

1 1. Specific Heat Ratio

The specific heat ratio is defined as $k = C_p/C_v$. Based on what you know regarding the differences between the types of specific heats do you anticipate it to be greater than, equal to, or less than one. Why?

Solution

I expect it to be greater than one. C_p accounts for both internal energy changes and boundary work as a result of volume changes. C_v accounts for only changes to internal energy. Therefore C_p is greater than C_v , and the ratio is > 1 .

2 2. Isentropic Compressor/Turbine Relationships

Using the T-ds equations and ideal-gas relationships and assuming constant specific heats, derive the predictive relationships for the performance of isentropic compressors / turbines for ideal gases. Additionally comment on the weaknesses of this prediction and where you would differ.

Key relationships

- $\frac{T_2}{T_1} = f(P_2, P_1, k)$
- $\frac{T_2}{T_1} = f(v_2, v_1, k)$
- $\frac{P_2}{P_1} = f(v_1, v_2, k)$

Derivation (sketch)

From the T-ds equation for an ideal gas with constant specific heats and $ds = 0$ for an isentropic process:

$$\begin{aligned} 0 &= C_v dT + P dv \\ &= C_v dT + \frac{RT}{v} dv \end{aligned}$$

Integrating and manipulating with $R = C_p - C_v$ and $k = C_p/C_v$ yields the standard results:

$$\begin{aligned} \frac{T_2}{T_1} &= \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} \\ \frac{T_2}{T_1} &= \left(\frac{v_1}{v_2} \right)^{k-1} \\ \frac{P_2}{P_1} &= \left(\frac{v_1}{v_2} \right)^k \end{aligned}$$

Commentary

These results require constant specific heats and reversible (isentropic) operation — both are approximations. Real compressors and turbines deviate due to friction, heat transfer, and variable thermophysical properties.

3 3. Insulated Box Equilibrium

An insulated box is initially divided into two halves by a frictionless, thermally conductive actuated wall. The piston is suddenly released and equilibrium between the two gases is fully attained.

Initial Conditions

- **Left side (Air):** $V = 1.5 \text{ m}^3, P = 4\text{bar}, T = 400K$
- **Right side (Also Air):** $V = 1.5 \text{ m}^3, P = 2\text{bar}, T = 400K$
- Wall is suddenly removed

Part A: Final Temperature

The system is in free expansion (no work) and insulated (no heat) therefore internal energy is conserved and the temperature is constant: $T_f = 400 \text{ K}$.

Part B: Final Pressure

Part C: Energy Production

$$\Delta U = Q - W = 0$$

Therefore, the energy production is zero.

Part D: Entropy Production

Since T is constant,

$$\Delta S = nR \ln \left(\frac{V_f}{V_i} \right)$$

Numerical values give $\Delta S_{total} \approx 0.0156 \text{ kJ/K}$, arising from irreversible mixing and pressure equalization.

4 4. Compressor and Heat Exchanger

Air flows through a compressor and heat exchanger. A separate liquid water stream also flows through the heat exchanger. The system operates at pseudo-steady state. Assume individual components are well insulated relative to the environment.

Given Conditions Air Stream:

- Inlet: $P = 96 \text{ kPa}, T = 27^\circ\text{C}, \dot{Q} = 26.91 \text{ m}^3/\text{min}$
- Compressor outlet: $P = 230 \text{ kPa}, T = 127^\circ\text{C}$
- Heat exchanger outlet: Assume Isobaric, $T = 77^\circ\text{C}$

Water Stream:

- Inlet: $P_{atm}, T = 25^\circ\text{C}$
- Outlet: Assume Isobaric, $T = 40^\circ\text{C}$

PART A

Determine the required compressor power and the mass flow rate of cooling water.

Key formulas

$$\dot{W}_{comp} = \dot{m}_{air}(h_2 - h_1)$$

$$\dot{m}_{water} = \frac{\dot{m}_{air}(h_2 - h_3)}{c_{p,water}(T_{w,out} - T_{w,in})}$$

Numerical values used:

- $h_1 = 426,309.837 \text{ J/kg}$
- $h_2 = 527,107.689 \text{ J/kg}$
- $h_3 = 476,472.292 \text{ J/kg}$
- $\rho_{inlet} = 1.115122 \text{ kg/m}^3, \dot{Q} = 26.91 \text{ m}^3/\text{min} \Rightarrow \dot{m}_{air} = 0.5 \text{ kg/s}$
- $h_{w,i} = 104,920.1198 \text{ J/kg}, h_{w,o} = 167,616.2859 \text{ J/kg}$
- $c_{p,water} = 4180 \text{ J/(kg K)}$

Results

$$\dot{m}_{water} = 0.4403 \text{ kg/s}$$

$$\dot{W}_{comp} = 50.4 \text{ kW}$$

PART B

Entropy production rates:

$$\dot{S}_{gen,comp} = \dot{m}_{air}(s_2 - s_1) = 19.5535 \text{ W/K}$$

$$\dot{S}_{gen,HX} = \dot{m}_{air}(s_3 - s_2) + \dot{m}_w(s_{w,o} - s_{w,i}) \approx 22.724 \text{ W/K}$$

PART C

- Compressor irreversibilities: friction, non-ideal compression, finite temperature differences.
- Heat exchanger irreversibilities: finite temperature differences driving heat transfer.
Dead state ($P_0 = 100 \text{ kPa}, T_0 = 25^\circ\text{C} = 298.15 \text{ K}$):

$$h_0 = 424.55 \text{ kJ/kg}$$

Specific exergy calculation:

$$\begin{aligned}\psi_{R134a} &= (h_1 - h_0) - T_0(s_1 - s_0) \\ &= (570.78 - 424.55) - 298.15(2.1283 - 1.9017) \\ &= 146.23 - 298.15(0.2266) \\ &= 146.23 - 67.57 \\ &= 78.66 \text{ kJ/kg}\end{aligned}$$

Conclusion

Steam has significantly higher exergy:

- Steam: $\psi = 794.55 \text{ kJ/kg}$
- R-134a: $\psi = 78.66 \text{ kJ/kg}$

Answer: Steam (water) has the capability to produce approximately 10 times more work than R-134a when brought to the environmental dead state.

This is because steam at these conditions has a much larger enthalpy difference from the dead state, despite both substances starting at the same pressure and temperature. This is largely due to water's specific heat capacity being so large.

5 5.

6 5. Turbine Between Two Tanks

Entropy and Other Things

A turbine is located between two tanks. Initially the smaller tank is pressurized while the larger tank is fully evacuated. Assume the heat transfer with the surroundings is negligible. Steam is allowed to flow from the smaller tank, through the turbine, and into the larger tank until equilibrium is attained. If the turbine is ideal during its expansion process, determine:

Initial Conditions

- Small Tank (100 m^3):
 - At $t = 0$: Steam at 3.0 MPa and 280°C
- Large Tank (1000 m^3):
 - At $t = 0$: Pure Vacuum

PART A

What are the equilibrium conditions in both tanks (T, P, m_s)?

Detailed Calculations with Proper Notation

Step 1: Initial Properties of Small Tank

$$T_1 = 280^\circ\text{C} = 553.15 \text{ K}$$

$$P_1 = 3.0 \text{ MPa} = 3,000,000 \text{ Pa}$$

$$V_{\text{small}} = 100 \text{ m}^3$$

$$\rho_1 = 12.9598 \text{ kg/m}^3$$

$$u_1 = 2,710,698.33 \text{ J/kg}$$

$$v_1 = \frac{1}{\rho_1} = \frac{1}{12.9598} = 0.07716 \text{ m}^3/\text{kg}$$

Step 2: Initial Mass in Small Tank

$$m_1 = \frac{V_{\text{small}}}{v_1} = \frac{100}{0.07716} = 1,296.0 \text{ kg}$$

Step 3: Conservation Principles

Conservation of Mass:

$$m_{\text{total}} = m_1 = 1,296.0 \text{ kg}$$

Conservation of Energy (adiabatic system, rigid tanks):

$$U_{\text{initial}} = U_{\text{final}}$$

$$m_1 u_1 = m_{\text{total}} u_f$$

$$u_f = \frac{m_1 u_1}{m_{\text{total}}} = u_1 = 2,710,698.33 \text{ J/kg}$$

Step 4: Final Specific Volume

$$V_{\text{total}} = V_{\text{small}} + V_{\text{large}} = 100 + 1,000 = 1,100 \text{ m}^3$$

$$v_f = \frac{V_{\text{total}}}{m_{\text{total}}} = \frac{1,100}{1,296.0} = 0.8488 \text{ m}^3/\text{kg}$$

$$\rho_f = \frac{1}{v_f} = \frac{1}{0.8488} = 1.1781 \text{ kg/m}^3$$

$$\rho = 1.1781 \text{ kg/m}^3$$

$$u = 2,710,698.33 \text{ J/kg}$$

$$T_f = 510.9918 \text{ K} = 237.84^\circ\text{C}$$

$$P_f = 274,926.80 \text{ Pa} = 0.27493 \text{ MPa}$$

$$m_{\text{small,final}} = \frac{V_{\text{small}}}{v_f} = \frac{100}{0.8488} = 117.8 \text{ kg}$$

$$m_{\text{large,final}} = \frac{V_{\text{large}}}{v_f} = \frac{1,000}{0.8488} = 1,178.2 \text{ kg}$$

Check:

$$m_{\text{small,final}} + m_{\text{large,final}} = 117.8 + 1,178.2 = 1,296.0 \text{ kg}$$

PART B

What is the maximum theoretical work that could be extracted from the turbine?

Step 1: Identify the Process

For an ideal turbine operating adiabatically and reversibly, the process is isentropic (constant entropy).

The maximum theoretical work occurs when the turbine operates reversibly between the initial state and the final state at the same specific entropy.

Step 2: Initial State Properties

From the initial conditions in the small tank:

$$P_1 = 3.0 \text{ MPa} = 3,000,000 \text{ Pa}$$

$$T_1 = 280^\circ\text{C} = 553.15 \text{ K}$$

$$u_1 = 2,710,698.33 \text{ J/kg}$$

$$s_1 = 6,448.57 \text{ J/(kg K)}$$

Step 3: Final State Properties (Isentropic Expansion)

The steam expands isentropically to the final density:

$$\rho_f = 1.1781 \text{ kg/m}^3$$

$$s_2 = s_1 = 6,448.57 \text{ J/(kg K)}$$

From CoolProp output at $s = 6,448.57 \text{ J/(kg K)}$ and $\rho = 1.1781 \text{ kg/m}^3$:

$$T_2 = 390.30 \text{ K} = 117.15^\circ\text{C}$$

$$P_2 = 181,372.43 \text{ Pa} = 0.18137 \text{ MPa}$$

$$u_2 = 2,270,315.48 \text{ J/kg}$$

Step 4: Calculate Maximum Work per Unit Mass

$$w_{\text{turbine}} = u_1 - u_2$$

$$w_{\text{turbine}} = 2,710,698.33 - 2,270,315.48$$

$$w_{\text{turbine}} = 440,382.85 \text{ J/kg}$$

Step 5: Total Mass Flow Through Turbine

$$m_{\text{flow}} = m_1 = 1,296.0 \text{ kg}$$

Step 6: Maximum Total Work Output

$$W_{\max} = m_{\text{flow}} \times w_{\text{turbine}}$$

$$W_{\max} = 1,296.0 \times 440,382.85$$

$$W_{\max} = 570,736,254 \text{ J}$$

$$W_{\max} = 570.74 \text{ MJ}$$

Final Answer

The maximum quantity of theoretical work that could be extracted from the turbine is 570.74 MJ (or approximately 571 MJ).

7 6. Desuperheating Process

Liquid water is injected into a superheated vapor to produce a saturated vapor.

Given Conditions

- Injected Water:

- P = 0.3 MPa
- $\dot{m} = 6.37 \text{ kg/min}$

- Steam Inlet:

- P = 0.3 MPa
- T = 200 °C

- Vapor Outlet:

- P = 0.3 MPa (Saturated Vapor)

PART A

Determine the mass flow rate of the superheated vapor stream

$$\begin{aligned}\dot{m}_{steam} &= \frac{\dot{m}_{water} \times (h_{sat,vapor} - h_{water})}{h_{steam} - h_{sat,vapor}} \\ &= \frac{6.37 \text{ kg/min} \times (2724882.6303 - 561426.6778) \text{ J/kg}}{(2865890.6540 - 2724882.6303) \text{ J/kg}} \\ &= 97.7 \text{ kg/min}\end{aligned}$$

PART B

Determine the rate of entropy production within the system.

$$\begin{aligned}\dot{S}_{gen} &= (\dot{m}_{water} + \dot{m}_{steam}) s_{sat,vapor} - \dot{m}_{water} s_{water} - \dot{m}_{steam} s_{steam} \\ &= (6.37 + 97.7) \text{ kg/min} \times 6991.6171 \text{ J/(kg K)} \\ &\quad - 6.37 \text{ kg/min} \times 1671.7187 \text{ J/(kg K)} \\ &\quad - 97.7 \text{ kg/min} \times 7313.1252 \text{ J/(kg K)} \\ &= 2.48 \times 10^3 \text{ J/(min K)} \\ \dot{S}_{gen} &= 0.0413 \text{ kW/K}\end{aligned}$$

8 7.

9 8. Mixing Chamber

Exergy

Liquid water is heated using a mixing chamber that combines it with superheated steam at a constant pressure. Heat loss to the environment is tracked.

Given Conditions

- **Liquid Water Inlet:**

- $T = 15 \text{ } ^\circ\text{C}$
- $\dot{m} = 4 \text{ kg/s}$

- **Steam Inlet:**

- $T = 200 \text{ } ^\circ\text{C}$
- $P = 200 \text{ kPa}$ (mixing chamber pressure)

- **Mixed Outlet:**

- $T = 80 \text{ } ^\circ\text{C}$
- $P = 200 \text{ kPa}$

- **Heat Loss:**

- $\dot{Q} = 600 \text{ kJ/min}$

PART A

Determine the mass flow rate of the superheated steam.

1. Control volume + assumptions

- Steady state mixing chamber at constant pressure, $P = 200 \text{ kPa}$
- Neglect KE/PE changes
- One heat interaction (loss): $\dot{Q} = -600 \text{ kJ/min} = -10 \text{ kW}$

2. Property states (use steam tables / IF97)

- Liquid inlet (compressed liquid, use saturated-liquid approx at 15°C):
 - $h_1 \approx h_f(15^\circ\text{C})$
 - $s_1 \approx s_f(15^\circ\text{C})$
- Steam inlet (superheated at $T_2 = 200^\circ\text{C}$, $P_2 = 200 \text{ kPa}$):
 - $h_2 = h(200^\circ\text{C}, 200 \text{ kPa})$
 - $s_2 = s(200^\circ\text{C}, 200 \text{ kPa})$
- Mixed outlet (at $T_3 = 80^\circ\text{C}$, $P_3 = 200 \text{ kPa}$):
 - Check phase: at 200 kPa , $T_{sat} \approx 120.2^\circ\text{C}$, so $T_3 < T_{sat} \Rightarrow$ **compressed (subcooled) liquid**
 - $h_3 \approx h_f(80^\circ\text{C})$
 - $s_3 \approx s_f(80^\circ\text{C})$

3. Mass balance

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

4. Steady-flow energy balance

$$0 = \dot{Q} + \dot{m}_1 h_1 + \dot{m}_2 h_2 - \dot{m}_3 h_3$$

Substitute $\dot{m}_3 = \dot{m}_1 + \dot{m}_2$:

$$0 = \dot{Q} + \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3$$

Solve for \dot{m}_2 :

$$\dot{m}_2(h_2 - h_3) = \dot{m}_1(h_3 - h_1) - \dot{Q}$$

$$\boxed{\dot{m}_2 = \frac{\dot{m}_1(h_3 - h_1) - \dot{Q}}{h_2 - h_3}}$$

Convert to kJ-based units:

- $h_1 = 63.171 \text{ kJ/kg}$
- $h_2 = 2870.730 \text{ kJ/kg}$
- $h_3 = 335.134 \text{ kJ/kg}$

Compute:

$$\dot{m}_1(h_3 - h_1) = 4(335.134 - 63.171) = 4(271.963) = 1087.85 \text{ kW}$$

Since $\dot{Q} = -10 \text{ kW}$ (heat loss),

$$\dot{m}_1(h_3 - h_1) - \dot{Q} = 1087.85 - (-10) = 1097.85 \text{ kW}$$

Denominator:

$$h_2 - h_3 = 2870.730 - 335.134 = 2535.596 \text{ kJ/kg}$$

Therefore:

$$\boxed{\dot{m}_2 = \frac{1097.85}{2535.596} = 0.433 \text{ kg/s}}$$

PART B

Determine the rate of lost work potential caused by this mixing process.
The **lost work potential rate** is the **exergy destruction rate**:

$$\dot{W}_{lost} = \dot{X}_{dest} = T_0 \dot{S}_{gen}$$

where T_0 is the environment (dead-state) temperature in K.

1. Choose dead state

$$T_0 = 25^\circ C = 298.15 \text{ K}, \quad p_0 = 100 \text{ kPa} \text{ (or local atmospheric)}$$

2. Entropy rate balance (steady CV)

$$0 = \sum \frac{\dot{Q}_j}{T_j} + \sum \dot{m}_{in} s_{in} - \sum \dot{m}_{out} s_{out} + \dot{S}_{gen}$$

Here there is one heat loss to the surroundings. If the heat crosses the boundary at approximately the ambient boundary temperature, take $T_b \approx T_0$:

$$\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 - \frac{\dot{Q}}{T_b}$$

with $\dot{m}_3 = \dot{m}_1 + \dot{m}_2$ and $\dot{Q} = -10$ kW. **Numerical evaluation (using your Cool-Prop results at $P = 200$ kPa)**

- $s_1 = 0.224433$ kJ/kg-K
- $s_2 = 7.508072$ kJ/kg-K
- $s_3 = 1.075478$ kJ/kg-K

Mass flow rates:

- $\dot{m}_1 = 4.000$ kg/s
- $\dot{m}_2 = 0.433$ kg/s (from Part A)
- $\dot{m}_3 = 4.433$ kg/s

Take $T_b \approx T_0 = 298.15$ K. Compute the entropy rate terms:

$$\dot{m}_3 s_3 = 4.433(1.075478) = 4.767 \text{ kW/K}$$

$$\dot{m}_1 s_1 = 4(0.224433) = 0.898 \text{ kW/K}$$

$$\dot{m}_2 s_2 = 0.433(7.508072) = 3.251 \text{ kW/K}$$

Heat-transfer entropy term:

$$-\frac{\dot{Q}}{T_b} = -\frac{-10}{298.15} = 0.0335 \text{ kW/K}$$

So

$$\dot{S}_{gen} = 4.767 - 0.898 - 3.251 + 0.0335 = 0.6515 \text{ kW/K}$$

Finally,

$\dot{W}_{lost} = T_0 \dot{S}_{gen} = 298.15(0.6515) = 194 \text{ kW}$
--

10 9. Heating Method Exergy Comparison

Problem Statement

The temperature of the air in a building can be maintained at a desirable level during winter by using different methods of heating. Compare indirectly heating this air in a heat exchanger unit with condensing steam to heating the air directly via an electric resistance heater. Play with exergy analyses to prove which heating method results in the least exergy destruction method. You will need to perform some research regarding temperatures and operating characteristics of these systems.

Given Information

- Building air heating requirement (winter)
- Two heating methods to compare:
 1. **Indirect heating:** Heat exchanger with condensing steam
 2. **Direct heating:** Electric resistance heater

Required

- Perform exergy analysis for both heating methods
- Determine which method has **least exergy destruction**
- Research typical temperatures and operating characteristics

Assumptions

- Steady state.
- Air behaves as an ideal gas.
- Indoor air is maintained at $T_{in} = 25^\circ\text{C} = 298.15 \text{ K}$.
- Environment (dead state): $T_0 = 0^\circ\text{C} = 273.15 \text{ K}$, $p_0 = 1 \text{ atm}$ (typical winter design condition).
- Same useful heating delivered in both cases: \dot{Q}_{load} is identical for both systems.
- Electric resistance heater has $\eta_{elec \rightarrow heat} = 1$ (all electrical work converted to heat).
- Steam heat source behaves as a condensing reservoir at constant T_s .
- Neglect KE and PE effects.
- No auxiliary work (fans, pumps) and no casing heat losses.
- **Point-of-use analysis only** — upstream losses (boiler efficiency, power plant losses) are not included.

Key equations

Exergy rate transfer with heat across a boundary at T_b :

$$\dot{X}_Q = \left(1 - \frac{T_0}{T_b}\right) \dot{Q}$$

Exergy destruction rate:

$$\dot{X}_{dest} = T_0 \dot{S}_{gen}$$

System 1: Steam condensing HX (heat input at T_s)

Exergy in with supplied heat:

$$\dot{X}_{in,steam} = \left(1 - \frac{T_0}{T_s}\right) \dot{Q}_{load}$$

Exergy out to the room:

$$\dot{X}_{out,room} = \left(1 - \frac{T_0}{T_{in}}\right) \dot{Q}_{load}$$

Exergy destruction in the heating process:

$$\dot{X}_{dest,steam} = \dot{X}_{in,steam} - \dot{X}_{out,room}$$

So,

$$\boxed{\dot{X}_{dest,steam} = \dot{Q}_{load} T_0 \left(\frac{1}{T_{in}} - \frac{1}{T_s} \right)}$$

System 2: Electric resistance heater (work input)

$$\dot{W}_{elec,in} = \dot{Q}_{load}$$

Exergy in:

$$\dot{X}_{in,elec} = \dot{W}_{elec,in} = \dot{Q}_{load}$$

Exergy destruction:

$$\boxed{\dot{X}_{dest,elec} = \dot{Q}_{load} - \left(1 - \frac{T_0}{T_{in}}\right) \dot{Q}_{load} = \dot{Q}_{load} \frac{T_0}{T_{in}}}$$

Difference

$$\boxed{\Delta \dot{X}_{dest} = \dot{X}_{dest,elec} - \dot{X}_{dest,steam}}$$

Energy (equal \dot{Q}_{load})

$$\boxed{\dot{E}_{in,elec} = \dot{E}_{in,steam} = \dot{Q}_{load}}$$

Analysis

To perform a simple comparison, we'll use typical operating parameters from real-world heating systems:

Typical Operating Parameters

- **Steam heating system:** Low-pressure steam condensing at approximately $T_s = 120^\circ\text{C} = 393.15 \text{ K}$ (common for building HVAC systems)
- **Indoor air temperature:** $T_{in} = 25^\circ\text{C} = 298.15 \text{ K}$ (already assumed)
- **Environment (dead state):** Winter ambient temperature $T_0 = 0^\circ\text{C} = 273.15 \text{ K}$ (typical winter design condition)

Normalized Comparison (per unit \dot{Q}_{load})

System 1: Steam Condensing HX

$$\begin{aligned}\frac{\dot{X}_{dest,steam}}{\dot{Q}_{load}} &= T_0 \left(\frac{1}{T_{in}} - \frac{1}{T_s} \right) \\ &= 273.15 \left(\frac{1}{298.15} - \frac{1}{393.15} \right) \\ &= 273.15 \times (0.003354 - 0.002544) \\ &= 273.15 \times 0.000810 = 0.221\end{aligned}$$

System 2: Electric Resistance Heater

$$\begin{aligned}\frac{\dot{X}_{dest,elec}}{\dot{Q}_{load}} &= \frac{T_0}{T_{in}} \\ &= \frac{273.15}{298.15} = 0.916\end{aligned}$$

Comparison

$$\begin{aligned}\frac{\Delta\dot{X}_{dest}}{\dot{Q}_{load}} &= 0.916 - 0.221 = 0.695 \\ \frac{\dot{X}_{dest,elec}}{\dot{X}_{dest,steam}} &= \frac{0.916}{0.221} \approx 4.1\end{aligned}$$

Conclusion: The electric resistance heater destroys **69.5%** of \dot{Q}_{load} as exergy versus **22.1%** for steam, roughly **4.1 times more**, independent of system scale. This is because electricity is pure work (exergy = energy), so converting it to low-temperature heat is inherently wasteful, while steam at 120°C supplies heat much closer to the delivery temperature.

Result: Steam condensing heat exchanger heating results in the least exergy destruction.

Contents

1 1. Specific Heat Ratio	1
2 2. Isentropic Compressor/Turbine Relationships	1
3 3. Insulated Box Equilibrium	2

4	4. Compressor and Heat Exchanger	2
5	5.	4
6	5. Turbine Between Two Tanks	4
7	6. Desuperheating Process	7
8	7.	7
9	8. Mixing Chamber	7
10	9. Heating Method Exergy Comparison	11