

(Chapter 1-3) Zeroth law: if 2 bodies are in thermal equilibrium with a 3rd body, then they will both have the same temperature.  
 $T_A = T_C$ ;  $T_B = T_C \rightarrow$  then  $T_A = T_B$ . Unit.  $\rightarrow$  Energy: Joules  $\Rightarrow N \cdot M$ ; Work  $\Rightarrow F \cdot d \Rightarrow N \cdot M = J$ ; PE:  $mgh = Mgh \Rightarrow (kg)(\frac{m}{s^2})(m) \rightarrow$  Pressure  $\Rightarrow$  unit  $\rightarrow [N] = [P_A], [1b] = [J/K]$ .

$E \Rightarrow \frac{1}{2}mv^2 \Rightarrow (kg)(\frac{m}{s})^2 = J$ ; Heat Energy: J; Density  $\Rightarrow \rho = (SG)(\rho_{H_2O})$ ; Pressure  $\Rightarrow P = \left[\frac{N}{m^2}\right] = \left[\frac{kg}{m \cdot s^2}\right]$ ; Pressure  $\Rightarrow P = \left[\frac{N}{m^2}\right] = \left[\frac{\frac{kg}{s^2}}{m^2}\right] = \left[\frac{kg}{m \cdot s^2}\right]$

Abs pressure  $\Rightarrow$  Actual pressure relative to O. Gage pressure  $\Rightarrow$  Pressure relative to local atmospheric pressure. Vacuum Pressure  $\Rightarrow$  a negative gage pressure.

$$\Delta P = \rho g h$$

Change in pressure when vessel goes down (same fluid)  $\Rightarrow P_1 + \rho gh = P_2$

Pressure with same h  $\Rightarrow P_2 = P_1$ ; Pressure change of gas is negligible

Flow Work  $= Fd = PAh$

**PV ENERGY**  $\Rightarrow E = U + PE + KE + FW [KJ]$

$E = VPE + KE + FW [KJ] \Rightarrow E = \dot{V} \dot{P} E + \dot{K} E + \dot{F} \dot{W}$

$$\text{Mass flow rate: } \dot{m} = \rho \dot{V} \Rightarrow \rho V A; \quad \text{Energy flow rate: } \dot{E} = \dot{m} E \Rightarrow \dot{E} = \dot{m} (v_f + P_f) \quad [kW]; \quad \dot{E} = \dot{m} (v_f + P_f) + KE + PE + PW_{friction} [W]$$

$$\text{Electrical Work: } W = VI \rightarrow \text{Work} = F \cdot d \rightarrow W = EV$$

$$\text{Heat} \rightarrow Q [KJ] ; \text{Work} \rightarrow P.d \rightarrow W = FV$$

shaft Work:

$$W = \frac{F}{\text{value}} \cdot d$$

First Law of thermodynamics

$$Q_{in} + Q_{out} = \Delta KE + W_{in} - W_{out} = \Delta PE + \Delta U$$

$\eta = \frac{Q_{out}}{Q_{in}}$  (ideal engine)

$$\eta_{motor} = \frac{W_{shaft}}{W_{electric}}$$

Overall efficiency of motor  
=  $\eta_{motor} \times \eta_{pump}$

$$\text{Partial pressure of Water Vapor} = P_v = \phi P_{\text{sat}} @ T$$

Phase equilibrium  $\rightarrow V_{\text{gas}} = V_{\text{liquid}}$

Pure Substance  
or a substance that contains  
Chapter 4

**Ideal Gas Eq.**  $PV = nRT$  Universal gas constant  $R = 8.314 \text{ J/K mol}$

**ideal gas Eq.**  $PV = MRT$

**Gas**  $P_v = RT$   $R = \frac{k}{m}$   $PV = Nk_{\text{B}}T$

**Boundary work**  $\rightarrow$  Expansion  $\rightarrow W_i = -P_i dV$   $\rightarrow$  "Work out"

**Compression**  $\rightarrow W_o = P_o dV$   $\rightarrow$  "Work in"

$W_{\text{cycle}} = \int P dV$   $\rightarrow$  for "Cycle" If constant  $P$

**Volume** > Boundary Work = 0  
 $W_{\text{out}} = W_{\text{out}} - W_{\text{in}}$   
 If constant Pressure  $\rightarrow W_{\text{out}} = P \Delta V$

**Polytropic process**

$$W_0 = P_0 V_0 - P_1 V_1 \quad \text{If Const Pressure} \rightarrow W_0 = P_0 V_0$$

$$W_0 = M R (T_1 - T_0) \quad \text{Ideal Gas}$$

Mass and Energy of Control Volume

$$T_1 \quad T_2 \quad T_3 = \frac{T_1 - T_2}{T_2}$$

system.  $\xrightarrow{\text{isobaric Process}}$   $P_1 = P_2$

Frr Ideal Gas  $\xrightarrow{\text{isothermal Process (constant temp)}}$   $T_3 = T_2 \Rightarrow P_3 V_3 = P_2 V_2$

$\frac{V_3}{V_1} = \frac{T_3}{T_1}$

$W = R(T_3 - T_1)$

Specific heat  $\xrightarrow{\text{constant volume}}$  usually  $\eta = U + PV$

$R T \ln\left(\frac{P_3}{P_1}\right)$

$$\Delta H = \int_{T_1}^{T_2} C_p dT = C_p(T_2 - T_1) \quad \text{or} \quad \Delta U = \int_{T_1}^{T_2} C_v dT = C_v(T_2 - T_1)$$

**When to Use internal energy (U) or enthalpy (H)**

$\Delta U = \Sigma \Delta H_f - \Sigma \Delta H_i$

$\Delta H = \Sigma \Delta U + \Sigma P \Delta V$

**Flow Energy**

**1) For a constant Pressure  $\rightarrow Q + W = \Delta H$**

**2) For other Process  $\rightarrow Q + W_1 + W_2 = \Delta H$**

Ways to find  $U$  &  $H$  (Table)

For a non flowing fluid

$$e = Pv + \frac{1}{2}V^2 + gZ$$

$$\Delta e = \Delta Pv + \frac{1}{2}\Delta V^2 + g\Delta Z$$

Energy of a flowing fluid

$$e = Pv + \frac{1}{2}V^2 + gZ + \frac{P}{\rho g} + \frac{V^2}{2g}$$

$$\Delta e = \Delta Pv + \frac{1}{2}\Delta V^2 + g\Delta Z + \frac{\Delta P}{\rho g} + \frac{\Delta V^2}{2g}$$

For a constant process  $\Rightarrow Q = \Delta H = \Delta U$

For other Process  $\Rightarrow Q + W_h + W = \Delta U$

$$\Delta U = \dot{Q}_v - \dot{W}_h$$

$\Delta U = U_2 - U_1$  (table)

$\Delta U = \dot{Q}_v - \dot{W}_h$

$\Delta U = \int_{U_1}^{U_2} C_v dT - \int_{W_h}^{W_2} \dot{W}_h dT$

$$C = U + \frac{V^2}{2} + \frac{P}{\rho g} \quad \text{Also } \frac{dC}{dt} = \frac{dU}{dt} + \frac{dV^2}{2dt} + \frac{dP}{\rho g dt} = 0 \quad \Rightarrow \quad \frac{dU}{dt} + \frac{V^2}{2} + \frac{dP}{\rho g} = 0$$

Nozzle & Orifice  $\rightarrow$  start with (3)  $\Rightarrow$  In general  $\Delta Q = 0; \Delta W = 0; \Delta F_E = 0$  Hence eq becomes  $\Delta h = C_{AVG} \Delta T$

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**3) Energy Analysis for steady flow system.** **Steady Flow Device**

Turbines - Compressors - Work carry's in pressure increase in pressure  
 turbines: Put fluid through doing  $\rightarrow$  found  $\rightarrow$  compression: WORK IN

$\sum \dot{m}_i = \sum \dot{m}_o$

$\dot{Q} + \dot{W} = \dot{m}_i (h_{in} + v_{in}^2/2) - \dot{m}_o (h_{out} + v_{out}^2/2)$

Work on turbines:  $W_{turbines} = \dot{m}_i h_i - \dot{m}_o h_o$

Work on pump:  $W_{pump} = \dot{m}_o h_o - \dot{m}_i h_i$

Thrusting Value:  $P = \frac{\dot{m}}{M} \cdot U = \frac{U}{M} \cdot \dot{m}$

$K = \frac{C_p}{C_v} \cdot \frac{C_p T R}{C_v}$

$$\begin{aligned} d/dt = 0 & \quad \sum M_{\text{ext}} = \sum M_{\text{int}} \\ \sum E_{\text{ext}} = \sum E_{\text{int}} & \quad Q + W = \dot{M}_{\text{ext}}(h_{\text{inlet}} + \frac{V^2}{2} + gZ) - \dot{M}_{\text{int}}(h_{\text{exit}} + \frac{V^2}{2} + gZ) \rightarrow \text{Work on blades, Work Output (W)} \\ \dot{Q} + \dot{W} & \quad \text{Mixing, Heat Addition, Heat Rejection} \end{aligned}$$

$$Q + W = M \dot{h}_{in} - h_{in} + \frac{1}{2} \rho \left( \frac{V^2}{2} - \frac{V_1^2}{2} \right) + g(z_2 - z_1)$$

real initial enthalpy

$$\sum \dot{m}_i \dot{h}_{in,i} = \sum \dot{m}_{out,i} \dot{h}_{out,i}$$

$$\sum \dot{m}_i \dot{h}_{out,i} = \sum \dot{m}_{in,i} \dot{h}_{in,i}$$

real final enthalpy

$$\dot{m}_1 \dot{h}_{in,1} - \dot{m}_1 \dot{h}_{out,1} > 0$$

Mass balance

$$Q + W = \dot{m}_1 \dot{h}_{in,1} - \dot{m}_1 \dot{h}_{out,1} > 0$$

Q = heat transfer, W = work,  $\dot{m}_1 = \dot{m}_2 = \dot{m}$ ,  $\dot{h}_{in,1} = \dot{h}_{out,1}$

Entropy → Measure of disorder

**for 1 inlet ;  $\dot{m}_1 = \dot{m}_2 = \dot{m}$  varies change with time**

$E_i - E_{out} = E_i - E_{in}$  for Unsteady Flow

$dS = \frac{\delta Q}{T_{env}}$ ;  $dS = S - S_i = \frac{\delta Q}{T_{env}}$

**4. Energy Analysis of unsteady flow** → Properties change with time  $E_i + E_{out} = E_{in}$  for Unsteady flow  
 Loss of Mass →  $M_{in} - M_{out} = M_1 - M_2$   $\rightarrow Q + W + \Sigma M_{in,0,i} - \Sigma M_{out,0,j} = M_2 - M_1$

Chapter 6. → 2nd Law  $\Rightarrow$  Process Occur at a certain direction and Energy has a quality // quantity

reservoir that supplies energy: Source; absorbs energy: Sink.

**3. Heat engine** → it's not easy to convert heat to work  $K \rightarrow Q_{in} - Q_{out} = W_{act}$

Efficiency and heat engines  $\rightarrow h = \text{Wheel} = 1 - \frac{Q_{out}}{Q_{in}}$

To check if Safety 2nd Law is violated  $\rightarrow S_{final} > S_{initial}$

**Isochoric** → Constant Volume  $\rightarrow$  Increasing  $T$   $\rightarrow S = \frac{1}{T} \ln(V_f/V_i)$

**Isothermal** → Constant Temperature  $\rightarrow$  Increasing  $V$   $\rightarrow S = \ln(V_f/V_i)$

**Iso-entropic** → Constant Entropy  $\rightarrow$  Increasing  $P$   $\rightarrow S = \ln(P_f/P_i)$

Some efficiency and heat rejection  $\rightarrow$   $h_1/h_2 = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$

**Internal Combustion Engines**  $Q_{in} = h_1 \cdot \dot{m}_{fuel} \cdot \text{Heat of combustion}$

**Heat Engines**  $Q_{in} = Q_{out} + \dot{m}_{coolant} \cdot \Delta T_{coolant}$

$h_1/h_2 = 1 - \frac{Q_{out}}{Q_{in}} \rightarrow h_1/h_2 = 1 - \frac{Q_{out}}{Q_{out} + \dot{m}_{coolant} \cdot \Delta T_{coolant}}$

$h_1/h_2 = 1 - \frac{1}{1 + \frac{\dot{m}_{coolant} \cdot \Delta T_{coolant}}{Q_{out}}}$

For  $\dot{m}_{coolant} \rightarrow 0$ ,  $h_1/h_2 \rightarrow 1$  (heat rejection is 0%)

For  $\dot{m}_{coolant} \rightarrow \infty$ ,  $h_1/h_2 \rightarrow 0$  (heat rejection is 100%)

**b) Temp Scale**

**Isentropic Process**  $Q_{isent} = 0 \rightarrow T_2/T_1 = P_2/P_1 = \sqrt{P_2/P_1}$

**Turbine**  $T_2 = T_1 - \Delta T_{turbine}$

**Compressor**  $T_3 = T_2 + \Delta T_{compressor}$

**Efficiency**  $\eta = \frac{T_3 - T_4}{T_3 - T_2}$

**Refrigeration Cycle**  $Q_{in} = h_1 - h_2$

**Properties Diagram**

**Intercooler Process**  $Q_{inter} = 0 \rightarrow T_2 = T_3$

**Evaporator Process**  $Q_{evap} = 0 \rightarrow T_4 = T_1$

**Condenser Process**  $Q_{cond} = 0 \rightarrow T_3 = T_2$

**Subcooling**  $Q_{sub} = 0 \rightarrow T_4 < T_1$

**Superheating**  $Q_{super} = 0 \rightarrow T_3 > T_2$

**Reheat**  $Q_{reheat} = 0 \rightarrow T_2 > T_3$

**Regeneration**  $Q_{regen} = 0 \rightarrow T_3 = T_2$

**Two-Stage Compression**  $Q_{inter} = 0 \rightarrow T_2 = T_3$

**Two-Stage Expansion**  $Q_{cond} = 0 \rightarrow T_3 = T_2$

**Two-Stage Subcooling**  $Q_{sub} = 0 \rightarrow T_4 < T_1$

**Two-Stage Superheating**  $Q_{super} = 0 \rightarrow T_3 > T_2$

**Two-Stage Reheat**  $Q_{reheat} = 0 \rightarrow T_2 > T_3$

**Two-Stage Regeneration**  $Q_{regen} = 0 \rightarrow T_3 = T_2$

**6.5 Resistive and IAR variable**

**-f. Refrigerator / Heat Pump**

A process that can be reversed without leaving any trace on the surroundings.

Process:  $Q_1 - Q_2 = \Delta Q$

For a reversible process:  $\Delta Q = \int_{\text{Initial}}^{\text{Final}} T dS$

For an irreversible process:  $\Delta Q > \int_{\text{Initial}}^{\text{Final}} T dS$

For a Carnot cycle:  $\Delta Q = \int_{\text{Initial}}^{\text{Final}} T dS = \int_{V_1}^{V_2} \frac{RT}{P} dV = R(T_2 - T_1)$

Efficiency:  $\eta = \frac{\Delta Q_{\text{useful}}}{\Delta Q_{\text{input}}} = \frac{T_2 - T_1}{T_2}$

Efficiency of a Carnot cycle:  $\eta = 1 - \frac{T_1}{T_2}$

Efficiency of a refrigerator:  $\eta = \frac{T_1}{T_2 - T_1}$

Efficiency of a heat pump:  $\eta = \frac{T_2}{T_2 - T_1}$



**Chapter 9. Carnot Cycle** → 4 totally reversible process.

$\eta_{\text{Carnot}} = 1 - \frac{T_L}{T_H}$  Carnot is impractical as reversible isothermal heat transfer is quite difficult to achieve in reality.

MED →  $\frac{W_{\text{net}}}{V_{\text{max}} - V_{\text{min}}}$  Otto cycle → 1) Frictionless compression, 2) Constant Volume heat addition, 3) Isentropic expansion, 4) Constant volume heat rejection.

$W_{\text{out},1-0} = P_0(V_1 - V_0)$ ,  $W_{\text{in},1-0} = P_0(V_1 - V_0)$ ,  $q_{\text{in}} = V_3 - V_2$ ,  $q_{\text{out}} = V_4 - V_1$

$\eta_{\text{Otto}} = 1 - \frac{1}{r-1}$

Diesel Cycle → CI engine; Fuel: Diesel; higher compression ratio

Wtof ratio →  $r_o = \frac{V_3}{V_0} = \frac{V_3}{V_1}$

$\eta_{\text{Diesel}} = 1 - \frac{1}{r-1} \left( \frac{r_o - 1}{r_o - 1 + \frac{1}{r_o - 1}} \right)$

Gas-Turbines → Stagnation flow device,  $T \uparrow$

$(h_{10} - g_{\text{out}}) + (w_{10} - w_{\text{out}}) = h_e - h_i$ ,  $q_{\text{in}} = h_3 - h_2$ ,  $q_{\text{out}} = h_4 - h_1$

$\eta_{\text{Gas Turbine}} = 1 - \frac{1}{r-1} \frac{h_e - h_i}{h_e - h_i}$

Isentropic efficiencies:

$\eta_b = \frac{W_s}{W_q} \approx \frac{h_{2s} - h_1}{h_{2a} - h_1}$

$\eta_T = \frac{W_s}{W_q} \approx \frac{h_3 - h_{fa}}{h_3 - h_{is}}$

$\eta_{\text{regen}} = 1 - \left( \frac{T_h}{T_d} \right)^{\eta_b(K-1)/K}$

$E = \frac{h_5 - h_2}{h_4 - h_2} \approx \frac{T_5 - T_2}{T_4 - T_2}$

Second law Analysis

$X_{\text{dest}} = T_0 S_{\text{gen}} = T_0 (A S_{\text{gen}} - S_{\text{in}} + S_{\text{out}}) = T_0 \left[ (S_3 - S_1)_{\text{regen}} - \frac{Q_{\text{in}}}{T_{\text{source}}} + \frac{Q_{\text{out}}}{T_{\text{sink}} \text{ (outdoor temp)}} \right]$

for closed system energy destruction

$\dot{X}_{\text{dest}} = T_0 \dot{S}_{\text{gen}} = T_0 (\dot{S}_{\text{out}} - \dot{S}_{\text{in}}) = T_0 \left( \sum_{\text{out}} m s - \sum_{\text{in}} m s - \frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} \right)$

rate form for steady flow system

→ The transfer of heat from a low temp region to a higher temp requires a special device call **Refrigerator**.

2) The transfer of heat from a low temp medium to a high temp one is the **heat pump**. ( $COP_R = \frac{Q_L}{W_{\text{net,in}}}$ ,  $COP_H = \frac{Q_H}{W_{\text{net,in}}}$ )

Carnot Cycle:  $\eta_{\text{Carnot}} = \frac{1}{r-1}$ ;  $COP_H = \frac{1}{1 - \frac{T_L}{T_H}}$

Second Law Analysis

Compressor:  $\dot{X}_{\text{dest},1,2} \rightarrow m T_0 (S_2 - S_1)$ ,  $COP_R = \frac{q_L}{W_{\text{net,in}}} = \frac{h_1 - h_2}{W_{\text{net,in}}}$

Condenser:  $\dot{X}_{\text{des},1,3} \rightarrow T_0 [m (S_3 - S_1) + \dot{Q}_{\text{in}}]$ ,  $COP_H = \frac{q_H}{W_{\text{net,in}}} = \frac{h_2 - h_3}{W_{\text{net,in}}}$

$\eta_{\text{ideal}} = \frac{\dot{X}_{\text{ideal}}}{\dot{X}_{\text{des},1,3}} = 1 - \frac{\dot{X}_{\text{des},1,3}}{\dot{X}_{\text{des},1,2}}$

$\eta_{\text{expander}} = 1 - \frac{\dot{X}_{\text{des},1,3}}{\dot{X}_{\text{des},1,2}} = 1 - 1 = 0$

Evaporator:  $\dot{X}_{\text{des},1,4} \rightarrow T_0 [m (S_4 - S_1) - \dot{Q}_{\text{in}}]$

$\eta_{\text{expander}} = 1 - \frac{\dot{X}_{\text{des},1,4}}{\dot{X}_{\text{des},1,3}} = 1 - \frac{\dot{X}_{\text{des},1,4}}{\dot{X}_{\text{des},1,3}} = 1 - 1 = 0$

Energy destruction:  $\dot{X}_{\text{dest}} = 1 - \frac{\dot{X}_{\text{des},1,2}}{W_{\text{in}}}$

$\dot{X}_{\text{dest,bol}} = \dot{X}_{\text{dest},1,2} + \dot{X}_{\text{des},1,3} + \dot{X}_{\text{des},1,4} + \dot{X}_{\text{des},1,1}$

A reversed Brayton cycle:  $\dot{W}_{\text{net}} = Q_H - Q_L$

$COP_{\text{ref}} = \frac{COP_R}{COP_{\text{ref,in}}}$ , Gas refrigeration cycle

$\dot{W}_{\text{net}} = Q_H - Q_L$

Chapter 12 - 13. Chapter 12 →  $dU = Tds - PdV$

Clapeyron eq.:  $\left( \frac{dP}{dT} \right)_{\text{sat}} = \frac{h_{fg}}{T_{\text{vfg}}}$

Maxwell relation:  $\left( \frac{\partial T}{\partial V} \right)_S = - \left( \frac{\partial P}{\partial S} \right)_V$

$\left( \frac{\partial T}{\partial P} \right)_S = \left( \frac{\partial V}{\partial S} \right)_P = - \left( \frac{\partial T}{\partial P} \right)_V$

$u_2 - u_1 = \int_{T_1}^{T_2} c_v dT + \int_{V_1}^{V_2} \left[ T \left( \frac{\partial P}{\partial T} \right)_V - P \right] dV$

$h_2 - h_1 = \int_{T_1}^{T_2} c_p dT + \int_{P_1}^{P_2} \left[ V - T \left( \frac{\partial V}{\partial T} \right)_P \right] dP$

$s_2 - s_1 = \int_{T_1}^{T_2} \frac{c_p}{T} dT + \int_{V_1}^{V_2} \left[ \frac{\partial P}{\partial T} \right]_V dV$

$\left( \frac{P_1}{P_m} - c_{p0} \right)_T = - T \int_0^P \left( \frac{\partial^2 V}{\partial T^2} \right)_P dP$

$\frac{P_1}{P_m} = \frac{V_1}{V_m} = y_1$

$S_m = \sum_i m_i S_i = \sum_i m_i S_{\text{std}}$

$(c_p - c_{p0})_T = - T \int_0^P \left( \frac{\partial^2 V}{\partial T^2} \right)_P dP$

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$S_m = \sum_i m_i S_i = \sum_i m_i S_{\text{std}}$

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**Chapter 14.** → Air in the atmosphere usually contain Water Vapor referred as atmospheric air;  $P_v$  = Partial Pressure of Water Vapor in the air (depends on how much vapor is present); !!! Air doesn't just turn to Vapor by heating Air that contains no water vapor is called dry air. If water vapor can be treated as ideal gas.  $\rightarrow$  If isentropically compressed then  $w_1 = w_2 \rightarrow$  For heat Pump; the ratio of heat supply to the room is  $(Q_H/C_p)$

Absolute or specific humidity  $\Rightarrow$   $W = \frac{M_v}{M_a} \rightarrow$   $W = \frac{M_v}{M_a} \rightarrow$  kg dry air

Saturated air  $\rightarrow$  Air cannot hold more moisture  $\Rightarrow$   $P_v = P_{sat}$

$\dot{X}_{dest, total} = \dot{W}_{in} - \dot{X}_{Q_L}$  Second Law of Thermodynamics  $\rightarrow$  The entropy of the universe must increase

Rate of Degree Subcool  $\rightarrow \Delta T_{subcool} = T_{sat} - T_{inlet temp}$

$P_{water} = 1.08 \text{ kg/K}$

Temp rise / decrease  $= \Delta T$

$T_m = 1000\text{L}$

A heat pump using refrigerant R-134a maintains a house at 20°C in cold winter. The cold outdoor air at -10°C infiltrates into the house through the cracks of windows, doors, and

More moisture. Relative humidity  $\rightarrow \phi = \frac{M}{M_0} = \frac{P_1}{P_0} C_p = \frac{P_1}{P_0} \cdot \text{sat}(T)$  UNIVERSITY never deviates  
 $\phi = \frac{W P}{(0.622 + W) P_0}$  and  $W = \frac{0.622 \phi P_1}{P - \phi P_0}$   
 Even if Absolute humidity remains constant, relative humidity can change as it is  
 $N = 1 - \frac{T_b}{T_f}$  The increase in Work produce will equal to the work consumed by dev.  
 What happens  $\rightarrow$  No because the 9-116C A simple ideal Brayton cycle without regen-

**Enthalpy of Atmospheric air**  $\rightarrow h = h_a + w h_g$   $\rightarrow h_a$  can be converted to  $c_p T$   $\rightarrow T = \text{constant}$

for  $T$  we choose not Kelvin

6-30C In a refrigerator, heat is transferred from a lower-temperature medium (the refrigerated space) to a higher-temperature one (the kitchen space). This is a violation of the first law of thermodynamics.

Thermal resistance  $R = \frac{\Delta T}{Q}$

Refrigerator condenser heat exchanger  $\rightarrow$  to accomplish this task

the task is modified to incorporate multistage compression with intercooling and multistage expansion with reheat, without changing the pressure or temperature limits of the cycle. As a result of these two modifications,

- 2) Does the net work output increase, decrease, or remain as it flows through the heat exchanger. Assuming constant specific heats for air and water. ( $c_p = 1.005 \text{ kJ/kg.K}$ ; water:  $C_p = 4.18 \text{ kJ/kg.K}$ ). Determine
  - the total heat supply to the house by this heat pump, [5 points]
  - the power input to the heat pump, [5 points]
  - the temperature at the exit of the condenser, [3 points]

**dew-point temp** → Temp which condensation begin when the air is saturated.

**dry-bulb temp** → Ordinary temp of atmospheric air.

second law of thermodynamics? Explain.

b) Does a heat pump is a device that absorbs energy from the cold outdoor air and transfers it to the warmer indoors. Is this a violation of the second law of thermodynamics? Explain.

c) Define the coefficient of performance of a refrigerator in words. Can it be greater than unity?

the minimum mass flow rate of lake water. [5 points]

Energy balance for the house  

$$\dot{Q}_1 + \dot{Q}_{\text{loss}} = \dot{Q}_{\text{out}} + \dot{Q}_{\text{in}}$$

3 points

a) Energy Balance

**6-33C** Define the coefficient of performance of a heat pump in words. Can it be greater than unity?

**6-34C** A heat pump that is used to heat a house has a COP of 2.5. That is, the heat pump delivers 2.5 kWh of energy to the house for each 1 kWh of electricity it consumes. Is this a violation of the first law of thermodynamics? Explain.

**117C** A simple ideal Brayton cycle is modified to incorporate multistage compression with intercooling, multistage expansion with reheating, and regeneration without changing the pressure limits of the cycle. As a result of these modifications,

At Process 1 280 KPa and 0°C

Actual humidity ratio of incoming air at the wet bulb.  $h_g - h_f$

Humidity ratio of saturated air at the wet bulb.

Work to accomplish this task: 6-32. Answer: 6-36. What is Clausius expression of the second law of thermodynamics? Explain.

Wet bulb temp  $\rightarrow$  A thermometer whose bulb is covered with a cotton wick saturated with water.  $T_w = 23.7^\circ\text{C}$

The heat pump curve  $\rightarrow$   $T_w = 35^\circ\text{C}$ . A refrigerator with a COP of 1.5. That is, the refrigerator removes 1.5 kWh of energy from the refrigerated space for each 1 kWh of electricity it consumes. Is this a violation of the first law of thermodynamics? Explain.

Does the net work output increase, decrease, or remain the same?

Does the back work ratio increase, decrease, or remain the same?

(c) Does the thermal efficiency increase, decrease, or remain the same?

$S_1 = 0.9362 \frac{\text{KJ}}{\text{kg K}}$

**Water:** Chapter 15. → In combustion → dry air  $\Rightarrow$  21% Oxygen and 79% Nitrogen. Yes! (Can be useful) amount of heat supplied to the system. Air-fuel ratio  $\Rightarrow M_a = M_f \Rightarrow \frac{M_{air} + M_f}{M_{air}} = 1 + \frac{M_f}{M_{air}}$  supplied to the system. For each unit of heat removed from the refrigeration plant  $\Rightarrow$  the AT Process 2. Does the heat rejected increase, decrease, or remain the same?

Enthalpy of formation  $\bar{h}_f$  → Chemical energy at some reference state; Heating Value → The amount of heat released when a fuel is burned completely; First Law analysis → study flow system →  $\text{Enthalpy} = \bar{h}_f + (\bar{h} - \bar{h}_0)$

for combustion chamber:  $Q_{in} = \sum N_p (\bar{h}_p^o + \bar{h} - \bar{h}^o)_p$ ,  $\bar{h}$  is lost as heat or friction.  $\Rightarrow$  No entropy is lost due to friction.

Adiabatic flame temp.  $\rightarrow$  when there's no heat loss to the surroundings the temp. of product reaches a maximum.

3) Every step is in partial equilibrium  $\rightarrow$  Only true if the process is reversible.

7-8C When a system is adiabatic, what can be said about the entropy change of the substance in the system?

7-9C Work is entropy free, and sometimes the claim is made that electric resistors are also entropy free. Is this true?

6-10C Baseboard heaters are basically electric resistance heaters. If specific resistance is constant, how does the voltage drop across the heater vary with current?

**Closed or steady-flow system**

$$\sum \frac{\partial Q}{T_K} + S_{gen} = S_{prod} - S_{result}$$

$$S_{in} - S_{out} + S_{gen} = \Delta S_{system}$$

If it's a reversible process, then entropy change, but for irreversible processes, the entropy of the helium increases, meaning entropy increase is due to irreversibility.

2 points

$\bar{s}_1(T, P_1) = \bar{s}_1^o(T, P_0) - R_u \ln \frac{T^* m}{P_0}$  → how to find  $P_0$   
~~If all temp are  $T_0 = 25^\circ\text{C}$~~   
 $W_{\text{rec}} = \sum N_i (\bar{h}_f^o + \bar{h} - \bar{h}^o - T_0 \bar{s}_f)$   $\Delta s > 0$   
 $W_{\text{rec}} = \sum N_i \bar{h}_{0,0}^o - \sum N_i \bar{h}_{0,0}^o$  (both reactant and products at T0)

**Energy Analysis**  $\dot{W}_{rev} = \sum N_p \dot{g}_p^o - \sum N_p \dot{g}_p^e$  Gross function of formation

**Enthalpy of Combustion**  $\dot{Q}_c = S_{gen} + Q_{Turb}$

**Steam**: Doing an actual adiabatic process, the entropy of the steam will (**never, sometimes, always**) increase.

**T-1C**: The entropy of steam will (**increase, decrease, remain the same**) as it flows through an actual adiabatic turbine.

**6-10 no as 100** T-1C: The entropy of the working fluid of the ideal Carnot cycle (**decreases, increases, remains the same**) during the

!! Average Molar Mass of Air is equal to  $29 \text{ kg/Kmol}$

$1 \text{ atm} \Rightarrow 101,325 \text{ kPa}$  final answer if want per unit mass then divide by Molar mass  $\frac{\text{kPa}}{\text{kmol}}$

**14-9C** Consider a tank that contains moist air at 3 atm and whose walls are permeable to water vapor. The surrounding air at 1 atm pressure also contains some moisture. Is it possible for the tank to reach equilibrium with the surroundings?

**11-12** An air-conditioner operates on the vapor-compression refrigeration cycle with refrigerant-134a as the refrigerant. The air conditioner is used to keep a space at  $21^\circ\text{C}$  while rejecting the heat to the outside air at  $32^\circ\text{C}$ .  
 (a) Sketch the cycle.  
 (b) Calculate the coefficient of performance.

**15-89** A steady-flow combustion chamber is supplied with  $\text{CO}_2$  at  $273^\circ\text{C}$ ,  $110 \text{ kPa}$ , and  $5.0 \text{ mol}/\text{m}^3$ .

sible for the water vapor to flow into the tank from surroundings? Explain.  $\rightarrow$  Yes, if the vapor pressure in the storage drum is greater than the vapor pressure in the tank.

**14-18** Humid air at 100 kPa, 20°C, and 90 percent relative humidity is heated at constant pressure until its temperature reaches 40°C. Calculate the final specific volume and the final dew point.  $\rightarrow$  0.84 m<sup>3</sup>/kg and 27°C

**8-6** The System that Have the same temp. as the environment. Consider two systems that are at the same pressure as the environment. The first system is at the same temperature as the environment, whereas the second system is at a lower temperature than the environment. How would you compare the energies of these two systems?

$\omega_2 = \omega_1 = \frac{0.622 P_{r,1}}{P - P_{r,1}} = \frac{(0.622)(2.105 \text{ kPa})}{(100 - 2.105) \text{ kPa}} = 0.0134 \text{ kg H}_2\text{O/kg dry air}$

$h_{2s} = 285.34; h_2 = 289.66$  At  $P_{atm} = 100 \text{ kPa}$ ;  $T_{atm} = 26.24 - 1.3 = 24.94^\circ\text{C}$

give  $h_{3p} = h_{2g,70^\circ\text{C}} = 108.27; s_{34} = 0.3944; h_3 = h_1 = 108.27 \rightarrow s = 0.229 \text{ kJ/kg}$

Q: Then we actually have from the reacting hence we need to find Airflow Ratio to find how much mole of air use per mole of fuel.

$$T_2 = T_1 \left( \frac{P_2}{P_1} \right)^{(k-1)/k} = (293 \text{ K}) \left( \frac{800 \text{ kPa}}{100 \text{ kPa}} \right)^{0.4/1.4} = 531 \text{ K}$$

$\dot{Q}_H = \dot{m}(h_2 - h_1) = 0.06(208.96 - 100.27) = 10.88 \text{ kW}$

$\dot{W}_{in} = \dot{m}(h_2 - h_1) = 2.670 \text{ kW}$

$$P_{v,2} = \frac{\omega_2 P_2}{\omega_2 + \omega_3} = \frac{(0.0134)(800)}{0.0134 + 0.22} = 16.87 \text{ kPa}$$

(b) Rate of heat transfer  $\rightarrow Q_{out} = \sum M_i (h_i^* + \bar{h}_i - h_i)$

saturated vapor at the compressor inlet. If the refrigerant enters the compressor at a rate of 100 L/min and the isentropic efficiency of the compressor is 75 percent, determine (a) the temperature of the refrigerant at the compressor exit, (b) the rate of heat generation by the people in the room, (c) the COP of the air conditioner, and (d) the minimum rate of heat rejection by the condenser.

**11-44** Refrigerant-134a enters the condenser of a residential heat pump at 800 kPa and 30°C at a rate of 0.002 kg/s and leaves at 750 kPa subcooled by 3°C. The refrigerant enters  $T_1 = 7^{\circ}\text{C}$ ,  $T_{\text{exit}} = 29.0^{\circ}\text{C}$ ,  $P_1 = 750\text{ kPa}$ .

$\dot{m}_1 = \frac{P_1}{R_1 T_1} = \frac{750}{0.4547} = 177.9\text{ kg/kg}$

$P_2 = 800\text{ kPa}$ ,  $b_2 = 286.71\text{ kJ/kg}$ ,  $T_2 = 50^{\circ}\text{C}$

$\eta_{\text{C}} = \frac{h_2 - h_3}{h_2 - h_1}$

$0.75 = \frac{277.45 - 259.36}{277.45 - h_1} \rightarrow h_1 = 282.65\text{ kJ/kg}$

$T_1 = 26^{\circ}\text{C}$

$h_4 = h_3 + \Delta h_{\text{fg}}$

$h_4 = h_3 + \frac{P_4}{P_3} \Delta h_{\text{fg}}$

$h_4 = 282.65 + \frac{750}{800} \cdot 200 = 350\text{ kJ/kg}$

$Q_{\text{out}} = \dot{m}_{\text{out}} \rightarrow 0.978(7762\text{ K}) \rightarrow 3567\text{ kJ}$

**11-45** The rate of exergy destruction  $\dot{E}_{\text{destr}}$

(b) The mass flowrate of the refrigerant is

$$\dot{m}_r = \frac{1}{\rho_r} \left( \frac{1}{t_1} - \frac{1}{t_2} \right) = \frac{1}{0.55} \left( \frac{1}{29.06} - \frac{1}{27.93} \right) = 87.93 \text{ kg/kJ}$$

The compressor is at 200 kPa superheated by 4°C. Determining (a) the isentropic efficiency of the compressor, (b) the rate of heat supplied to the heated room, and (c) the COP of the heat pump. Also, determine (d) the COP and the rate of heat supplied to the heated room if this heat pump operates at the ideal vapor-compression cycle between the pressure limits of

The ideal vapor-compression cycle analysis of the cycle is as follows:

The minimum mass and volume flow rates are

$$\dot{m}_{min} = \frac{\dot{V}_{min} \rho_{min}}{h_2 - h_1} = \frac{0.1533 \text{ kW}}{273.29 - 259.36 \text{ kJ/kg}} = 0.006358 \text{ kg/s}$$

$$\dot{V}_{min} = \dot{m}_{min} V_1 = (0.006358 \text{ kg/s}) / (0.04117 \text{ m}^3/\text{kg}) = (0.002618 \text{ m}^3/\text{s}) = 15.71 \text{ l/min}$$