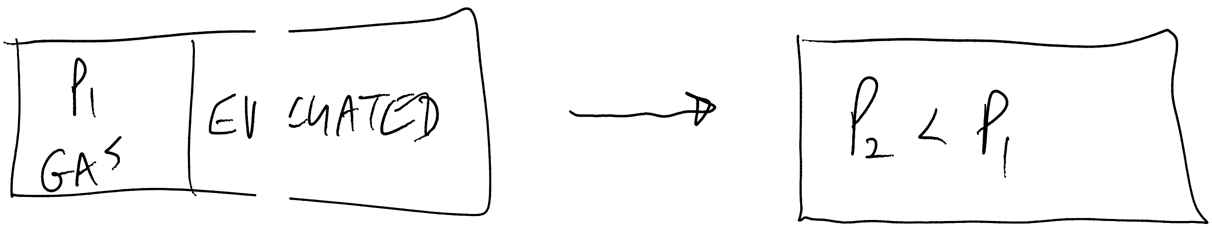
Entropy

REVISIT UNRESTRAINED EXPANSION:



ENTROPY INCREASES (MORE DISORDERED)

2ND LAW: ENTROPY MUST ALLWAYS INCREASE

OR REMAIN CONSTANT FOR AN ISOLATED SYSTEM.

$$dS \geq 0 \quad \text{OR} \quad S_f - S_i \geq 0$$

THE INEQUALITY CAN BE ELIMINATED BY DEFINING

AN ENTROPY GENERATION TERM:

$$S_f - S_i = S_{GEN} \geq 0$$

IN A REVERSIBLE PROCESS (FRICTIONLESS PISTON) $= 0$

$$S_f = S_i \quad S_{GEN}$$

IN AN IRREVERSIBLE PROCESS (PISTON W/ FRICTION)

$S_f - S_i > 0$ OR $S_{GEN} > 0$

CONTROL VOLUME ANALYSIS: RATE FORM

THE CHANGE IN ENTROPY WITHIN THE C.V. MINUS THE NET ENTROPY TRANSPORTED INTO THE C.V MINUS THE SUM OF THE HEAT TRANSFERS DIVIDED BY THE CORRESPONDING ABSOLUTE BOUNDARY TEMPERATURES IS GREATER THAN OR EQUAL TO ZERO.

CONTROL VOLUME   CONVECTION   HEAT TRANSFER

$$\frac{dS_{CV}}{dt} - \dot{S}_{NET} - \sum_{n=1} \left(\frac{\dot{Q}_i}{T_i}\right)_{CV} \; 0$$

$$\dot{S}_{NET} = \dot{S}_{IN} - \dot{S}_{OUT} = \sum (s\dot{m})_{in} - \sum (s\dot{m})_{OUT}$$

$[S] =$ ENTROPY $= \left(\frac{KJ}{K}\right)$, $[s] =$ SPECIFIC ENTROPY

$$[s] = \frac{kJ}{kg\text{-}K}$$

$$\dot{S}_{GEN} = \frac{dS_{CV}}{dt} - \sum(s\dot{m})_{IN} + \sum(s\dot{m})_{OUT} - \sum\left(\frac{\dot{Q}_i}{T_i}\right) \geq 0$$

FOR A STEADY FLOW PROCESS, $dS/dt = 0$

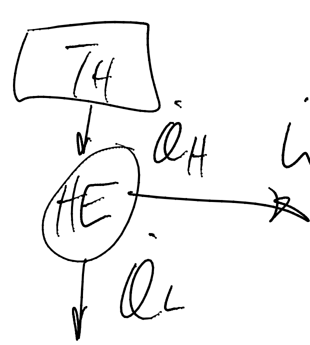
$$\dot{S}_{GEN} = -\sum(s\dot{m})_{in} + \sum(s\dot{m})_{OUT} - \sum\left(\frac{\dot{Q}_i}{T_i}\right) \geq 0$$

FOR A STEADY FLOW ADIABATIC PROCESS: $\dot{Q} = 0$

$$\dot{S}_{GEN} = -\sum(s\dot{m})_{in} + \sum(s\dot{m})_{OUT} \geq 0$$

FOR A STEADY FLOW ADIABATIC, ONE INLET/OUTLET

$$\dot{S}_{GEN} = -\dot{m}(s_{in} - s_{OUT}) = \dot{m}(s_e - s_i)$$



CARNOT CYCLE ANALYSIS

1ST LAW FOR A CYCLE:

$$\dot{Q}_H = \dot{W} + \dot{Q}_L \qquad \oint \delta W = \oint \delta Q$$

$$\left[\frac{\dot{Q}_i}{T_i} \right] \geq 0$$

2ND LAW:

$$\oint \left[(S_2 - S_1) - \sum (s\dot{m})_{in} + \sum (s\dot{m})_{out} - \sum (\right.$$

FOR $\left(\frac{\dot{Q}_H}{T_H} \text{ CYC } \frac{\dot{Q}_L}{T_L} \right)$ $S_2 = S_1$, $\dot{m}_i = \dot{m}_{out} = 0$

$$\frac{\dot{Q}_L}{T_L} \geq \frac{\dot{Q}}{T}, \qquad \boxed{\frac{\dot{Q}_L}{\dot{Q}_H} \geq \frac{T_L}{T_H}}$$

2ND LAW FOR A CYCLE

FOR A CARNOT HEAT ENGINE, $\frac{\dot{Q}_L}{\dot{Q}_H} = \frac{T_L}{T_H}$

$$\boxed{\eta = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} = 1 - \frac{T_L}{T_H}}$$

THE 2ND LAW REQUIRES US TO CALCULATE CHANGES IN ENTROPY. WE MUST FIND SOME RELATION BETWEEN THE PROPERTY ENTROPY

AND THE OTHER THERMODYNAMIC PROPERTIES WE KNOW.

LET'S SPECIFY INTERNAL ENERGY IN TERMS OF TWO INDEPENDENT PROPERTIES:

$$\mathcal{U} = \mathcal{U}(S, V)$$

CHAIN RULE TO FIND THE DIFFERENTIAL:

$$d\mathcal{U} = \left(\frac{\partial \mathcal{U}}{\partial S}\right)_V dS + \left(\frac{\partial \mathcal{U}}{\partial V}\right)_S dV$$

THERMODYNAMIC DEFINITIONS OF TEMPERATURE AND PRESSURE:

$$T \equiv \left(\frac{\partial \mathcal{U}}{\partial S}\right)_V , \quad P \equiv -\left(\frac{\partial \mathcal{U}}{\partial V}\right)_S$$

$$d\mathcal{U} = T\,dS - P\,dV \quad \text{(GIBB'S EQUATION)}$$

THIS EQUATION RELATES EQUILIBRIUM

THERMODYNAMIC PROPERTIES.

SIMILARLY,

$$dH = T\,dS + V\,dP$$

SOLVING FOR dS GIVES

$$dS = \frac{dU}{T} + \frac{P}{T}dV = \frac{dH}{T} - \frac{V}{T}dP$$

FOR IDEAL GASES, $Pv = RT$ OR $PV = mRT$

$du = C_v\,dT$, $u = \frac{U}{m}$, $dU = mC_v\,dT$

$dH = mC_p\,dT$, $S = s \cdot m$, $V = v \cdot m$

$$ds = C_v \frac{dT}{T} + R\frac{dv}{v} = C_p\frac{dT}{T} - R\frac{dP}{P} \quad \text{DIFF. EQN.}$$

INTEGRATE BOTH SIDES

$$s_2 - s_1 = \int_{T_1}^{T_2} C_v \frac{dT}{T} + R\ln\frac{v_2}{v_1} = \int_{T_1}^{T_2} C_p \frac{dT}{T} - R\ln\frac{P_2}{P_1}$$

$C_v = C_v(T)$, $C_p = C_p(T)$

FOR C_v AND C_p CONSTANT

FOR C_v AND C_p CONSTANT,

$$s_2 - s_1 = C_v \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1} = C_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}$$

FOR INCOMPRESSIBLE LIQUIDS OR SOLIDS,

$C_p = C_v = C =$ CONSTANT AND $v =$ CONSTANT

$$dS = \frac{dU}{T} + \frac{P}{T} dV = \frac{mC\,dT}{T}$$

($dV \to 0$)

$$ds = C \frac{dT}{T}$$

$$s_2 - s_1 = C \ln \frac{T_2}{T_1}$$

LIQUIDS/SOLIDS

FOR SATURATED SYSTEM: $s = s_f + x s_{fg}$

VARIABLE SPECIFIC HEATS: Δs FOR IDEAL GASES

$$s_2 - s_1 = \int_{T_1}^{T_2} C_p \frac{dT}{T} - R \ln \frac{P_2}{P_1}$$

DEFINE $s^\circ = \int_0^T C_p \frac{dT}{T}$ ABSOLUTE ZERO TEMP. IS THE REFERENCE STATE

STATE

$$\int_{T_1}^{T_2} C_p \frac{dT}{T} = s_2^{\circ} - s_1^{\circ} = \int_0^{T_2} C_p \frac{dT}{T} - \int_0^{T_1} C_p \frac{dT}{T}$$

$$s_2 - s_1 = s_2^{\circ} - s_1^{\circ} - R \ln \frac{P_2}{P_1}$$

$\Delta s = \text{FCN}(T, P)$, UNLIKE $u = u(T)$, $h = h(T)$

THE ENTROPY OF A FIXED MASS DOES NOT CHANGE DURING A PROCESS THAT IS INTERNALLY REVERSIBLE AND ADIABATIC: ISENTROPIC PROCESS

LET'S EXAMINE $s_2 - s_1$ FOR AN IDEAL GAS UNDERGOING AN ISENTROPIC PROCESS:

$$s_2 - s_1 = 0 = \int_{T_1}^{T_2} C_v \frac{dT}{T} + R \ln \frac{v_2}{v_1}$$

$$0 = \int_{T_1}^{T_2} C_p \frac{dT}{T} - R \ln \frac{P_2}{P_1}$$

FOR CONSTANT SPECIFIC HEATS,

$$C_v \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1} = 0 \quad , \quad C_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} = 0$$

REARRANGING GIVES:

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$k = \frac{C_p}{C_v}$ = RATIO OF SPECIFIC HEATS

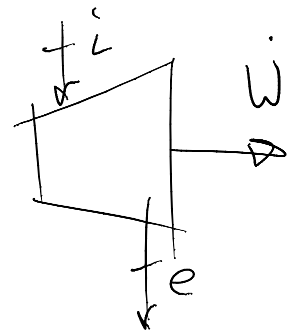
$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$\left(\frac{P_2}{P_1}\right) = \left(\frac{v_1}{v_2}\right)^{k}$$

THESE RELATIONS ARE USED BY AERODYNAMICISTS TO SOLVE FLOWS INVOLVING COMPRESSIBILITY (SHOCK WAVES)

ISENTROPIC EFFICIENCIES

WE CAN NOW EXAMINE COMPONENTS AND DETERMINE THEIR EFFICIENCIES IN COMPARISON TO ONE WHICH IS ISENTROPIC.



TURBINES:

$$\eta_T = \frac{\text{ACTUAL TURBINE WORK}}{\text{ISENTROPIC WORK}} = \frac{\dot{W}_a}{\dot{W}_s} < 1.0$$

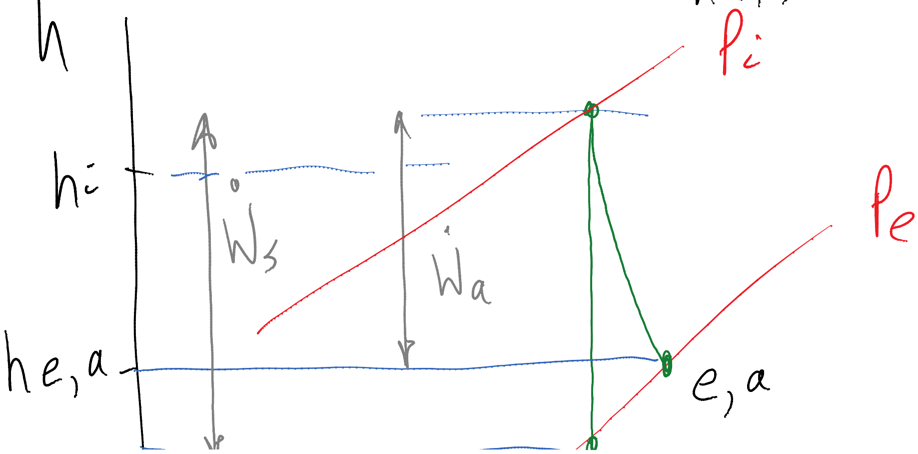
FOR NEGLIGIBLE CHANGES IN KE + PE,

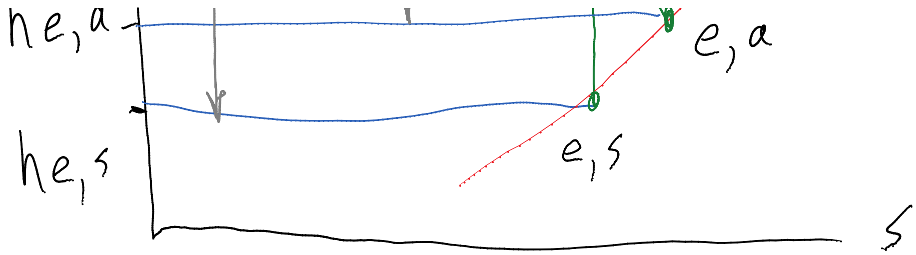
FIRST LAW: $\dot{W} = \dot{m}(h_i - h_e)$

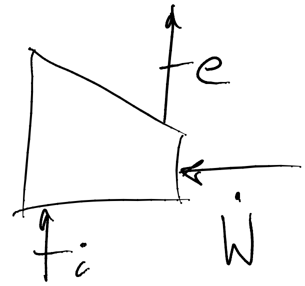
$$\eta_T \cong \frac{h_i - h_{e,a}}{h_i - h_{e,s}}$$

$h_{e,a}$ = ACTUAL EXIT ENTHALPY

$h_{e,s}$ = ISENTROPIC EXIT ENTHALPY







COMPRESSORS:

$$\eta_c = \frac{\text{ISENTROPIC COMPRESSOR WORK}}{\text{ACTUAL COMPRESSOR WORK}} = \frac{\dot{W}_s}{\dot{W}_a}$$

$$\eta_c \simeq \frac{h_i - h_{e,s}}{h_i - h_{e,a}}$$

LIQUID PUMPS:

$$\eta_p = \frac{\dot{W}_s}{\dot{W}_a}$$ ☆

FIRST LAW: $\dot{W} = \dot{m}(h_i - h_e)$

GIBB'S EQUATION: $dh = T ds + v dP$ ☆ (with $T ds$ crossed out, $\to 0$)

FOR AN ISENTROPIC PUMP, $ds = 0$

FOR AN INCOMPRESSIBLE LIQUID, v = CONSTANT

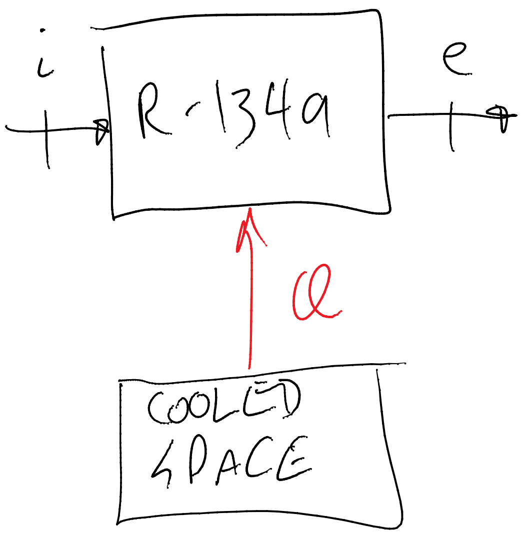
$dh = v\,dP$: INTEGRATE BOTH SIDES

$$h_i - h_{e,s} = v(P_i - P_e)$$

$$\eta = \frac{\dot{W}_s}{\dot{W}_a} = \frac{\dot{m}(h_i - h_{e,s})}{\dot{m}(h_i - h_{e,a})} = \frac{v(P_i - P_e)}{(h_i - h_{e,a})}$$

Prob. 6.29

6–29 Refrigerant-134a enters the coils of the evaporator of a refrigeration system as a saturated liquid–vapor mixture at a pressure of 200 kPa. The refrigerant absorbs 120 kJ of heat from the cooled space, which is maintained at −5°C, and leaves as saturated vapor at the same pressure. Determine (*a*) the entropy change of the refrigerant, (*b*) the entropy change of the cooled space, and (*c*) the total entropy change for this process.



$P_i = P_e = 200$ kPa

X_i : SAT. MIXTURE

$X_e = 1.0$ SAT. VAPOR

$T_{CS} = -5°C$, $Q = 120$ kJ

FIND ΔS_R, ΔS_{CS}, ΔS_{TOTAL}

• THE ENTROPY CHANGE OF A SYSTEM DURING AN INTERNALLY REVERSIBLE ISOTHERMAL HEAT TRANSFER PROCESS IS:

$\Delta S = \frac{Q}{\ }$ EQN. (7-6)

$$\Delta S = \frac{Q}{T_0} \quad \text{EQN. } (7\text{-}6)$$

$$T_R = T_{sat}(P_{sat} = 200\ \text{kPa}) = -10.09^\circ C + 273 = 263\ \text{K}$$

$$T_{CS} = -5^\circ C + 273 = 268\ \text{K}$$

$$\Delta S_R = \frac{Q}{T_R} = \frac{(120\ \text{kJ})}{(263\ \text{K})} = 0.456\ \frac{\text{kJ}}{\text{K}}$$

$$\Delta S_{CS} = \frac{Q}{T_{CS}} = \frac{(-120\ \text{kJ})}{(268\ \text{K})} = -0.448\ \frac{\text{kJ}}{\text{K}}$$

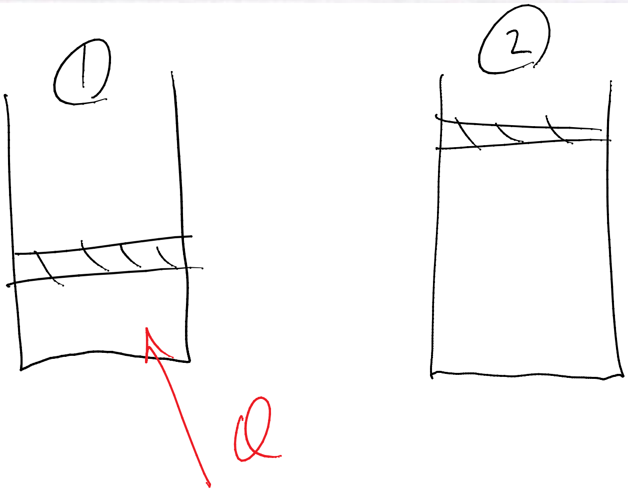
$$S_{GEN} = \Delta S_{TOTAL} = \Delta S_R + \Delta S_{CS} = \left(0.456 - 0.448\ \frac{\text{kJ}}{\text{K}}\right)$$

$$S_{GEN} = +0.008\ \frac{\text{kJ}}{\text{K}}$$

ENTROPY CHANGE OF A SYSTEM <u>CAN</u> BE NEGATIVE DURING A PROCESS, BUT THE ENTROPY GENERATION CANNOT BE NEGATIVE.

Prob. 6.46

6–46 A piston-cylinder device contains 1.2 kg of saturated water vapor at 200°C. Heat is now transferred to steam, and steam expands reversibly and isothermally to a final pressure of 800 kPa. Determine the heat transferred and the work done during this process.



$m = 1.2\ \text{kg}\ H_2O$

$X_1 = 1.0$ SAT. VAPOR

$T_1 = 200°C$

$P_2 = 800\ \text{kPa}$,

$T_2 = T_1 = 200°C$

FIND Q, W

REVERSIBLE AND ISOTHERMAL:

$$\Delta S = \frac{Q}{T_0} \quad \text{(2ND LAW OF THERMO)}$$

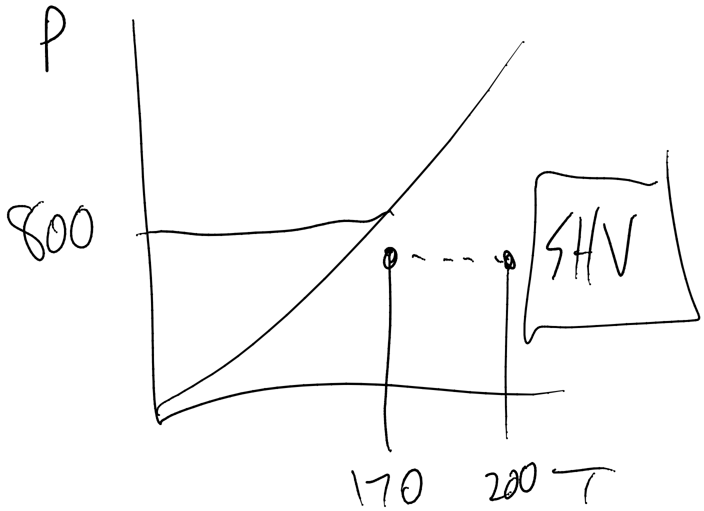
$$Q = T_0 \cdot \Delta S = T_0 (S_2 - S_1) = T_0\, m (s_2 - s_1)$$

$$s = \text{SPECIFIC ENTROPY} = \frac{S}{m}$$

$$s = s_g @ T_1 = 200°C = 6.4323\ \frac{\text{kJ}}{\text{kg-K}}$$

$u_1 = u_g = 2595.3 \frac{kJ}{kg}$

FOR $P_2 = 800$ kPa, $T_2 = 200°C$



@ $P_{sat} = 800$ kPa

$T_{sat} = 170.43°C$

$s_2 = 6.8158 \frac{kJ}{kg\text{-}K}$

$u_2 = 2630.6 \frac{kJ}{kg}$

$Q = T_0 m (s_2 - s_1)$

$Q = (200 + 273^{K})(1.2^{kg})(6.8158 - 6.4323 \frac{kJ}{kg\text{-}K}) = 217.7 \; kJ$

FIRST LAW: $Q - W = m(u_2 - u_1)$

$W = Q - m(u_2 - u_1)$

$W = (217.7 \; kJ) - (1.2^{kg})(2630.6 - 2595.3 \frac{kJ}{kg}) = 175.3 \; kJ$

Prob. 6.72

6–72 Helium gas is compressed from 90 kPa and 30°C to 450 kPa in a reversible, adiabatic process. Determine the final temperature and the work done, assuming the process takes place (*a*) in a piston-cylinder device and (*b*) in a steady-flow compressor.

= ISENTROPIC

He, $P_1 = 90$ kPa, $T_1 = 30°C$

$P_2 = 450$ kPa

REV + ADIAB = ISENTROPIC

FIND T_2, W FOR PISTON-CYLINDER AND

FIND T_2, $\dot{W}$ FOR A STEADY-FLOW COMPRESSOR

ISENTROPIC RELATION:

$$\left(\frac{T_{2,s}}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

EQN. (7-43)

FOR IDEAL GASES ONLY!

$$k = \frac{c_p}{c_v}$$

$$T_{2,s} = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

FROM TABLE A-2, $k(He) = 1.667$

$$T_{2,s} = (30 + 273\,K)\left(\frac{450}{90}\right)^{[(1.667-1)/1.667]} = 576.9\,K$$

FOR A PISTON/CYLINDER, FIRST LAW:

$$\cancel{Q}^{\,0} - W = m(u_2 - u_1), \quad W = m(u_1 - u_2)$$

$$w = \frac{W}{m} = u_1 - u_2 \qquad w = \text{SPECIFIC WORK}$$

FOR AN IDEAL GAS, CONSTANT C_v

$$w = C_v(T_1 - T_2)$$

$$w = \left(3.1156\,\frac{kJ}{kg\text{-}K}\right)(303 - 576.7\,K) = -853.4\,\frac{kJ}{kg}$$

FOR A STEADY FLOW COMPRESSOR, FIRST LAW:

$$\dot{Q} - \dot{W} = \dot{m}(h_e - h_i)$$

(the $\dot{Q}$ term is crossed out in red)

$$\dot{w} = \frac{\dot{W}}{\dot{m}} = h_i - h_e \quad \text{SPECIFIC POWER}$$

$h = u + Pv$

FOR AN IDEAL GAS, CONSTANT C_p

$$h_i - h_e = C_p (T_i - T_e)$$

$$\dot{w} = C_p (T_i - T_e)$$

$$\boxed{\dot{w}} = \left(5.1926 \frac{kJ}{kg\text{-}K}\right)(303 - 576.9\,K) = \boxed{-1422 \frac{kJ}{kg}}$$

Prob. 6.97

6–97 Steam enters an adiabatic turbine at 6 MPa, 600°C, and 80 m/s and leaves at 50 kPa, 100°C, and 140 m/s. If the power output of the turbine is 8 MW, determine (*a*) the mass flow rate of the steam flowing through the turbine and (*b*) the isentropic efficiency of the turbine.

Answers: (*a*) 8.25 kg/s, (*b*) 83.7 percent

H_2O, ADIABATIC $\Rightarrow \dot{Q}=0$, $P_i = 6$ MPa, $T_i = 600°C$

$\vec{V}_i = 80\ \frac{m}{s}$, $P_e = 50$ kPa, $T_e = 100°C$, $\vec{V}_e = 140\ \frac{m}{s}$

$\dot{W}_a = 8$ MW, FIND $\dot{m}$, η_T

FIRST LAW: $\dot{Q} - \dot{W}_a = \dot{m}\left[(h_{e,a} - h_i) + \frac{1}{2}(V_e^2 - V_i^2)\right]$

$$\dot{m} = -\dot{W}_a \Big/ \left[(h_{e,a} - h_1) + \frac{1}{2}(V_e^2 - V_i^2)\right]$$

$h_1 = 3658.4\ \frac{kJ}{kg}$, $s_1 = 7.1677\ \frac{kJ}{kg\text{-}K}$

$h_{e,a} = 2682.5\ \frac{kJ}{kg}$

$(h_{e,a} - h_1) = -975.9\ \frac{kJ}{kg}$

$$\frac{1}{2}\left(V_e^2 - V_i^2\right) = \frac{1}{2}\left[\left(140\,\frac{m}{s}\right)^2 - \left(80\,\frac{m}{s}\right)^2\right]\left(\frac{1\ kJ/kg}{1000\ m^2/s^2}\right) = 6.6$$

$$\dot{m} = -\left(8000\ kW\right) / \left(-975.9\ \frac{kJ}{kg} + 6.6\ \frac{kJ}{kg}\right) = 8.253\ \frac{kg}{s}$$

EFFICIENCY:

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{\text{ACTUAL POWER}}{\text{ISENTROPIC POWER}}$$

$$\dot{W}_s = -\dot{m}\left[\left(h_{es} - h_i\right) + \frac{1}{2}\left(V_e^2 - V_i^2\right)\right]$$

FOR AN ISENTROPIC PROCESS,

$$s_{es} = s_i = 7.1677\ \frac{kJ}{kg\text{-}K}$$

$$P_2 = 50\ kPa$$

NO CHANGE IN ENTROPY

IF-THEN STATEMENTS TO FIND EXIT PROPERTIES

$$s_f = 1.09,\quad s_g = 7.59\ \frac{kJ}{kg\text{-}K}$$

SINCE $s_f \le s_{es} \le s_g$ $\therefore$ SATURATED MIXTURE

$$s_{es} = s_f + X_{es} s_{fg}$$

$$X_{es} = \frac{s_{es} - s_f}{s_g - s_f}$$ QUALITY AT THE EXIT FOR THE ISENTROPIC TURBINE

$$X_{es} = \frac{7.1677 - 1.091}{6.5029} = 0.934$$

$$h_{e,s} = h_f + X_{es} h_{fg} = (340.49) + (0.934)(2305.4) = 2493.7 \ \frac{kJ}{kg}$$

$$\dot{W}_s = -\left(8.253 \frac{kg}{s}\right)\left[\left(2493.7 - 3658 \frac{kJ}{kg}\right) + \left(6.6 \frac{kJ}{kg}\right)\right] = 9558 \ kW$$

$$\eta = \frac{\dot{W}_a}{\dot{W}_s} = \frac{8000 \ kW}{9558 \ kW} = 0.837$$

$$X_{e,a} = \frac{h_{e,a} - h_f}{h_g - h_f} = \frac{h_{e,a} - h_f}{h_{fg}} = \frac{2682 - 340.5}{2304}$$

$X_{e,a} \geq 1.0$ SUPERHEATED VAPOR

Prob. 6.32

6–32 A 0.5-m^3 rigid tank contains refrigerant-134a initially at 200 kPa and 40 percent quality. Heat is transferred now to the refrigerant from a source at 35°C until the pressure rises to 400 kPa. Determine (*a*) the entropy change of the refrigerant, (*b*) the entropy change of the heat source, and (*c*) the total entropy change for this process.

Answers: (*a*) 3.783 kJ/K, (*b*) −3.432 kJ/K, (*c*) 0.441 kJ/K

Prob. 6.39

6–39 An insulated piston-cylinder device contains 0.05 m^3 of saturated refrigerant-134a vapor at 0.8-MPa pressure. The refrigerant is now allowed to expand in a reversible manner until

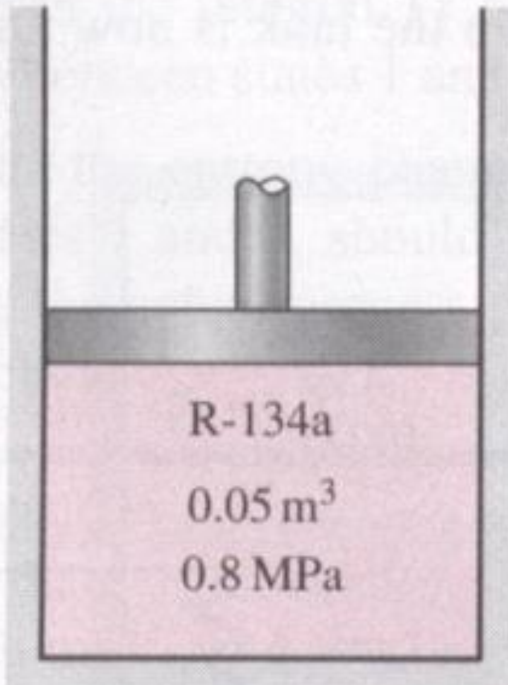


FIGURE P6–39

the pressure drops to 0.4 MPa. Determine (*a*) the final temperature in the cylinder and (*b*) the work done by the refrigerant.

Prob. 6.73

6–73 An insulated, rigid tank contains 4 kg of argon gas at 450 kPa and 30°C. A valve is now opened, and argon is allowed to escape until the pressure inside drops to 150 kPa. Assuming the argon remaining inside the tank has undergone a reversible, adiabatic process, determine the final mass in the tank. *Answer:* 2.07 kg

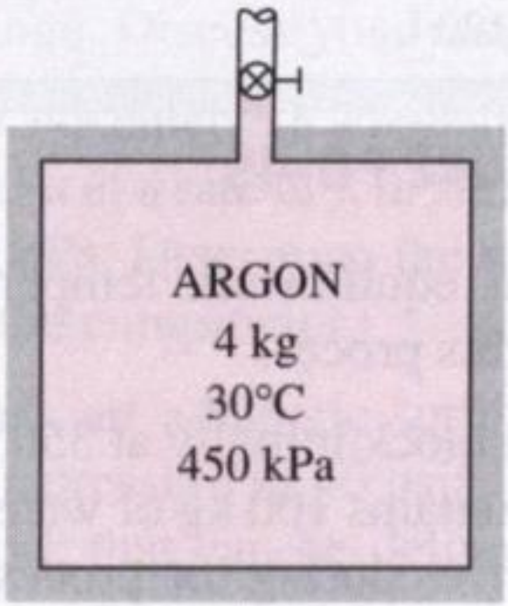


FIGURE P6–73

Prob. 6.95

6–95 Steam enters an adiabatic turbine at 8 MPa and 500°C with a mass flow rate of 3 kg/s and leaves at 30 kPa. The isentropic efficiency of the turbine is 0.90. Neglecting the kinetic energy change of the steam, determine (*a*) the temperature at the turbine exit and (*b*) the power output of the turbine. *Answers:* (*a*) 69.1°C, (*b*) 3052 kW

Prob. 6.100

6–100 EES Refrigerant-134a enters an adiabatic compressor as saturated vapor at 120 kPa at a rate of 0.3 m^3/min and exits at 1-MPa pressure. If the isentropic efficiency of the compressor is 80 percent, determine (*a*) the temperature of the refrigerant at the exit of the compressor and (*b*) the power input, in kW. Also, show the process on a *T-s* diagram with respect to saturation lines.

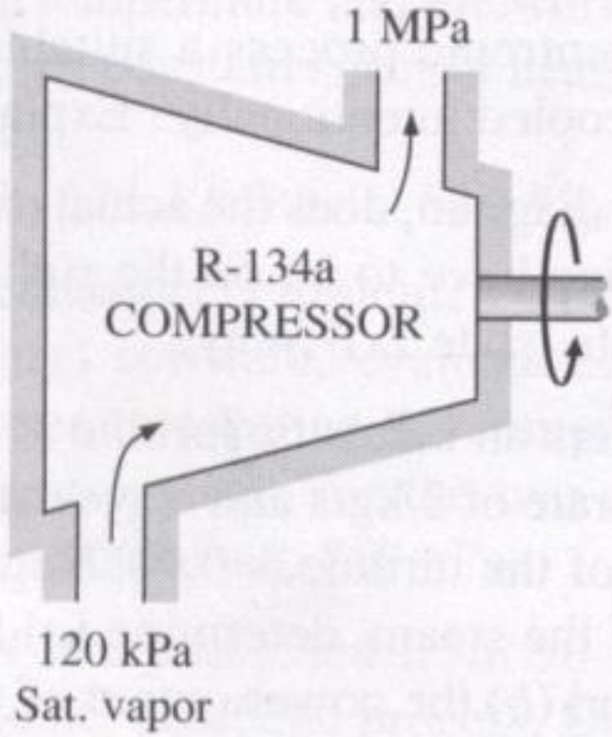


FIGURE P6–100

Prob. 6.102

6–102 Air enters an adiabatic compressor at 100 kPa and 17°C at a rate of 2.4 m^3/s, and it exits at 257°C. The compressor has an isentropic efficiency of 84 percent. Neglecting the changes in kinetic and potential energies, determine (*a*) the exit pressure of air and (*b*) the power required to drive the compressor.

Prob. 6.110

6–110 Steam is to be condensed in the condenser of a steam power plant at a temperature of 50°C with cooling water from a nearby lake, which enters the tubes of the condenser at 18°C at a rate of 101 kg/s and leaves at 27°C. Assuming the condenser to be perfectly insulated, determine (*a*) the rate of condensation of the steam and (*b*) the rate of entropy generation in the condenser. *Answers:* (*a*) 1.595 kg/s, (*b*) 1.10 kW/K

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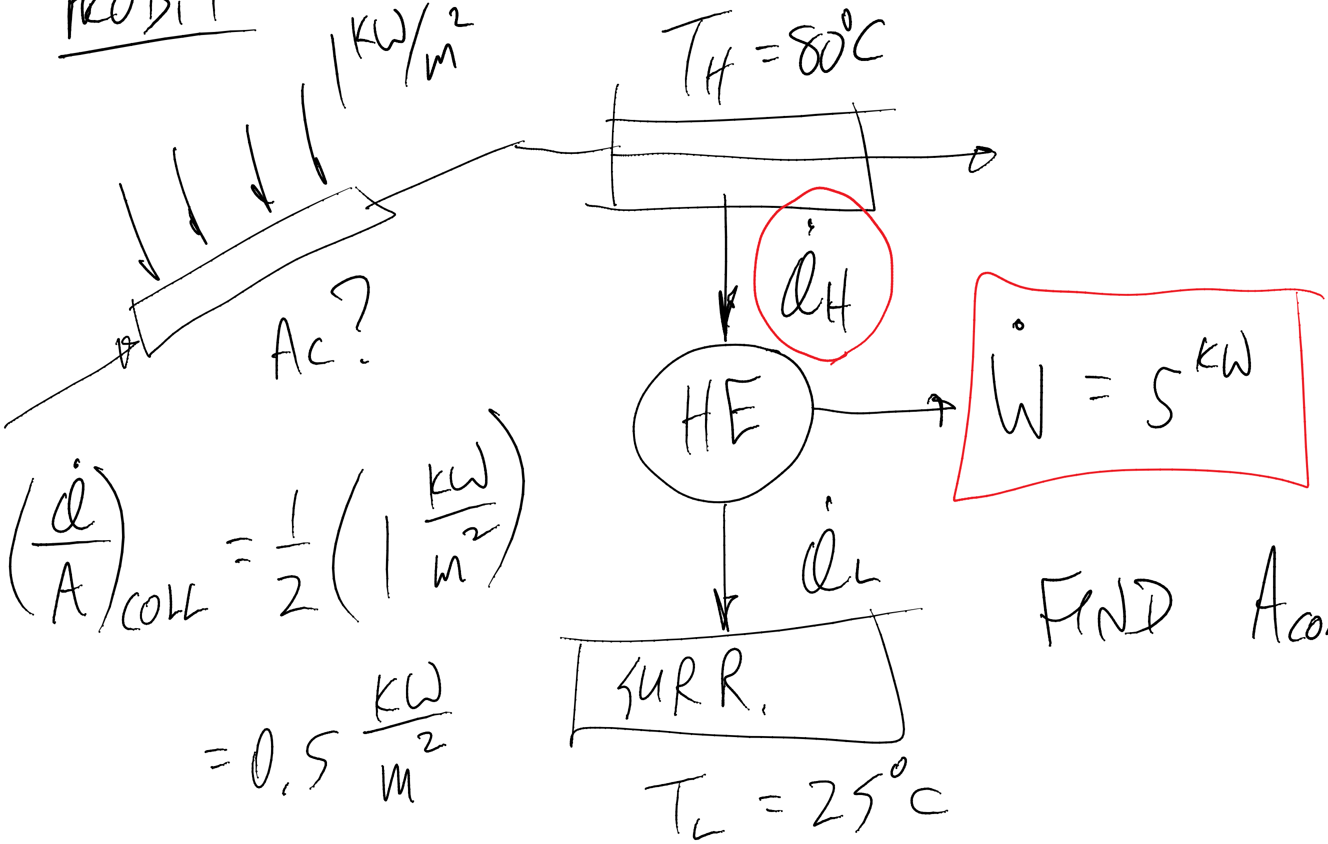
PROB. 1



$$\left(\frac{\dot{Q}}{A}\right)_{COLL} = \frac{1}{2}\left(1\ \frac{kW}{m^2}\right) = 0.5\ \frac{kW}{m^2}$$

CARNOT H.E.: $\eta = \frac{\dot{W}}{\dot{Q}_H} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} = 1 - \frac{T_L}{T_H}$

$$\eta = 1 - \left(\frac{25+273^K}{80+273^K}\right) = 0.1558$$

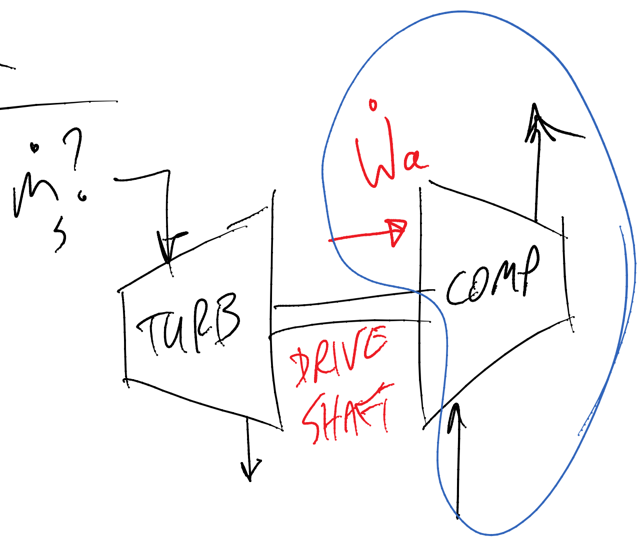
$$\eta = \frac{\dot{W}}{\dot{Q}_H}, \quad \dot{Q}_H = \frac{\dot{W}}{\eta} = \frac{(5^{kW})}{(0.1558)} = 32.09^{kW}$$

$$\left(\frac{\dot{Q}_H}{A}\right) = 0.5\ \frac{kW}{m^2}$$

$$\left(\frac{Q_H}{A}\right) = 0.5 \frac{kW}{m^2}$$

$$A = \frac{(32.09\ kW)}{\left(0.5 \frac{kW}{m^2}\right)} = 64.18\ m^2$$

PROB. 2



STEAM

$P_i = 2\ MPa$, $T_i = 350°C$

$P_e = 8\ kPa$

AIR

$P_i = 1\ atm$, $T_i = 27°C$ ☆

$P_e = 5\ atm$, $\dot{m}_{air} = 0.1 \frac{kg}{s}$ FIND $\dot{m}_s$

$\eta_T = 0.70$, $\eta_c = 0.78$

COMPRESSOR : AIR

ISENTROPIC EFFICIENCY:

$$\eta_c = \frac{\dot{W}_s}{\dot{W}_a}, \quad \dot{W}_a = \frac{\dot{W}_s}{\eta_c} = \frac{\dot{m}_{air}(h_i - h_{2,s})}{\eta_c}$$

$$\dot{W}_a = \frac{\dot{m}_{air} C_p (T_i - T_{e,s})}{\eta_c}$$

$dh = C_p dT$ FOR IDEAL GAS

FIND $T_{e,s}$ USING ISENTROPIC RELATION:

$$\left(\frac{T_{e,s}}{T_i}\right) = \left(\frac{P_e}{P_i}\right)^{(k-1)/k}$$

$ds = s_2 - s_1 = 0$

$$T_{e,s} = T_1 \left(\frac{P_e}{P_i}\right)^{(k-1)/k} = (27+273\,K)\left(\frac{5\text{ atm}}{1\text{ atm}}\right)^{\frac{0.4}{1.4}} = 475.1\,K$$

$$\dot{W}_a = \frac{\left(0.1\,\frac{kg}{s}\right)\left(1.005\,\frac{kJ}{kg\text{-}K}\right)\left[(27+273\,K) - (475.1\,K)\right]}{(0.78)}$$

$$\dot{W}_a = -22.56\ kW$$

TURBINE (STEAM) $P_i = 2\ MPa$, $T_i = 350°C$, $P_e = 8\ kPa$

TURBINE: $P_i = 2$ MPa, $T_i = 350$ C, $P_e = 8$

$\dot{W}_a = +22.56$ KW

ISENTROPIC EFFICIENCY:

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{\dot{W}_a}{\dot{m}_s(h_i - h_{e,s})}$$

$h_{e,s}$ = ISENTROPIC ENTHALPY AT THE EXIT

$h_i(2^{MPa}, 350°C) = 3137.7 \frac{KJ}{kg}$, $s_i = 6.9583 \frac{KJ}{kg-K}$

FIND $h_{e,s}$ BY SETTING $s_e = s_i = 6.9583 \frac{KJ}{kg-K}$

$P_e = 8^{KPa}$

@ $P_{sat} = 8^{KPa}$, $s_f = 0.5909 \frac{KJ}{kg-K}$, $s_g = 8.230 \frac{KJ}{kg-K}$

SINCE $s_f \leq s_e \leq s_g$ ∴ SAT. MIXTURE

ISENTROPIC EXIT QUALITY:

$$s_e = s_f + X_{e,s} s_{fg}, \quad X_{e,s} = \frac{s_e - s_f}{s_{fg}} = \frac{s_e - s_f}{s_g - s_f}$$

$$X_{e,s} = \frac{(6.9583) - (0.5909)}{\ldots} = 0.8336$$

$$(8.230) - (0.5909) \qquad 0.0335$$

$$h_{e,s} = h_f + X_{e,s} h_{fg} = (173.4) + (0.8335)(2402) = 2176 \frac{kJ}{kg}$$

$$\dot{m}_s = \frac{\dot{W}_a}{\eta_T (h_i - h_{e,s})} = \frac{\left(22.56 \frac{kJ}{s}\right)}{(0.70)\left(3137 - 2176 \frac{kJ}{kg}\right)} =$$

$$\boxed{\dot{m}_s = 0.03351 \frac{kg}{s}}$$

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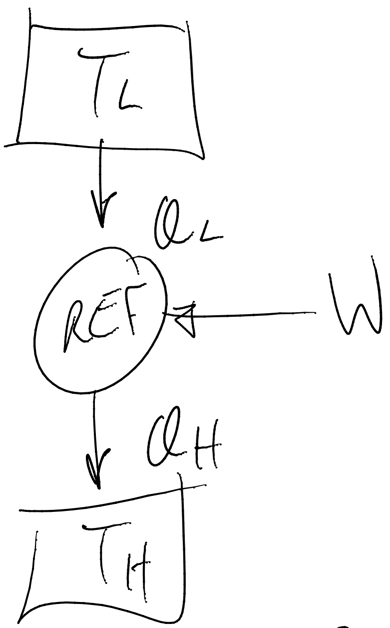
PROB. 3



$Q_H = 2.5 \times 10^3$ KJ

$T_H = 80 + 273 = 353^K$

$W = 1.1 \times 10^3$ KJ

FIND COP_{REF}

$$COP_{REF} = \frac{1}{Q_H/Q_L - 1}$$

$$COP_{REF,CAR} = \frac{1}{T_H/T_L - 1}$$

FIRST LAW: $Q_L + W = Q_H$

$Q_L = Q_H - W = 2.5 \times 10^3 - 1.1 \times 10^3 = 1.4 \times 10^3$ KJ

FOR A CARNOT CYCLE, $\frac{Q_H}{Q_L} = \frac{T_H}{T_L}$

$T_L = T_H \left(\frac{Q_L}{\ldots} \right) \ldots \left(\frac{1.4 \times 10^3}{\ldots} \right) = 198^K$

$$T_L = T_H\left(\frac{Q_L}{Q_H}\right) = (353^K)\left(\frac{1.4\times10^3}{2.5\times10^3}\right) = 198^K$$

$$\boxed{COP} = \frac{1}{(353/198)-1} = \boxed{1.277}$$

PROB. 4

$$\dot{m} = \left(40{,}000\ \frac{kg}{HR}\right)\left(\frac{HR}{3600^s}\right) = 11.11\ \frac{kg}{s},\ H_2O$$

$P_i = 8^{MPa}$, $T_i = 500°C$, $P_e = 40^{kPa}$, $X_e = 1.0$

$\dot{W}_a = 8.2^{MW} = 8200^{kW}$, $T_{SURR} = 25 + 273 = 298^K$

FIND ENTROPY GENERATION: 2ND LAW OF THERMO

$$\dot{S}_{GEN} = \sum \dot{m}_e s_e - \sum \dot{m}_i s_i - \sum \frac{\dot{Q}}{T_o}$$

$$\dot{S}_{GEN} = \dot{m}(s_e - s_i) - \frac{\dot{Q}}{T_{SURR}} \ \bigstar$$

$s_i = 6.7266\ \frac{kJ}{kg-K}$, $s_e = s_g = 7.6691\ \frac{kJ}{kg-K}$

FIND $\dot{Q}$ USING THE FIRST LAW:

$$\dot{Q} - \dot{W}_a = \dot{m}(h_{e,a} - h_i)$$

$$h_i = 3399.5 \frac{kJ}{kg}, \quad h_{e,a} = 2636.1 \frac{kJ}{kg}$$

$$\dot{Q} = (8200\ kW) + \left(11.11 \frac{kg}{s}\right)\left(2636 - 3399 \frac{kJ}{kg}\right) = -281.4\ kW$$

$$\dot{S}_{GEN} = \left(11.11 \frac{kg}{s}\right)\left(7.6691 - 6.7266 \frac{kJ}{kg\text{-}K}\right) - \frac{(-281.4\ kW)}{(298\ K)}$$

$$\dot{S}_{GEN} = 11.41 \frac{kW}{K}$$