (kJ/kg-K)	6.923	7.213	+0.290
x_2 (quality)	0.928	0.976	+0.048
T_2 (K)	372.8	372.8	0

The key observation is that the real turbine has higher quality ($x_2 > x_{2s}$), higher entropy ($s_2 > s_1$), and higher enthalpy at exit ($h_2 > h_{2s}$). This provides a benefit: less liquid content in the exhaust (higher quality means more vapor), which reduces water droplet erosion on turbine blades. The non-ideal operation trades work output for blade lifespan.

Problem 11: Steam Quality Change in Piston-Cylinder

Given:

- System: Free piston-cylinder (constant pressure process)
- Working fluid: $m = 3$ kg of water/steam mixture
- Pressure: $P = 200$ kPa (constant throughout process)
- Initial state: $x_1 = 0.60$ (40% liquid, 60% vapor)
- Final state: $x_2 = 1.0$ (all liquid evaporated to saturated vapor)
- Process: Adiabatic heating with electric resistance heater

Relevant Equations

- For specific entropy in a two-phase mixture:

$$s = s_f + x \cdot s_{fg}$$

- Specific entropy change:

$$\Delta s = s_2 - s_1$$

For this entropy calculation, the initial state (1) has specific entropy $s_1 = s_f + x_1 \cdot s_{fg} = s_f + (0.60) \cdot s_{fg}$, while the final state (2) has $s_2 = s_g$ since it's saturated vapor. The change in specific entropy is therefore $\Delta s = s_2 - s_1 = s_g - [s_f + (0.60) \cdot s_{fg}]$.

Solution

From saturated steam properties at $P = 200$ kPa: $s_f = 1.530$ kJ/kg·K, $s_g = 7.127$ kJ/kg·K, and $s_{fg} = s_g - s_f = 5.597$ kJ/kg·K.

The initial specific entropy is:

$$\begin{aligned} s_1 &= s_f + x_1 \cdot s_{fg} = 1.530 + (0.60)(5.597) \\ &= 1.530 + 3.358 = \boxed{4.888 \text{ kJ/kg} \cdot \text{K}} \end{aligned}$$

The final specific entropy is $s_2 = s_g = 7.127$ kJ/kg·K for saturated vapor. Therefore, the specific entropy change is:

$$\Delta s = s_2 - s_1 = 7.127 - 4.888 = \boxed{2.239 \text{ kJ/kg} \cdot \text{K}}$$

The positive Δs reflects increased molecular disorder as liquid evaporates to vapor.

Problem 12: Piston-Cylinder Total Entropy Change

Given:

- System: Well-insulated piston-cylinder with freely moving piston
- Working fluid: Water (saturated liquid initially)
- Initial state: $V_1 = 4.8 \text{ L}$, $P_1 = 150 \text{ kPa}$, $x_1 = 0$
- Energy input: $Q_{in} = 1700 \text{ kJ}$ from electric resistance heater
- Process: Constant pressure heating (piston moves freely)

Assumptions

- Well-insulated system \rightarrow no heat loss to surroundings
- Piston moves freely \rightarrow constant pressure process
- Neglect kinetic and potential energy changes
- Quasi-equilibrium process

Relevant Equations

- Energy Balance (First Law):

$$Q_{in} - W_{out} = \Delta U = m(u_2 - u_1)$$

For constant pressure:

$$\begin{aligned} W_{out} &= P(V_2 - V_1) = mP(v_2 - v_1) \\ Q_{in} &= m(u_2 - u_1) + mP(v_2 - v_1) = m(h_2 - h_1) \end{aligned}$$

- Entropy Change:

$$\begin{aligned} \Delta S &= m(s_2 - s_1) \\ \Delta s &= s_2 - s_1 \end{aligned}$$

Solution

The initial state properties at 150 kPa saturated liquid are: $h_1 = h_f = 467.1 \text{ kJ/kg}$, $s_1 = s_f = 1.434 \text{ kJ/kg}\cdot\text{K}$, $\rho_1 = 949.9 \text{ kg/m}^3$, and specific volume $v_1 = 1/\rho_1 = 0.001053 \text{ m}^3/\text{kg}$.

The mass of water is:

$$m = \frac{V_1}{v_1} = \frac{0.0048 \text{ m}^3}{0.001053 \text{ m}^3/\text{kg}} = \boxed{4.558 \text{ kg}}$$

Applying the energy balance for constant pressure:

$$\begin{aligned}
 Q_{in} &= m(h_2 - h_1) \\
 h_2 &= h_1 + \frac{Q_{in}}{m} = 467.1 + \frac{1700}{4.558} \\
 &= 467.1 + 373.0 = \boxed{840.1 \text{ kJ/kg}}
 \end{aligned}$$

Since $h_1 = 467.1 < h_2 = 840.1 < h_g = 2693.1$ kJ/kg (saturated vapor enthalpy), the final state is a two-phase mixture with quality $x_2 = 0.168$ (16.8% vapor) and final entropy $s_2 = 2.404$ kJ/kg·K.

The entropy changes are:

$$\begin{aligned}
 \Delta s &= s_2 - s_1 = 2.404 - 1.434 = \boxed{0.970 \text{ kJ/kg}\cdot\text{K}} \\
 \Delta S &= m \times \Delta s = 4.558 \times 0.970 = \boxed{4.42 \text{ kJ/K}}
 \end{aligned}$$

Problem 13: Solar Energy Storage Entropy Analysis

Given:

- **Ceramic Particles (CARBOBEAD CP):**
 - Density: $\rho = 3270$ kg/m³
 - Specific heat: $C = 1.3$ kJ/kg · K
 - Temperature range: $T_1 = 250^\circ\text{C}$ to $T_2 = 1100^\circ\text{C}$
- **Compressed Air System:**
 - Mass flow rate: $\dot{m} = 0.75$ kg/s
 - Inlet conditions: $T_3 = 250^\circ\text{C}$, $P_3 = 4000$ kPa
 - Heating: $T_4 = 1100^\circ\text{C}$ at constant pressure
 - Turbine expansion: Isentropic to $P_5 = 100$ kPa
- **Air Properties:** Gas constant $R = 0.287$ kJ/kg · K

Assumptions:

- **Ceramic Particles (CARBOBEAD CP):**
 - Incompressible solid with constant specific heat ($C = 1.3$ kJ/kg·K)
 - Uniform heating from 250°C to 1100°C
 - Negligible thermal expansion effects
- **Compressed Air System:**

- Ideal gas behavior with constant gas constant ($R = 0.287 \text{ kJ/kg}\cdot\text{K}$)
- Negligible pressure drop in solar receiver ($P_4 = P_3 = 4000 \text{ kPa}$)
- Constant pressure heating process (3→4)
- Reversible adiabatic (isentropic) turbine expansion (4→5)
- Steady flow conditions throughout

• **Reference State:** Table A-17 reference at 0 K and 100 kPa

• **Negligible Effects:** Kinetic and potential energy changes, heat losses

2 PART A: Change in specific entropy (kJ/kg·K) of ceramic particles from 1 → 2

For an incompressible substance with constant specific heat:

$$\begin{aligned}
 \Delta s_{1 \rightarrow 2} &= C \ln \left(\frac{T_2}{T_1} \right) \\
 &= 1.3 \text{ kJ/kg} \cdot \text{K} \times \ln \left(\frac{1373.15}{523.15} \right) \\
 &= 1.3 \times \ln(2.625) = 1.3 \times 0.965 \\
 &= \boxed{1.254 \text{ kJ/kg} \cdot \text{K}}
 \end{aligned}$$

PART B: Change in specific entropy (kJ/kg·K) of compressed air from 3 → 4

Using CoolProp for air property calculations with state 3 at $T_3 = 523.15 \text{ K}$, $P_3 = 4000 \text{ kPa}$ and state 4 at $T_4 = 1373.15 \text{ K}$, $P_4 = 4000 \text{ kPa}$ (constant pressure heating):

$$\begin{aligned}
 \Delta s_{3 \rightarrow 4} &= s_4 - s_3 \\
 s_3 &= 3389.9 \text{ J/kg} \cdot \text{K} = 3.390 \text{ kJ/kg} \cdot \text{K} \\
 s_4 &= 4468.7 \text{ J/kg} \cdot \text{K} = 4.469 \text{ kJ/kg} \cdot \text{K} \\
 \Delta s_{3 \rightarrow 4} &= 4.469 - 3.390 = \boxed{1.079 \text{ kJ/kg} \cdot \text{K}}
 \end{aligned}$$

PART C: Specific entropy (kJ/kg·K) of air at state 4

From Part B calculation:

$$s_4 = \boxed{4.469 \text{ kJ/kg} \cdot \text{K}}$$

PART D: Work (kW) produced via turbine from 4 → 5

For isentropic expansion ($s_5 = s_4$), we solve for T_5 at $P_5 = 100 \text{ kPa}$ using property_solver.py to find $T_5 = 529.7 \text{ K}$ where $s_5 = 4468.7 \text{ J/kg}\cdot\text{K}$.

Using CoolProp for enthalpy property lookup:

$$h_4 = 1612.8 \text{ kJ/kg} \quad (\text{at } 1373.15 \text{ K, } 4000 \text{ kPa})$$

$$h_5 = 660.1 \text{ kJ/kg} \quad (\text{at } 529.7 \text{ K, } 100 \text{ kPa})$$

$$w_t = h_4 - h_5 = 1612.8 - 660.1 = 952.7 \text{ kJ/kg}$$

$$\dot{W}_{turb} = \dot{m} \times w_t = 0.75 \times 952.7 = \boxed{714.5 \text{ kW}}$$

Problem 14: Concentrated Solar Power Carnot Analysis

Given:

- **Solar Concentrator System:**

- Concentration ratios: $C = 100, 500, 1000, 2000, 3000$ suns
- Solar irradiance: $I = 1000 \text{ W/m}^2$
- Stefan-Boltzmann constant: $\sigma = 5.67 \times 10^{-8} \text{ W/(m}^2 \cdot \text{K}^4)$

- **Carnot Heat Engine:**

- Cold reservoir: $T_L = 300 \text{ K}$ (ambient temperature)
- Hot reservoir: T_H varies (300 K to 2600 K)

Assumptions:

- **Ideal Carnot cycle:** Reversible processes, no irreversibilities in heat engine
- **Steady-state operation:** Constant solar irradiance and ambient conditions
- **Perfect solar concentration:** All concentrated solar energy reaches receiver surface
- **Black-body radiation:** Receiver emits according to Stefan-Boltzmann law
- **Ambient cold reservoir:** $T_L = 300 \text{ K}$ (environmental heat sink)
- **Negligible convection losses:** Only radiative losses from receiver considered
- **Uniform receiver temperature:** Isothermal hot reservoir at T_H

PART A: Plot Total Efficiency

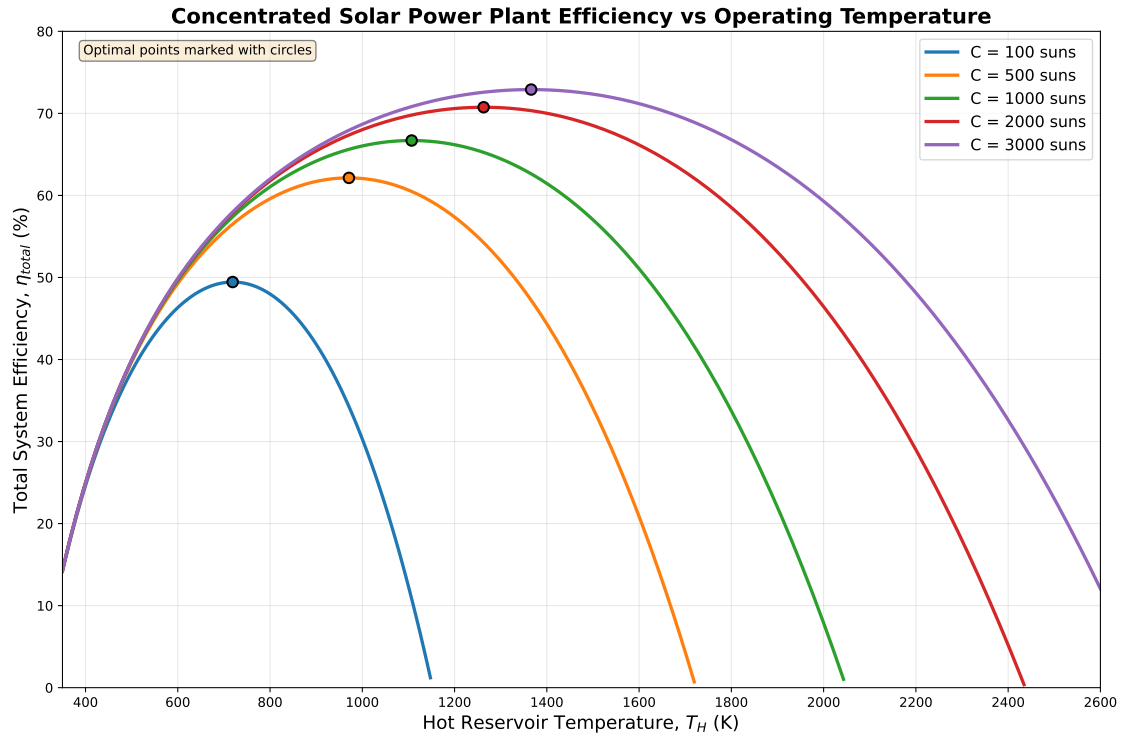


Figure 1: Total system efficiency vs hot reservoir temperature for different concentration ratios. Optimal operating points are marked with circles.

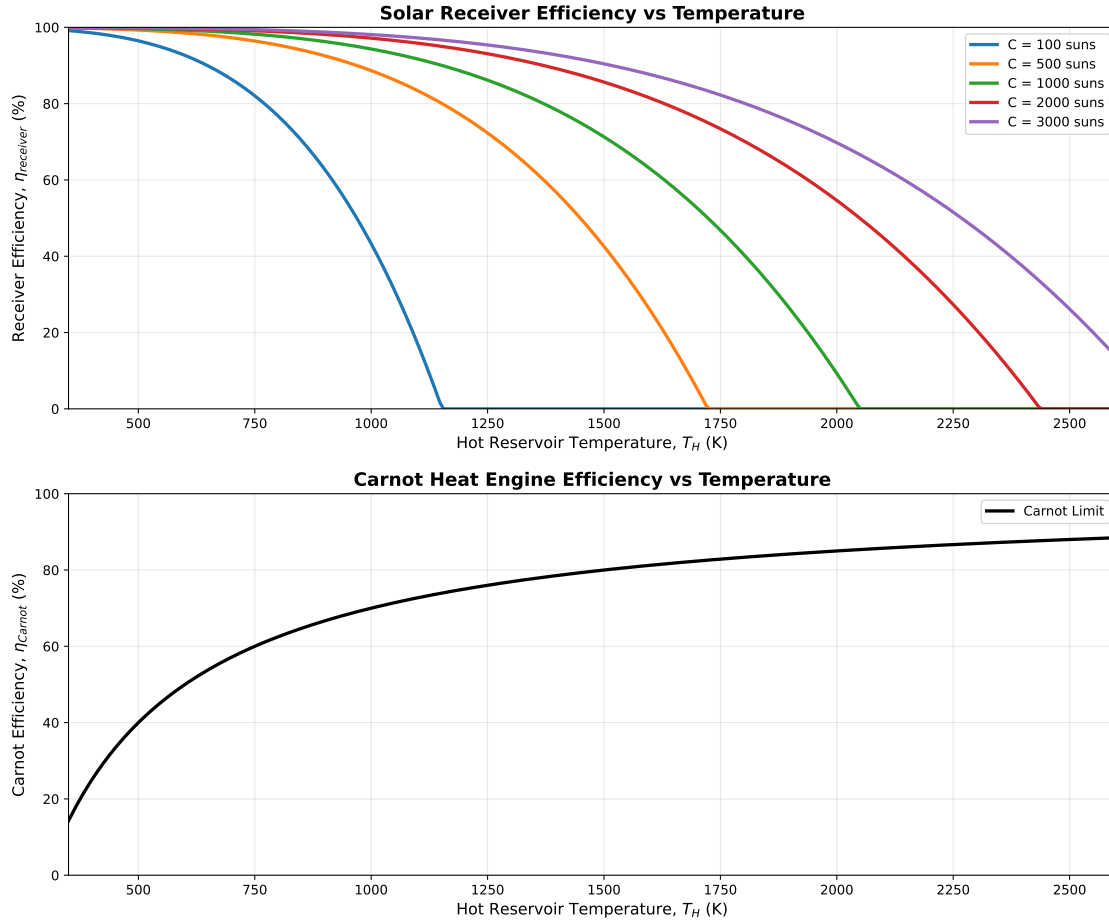


Figure 2: Component efficiencies: (top) Solar receiver efficiency decreases with temperature due to radiation losses, (bottom) Carnot heat engine efficiency increases with temperature.

Relevant Equations

- Receiver Efficiency:

$$\eta_{receiver} = 1 - \frac{\sigma T_H^4}{C \cdot I}$$

- Carnot Heat Engine Efficiency:

$$\eta_{Carnot, HE} = 1 - \frac{T_L}{T_H}$$

- Total System Efficiency:

$$\begin{aligned} \eta_{Total} &= \eta_{receiver} \cdot \eta_{Carnot, HE} \\ &= \left(1 - \frac{5.67 \times 10^{-8} \cdot T_H^4}{C \cdot 1000} \right) \cdot \left(1 - \frac{300}{T_H} \right) \end{aligned}$$

Solution using Python Analysis

- **Python Scripts:** `carnot_simple.py` for optimization, `carnot_plotter.py` for visualization
- **Method:** Temperature optimization scan to find maximum η_{total} for each concentration ratio

Results - Optimal Operating Conditions:

C [suns]	$T_{H,opt}$ [K]	$T_{H,opt}$ [°C]	$\eta_{total,max}$	$\eta_{receiver}$	η_{Carnot}
100	719	446	49.4%	84.8%	58.3%
500	971	698	62.1%	89.9%	69.1%
1000	1107	834	66.7%	91.5%	72.9%
2000	1263	990	70.7%	92.8%	76.2%
3000	1366	1093	72.9%	93.4%	78.0%

Sample Data for Plotting:

T_H [K]	$C = 100$	$C = 500$	$C = 1000$	$C = 2000$	$C = 3000$
500	0.386	0.397	0.399	0.399	0.400
750	0.492	0.578	0.589	0.595	0.596
1000	0.303	0.621	0.660	0.680	0.687
1250	–	0.550	0.655	0.707	0.725
1500	–	0.341	0.570	0.685	0.723
1750	–	–	0.388	0.608	0.682
2000	–	–	0.079	0.464	0.593

PART B: Identify Optimal Receiver Temperature

Key Trends Observed:

1. Optimal Temperature Increases with Concentration:

- $C = 100$ suns: $T_{H,opt} = 719$ K (446°C)
- $C = 3000$ suns: $T_{H,opt} = 1366$ K (1093°C)
- Higher concentration allows operation at higher temperatures

2. Maximum Efficiency Increases with Concentration:

- $C = 100$ suns: $\eta_{max} = 49.4\%$
- $C = 3000$ suns: $\eta_{max} = 72.9\%$
- Improvement of 23.5 percentage points

3. **Physical Trade-offs:** Carnot efficiency increases with hot reservoir temperature, while receiver efficiency decreases due to radiative losses scaling as T_H^4 . The optimal operating point balances these competing effects, and higher concentration ratios shift this optimum to higher temperatures and system efficiencies.