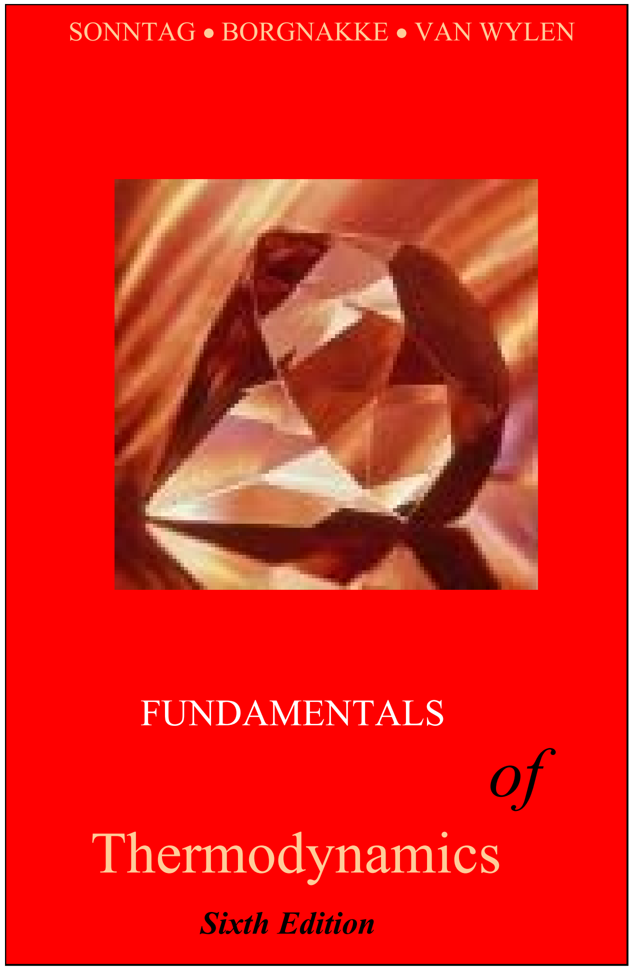
SOLUTION MANUAL
ENGLISH UNIT PROBLEMS
CHAPTER 9



CONTENT

This problem set compared to the fifth edition chapter 9 set and the current SI unit problems.

New	5th	SI	New	5th	SI	New	5th	SI
134	new	6	147	new	33	160	105	82
135	new	8	148	new	35	161	106	84
136	new	9	149	97	39	162	new	94
137	new	14	150	94	47	163	107	90
138	new	17	151	95	52	164	109	96
139	new	18	152	108	54	165	110	101
140	new	19	153	new	53	166	111	107
141	new	20	154	98	56	167	117	111
142	91mod	22	155	99	60	168	115	-
143	29	24	156	101	69	169	114	114
144	96	30	157	100	73	170	118	-
145	new	31	158	103	77	171	102	131
146	93	37	159	104	79	172	112	133

Concept Problems

9.134E

A compressor receives R-134a at 20 F, 30 psia with an exit of 200 psia, x = 1. What can you say about the process?

Solution:

Properties for R-134a are found in Table F.10

Inlet state: $s_i = 0.4157$ Btu/lbm R

Exit state: $s_e = 0.4080$ Btu/lbm R

Steady state single flow: $$s_e = s_i + \int_i^e \frac{dq}{T} + s_{gen}$$

Since s decreases slightly and the generation term can only be positive, it must be that the heat transfer is negative (out) so the integral gives a contribution that is smaller than $-s_{gen}$.

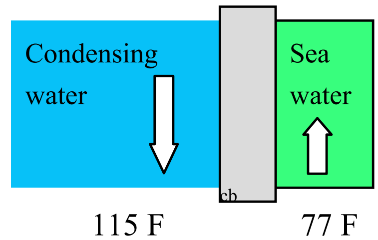
9.135E

A large condenser in a steam power plant dumps 15 000 Btu/s at 115 F with an ambient at 77 F. What is the entropy generation rate?

Solution:

This process transfers heat over a finite temperature difference between the water inside the condenser and the outside ambient (cooling water from the sea, lake or river or atmospheric air)

C.V. The wall that separates the inside 115 F water from the ambient at 77 F.



Entropy Eq. 9.1 for steady state operation:

$$\frac{dS}{dt} = 0 = \sum \frac{\dot{Q}}{T} + \dot{S}_{gen} = \frac{\dot{Q}}{T_{115}} - \frac{\dot{Q}}{T_{77}} + \dot{S}_{gen}$$

$$\dot{S}_{gen} = \left[\frac{15\ 000}{536.7} - \frac{15\ 000}{115 + 459.7} \right] \frac{\text{Btu}}{\text{s R}} = \mathbf{1.85}\ \frac{\textbf{Btu}}{\textbf{s R}}$$

9.136E

Air at 150 psia, 540 R is throttled to 75 psia. What is the specific entropy generation?

Solution:

C.V. Throttle, single flow, steady state. We neglect kinetic and potential energies and there are no heat transfer and shaft work terms.

Energy Eq. 6.13: $h_i = h_e \quad \Rightarrow \quad T_i = T_e$ (ideal gas)

Entropy Eq. 9.9: $s_e = s_i + \int_i^e \frac{dq}{T} + s_{gen} = s_i + s_{gen}$

Change in s Eq.8.24: $s_e - s_i = \int_i^e C_p \frac{dT}{T} - R \ln \frac{P_e}{P_i} = - R \ln \frac{P_e}{P_i}$

$$s_{gen} = s_e - s_i = -\frac{53.34}{778} \ln\left(\frac{75}{150}\right) = \mathbf{0.0475 \ \frac{Btu}{lbm\ R}}$$

9.137E

A pump has a 2 kW motor. How much liquid water at 60 F can I pump to 35 psia from 14.7 psia?

Incompressible flow (liquid water) and we assume reversible. Then the shaftwork is from Eq.9.18

$$w = -\int v\, dP = -v\, \Delta P = -0.016\ \text{ft}^3\text{/lbm}\ (35 - 14.7)\ \text{psia}$$

$$= -\,46.77\ \text{lbf-ft/lbm} = -0.06\ \text{Btu/lbm}$$

$$\dot{W} = 2\ \text{kW} = 1.896\ \text{Btu/s}$$

$$\dot{m} = \frac{\dot{W}}{-w} = \frac{1.896}{0.06} = \mathbf{31.6\ lbm/s}$$

9.138E

A steam turbine inlet is at 200 psia, 900 F. The exit is at 40 psia. What is the lowest possible exit temperature? Which efficiency does that correspond to?

We would expect the lowest possible exit temperature when the maximum amount of work is taken out. This happens in a reversible process so if we assume it is adiabatic this becomes an isentropic process.

Exit: 40 psia, $s = s_{in} = 1.8055$ Btu/lbm R $\Rightarrow$ **T = 483.7 F**

The efficiency from Eq.9.27 measures the turbine relative to an isentropic turbine, so the **efficiency** will be **100%**.

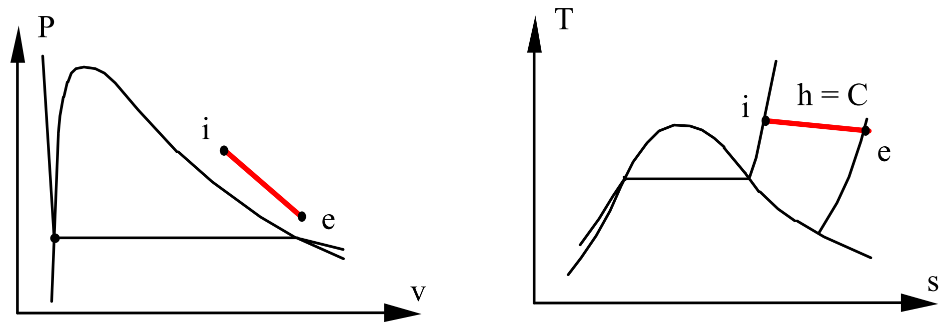
9.139E

A steam turbine inlet is at 200 psia, 900 F. The exit is at 40 psia. What is the highest possible exit temperature? Which efficiency does that correspond to?

The highest possible exit temperature would be if we did not get any work out, i.e. the turbine broke down. Now we have a throttle process with constant h assuming we do not have a significant exit velocity.

Exit: 40 psia, $h = h_{in} = 1477.04$ Btu/lbm $\Rightarrow$ **T = 889 F**

Efficiency: $\eta = \frac{w}{w_s} = \mathbf{0}$



Remark: Since process is irreversible there is no area under curve in T-s diagram that correspond to a q, nor is there any area in the P-v diagram corresponding to a shaft work.

9.140E

A steam turbine inlet is at 200 psia, 900 F. The exit is at 40 psia, 600 F. What is the isentropic efficiency?

from table F.7.2

Inlet: h_{in} = 1477.04 Btu/lbm, s_{in} = 1.8055 Btu/lbm R

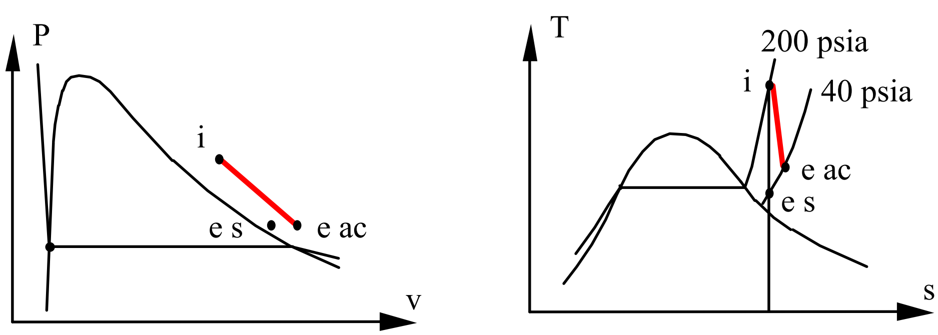
Exit: h_{ex} = 1333.43 Btu/lbm, s_{ex} = 1.8621 Btu/lbm R

Ideal Exit: 40 psia, $s = s_{in}$ = 1.8055 Btu/lbm R $\Rightarrow$ h_s = 1277.0 Btu/lbm

$$w_{ac} = h_{in} - h_{ex} = 1477.04 - 1333.43 = 143.61 \text{ Btu/lbm}$$

$$w_s = h_{in} - h_s = 1477.04 - 1277.0 = 200 \text{ Btu/lbm}$$

$$\eta = \frac{w_{ac}}{w_s} = \frac{143.61}{200} = \mathbf{0.718}$$



9.141E

The exit velocity of a nozzle is 1500 ft/s. If η_{nozzle} = 0.88 what is the ideal exit velocity?

The nozzle efficiency is given by Eq. 9.30 and since we have the actual exit velocity we get

$$\mathbf{V}_{e\,s}^2 = \mathbf{V}_{ac}^2/\eta_{nozzle} \quad \Rightarrow$$

$$\mathbf{V}_{e\,s} = \mathbf{V}_{ac}/\sqrt{\eta_{nozzle}} = 1500 / \sqrt{0.88} = \mathbf{1599\ ft/s}$$

Steady Single Flow Devices

9.142E

Steam enters a turbine at 450 lbf/in.2, 900 F, expands in a reversible adiabatic process and exhausts at 130 F. Changes in kinetic and potential energies between the inlet and the exit of the turbine are small. The power output of the turbine is 800 Btu/s. What is the mass flow rate of steam through the turbine?

Solution:

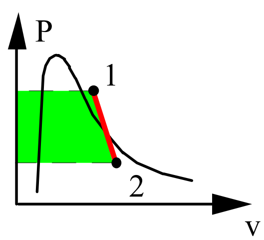
C.V. Turbine, Steady single inlet and exit flows. Adiabatic: $\dot{Q} = 0$.

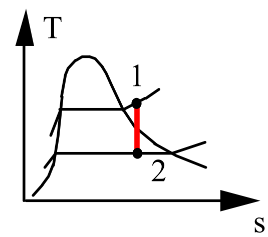
Continuity Eq.6.11: $\dot{m}_i = \dot{m}_e = \dot{m}$,

Energy Eq.6.12: $\dot{m}h_i = \dot{m}h_e + \dot{W}_T$,

Entropy Eq.9.8: $\dot{m}s_i + \emptyset = \dot{m}s_e$ (Reversible $\dot{S}_{gen} = 0$)

Explanation for the work term is in Sect. 9.3, Eq.9.18





Inlet state: Table F.7.2 $h_i = 1468.3$ btu/lbm, $s_i = 1.7113$ btu/lbm R

Exit state: $s_e = 1.7113$ Btu/lbm R, $T_e = 130$ F $\Rightarrow$ saturated

$x_e = (1.7113 – 0.1817)/1.7292 = 0.8846$,

$h_e = 97.97 + x_e\ 1019.78 = 1000$ Btu/lbm

$w = h_i - h_e = 1468.3 – 1000 = 468.31$ Btu/lbm

$\dot{m} = \dot{W} / w = 800 / 468.3 =$ **1.708 lbm/s**

9.143E

In a heat pump that uses R-134a as the working fluid, the R-134a enters the compressor at 30 lbf/in.^{2}, 20 F at a rate of 0.1 lbm/s. In the compressor the R-134a is compressed in an adiabatic process to 150 lbf/in.^{2}. Calculate the power input required to the compressor, assuming the process to be reversible.

Solution:

C.V. Compressor, Steady single inlet and exit flows. Adiabatic: $\dot{Q} = 0$.

Continuity Eq.6.11: $\dot{m}_1 = \dot{m}_2 = \dot{m}$,

Energy Eq.6.12: $\dot{m}h_1 = \dot{m}h_2 + \dot{W}_C$,

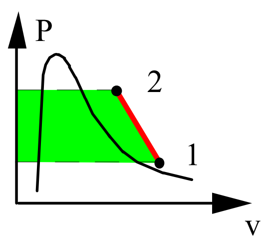
Entropy Eq.9.8: $\dot{m}s_1 + \emptyset = \dot{m}s_2$ (Reversible $\dot{S}_{gen} = 0$)

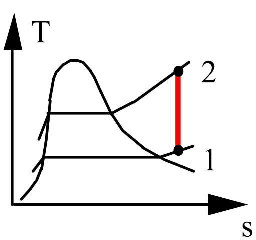
Inlet state: Table F.10.2 $h_1 = 169.82$ Btu/lbm, $s_1 = 0.4157$ Btu/lbm R

Exit state: $P_2 = 150$ psia & s_2 $\Rightarrow$ $h_2 = 184.46$ Btu/lbm

$\dot{W}_c = \dot{m}w_c = \dot{m}(h_1 - h_2) = 0.1 \times (169.82 - 184.46) =$ **-1.46 btu/s**

Explanation for the work term is in Sect. 9.3 Eq.9.18





9.144E

A diffuser is a steady-state, steady-flow device in which a fluid flowing at high velocity is decelerated such that the pressure increases in the process. Air at 18 lbf/in.^{2}, 90 F enters a diffuser with velocity 600 ft/s and exits with a velocity of 60 ft/s. Assuming the process is reversible and adiabatic what are the exit pressure and temperature of the air?

C.V. Diffuser, Steady single inlet and exit flow, no work or heat transfer.

Energy Eq.: $h_i + \mathbf{V}_i^2/2g_c = h_e + \mathbf{V}_e^2/2g_c$, $\Rightarrow$ $h_e - h_i = C_{Po}(T_e - T_i)$

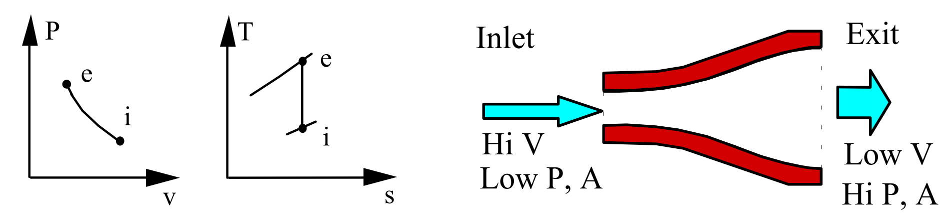
Entropy Eq.: $s_i + \int dq/T + s_{gen} = s_i + 0 + 0 = s_e$ (Reversible, adiabatic)

Energy equation then gives (conversion 1 Btu/lbm = 35 037 ft^2/s^2 from A.1):

$$C_{Po}(T_e - T_i) = 0.24(T_e - 549.7) = \frac{600^2 - 60^2}{2 \times 25\,037}$$

T_e = **579.3 R**

$$P_e = P_i(T_e/T_i)^{\frac{k}{k-1}} = 18\left(\frac{579.3}{549.7}\right)^{3.5} = \mathbf{21.6\ lbf/in^2}$$



9.145E

The exit nozzle in a jet engine receives air at 2100 R, 20 psia with neglible kinetic energy. The exit pressure is 10 psia and the process is reversible and adiabatic. Use constant heat capacity at 77 F to find the exit velocity.

Solution:

C.V. Nozzle, Steady single inlet and exit flow, no work or heat transfer.

Energy Eq.6.13: $h_i = h_e + \mathbf{V}_e^2/2 \quad (Z_i = Z_e)$

Entropy Eq.9.8: $s_e = s_i + \int dq/T + s_{gen} = s_i + 0 + 0$

Use constant specific heat from Table F.4, $C_{Po} = 0.24 \frac{\text{Btu}}{\text{lbm R}}$, $k = 1.4$

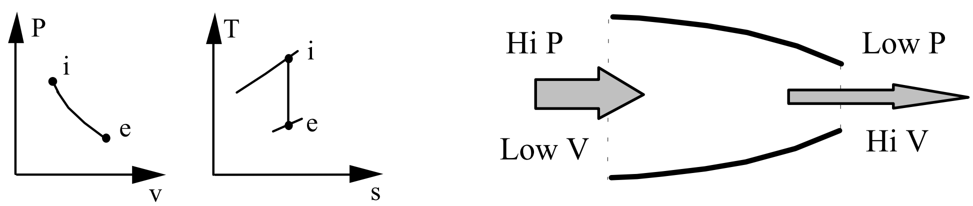
The isentropic process ($s_e = s_i$) gives Eq.8.32

$$=> \quad T_e = T_i(P_e/P_i)^{\frac{k-1}{k}} = 2100\ (10/20)^{0.2857} = 1722.7 \text{ R}$$

The energy equation becomes (conversion 1 Btu/lbm = 25 037 ft^2/s^2 in A.1)

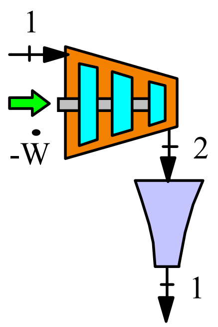
$$\mathbf{V}_e^2/2 = h_i - h_e \cong C_P(T_i - T_e)$$

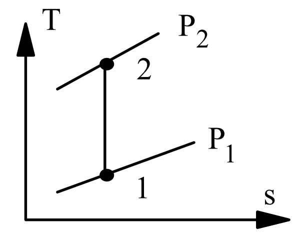
$$\mathbf{V}_e = \sqrt{2\ C_P(T_i - T_e)} = \sqrt{2 \times 0.24(2100\text{-}1722.7) \times 25\ 037} = \mathbf{2129\ ft/s}$$



9.146E

Air at 1 atm, 60 F is compressed to 4 atm, after which it is expanded through a nozzle back to the atmosphere. The compressor and the nozzle are both reversible and adiabatic and kinetic energy in/out of the compressor can be neglected. Find the compressor work and its exit temperature and find the nozzle exit velocity.





Separate control volumes around compressor and nozzle. For ideal compressor we have inlet : 1 and exit : 2

Adiabatic : $q = 0$.
Reversible: $s_{gen} = 0$

Energy Eq.6.13: $h_1 + 0 = w_C + h_2$;

Entropy Eq.9.8: $s_1 + 0/T + 0 = s_2$

$$- w_C = h_2 - h_1 , \quad s_2 = s_1$$

The constant s from Eq. 8.25 gives

$$T_2 = T_1 (P_2/P_1)^{\frac{k-1}{k}} = (459.7 + 60) \times (4/1)^{0.2857} = \mathbf{772\ R}$$

$$\Rightarrow \quad -w_C = h_2 - h_1 = C_P(T_2 - T_1) = 0.24\ (772 - 519.7) = \mathbf{60.55\ Btu/lbm}$$

The ideal nozzle then expands back down to state 1 (constant s) so energy equation gives:

$$\frac{1}{2}\mathbf{V}^2 = h_2 - h_1 = -w_C = 60.55 \text{ Btu/lbm}$$

$$\Rightarrow \quad \mathbf{V} = \sqrt{2 \times 60.55 \times 25\ 037} = \mathbf{1741\ ft/s}$$

Remember conversion 1 Btu/lbm = 25 037 ft^2/s^2 from Table A.1.

9.147E

An expander receives 1 lbm/s air at 300 psia, 540 R with an exit state of 60 psia, 540 R. Assume the process is reversible and isothermal. Find the rates of heat transfer and work neglecting kinetic and potential energy changes.

Solution:

C.V. Expander, single steady flow.

Energy Eq.: $\dot{m}h_i + \dot{Q} = \dot{m}h_e + \dot{W}$

Entropy Eq.: $\dot{m}s_i + \dot{Q}/T + \dot{m}s_{gen} = \dot{m}s_e$

Process: T is constant and $s_{gen} = 0$

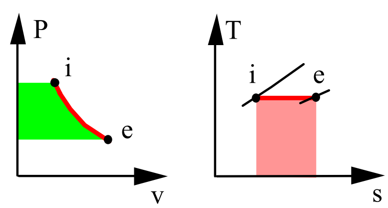
Ideal gas and isothermal gives a change in entropy by Eq. 8.24, so we can solve for the heat transfer

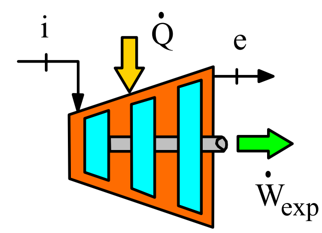
$$\dot{Q} = T\dot{m}(s_e - s_i) = -\dot{m}RT \ln \frac{P_e}{P_i}$$

$$= - 1 \times 540 \times \frac{53.34}{778} \times \ln \frac{60}{300} = \mathbf{59.6\ Btu/s}$$

From the energy equation we get

$$\dot{W} = \dot{m}(h_i - h_e) + \dot{Q} = \dot{Q} = \mathbf{59.6\ Btu/s}$$



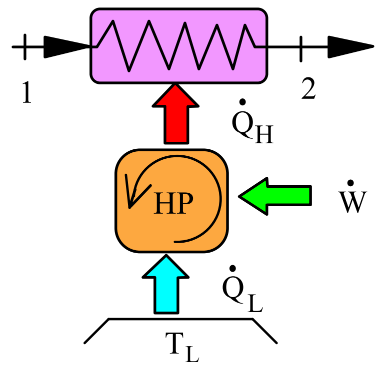


9.148E

A flow of 4 lbm/s saturated vapor R-22 at 100 psia is heated at constant pressure to 140 F. The heat is supplied by a heat pump that receives heat from the ambient at 540 R and work input, shown in Fig. P9.35. Assume everything is reversible and find the rate of work input.

Solution:

C.V. Heat exchanger



Continuity Eq.: $\dot{m}_1 = \dot{m}_2$;

Energy Eq.: $\dot{m}_1 h_1 + \dot{Q}_H = \dot{m}_1 h_2$

Table F.9.2:

$h_1 = 109.01$ Btu/lbm,

$s_1 = 0.2179$ Btu/lbm R

$h_2 = 125.08$ Btu/lbm,

$s_2 = 0.2469$ Btu/lbm R

Notice we can find $\dot{Q}_H$ but the temperature T_H is not constant making it difficult to evaluate the COP of the heat pump.

C.V. Total setup and assume everything is reversible and steady state.

Energy Eq.: $\dot{m}_1 h_1 + \dot{Q}_L + \dot{W} = \dot{m}_1 h_2$

Entropy Eq.: $\dot{m}_1 s_1 + \dot{Q}_L/T_L + 0 = \dot{m}_1 s_2$ (T_L is constant, $s_{gen} = 0$)

$$\dot{Q}_L = \dot{m}_1 T_L [s_2 - s_1] = 4 \times 540 [0.2469 - 0.2179] = 62.64 \text{ Btu/s}$$

$$\dot{W} = \dot{m}_1 [h_2 - h_1] - \dot{Q}_L = 4 (125.08 - 109.01) - 62.64 = \mathbf{1.64\ Btu/s}$$

9.149E

One technique for operating a steam turbine in part-load power output is to throttle the steam to a lower pressure before it enters the turbine, as shown in Fig. P9.39. The steamline conditions are 200 lbf/in.^{2}, 600 F, and the turbine exhaust pressure is fixed at 1 lbf/in.^{2}. Assuming the expansion inside the turbine to be reversible and adiabatic, determine

a. The full-load specific work output of the turbine
b. The pressure the steam must be throttled to for 80% of full-load output
c. Show both processes in a *T–s* diagram.

a) C.V. Turbine full-load, reversible.

$s_{3a} = s_1 = 1.6767$ Btu/lbm R $= 0.132\ 66 + x_{3a} \times 1.8453$

$x_{3a} = 0.8367$

$h_{3a} = 69.74 + 0.8367 \times 1036.0 = 936.6$ Btu/lbm

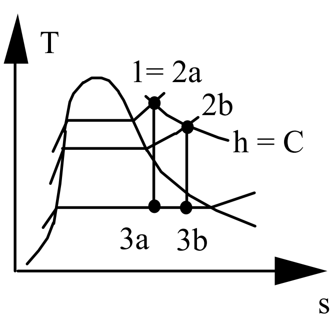
$w = h_1 - h_{3a} = 1322.1 - 936.6 =$ **385.5 Btu/lbm**

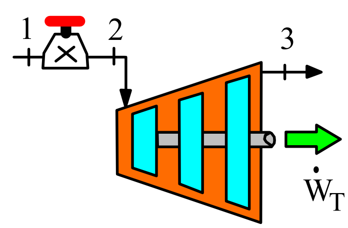
b) $w = 0.80 \times 385.5 = 308.4 = 1322.1 - h_{3b}$ $\Rightarrow$ $h_{3b} = 1013.7$ Btu/lbm

$1013.7 = 69.74 + x_{3b} \times 1036.0$ $\Rightarrow$ $x_{3b} = 0.9112$

$s_{3b} = 0.13266 + 0.9112 \times 1.8453 = 1.8140$ Btu/lbm R

$$\left.\begin{array}{l} s_{2b} = s_{3b} = 1.8140 \\ h_{2b} = h_1 = 1322.1 \end{array}\right\} \rightarrow \begin{array}{l} P_2 = \mathbf{56.6\ lbf/in^2} \\ T_2 = \mathbf{579\ F} \end{array}$$





Steady Irreversible Processes

9.150E

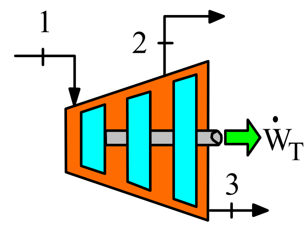
Analyse the steam turbine described in Problem 6.161. Is it possible?

C.V. Turbine. Steady flow and adiabatic.

Continuity Eq.6.9: $\dot{m}_1 = \dot{m}_2 + \dot{m}_3$;

Energy Eq.6.10: $\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}$

Entropy Eq.9.7: $\dot{m}_1 s_1 + \dot{S}_{gen} = \dot{m}_2 s_2 + \dot{m}_3 s_3$



States from Table F.7.2: s_1 = 1.6398 Btu/lbm R, s_2 = 1.6516 Btu/lbm R,

$s_3 = s_f + x\ s_{fg} = 0.283 + 0.95 \times 1.5089 = 1.71$ Btu/lbm R

$\dot{S}_{gen} = 40 \times 1.6516 + 160 \times 1.713 - 200 \times 1.6398 =$ **12.2 Btu/s ·R**

Since it is positive => possible.

Notice the entropy is increasing through turbine: $s_1 < s_2 < s_3$

9.151E

Two flowstreams of water, one at 100 lbf/in.^{2}, saturated vapor, and the other at 100 lbf/in.^{2}, 1000 F, mix adiabatically in a steady flow process to produce a single flow out at 100 lbf/in.2, 600 F. Find the total entropy generation for this process.

Solution:

Continuity Eq.6.9: $\dot{m}_3 = \dot{m}_1 + \dot{m}_2,$

Energy Eq.6.10: $\dot{m}_3 h_3 = \dot{m}_1 h_1 + \dot{m}_2 h_2$

State properties from Table F.7.2

$h_1 = 1187.8$, $h_2 = 1532.1$, $h_3 = 1329.3$ all in Btu/lbm

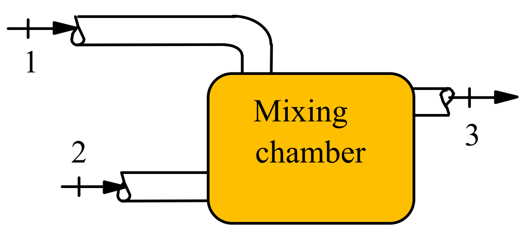
$s_1 = 1.6034$, $s_2 = 1.9204$, $s_3 = 1.7582$ all in Btu/lbm R

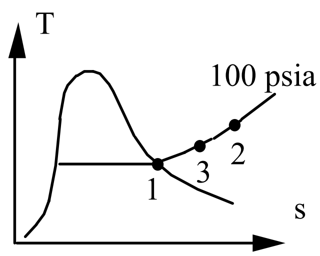
$\Rightarrow \dot{m}_1/\dot{m}_3 = (h_3 - h_2) / (h_1 - h_2) = 0.589$

Entropy Eq.9.7: $\dot{m}_3 s_3 = \dot{m}_1 s_1 + \dot{m}_2 s_2 + \dot{S}_{gen}$ $\Rightarrow$

$\dot{S}_{gen}/\dot{m}_3 = s_3 - (\dot{m}_1/\dot{m}_3)\, s_1 - (\dot{m}_2/\dot{m}_3)\, s_2$

$= 1.7582 - 0.589 \times 1.6034 - 0.411 \times 1.9204 = \mathbf{0.0245\ \frac{Btu}{lbm\ R}}$





9.152E

A mixing chamber receives 10 lbm/min ammonia as saturated liquid at 0 F from one line and ammonia at 100 F, 40 lbf/in.^{2} from another line through a valve. The chamber also receives 340 Btu/min energy as heat transferred from a 100-F reservoir. This should produce saturated ammonia vapor at 0 F in the exit line. What is the mass flow rate at state 2 and what is the total entropy generation in the process?

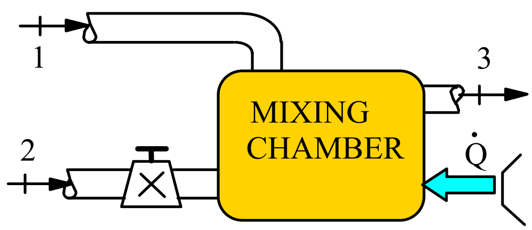
Solution:

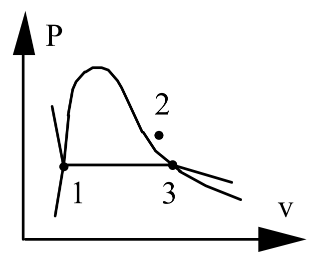
CV: Mixing chamber out to reservoir

Continuity Eq.6.9: $\dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy Eq.6.10: $\dot{m}_1 h_1 + \dot{m}_2 h_2 + \dot{Q} = \dot{m}_3 h_3$

Entropy Eq.9.7: $\dot{m}_1 s_1 + \dot{m}_2 s_2 + \dot{Q}/T_{res} + \dot{S}_{gen} = \dot{m}_3 s_3$





From Table F.8.1: $h_1 = 42.6$ Btu/lbm, $s_1 = 0.0967$ Btu/lbm R

From Table F.8.2: $h_2 = 664.33$ Btu/lbm, $s_2 = 1.4074$ Btu/lbm R

From Table F.8.1: $h_3 = 610.92$ Btu/lbm, $s_3 = 1.3331$ Btu/lbm R

From the energy equation:

$$\dot{m}_2 = \frac{\dot{m}_1(h_1 - h_3) + \dot{Q}}{h_3 - h_2} = \frac{10(42.6 - 610.92) + 340}{610.92 - 664.33} = \mathbf{100.1\ lbm/min}$$

$$\Rightarrow \qquad \dot{m}_3 = 110.1 \text{ lbm/min}$$

$$\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 - \dot{Q}/T_{res}$$

$$= 110.1\times1.3331 - 10\times0.0967 - 100.1\times1.4074 - \frac{340}{559.67} = \mathbf{4.37\ \frac{Btu}{R\ min}}$$

9.153E

A condenser in a power plant receives 10 lbm/s steam at 130 F, quality 90% and rejects the heat to cooling water with an average temperature of 62 F. Find the power given to the cooling water in this constant pressure process and the total rate of enropy generation when condenser exit is saturated liquid.

Solution:

C.V. Condenser. Steady state with no shaft work term.

Energy Eq.6.12: $\dot{m}\, h_i + \dot{Q} = \dot{m} h_e$

Entropy Eq.9.8: $\dot{m}\, s_i + \dot{Q}/T + \dot{S}_{gen} = \dot{m}\, s_e$

Properties are from Table F.7.1

$h_i = 98.0 + 0.9 \times 1019.8 = 1015.8$ Btu/lbm, $h_e = 98.0$ Btu/lbm

$s_i = 0.1817 + 0.9 \times 1.7292 = 1.7380$ Btu/lbm R, $s_e = 0.1817$ Btu/lbm R

$\dot{Q}_{out} = -\dot{Q} = \dot{m}\,(h_i - h_e) = 10(1015.8 - 98.0) =$ **9178 btu/s**

$\dot{S}_{gen} = \dot{m}\,(s_e - s_i) + \dot{Q}_{out}/T$

$= 10(0.1817 - 1.738) + 9178/(459.7 + 62)$

$= -15.563 + 17.592 =$ **2.03 Btu/s-R**

9.154E

Air at 540 F, 60 lbf/in.^{2} with a volume flow 40 ft^3/s runs through an adiabatic turbine with exhaust pressure of 15 lbf/in.^{2}. Neglect kinetic energies and use constant specific heats. Find the lowest and highest possible exit temperature. For each case find also the rate of work and the rate of entropy generation.

$T_i = 540 \text{ F} = 1000 \text{ R}$

$v_i = RT_i / P_i = 53.34 \times 1000/(60 \times 144) = 6.174 \text{ ft}^3 / \text{lbm}$

$\dot{m} = \dot{V} / v_i = 40/6.174 = 6.479 \text{ lbm/s}$

a. **lowest exit T**, this must be reversible for maximum work out.

$T_e = T_i(P_e/P_i)^{\frac{k-1}{k}} = 1000\ (15/60)^{0.286} = \mathbf{673\ R}$

$w = 0.24\ (1000 - 673) = 78.48 \text{ Btu/lbm}$; $\dot{W} = \dot{m}w = \mathbf{508.5\ Btu/s}$

$\dot{S}_{gen} = \mathbf{0}$

b. **Highest exit T**, for no work out. $T_e = T_i = \mathbf{1000\ R}$

$\dot{W} = \mathbf{0}$

$\dot{S}_{gen} = \dot{m}\ (s_e - s_i) = - \dot{m}R \ln (P_e / P_i)$

$= - 6.479 \times \frac{53.34}{778} \ln (15/60) = \mathbf{0.616\ Btu/s{\cdot}R}$

9.155E

A supply of 10 lbm/s ammonia at 80 lbf/in.^{2}, 80 F is needed. Two sources are available one is saturated liquid at 80 F and the other is at 80 lbf/in.^{2}, 260 F. Flows from the two sources are fed through valves to an insulated mixing chamber, which then produces the desired output state. Find the two source mass flow rates and the total rate of entropy generation by this setup.

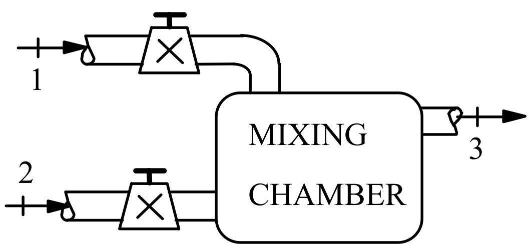
Solution:

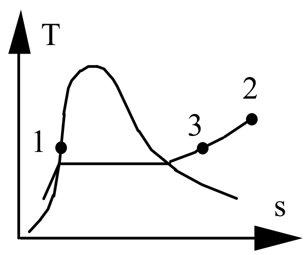
C.V. mixing chamber + valve. Steady, no heat transfer, no work.

Continuity Eq.6.9: $\dot{m}_1 + \dot{m}_2 = \dot{m}_3$;

Energy Eq.6.10: $\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3$

Entropy Eq.9.7: $\dot{m}_1 s_1 + \dot{m}_2 s_2 + \dot{S}_{gen} = \dot{m}_3 s_3$





State 1: Table F.8.1 h_1 = 131.68 Btu/lbm, s_1= 0.2741 Btu/lbm R

State 2: Table F.8.2 h_2 = 748.5 Btu/lbm, s_2 = 1.4604 Btu/lbm R

State 3: Table F.8.2 h_3= 645.63 Btu/lbm, s_3= 1.2956 Btu/lbm R

As all states are known the energy equation establishes the ratio of mass flow rates and the entropy equation provides the entropy generation.

$$\dot{m}_1 h_1 + (\dot{m}_3 - \dot{m}_1) h_2 = \dot{m}_3 h_3 \;\Rightarrow\; \dot{m}_1 = \dot{m}_3 \frac{h_3 - h_2}{h_1 - h_2} = 10 \times \frac{-102.87}{-616.82} = 1.668 \text{ lbm/s}$$

$$\Rightarrow \quad \dot{m}_2 = \dot{m}_3 - \dot{m}_1 = 8.332 \text{ lbm/s}$$

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

$$= 10 \times 1.2956 - 1.668 \times 0.2741 - 8.332 \times 1.46 = \mathbf{0.331\ Btu/s{\cdot}R}$$

Transient Processes

9.156E

An old abandoned saltmine, 3.5 × 106 ft^3 in volume, contains air at 520 R, 14.7 lbf/in.^{2}. The mine is used for energy storage so the local power plant pumps it up to 310 lbf/in.^{2} using outside air at 520 R, 14.7 lbf/in.^{2}. Assume the pump is ideal and the process is adiabatic. Find the final mass and temperature of the air and the required pump work. Overnight, the air in the mine cools down to 720 R. Find the final pressure and heat transfer.

Solution:

C.V. The mine volume and the pump

Continuity Eq.6.15: $m_2 - m_1 = m_{in}$

Energy Eq.6.16: $m_2u_2 - m_1u_1 = {}_1Q_2 - {}_1W_2 + m_{in}h_{in}$

Entropy Eq.9.12: $m_2s_2 - m_1s_1 = \int dQ/T + {}_1S_{2\,gen} + m_{in}s_{in}$

Process: Adiabatic ${}_1Q_2 = 0$, Process ideal ${}_1S_{2\,gen} = 0$, $s_1 = s_{in}$

$\Rightarrow m_2s_2 = m_1s_1 + m_{in}s_{in} = (m_1 + m_{in})s_1 = m_2s_1 \Rightarrow s_2 = s_1$

Constant s $\Rightarrow$ Eq.8.28 $s^o_{T2} = s^o_{Ti} + R \ln(P_e / P_i)$

$$\text{Table F.4} \Rightarrow s^o_{T2} = 1.63074 + \frac{53.34}{778} \ln\left(\frac{310}{14.7}\right) = 1.83976 \text{ Btu/lbm R}$$

$\Rightarrow T_2 =$ **1221 R** , $u_2 = 213.13$ Btu/lbm

Now we have the states and can get the masses

$$m_1 = P_1V_1/RT_1 = \frac{14.7 \times 3.5{\times}10^6 \times 144}{53.34 \times 520} = 2.671{\times}10^5 \text{ lbm}$$

$$m_2 = P_2V_2/RT_2 = \frac{310 \times 3.5{\times}10^6 \times 144}{53.34 \times 1221} = 2.4{\times}10^6 \text{ kg}$$

$\Rightarrow m_{in} = m_2 - m_1 = \mathbf{2.1319{\times}10^6\ lbm}$

${}_1W_2 = m_{in}h_{in} + m_1u_1 - m_2u_2 = 2.1319{\times}10^6 \times 124.38 + 2.671{\times}10^5 \times 88.73 - 2.4{\times}10^6 \times 213.13 = -2.226 \times 10^8$ Btu = -pump work

$W_{pump} = \mathbf{2.23 \times 10^8\ Btu}$

${}_2W_3 = \emptyset$, $P_3 = P_2T_3/T_2 = 310{\times}720/1221 = \mathbf{182.8\ lbf/in^2}$

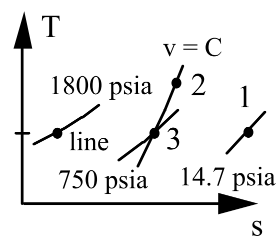
${}_2Q_3 = m_2(u_3 - u_2) = 2.4{\times}10^6(123.17 - 213.13) = \mathbf{-2.16 \times 10^8\ Btu}$

9.157E

Air from a line at 1800 lbf/in.^{2}, 60 F, flows into a 20-ft^3 rigid tank that initially contained air at ambient conditions, 14.7 lbf/in.^{2}, 60 F. The process occurs rapidly and is essentially adiabatic. The valve is closed when the pressure inside reaches some value, P_2. The tank eventually cools to room temperature, at which time the pressure inside is 750 lbf/in.^{2}. What is the pressure P_2? What is the net entropy change for the overall process?

CV: Tank. Mass flows in, so this is transient. Find the mass first

$$m_1 = P_1V/RT_1 = \frac{14.7 \times 144 \times 20}{53.34 \times 520} = 1.526 \text{ lbm}$$



Fill to P_2, then cool to T_3 = 60 F, P_3 = 750 psia

$$m_3 = m_2 = P_3V/RT_3$$

$$= \frac{750 \times 144 \times 20}{53.34 \times 520} = 77.875 \text{ lbm}$$

Cont. Eq.: $m_i = m_2 - m_1 = 77.875 - 1.526 = 76.349$ lbm

Consider the overall process from 1 to 3

Energy Eq.: $Q_{CV} + m_ih_i = m_2u_3 - m_1u_1 = m_2h_3 - m_1h_1 - (P_3 - P_1)V$

But, since $T_i = T_3 = T_1$, $m_ih_i = m_2h_3 - m_1h_1$

$\Rightarrow Q_{CV} = -(P_3 - P_1)V = -(750 - 14.7)\times 20 \times 144/778 = -2722$ Btu

$$\Delta S_{NET} = m_3s_3 - m_1s_1 - m_is_i - Q_{CV}/T_0 = m_3(s_3 - s_i) - m_1(s_1 - s_i) - Q_{CV}/T_0$$

$$= 77.875\left[0 - \frac{53.34}{778}\ln\left(\frac{750}{1800}\right)\right] - 1.526\left[0 - \frac{53.34}{778}\ln\left(\frac{14.7}{1800}\right)\right]$$

$$+ 2722/520 = \mathbf{9.406\ Btu/R}$$

The filling process from 1 to 2 ($T_1 = T_i$)

1-2 heat transfer = 0 so 1st law: $m_ih_i = m_2u_2 - m_1u_1$

$$m_iC_{P0}T_i = m_2C_{V0}T_2 - m_1C_{V0}T_1$$

$$T_2 = \frac{76.349 \times 0.24 + 1.526 \times 0.171}{77.875 \times 0.171} \times 520 = 725.7 \text{ R}$$

$$P_2 = m_2RT_2/V = 77.875 \times 53.34 \times 725.7 / (144 \times 20) = \mathbf{1047\ lbf/in^2}$$

Reversible Shaft Work, Bernoulli

9.158E

Liquid water at ambient conditions, 14.7 lbf/in.^{2}, 75 F, enters a pump at the rate of 1 lbm/s. Power input to the pump is 3 Btu/s. Assuming the pump process to be reversible, determine the pump exit pressure and temperature.

Solution:

C.V. Pump. Steady single inlet and exit flow with no heat transfer.

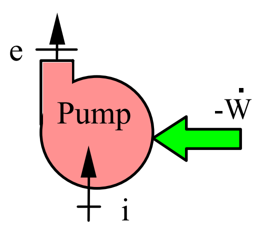
Energy Eq.6.13: $w = h_i - h_e = \dot{W}/\dot{m} = -3/1 = -3.0$ btu/lbm

Using also incompressible media we can use Eq.9.18

$$w_P = -\int v dP \approx -v_i(P_e - P_i) = -0.01606 \text{ ft/lbm}(P_e - 14.7 \text{ psia})$$

from which we can solve for the exit pressure

$$3 \cong 0.01606(P_e - 14.7) \times \frac{144}{778} \Rightarrow P_e = \mathbf{1023.9\ lbf/in^2}$$



$-\dot{W} = 3$ Btu/s, $P_i = 14.7$ psia

$T_i = 75$ F $\dot{m} = 1$ lbm/s

Energy Eq.: $h_e = h_i - w_P = 43.09 + 3 = 46.09$ Btu/lbm

Use Table F.7.3 at 1000 psia => $\mathbf{T_e = 75.3\ F}$

9.159E

A fireman on a ladder 80 ft above ground should be able to spray water an additional 30 ft up with the hose nozzle of exit diameter 1 in. Assume a water pump on the ground and a reversible flow (hose, nozzle included) and find the minimum required power.

Solution:

C.V.: pump + hose + water column, total height difference 35 m. Here **V** is velocity, not volume.

Continuity Eq.6.3, 6.11: $\dot{m}_{in} = \dot{m}_{ex} = (\rho A \mathbf{V})_{nozzle}$

Energy Eq.6.12: $\dot{m}(-w_p) + \dot{m}(h + \mathbf{V}^2/2 + gz)_{in} = \dot{m}(h + \mathbf{V}^2/2 + gz)_{ex}$

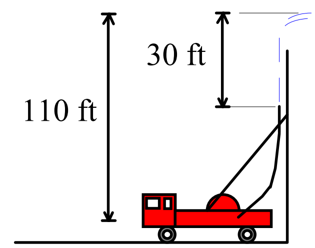
Process: $h_{in} \cong h_{ex}$, $\mathbf{V}_{in} \cong \mathbf{V}_{ex} = 0$, $z_{ex} - z_{in} = 110$ ft, $\rho = 1/v \cong 1/v_f$

$-w_p = g(z_{ex} - z_{in}) = 32.174 \times (110 - 0)/25\,037 = 0.141$ Btu/lbm

Recall the conversion 1 Btu/lbm = 25 037 ft^2/s^2 from Table A.1. The velocity in the exit nozzle is such that it can rise 30 ft. Make that column a C.V. for which Bernoulli Eq.9.17 is:

$$gz_{noz} + \frac{1}{2}\mathbf{V}^2_{noz} = gz_{ex} + 0$$

$$\mathbf{V}_{noz} = \sqrt{2g(z_{ex} - z_{noz})} = \sqrt{2 \times 32.174 \times 30} = 43.94 \text{ ft/s}$$



Assume: $v = v_{F,70F} = 0.01605\ ft^3/lbm$

$$\dot{m} = \frac{\pi}{v_f}\left(\frac{D}{2}\right)^2 \mathbf{V}_{noz} = (\pi/4)\,(1^2/144) \times 43.94 / 0.01605 = 14.92 \text{ lbm/s}$$

$$\dot{W}_{pump} = \dot{m}w_p = 14.92 \times 0.141 \times (3600/2544) = \mathbf{3\ hp}$$

9.160E

Saturated R-134a at 10 F is pumped/compressed to a pressure of 150 lbf/in.^{2} at the rate of 1.0 lbm/s in a reversible adiabatic steady flow process. Calculate the power required and the exit temperature for the two cases of inlet state of the R-134a:

a) quality of 100 %.
b) quality of 0 %.

Solution:

C.V.: Pump/Compressor, $\dot{m}$ = 1 lbm/s, R-134a

a) State 1: Table F.10.1, $x_1 = 1.0$ Saturated vapor, $P_1 = P_g = 26.79$ psia,

$h_1 = h_g = 168.06$ Btu/lbm, $s_1 = s_g = 0.414$ Btu/lbm R

Assume Compressor is isentropic, $s_2 = s_1 = 0.414$ Btu/lbm R

$h_2 = 183.5$ Btu/lbm, $\mathbf{T_2 = 116\ F}$

1st Law Eq.6.13: $q_c + h_1 = h_2 + w_c$; $q_c = 0$

$w_{cs} = h_1 - h_2 = 168.05 – 183.5 = - 15.5$ Btu/lbm;

=> $\dot{W}_C = \dot{m}w_C = \mathbf{-15.5\ Btu/s = 21.9\ hp}$

b) State 1: $T_1 = 10$ F, $x_1 = 0$ Saturated liquid. This is a pump.

$P_1 = 26.79$ psia, $h_1 = h_f = 79.02$ Btu/lbm, $v_1 = v_f = 0.01202$ ft^3/lbm

1st Law Eq.6.13: $q_p + h_1 = h_2 + w_p$; $q_p = 0$

Assume Pump is isentropic and the liquid is incompressible, Eq.9.18:

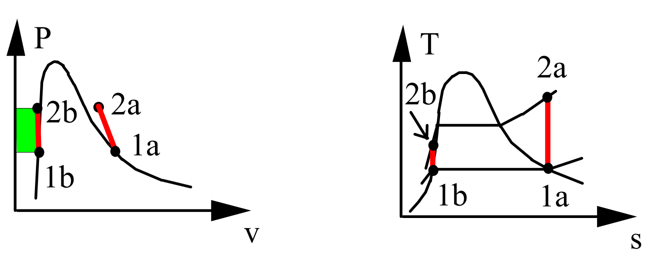
$w_{ps} = - \int v\ dP = -v_1(P_2 - P_1) = -0.01202\ (150 – 26.79)\ 144$

$= -213.3$ lbf-ft/lbm = - 0.274 Btu/lbm

$h_2 = h_1 - w_p = 79.02 - (- 0.274) = 187.3$ Btu/lbm,

Assume State 2 is approximately a saturated liquid => $\mathbf{T_2 \cong 10.9\ F}$

$\dot{W}_P = \dot{m}w_P = 1\ (- 0.274) = \mathbf{-0.27\ Btu/s = -0.39\ hp}$



9.161E

A small pump takes in water at 70 F, 14.7 lbf/in.^{2} and pumps it to 250 lbf/in.^{2} at a flow rate of 200 lbm/min. Find the required pump power input.

Solution:

C.V. Pump. Assume reversible pump and incompressible flow.

This leads to the work in Eq.9.18

$$w_p = -\int v dP = -v_i(P_e - P_i) = -0.01605(250 - 14.7) \times \frac{144}{778} = -0.7 \text{ Btu/lbm}$$

$$\dot{W}_{p\ in} = \dot{m}(-w_p) = \frac{200}{60}(0.7) = \mathbf{2.33\ Btu/s = 3.3\ hp}$$

9.162E

An expansion in a gas turbine can be approximated with a polytropic process with exponent n = 1.25. The inlet air is at 2100 R, 120 psia and the exit pressure is 18 psia with a mass flow rate of 2 lbm/s. Find the turbine heat transfer and power output.

Solution:

C.V. Steady state device, single inlet and single exit flow.

Energy Eq.6.13: $h_i + q = h_e + w$ Neglect kinetic, potential energies

Entropy Eq.9.8: $s_i + \int dq/T + s_{gen} = s_e$

Process Eq.8.37:

$$T_e = T_i (P_e/P_i)^{\frac{n-1}{n}} = 2100 (18/120)^{\frac{0.25}{1.25}} = 1436.9 \text{ R}$$

so the exit enthalpy is from Table F.5, $h_i = 532.6$ Btu/lbm

$$h_e = 343.0 + \frac{36.9}{40}(353.5 - 343.0) = 352.7 \text{ Btu/lbm}$$

The process leads to Eq.9.19 for the work term

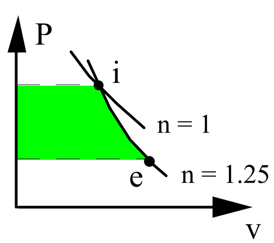
$$\dot{W} = \dot{m}w = -\dot{m}\frac{nR}{n-1}(T_e - T_i) = -2\,\frac{1.25 \times 53.34}{0.25 \times 778} \times (1436.9 - 2100)$$

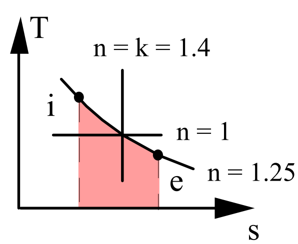
$= \mathbf{454.6\ Btu/s}$

Energy equation gives

$$\dot{Q} = \dot{m}q = \dot{m}(h_e - h_i) + \dot{W} = 2(352.7 - 532.6) + 454.6$$

$= -359.8 + 454.6 = \mathbf{94.8\ Btu/s}$





Notice:
dP < 0
so dw > 0

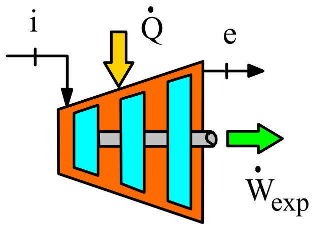
ds > 0
so dq > 0

Notice this process has some heat transfer in during expansion which is unusual. The typical process would have n = 1.5 with a heat loss.

9.163E

Helium gas enters a steady-flow expander at 120 lbf/in.^{2}, 500 F, and exits at 18 lbf/in.^{2}. The mass flow rate is 0.4 lbm/s, and the expansion process can be considered as a reversible polytropic process with exponent, *n* = 1.3. Calculate the power output of the expander.

Solution:



CV: expander, reversible polytropic process.
From Eq.8.37:

$$T_e = T_i \left(\frac{P_e}{P_i}\right)^{\frac{n-1}{n}} = 960 \left(\frac{18}{120}\right)^{\frac{0.3}{1.3}} = 619.6 \text{ R}$$

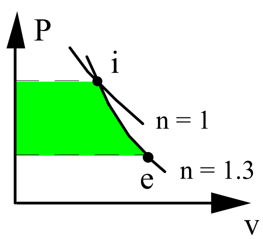
Table F.4: R = 386 lbf-ft/lbm-R

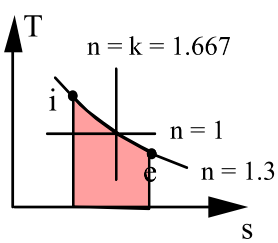
Work evaluated from Eq.9.19

$$w = - \int v dP = - \frac{nR}{n-1}(T_e - T_i) = - \frac{1.3 \times 386}{0.3 \times 778}(619.6 - 960)$$

$$= +731.8 \text{ Btu/lbm}$$

$$\dot{W} = \dot{m}w = 0.4 \times 731.8 \times \frac{3600}{2544} = \mathbf{414\ hp}$$





Device Efficiency

9.164E

A compressor is used to bring saturated water vapor at 103 lbf/in.^{2} up to 2000 lbf/in.^{2}, where the actual exit temperature is 1200 F. Find the isentropic compressor efficiency and the entropy generation.

Solution:

C.V. Compressor. Assume adiabatic and neglect kinetic energies.

Energy Eq.6.13: $w = h_1 - h_2$

Entropy Eq.9.8: $s_2 = s_1 + s_{gen}$

We have two different cases, the ideal and the actual compressor.

States: 1: F.7.1 $h_1 = 1188.36$ Btu/lbm, $s_1 = 1.601$ Btu/lbm R

2ac: F.7.2 $h_{2,AC} = 1598.6$ Btu/lbm, $s_{2,AC} = 1.6398$ Btu/lbm R

2s: F.7.2 $(P, s = s_1)$ $h_{2,s} = 1535.1$ Btu/lbm

IDEAL:
$-w_{c,s} = h_{2,s} - h_1 = 346.7$ Btu/lbm

ACTUAL:
$-w_{C,AC} = h_{2,AC} - h_1 = 410.2$ Btu/lbm

Definition Eq.9.28: $\eta_c = w_{c,s}/w_{c,AC} =$ **0.845 ~ 85%**

Entropy Eq.9.8:

$s_{gen} = s_{2\ ac} - s_1 = 1.6398 - 1.601 =$ **0.0388 Btu/lbm R**

9.165E

A small air turbine with an isentropic efficiency of 80% should produce 120 Btu/lbm of work. The inlet temperature is 1800 R and it exhausts to the atmosphere. Find the required inlet pressure and the exhaust temperature.

Solution:

C.V. Turbine actual energy Eq.6.13:

$$w = h_i - h_{e,ac} = 120$$

Table F.5: $h_i = 449.794$ Btu/lbm

$\Rightarrow$ $h_{e,ac} = h_i - 120 = 329.794$ Btu/lbm, $\mathbf{T_e = 1349\ R}$

C.V. Ideal turbine, Eq.9.27 and energy Eq.6.13:

$w_s = w/\eta_s = 120/0.8 = 150 = h_i - h_{e,s}$ $\Rightarrow$ $h_{e,s} = 299.794$ Btu/lbm

From Table F.5: $T_{e,s} = 1232.7$ R , $s^o_{Te} = 1.84217$ Btu/lbm R

Entropy Eq.9.8: $s_i = s_{e,s}$ adiabatic and reversible

To relate the entropy to the pressure use Eq.8.28 inverted and standard entropy from Table F.5:

$$P_e/P_i = \exp[\,(s^o_{Te} - s^o_{Ti}\,)/R\,] = \exp[(1.84217 - 1.94209)\frac{778}{53.34}] = 0.2328$$

$P_i = P_e\ /\ 0.2328 = 14.7/0.2328 =$ **63.14 psia**

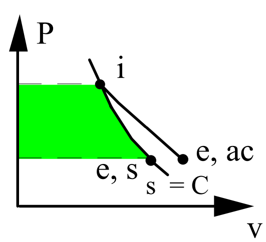
If constant heat capacity was used

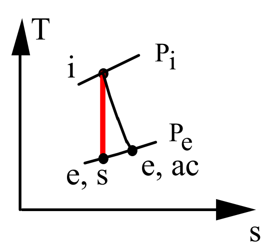
$T_e = T_i - w/C_p = 1800 - 120/0.24 = 1300$ R

$T_{e,s} = T_i - w_s/C_p = 1800 - 150/0.24 = 1175$ R

The constant s relation is Eq.8.32

$P_e/P_i = (T_e/T_i)^{k/(k-1)}$ $\Rightarrow$ $P_i = 14.7\ (1800/1175)^{3.5} = 65.4$ psia





9.166E

Air enters an insulated compressor at ambient conditions, 14.7 lbf/in.^{2}, 70 F, at the rate of 0.1 lbm/s and exits at 400 F. The isentropic efficiency of the compressor is 70%. What is the exit pressure? How much power is required to drive the compressor?

Solution:

C.V. Compressor: P_1, T_1, T_e(real), $\eta_{s\ COMP}$ known, assume constant C_{P0}

Energy Eq.6.13 for real: $-w = C_{P0}(T_e - T_i) = 0.24(400 - 70) = 79.2$ Btu/lbm

Ideal $-w_s = -w \times \eta_s = 79.2 \times 0.7 = 55.4$ Btu/lbm

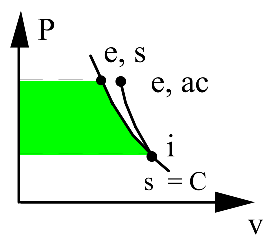
Energy Eq.6.13 for ideal:

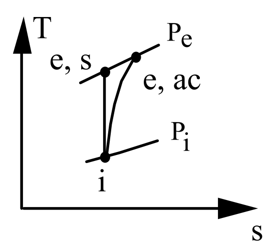
$55.4 = C_{P0}(T_{es} - T_i) = 0.24(T_{es} - 530)$, $T_{es} = 761$ R

Constant entropy for ideal as in Eq.8.32:

$P_e = P_i(T_{es}/T_i)^{\frac{k}{k-1}} = 14.7(761/530)^{3.5} = \mathbf{52.1\ lbf/in^2}$

$-\dot{W}_{REAL} = \dot{m}(-w) = 0.1 \times 79.2 \times 3600/2544 = \mathbf{11.2\ hp}$





9.167E

A watercooled air compressor takes air in at 70 F, 14 lbf/in.^{2} and compresses it to 80 lbf/in.^{2}. The isothermal efficiency is 80% and the actual compressor has the same heat transfer as the ideal one. Find the specific compressor work and the exit temperature.

Solution:

Ideal isothermal compressor exit 80 psia, 70 F

Reversible process: $dq = T\,ds \;\Rightarrow\; q = T(s_e - s_i)$

$$q = T(s_e - s_i) = T[s^o_{Te} - s^o_{T1} - R \ln(P_e / P_i)]$$

$$= - RT \ln (P_e / P_i) = - (460 + 70) \frac{53.34}{778} \ln \frac{80}{14} = - 63.3 \text{ Btu/lbm}$$

As same temperature for the ideal compressor $h_e = h_i \Rightarrow$

$w = q = -63.3$ Btu/lbm $\Rightarrow$ $w_{ac} = w / \eta =$ **- 79.2 Btu/lbm**, $q_{ac} = q$

Now for the actual compressor energy equation becomes

$$q_{ac} + h_i = h_{e\,ac} + w_{ac} \Rightarrow$$

$$h_{e\,ac} - h_i = q_{ac} - w_{ac} = - 63.3 - (-79.2) = 15.9 \text{ Btu/lbm} \approx C_p (T_{e\,ac} - T_i)$$

$T_{e\,ac} = T_i + 15.9/0.24 =$ **136 F**

9.168E

A nozzle is required to produce a steady stream of R-134a at 790 ft/s at ambient conditions, 15 lbf/in.^{2}, 70 F. The isentropic efficiency may be assumed to be 90%. What pressure and temperature are required in the line upstream of the nozzle?

C.V. Nozzle, steady flow and no heat transfer.

Actual nozzle energy Eq.: $h_1 = h_2 + \mathbf{V}_2^2/2$

State 2 actual: Table F.10.2 $\quad h_2 = 180.975$ Btu/lbm

$$h_1 = h_2 + \mathbf{V}_2^2/2 = 180.975 + \frac{790^2}{2 \times 25\,037} = 193.44 \text{ Btu/lbm}$$

Recall 1 Btu/lbm = 25 037 ft^2/s^2 from Table A.1.

Ideal nozzle exit: $h_{2s} = h_1 - KE_s = 193.44 - \frac{790^2}{2 \times 25\,037}/0.9 = 179.59$ Btu/lbm

State 2s: $(P_2, h_{2s}) \Rightarrow T_{2s} = 63.16$ F, $\quad s_{2s} = 0.4481$ Btu/lbm R

Entropy Eq. ideal nozzle: $s_1 = s_{2s}$

State 1: $(h_1, s_1 = s_{2s}) \Rightarrow$ Double interpolation or use software.

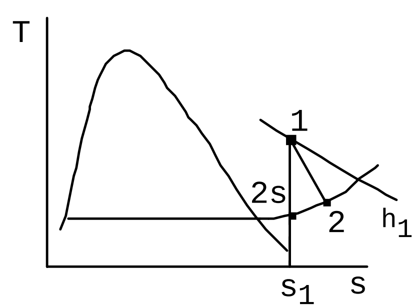
For 40 psia: given h_1 then $s = 0.4544$ Btu/lbm R, $\quad T = 134.47$ F

For 60 psia: given h_1 then $s = 0.4469$ Btu/lbm R, $\quad T = 138.13$ F

Now a linear interpolation to get P and T for proper s

$$P_1 = 40 + 20\,\frac{0.4481 - 0.4544}{0.4469 - 0.4544} = \mathbf{56.8\ psia}$$

$$T_1 = 134.47 + (138.13 - 134.47)\frac{0.4481 - 0.4544}{0.4469 - 0.4544} = \mathbf{137.5\ F}$$



9.169E

Redo Problem 9.159 if the water pump has an isentropic efficiency of 85% (hose, nozzle included).

Solution:

C.V.: pump + hose + water column, total height difference 35 m. Here **V** is velocity, not volume.

Continuity Eq.6.3, 6.11: $\dot{m}_{in} = \dot{m}_{ex} = (\rho A \mathbf{V})_{nozzle}$

Energy Eq.6.12: $\dot{m}(-w_p) + \dot{m}(h + \mathbf{V}^2/2 + gz)_{in} = \dot{m}(h + \mathbf{V}^2/2 + gz)_{ex}$

Process: $h_{in} \cong h_{ex}$, $\mathbf{V}_{in} \cong \mathbf{V}_{ex} = 0$, $z_{ex} - z_{in} = 110$ ft, $\rho = 1/v \cong 1/v_f$

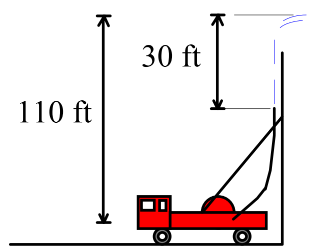
$$-w_p = g(z_{ex} - z_{in}) = 32.174 \times (110 - 0)/25\,037 = 0.141 \text{ Btu/lbm}$$

Recall the conversion 1 Btu/lbm = 25 037 ft^2/s^2 from Table A.1. The velocity in the exit nozzle is such that it can rise 30 ft. Make that column a C.V. for which Bernoulli Eq.9.17 is:

$$gz_{noz} + \frac{1}{2}\mathbf{V}^2_{noz} = gz_{ex} + 0$$

$$\mathbf{V}_{noz} = \sqrt{2g(z_{ex} - z_{noz})}$$

$$= \sqrt{2 \times 32.174 \times 30} = 43.94 \text{ ft/s}$$



Assume: $v = v_{F,70F} = 0.01605\ ft^3/lbm$

$$\dot{m} = \frac{\pi}{v_f}\left(\frac{D}{2}\right)^2 \mathbf{V}_{noz} = (\pi/4)\,(1^2/144) \times 43.94 / 0.01605 = 14.92 \text{ lbm/s}$$

$$\dot{W}_{pump} = \dot{m}w_p/\eta = 14.92 \times 0.141 \times (3600/2544)/0.85 = \mathbf{3.5\ hp}$$

9.170E

Repeat Problem 9.160 for a pump/compressor isentropic efficiency of 70%.

Solution:

C.V.: Pump/Compressor, $\dot{m}$ = 1 lbm/s, R-134a

a) State 1: Table F.10.1, $x_1 = 1.0$ Saturated vapor, $P_1 = P_g = 26.79$ psia,

$h_1 = h_g = 168.06$ Btu/lbm, $s_1 = s_g = 0.414$ Btu/lbm R

Assume Compressor is isentropic, $s_2 = s_1 = 0.414$ Btu/lbm R

$h_2 = 183.5$ Btu/lbm, $T_2 = 116$ F

1st Law Eq.6.13: $q_c + h_1 = h_2 + w_c$; $q_c = 0$

$w_{cs} = h_1 - h_2 = 168.05 – 183.5 = - 15.5$ Btu/lbm;

Now the actual compressor

$w_{c,\ AC} = w_{cs}/\eta = - 22.1 = h_1 – h_{2\ AC}$

$h_{2,\ AC} = 168.06 + 22.1 = 190.2 \Rightarrow T_2 =$ **141.9 F**

=> $\dot{W}_{C\ in} = \dot{m}(-w_C) =$ **22.1 Btu/s = 31.3 hp**

b) State 1: $T_1 = 10$ F, $x_1 = 0$ Saturated liquid. This is a pump.

$P_1 = 26.79$ psia, $h_1 = h_f = 79.02$ Btu/lbm, $v_1 = v_f = 0.01202$ ft^3/lbm

1st Law Eq.6.13: $q_p + h_1 = h_2 + w_p$; $q_p = 0$

Assume Pump is isentropic and the liquid is incompressible, Eq.9.18:

$w_{ps} = - \int v\ dP = -v_1(P_2 - P_1) = -0.01202\ (150 – 26.79)\ 144$

$= -213.3$ lbf-ft/lbm $= - 0.274$ Btu/lbm

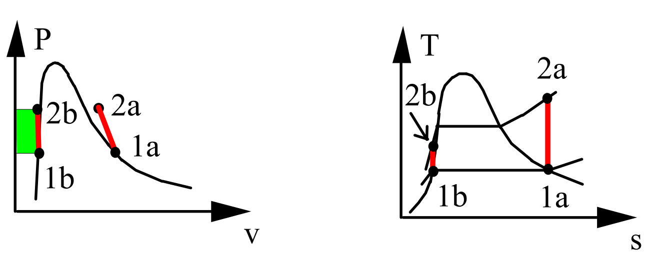
Now the actual pump

$w_{c,\ AC} = w_{cs}/\eta = - 0.391 = h_1 – h_{2\ AC}$

$h_2 = h_1 - w_p = 79.02 - (- 0.391) = 79.41$ Btu/lbm,

Assume State 2 is approximately a saturated liquid => $T_2 \cong$ **11.2 F**

$\dot{W}_{P\ in} = \dot{m}(-w_P) = 1\ (0.391) =$ **0.39 Btu/s** = **0.55 hp**



Review Problems

9.171E

A rigid 35 ft^3 tank contains water initially at 250 F, with 50 % liquid and 50% vapor, by volume. A pressure-relief valve on the top of the tank is set to 150 lbf/in.^{2} (the tank pressure cannot exceed 150 lbf/in.^{2} - water will be discharged instead). Heat is now transferred to the tank from a 400 F heat source until the tank contains saturated vapor at 150 lbf/in.^{2}. Calculate the heat transfer to the tank and show that this process does not violate the second law.

C.V. Tank. $v_{f1} = 0.017$ $v_{g1} = 13.8247$

$m_{LIQ} = V_{LIQ} / vf1 = 0.5 \times 35/0.017 = 1029.4$ lbm

$m_{VAP} = V_{VAP} / vg1 = 0.5 \times 35/13.8247 = 1.266$ lbm

$m = 1030.67$ lbm

$x = m_{VAP} / (m_{LIQ} + m_{VAP}) = 0.001228$

$u = u_f + x\ u_{fg} = 218.48 + 0.001228 \times 869.41 = 219.55$

$s = s_f + x\ s_{fg} = 0.3677 + 0.001228 \times 1.3324 = 0.36934$

state 2: $v_2 = v_g = 3.2214$ $u_2 = 1110.31$ $h_2 = 1193.77$

$s_2 = 1.576$ $m_2 = V/v_2 = 10.865$ lbm

$Q = m_2 u_2 - m_1 u_1 + m_e h_e + W$

$= 10.865 \times 1110.31 - 1030.67 \times 219.55 + 1019.8 \times 1193.77 = 1003187$ Btu

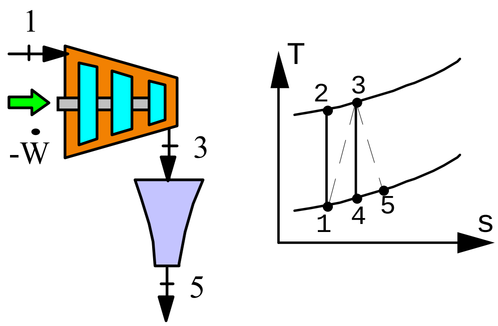
$\dot{S}_{gen} = m_2 s_2 - m_1 s_1 - m_e s_e - {}_1Q_2 / T_{source}$

$= 10.865 \times 1.576 - 1030.67 \times 0.36934 + 1019.8 \times 1.57 - 1003187/860$

$=$ **77.2 Btu/s · R**

9.172E

Air at 1 atm, 60 F is compressed to 4 atm, after which it is expanded through a nozzle back to the atmosphere. The compressor and the nozzle both have efficiency of 90% and kinetic energy in/out of the compressor can be neglected. Find the actual compressor work and its exit temperature and find the actual nozzle exit velocity.



Steady state separate control volumes around compressor and nozzle. For ideal compressor we have inlet : 1 and exit : 2

Adiabatic : $q = 0$.

Reversible: $s_{gen} = 0$

Energy Eq.: $h_1 + 0 = w_C + h_2$;

Entropy Eq.: $s_1 + 0/T + 0 = s_2$

Ideal compressor: $w_c = h_1 - h_2$, $s_2 = s_1$

The constant s from Eq. 8.25 gives

$$T_2 = T_1 (P_2/P_1)^{\frac{k-1}{k}} = (459.7 + 60) \times (4/1)^{0.2857} = 772 \text{ R}$$

$$\Rightarrow \quad -w_C = h_2 - h_1 = C_P(T_2 - T_1) = 0.24\,(772 - 519.7) = 60.55 \text{ Btu/lbm}$$

Actual compressor: $w_{c,AC} = w_{c,s}/\eta_c =$ **-67.3 Btu/lbm** $= h_1 - h_3$

$$\Rightarrow \quad T_3 = T_1 - w_{c,AC}/C_P = 519.7 + 67.3/0.24 = \mathbf{800\ R}$$

Ideal nozzle: $s_4 = s_3$ so use Eq.8.25 again

$$\Rightarrow T_4 = T_3 \times (P_4/P_3)^{\frac{k-1}{k}} = 800\,(1/4)^{0.2857} = 538.4 \text{ R}$$

$$\mathbf{V}_s^2/2 = h_3 - h_4 = C_P(T_3 - T_4) = 0.24(800 - 538.4) = 62.78 \text{ Btu/lbm}$$

$$\mathbf{V}_{AC}^2/2 = \mathbf{V}_s^2 \times \eta_{NOZ}/2 = 62.78 \times 0.9 = 56.5 \text{ Btu/lbm}$$

$$\mathbf{V}_{AC} = \sqrt{2 \times 56.5 \times 25\,037} = \mathbf{1682\ ft/s}$$

Remember conversion 1 Btu/lbm = 25 037 ft^2/s^2 from Table A.1.