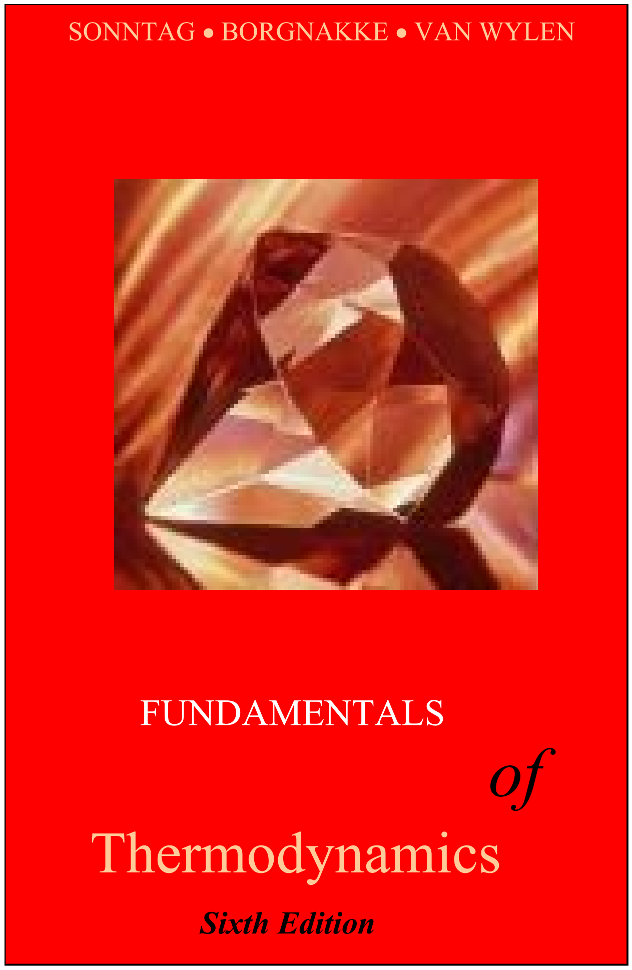
SOLUTION MANUAL
ENGLISH UNIT PROBLEMS
CHAPTER 11



CHAPTER 11

SUBSECTION	PROB NO.
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Otto, Diesel, Stirling and Carnot Cycles	188-201
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Correspondence List

The correspondence between the new English unit problem set and the previous 5th edition chapter 11 problem set and the current SI problems.

New	5th	SI	New	5th	SI	New	5th	SI
167	117 mod	21	184	new	73	201	148b	118
168	118 mod	22	185	133	74	202	new	-
169	new	24	186	136	86	203	149	120
170	119	26	187	137	89	204	150	121
171	120	27	188	138	93	205	151	125
172	new	32	189	139	95	206	new	130
173	121 mod	33	190	new	97	207	new	137
174	122 mod	35	191	new	98	208	new	146
175	123 mod	37	192	141	104	209	155	147
176	125 mod	45	193	140	105	210	new	148
177	124	48	194	142	107	211	new	150
178	127 mod	55	195	143	109	212	154	144
179	128 mod	57	196	144	112	213	126	-
180	129	60	197	145	113	214	130	157
181	131 mod	66	198	146	114	215	134	160
182	132	71	199	147	116	216	135	160
183	new	72	200	148a	117	217	153	134

Rankine cycles

11.167E

A steam power plant, as shown in Fig. 11.3, operating in a Rankine cycle has saturated vapor at 600 lbf/in.^{2} leaving the boiler. The turbine exhausts to the condenser operating at 2.225 lbf/in.^{2}. Find the specific work and heat transfer in each of the ideal components and the cycle efficiency.

Solution:

For the cycle as given:

1: $h_1 = 97.97$ Btu/lbm, $v_1 = 0.01625$ ft^3/lbm,

3: $h_3 = h_g = 1204.06$ Btu/lbm, $s_3 = s_g = 1.4464$ Btu/lbm R

C.V. Pump Reversible and adiabatic.

Energy: $w_p = h_2 - h_1$; Entropy: $s_2 = s_1$

since incompressible it is easier to find work (positive in) as

$$w_P = \int v\, dP = v_1(P_2 - P_1) = 0.01625(600 - 2.2)\frac{144}{778} = \mathbf{1.8\ Btu/lbm}$$

$$h_2 = h_1 + w_P = 97.97 + 1.8 = 99.77 \text{ Btu/lbm}$$

C.V. Boiler: $q_H = h_3 - h_2 = 1204.06 - 99.77 =$ **1104.3 Btu/lbm**

C.V. Tubine: $w_T = h_3 - h_4$, $s_4 = s_3$

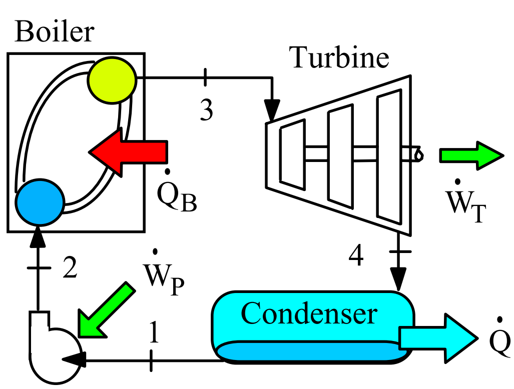
$$s_4 = s_3 = 1.4464 = 0.1817 + x_4 \times 1.7292 \quad \Rightarrow \quad x_4 = 0.7314,$$

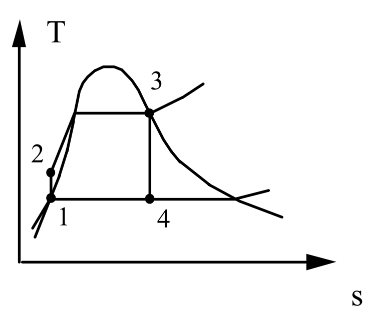
$$h_4 = 97.97 + 0.7314 \times 1019.78 = 843.84 \text{ Btu/lbm}$$

$$w_T = 1204.06 - 843.84 = \mathbf{360.22\ Btu/lbm}$$

$$\eta_{CYCLE} = (w_T - w_P)/q_H = (360.22 - 1.8)/1104.3 = \mathbf{0.325}$$

C.V. Condenser: $q_L = h_4 - h_1 = 843.84 - 97.97 =$ **745.9 Btu/lbm**





11.168E

Consider a solar-energy-powered ideal Rankine cycle that uses water as the working fluid. Saturated vapor leaves the solar collector at 350 F, and the condenser pressure is 0.95 lbf/in.^{2}. Determine the thermal efficiency of this cycle.

H_2O ideal Rankine cycle

CV: turbine

State 3: Table F.7.1 $h_3 = 1193.1$ Btu/lbm, $s_3 = 1.5793$ Btu/lbm R

$s_4 = s_3 = 1.5793 = 0.1296 + x_4 \times 1.8526 \quad => \quad x_4 = 0.7825$

$h_4 = 68.04 + 0.7825 \times 1036.98 = 879.5$ Btu/lbm

$w_T = h_3 - h_4 = 1193.1 - 879.5 = 313.6$ Btu/lbm

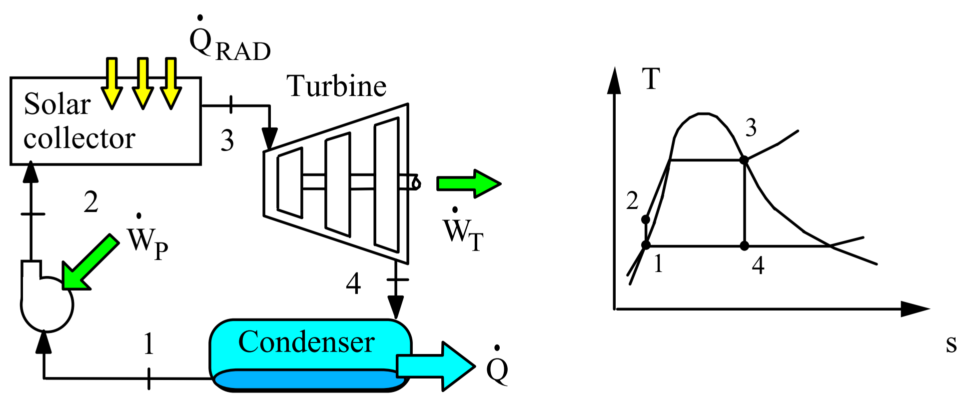
$$w_P = \int v dP \approx v_1(P_2 - P_1) = 0.01613(134.54 - 0.95)\frac{144}{778} = 0.4 \text{ Btu/lbm}$$

$\Rightarrow w_{NET} = w_T - w_P = 313.6 - 0.4 = 313.2$ Btu/lbm

$h_2 = h_1 + w_P = 68.04 + 0.4 = 68.44$ Btu/lbm

$q_H = h_3 - h_2 = 1193.1 - 68.44 = 1124.7$ Btu/lbm

$\eta_{TH} = w_{NET}/q_H = 313.2/1124.7 =$ **0.278**



11.169E

A Rankine cycle uses ammonia as the working substance and powered by solar energy. It heats the ammonia to 320 F at 800 psia in the boiler/superheater. The condenser is water cooled, and the exit is kept at 70 F. Find (T, P, and x if applicable) for all four states in the cycle.

NH_3 ideal Rankine cycle

State 1: Table F.8.1, T = 70 F, x = 0, P_1 = 128.85 psia,

h_1 = 120.21 Btu/lbm, v_1 = 0.2631 ft^3/lbm

CV Pump:

$$w_P = h_2 - h_1 = \int vdP \approx v_1(P_2 - P_1) = 0.02631(800 - 128.85)\frac{144}{778}$$

$$= 3.27 \text{ Btu/lbm}$$

$h_2 = h_1 + w_P$ = 120.21 + 3.27 = 123.48 Btu/lbm = h_f => T_2 = 72.8 F

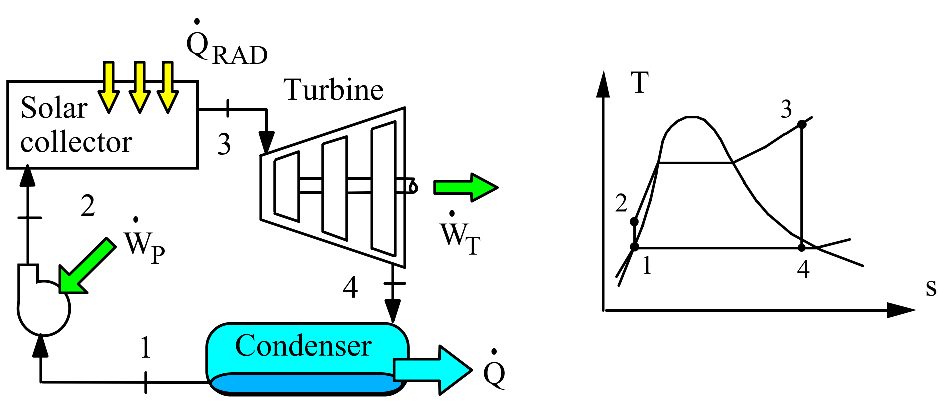
[we need the computer software to do better (P_2, $s_2 = s_1$)]

State 3: 320 F, 800 psia : superheated vapor, s_3 = 1.1915 Btu/lbm

CV: turbine

$s_4 = s_3$ = 1.1915 = 0.2529 + x_4 × 0.9589 => x_4 = 0.9788

$P_4 = P_1$ = 128.85 psia, $T_4 = T_1$ = 70 F



11.170E

A supply of geothermal hot water is to be used as the energy source in an ideal Rankine cycle, with R-134a as the cycle working fluid. Saturated vapor R-134a leaves the boiler at a temperature of 180 F, and the condenser temperature is 100 F. Calculate the thermal efficiency of this cycle.

Solution:

CV: Pump (use R-134a Table F.10)

$P_1 = 138.93$ psia, $P_2 = P_3 = 400.4$ psia

$h_3 = 184.36$ Btu/lbm, $s_3 = 0.402$ Btu/lbm R

$h_1 = 108.86$ Btu/lbm, $v_1 = 0.01387$ ft^3/lbm

$$w_P = h_2 - h_1 = \int_1^2 v dP \approx v_1(P_2 - P_1)$$

$$= 0.01387(400.4 - 138.93)\,\frac{144}{778} = 0.671 \text{ Btu/lbm}$$

$$h_2 = h_1 + w_P = 108.86 + 0.671 = 109.53 \text{ Btu/lbm}$$

CV: Boiler

$$q_H = h_3 - h_2 = 184.36 - 109.53 = 74.83 \text{ Btu/lbm}$$

CV: Turbine

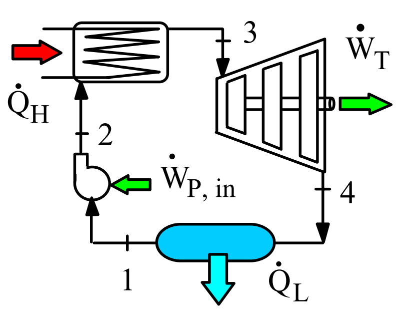
$$s_4 = s_3 = 0.402 \;\Rightarrow\; x_4 = (0.402 - 0.2819)/0.1272 = 0.9442$$

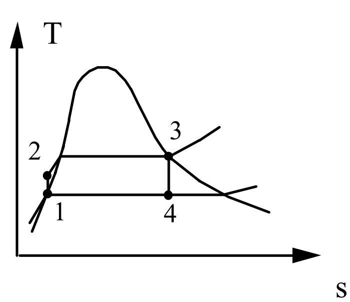
$h_4 = 176.08$ Btu/lbm,

Energy Eq.: $w_T = h_3 - h_4 = 8.276$ Btu/lbm

$$w_{NET} = w_T - w_P = 8.276 - 0.671 = 7.605 \text{ Btu/lbm}$$

$$\eta_{TH} = w_{NET} / q_H = 7.605/74.83 = \mathbf{0.102}$$





11.171E

Do Problem 11.170 with R-22 as the working fluid.

Standard Rankine cycle with properties from the R-22 tables,

h_1 = 39.267 Btu/lbm, v_1 = 0.01404 ft^3/lbm, P_1 = 210.6 psia,

$P_2 = P_3$ = 554.8 psia, h_3 = 110.07 Btu/lbm, s_3 = 0.1913 Btu/lbm R

CV: Pump $\quad w_P = v_1(P_2-P_1) = 0.01404\ (554.8-210.6)\frac{144}{778} = 0.894$ Btu/lbm

$h_2 = h_1 + w_P = 39.267 + 0.894 = 40.16$ Btu/lbm

CV: Turbine $\quad s_4 = s_3$

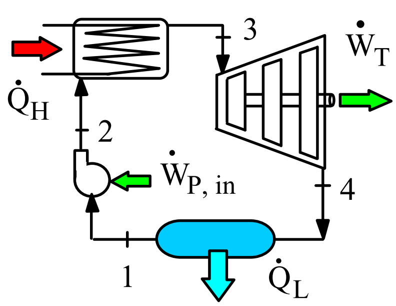
$\Rightarrow\ x_4 = (0.1913 - 0.07942)/0.13014 = 0.9442$

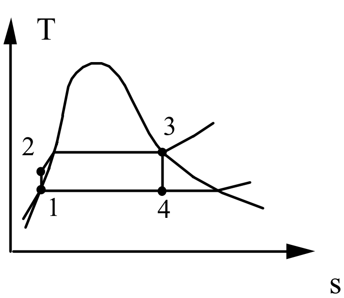
h_4 = 101.885 Btu/lbm, $w_T = h_3 - h_4 = 8.185$ Btu/lbm

CV: Boiler

$q_H = h_3 - h_2 = 110.07 - 40.16 = 69.91$ Btu/lbm

$\eta_{TH} = (w_T - w_P)/q_H = (8.185 - 0.894)/157.21 = \mathbf{0.104}$





11.172E

A smaller power plant produces 50 lbm/s steam at 400 psia, 1100 F, in the boiler. It cools the condenser with ocean water coming in at 55 F and returned at 60 F so that the condenser exit is at 110 F. Find the net power output and the required mass flow rate of the ocean water.

Solution:

The states properties from Tables F.7.1 and F.7.2

1: 110 F, x = 0: $h_1 = 78.01$ Btu/lbm, $v_1 = 0.01617$ ft^3/lbm, $P_{sat} = 1.28$ psia

3: 400 psia, 1100 F: $h_3 = 1577.44$ Btu/lbm, $s_3 = 1.7989$ Btu/lbm R

C.V. Pump Reversible and adiabatic.

Energy: $w_p = h_2 - h_1$; Entropy: $s_2 = s_1$

since incompressible it is easier to find work (positive in) as

$$w_p = \int v\, dP = v_1 (P_2 - P_1) = 0.01617 (400 - 1.3)\frac{144}{778} = 1.19 \text{ Btu/lbm}$$

C.V. Turbine : $w_T = h_3 - h_4$; $s_4 = s_3$

$$s_4 = s_3 = 1.7989 = 0.1473 + x_4 (1.8101) \quad => \quad x_4 = 0.9124$$

$$=> h_4 = 78.01 + 0.9124 (1031.28) = 1018.95 \text{ Btu/lbm}$$

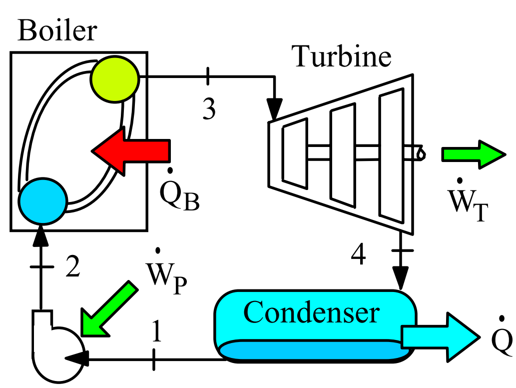
$$w_T = 1577.44 – 1018.95 = 558.5 \text{ Btu/lbm}$$

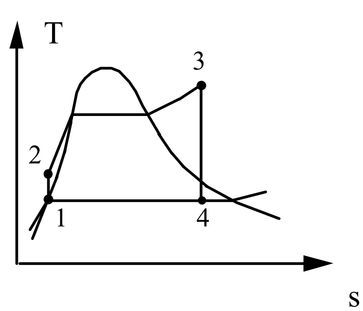
$$\dot{W}_{NET} = \dot{m}(w_T – w_p) = 50 (558.5 – 1.19) = \mathbf{27\,866\ Btu/s}$$

C.V. Condenser : $q_L = h_4 - h_1 = 1018.95 - 78.01 = 940.94$ Btu/lbm

$$\dot{Q}_L = \dot{m}q_L = 50 \times 940.94 = 47\,047 \text{ Btu/s} = \dot{m}_{ocean} C_p \Delta T$$

$$\dot{m}_{ocean} = \dot{Q}_L / C_p \Delta T = 47\,047 / (1.0 \times 5) = \mathbf{9409\ lbm/s}$$





11.173E

The power plant in Problem 11.167 is modified to have a superheater section following the boiler so the steam leaves the super heater at 600 lbf/in.^{2}, 700 F. Find the specific work and heat transfer in each of the ideal components and the cycle efficiency.

Solution:

For this cycle from Table F.7

State 3: Superheated vapor $h_3 = 1350.62$ Btu/lbm, $s_3 = 1.5871$ Btu/lbm R,

State 1: Saturated liquid $h_1 = 97.97$ Btu/lbm, $v_1 = 0.01625$ ft^3/lbm

C.V. Pump: Adiabatic and reversible. Use incompressible fluid so

$$w_P = \int v\, dP = v_1(P_2 - P_1) = 0.01625(600 - 2.2)\frac{144}{778} = \mathbf{1.8\ Btu/lbm}$$

$$h_2 = h_1 + w_P = 95.81 \text{ Btu/lbm}$$

C.V. Boiler: $q_H = h_3 - h_2 = 1350.62 - 97.97 =$ **1252.65 Btu/lbm**

C.V. Tubine: $w_T = h_3 - h_4$, $s_4 = s_3$

$$s_4 = s_3 = 1.5871 \text{ Btu/lbm R} = 0.1817 + x_4\, 1.7292 \Rightarrow x_4 = 0.8127,$$

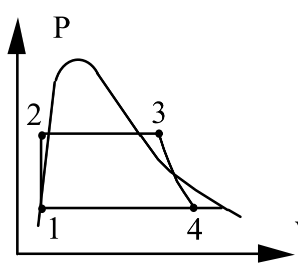
$$h_4 = 97.97 + 0.8127 \times 1019.78 = 926.75 \text{ Btu/lbm}$$

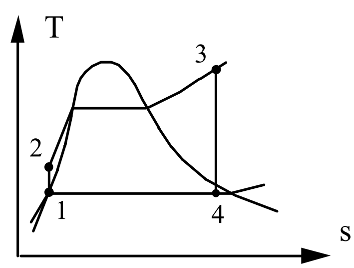
$$w_T = 1350.62 - 926.75 = \mathbf{423.87\ Btu/lbm}$$

$$\eta_{CYCLE} = (w_T - w_P)/q_H = (423.87 - 1.8)/1252.65 = \mathbf{0.337}$$

C.V. Condenser:

$$q_L = h_4 - h_1 = 926.75 - 97.97 = \mathbf{828.8\ Btu/lbm}$$





11.174E

Consider a simple ideal Rankine cycle using water at a supercritical pressure. Such a cycle has a potential advantage of minimizing local temperature differences between the fluids in the steam generator, such as the instance in which the high-temperature energy source is the hot exhaust gas from a gas-turbine engine. Calculate the thermal efficiency of the cycle if the state entering the turbine is 8000 lbf/in.^{2}, 1300 F, and the condenser pressure is 0.95 lbf/in.^{2}. What is the steam quality at the turbine exit?

Solution:

For the efficiency we need the net work and steam generator heat transfer.

State 1: $s_1 = 0.1296$ Btu/lbm R, $h_1 = 68.04$ Btu/lbm

State 3: $h_3 = 1547.5$ Btu/lbm, $s_3 = 1.4718$ Btu/lbm R

C.V. Pump. For this high exit pressure we use Table F.7.3 to get state 2.

Entropy Eq.: $s_2 = s_1$ => $h_2 = 91.69$ Btu/lbm

$w_p = h_2 - h_1 = 91.69 – 68.04 = 23.65$ Btu/lbm

C.V. Turbine. Assume reversible and adiabatic.

Entropy Eq.: $s_4 = s_3 = 1.4718 = 0.1296 + x_4 \times 1.8526$

$x_4 = \mathbf{0.7245}$ Very low for a turbine exhaust

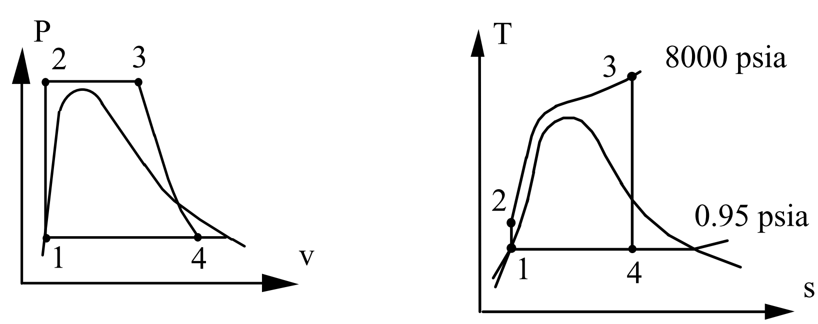
$h_4 = 68.04 + x_4 \times 1036.98 = 751.29$ Btu/lbm,

$w_T = h_3 - h_4 = 796.2$ Btu/lbm

Steam generator: $q_H = h_3 - h_2 = 1547.5 – 91.69 = 1455.8$ Btu/lbm

$w_{NET} = w_T – w_p = 796.2 – 23.65 = 772.6$ Btu/lbm

$\eta = w_{NET}/q_H = 772.6 / 1455.8 = \mathbf{0.53}$



11.175E

Consider an ideal steam reheat cycle in which the steam enters the high-pressure turbine at 600 lbf/in.^{2}, 700 F, and then expands to 150 lbf/in.^{2}. It is then reheated to 700 F and expands to 2.225 lbf/in.^{2} in the low-pressure turbine. Calculate the thermal efficiency of the cycle and the moisture content of the steam leaving the low-pressure turbine.

Solution:

Basic Rankine cycle with a reheat section. For this cycle from Table F.7

State 3: Superheated vapor $h_3 = 1350.62$ Btu/lbm, $s_3 = 1.5871$ Btu/lbm R,

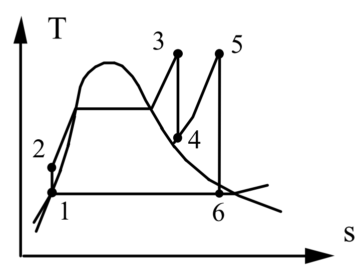
State 1: Saturated liquid $h_1 = 97.97$ Btu/lbm, $v_1 = 0.01625$ ft^3/lbm

C.V. Pump: Adiabatic and reversible. Use incompressible fluid so

$$w_P = \int v\, dP = v_1(P_2 - P_1)$$

$$= 0.01625(600 - 2.2)\frac{144}{778} = 1.8 \text{ Btu/lbm}$$

$$h_2 = h_1 + w_P = 95.81 \text{ Btu/lbm}$$



C.V. Tubine 1: $w_{T1} = h_3 - h_4$, $s_4 = s_3$

$s_4 = s_3 = 1.5871$ Btu/lbm R => $h_4 = 1208.93$ Btu/lbm

$w_{T1} = 1350.62 - 1208.93 = 141.69$ Btu/lbm

C.V. Tubine 2: $w_{T2} = h_5 - h_6$, $s_6 = s_5$

State 5: $h_5 = 1376.55$ Btu/lbm, $s_5 = 1.7568$ Btu/lbm R

State 6: $s_6 = s_5 = 1.7568 = 0.1817 + x_6 \times 1.7292$ => $x_6 = 0.9109$

$h_6 = 97.97 + 0.9109 \times 1019.78 = 1026.89$ Btu/lbm

$w_{T2} = 1376.55 - 1026.89 = 349.66$ Btu/lbm

$w_{T,tot} = w_{T1} + w_{T2} = 141.69 + 349.66 = 491.35$ Btu/lbm

C.V. Boiler: $q_{H1} = h_3 - h_2 = 1350.62 - 97.97 = 1252.65$ Btu/lbm

$q_H = q_{H1} + h_5 - h_4 = 1252.65 + 1376.55 - 1208.93 = 1420.3$ Btu/lbm

$\eta_{CYCLE} = (w_{T,tot} - w_P)/q_H = (491.35 - 1.8)/1420.3 = \mathbf{0.345}$

11.176E

Consider an ideal steam regenerative cycle in which steam enters the turbine at 600 lbf/in.^{2}, 700 F, and exhausts to the condenser at 2.225 lbf/in.^{2}. Steam is extracted from the turbine at 150 lbf/in.^{2} for an open feedwater heater. The feedwater leaves the heater as saturated liquid. The appropriate pumps are used for the water leaving the condenser and the feedwater heater. Calculate the thermal efficiency of the cycle and the net work per pound-mass of steam.

From Table F.7.2

$h_5 = 1350.62$ Btu/lbm,

$s_5 = 1.5871$ Btu/lbm R

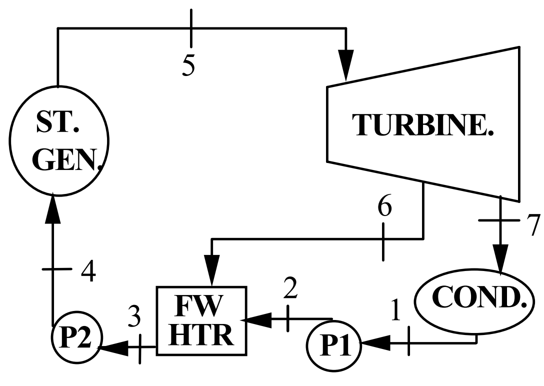
$h_1 = 97.97$ Btu/lbm,

$v_1 = 0.01625$ ft^3/lbm

Interpolate to get

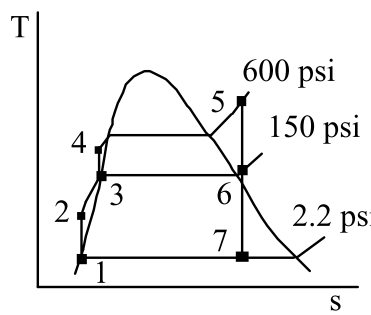
$h_3 = 330.67$ Btu/lbm,

$v_3 = 0.01809$ ft^3/lbm



C.V. Pump1:

$$w_{P12} = 0.01625(150 - 2.2)\frac{144}{778}$$
$$= 0.44 \text{ Btu/lbm} = h_2 - h_1$$
$$h_2 = h_1 + w_{P12} = 98.41 \text{ Btu/lbm}$$



C.V. Pump2:

$$w_{P34} = 0.01809(600 - 150)144/778 = 1.507 \text{ Btu/lbm}$$
$$\Rightarrow \quad h_4 = h_3 + w_{P34} = 332.18 \text{ Btu/lbm}$$

C.V. Turbine (high pressure section)

2nd law: $s_6 = s_5 = 1.5871$ Btu/lbm R => $h_6 = 1208.93$ Btu/lbm

CV: feedwater heater, call the extraction fraction $y = \dot{m}_6/\dot{m}_3$

Continuity Eq.: $\dot{m}_3 = \dot{m}_6 + \dot{m}_2$, Energy Eq.: $\dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3$

$$y_6 h_6 + (1 - y_6)h_2 = h_3 \quad \Rightarrow \quad y_6 = (h_3 - h_2)/(h_6 - h_2)$$

$$\Rightarrow \quad y_6 = (330.67 - 98.41)/(1208.93 - 98.41) = 0.2091$$

CV: Turbine from 5 to 7

$s_7 = s_5 \quad \Rightarrow \; x_7 = (1.5871 - 0.1817)/1.7292 = 0.8127$

$h_7 = 97.97 + 0.8127 \times 1019.78 = 926.75$ Btu/lbm

$w_T = (h_5 - h_6) + (1• - y_6)(h_6 - h_7)$

$= (1350.62 – 1208.93) + 0.7909(1208.93 - 926.75) = 364.87$ Btu/lbm

CV: pumps

$w_P = (1 - y_6)w_{P12} + w_{P34} = 0.7909 \times 0.44 + 1 \times 1.507 = 1.855$ Btu/lbm

$w_{NET} = w_T - w_P = 364.87 - 1.855 =$ **363.0 Btu/lbm**

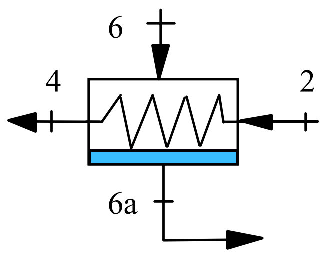
CV: steam generator

$q_H = h_5 - h_4 = 1350.62 – 332.18 = 1018.44$ Btu/lbm

$\eta_{TH} = w_{NET}/q_H = 363/1018.44 =$ **0.356**

11.177E

A closed feedwater heater in a regenerative steam power cycle heats 40 lbm/s of water from 200 F, 2000 lbf/in.^{2} to 450 F, 2000 lbf/in.^{2}. The extraction steam from the turbine enters the heater at 600 lbf/in.^{2}, 550 F and leaves as saturated liquid. What is the required mass flow rate of the extraction steam?



From the steam tables F.7:

F.7.3: $h_2 = 172.6$ Btu/lbm

F.7.3: $h_4 = 431.13$ Btu/lbm

F.7.2: $h_6 = 1255.36$ Btu/lbm

Interpolate for this state

F.7.1: $h_{6a} = 471.56$ Btu/lbm

C.V. Feedwater Heater

Energy Eq.: $\dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_2 h_4 + \dot{m}_6 h_{6a}$

Since all four state are known we can solve for the extraction flow rate

$$\dot{m}_6 = \dot{m}_2 \frac{h_2 - h_4}{h_{6a} - h_6} = 40 \frac{172.6 - 431.13}{471.56 - 1255.36} = \mathbf{13.2} \ \frac{\mathbf{lbm}}{\mathbf{s}}$$

11.178E

A steam power cycle has a high pressure of 600 lbf/in.^{2} and a condenser exit temperature of 110 F. The turbine efficiency is 85%, and other cycle components are ideal. If the boiler superheats to 1400 F, find the cycle thermal efficiency.

State 3: $h_3 = 1739.51$ Btu/lbm, $s_3 = 1.8497$ Btu/lbm R

State 1: $h_1 = 78.01$ Btu/lbm, $v_1 = 0.01617$ ft^3/lbm

C.V. Pump: $w_P = \int v dP \approx v_1(P_2 - P_1) = h_2 - h_1$

$= 0.01617(600 - 1.28)\ 144/778 = 1.79$ Btu/lbm

$h_2 = h_1 + w_P = 78.01 + 1.79 = 79.8$ Btu/lbm

C.V. Turb.: $w_T = h_3 - h_4$, $s_4 = s_3 + s_{T,GEN}$

Ideal: $s_{4S} = s_3 = 1.8497$ Btu/lbm R $= 0.1473 + x_{4S}\ 1.8101$

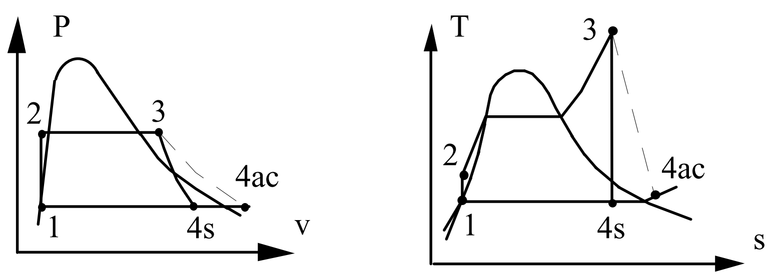
=> $x_{4S} = 0.9405$, $h_{4S} = 78.01 + x_{4S}\ 1031.28 = 1047.93$ Btu/lbm

=> $w_{T,S} = 1739.51 - 1047.93 = 691.58$ Btu/lbm

Actual: $w_{T,AC} = \eta \times w_{T,S} = 0.85 \times 691.58 = 587.8$ Btu/lbm

C.V. Boiler: $q_H = h_3 - h_2 = 1739.51 - 79.8 = 1659.7$ Btu/lbm

$\eta = (w_{T,AC} - w_P)/q_H = (587.8 - 1.79)/1659.7 =$ **0.353**



11.179E

The steam power cycle in Problem 11.167 has an isentropic efficiency of the turbine of 85% and that for the pump it is 80%. Find the cycle efficiency and the specific work and heat transfer in the components.

States numbered as in fig 11.3 of text.

CV Pump: $w_{P,S} = v_1(P_2 - P_1) = 0.01625(600 – 2.2)144/778 = 1.8$ Btu/lbm

$\Rightarrow\ w_{P,AC} = 1.8/0.8 =$ **2.245 Btu/lbm**

$h_2 = h_1 + w_{P,AC} = 97.97 + 2.245 = 100.2$ Btu/lbm

CV Turbine: $w_{T,S} = h_3 - h_{4s}$, $s_4 = s_3 = 1.4464$ Btu/lbm R

$s_4 = s_3 = 1.4464 = 0.1817 + x_4 \times 1.7292 \quad => \quad x_4 = 0.7314,$

$h_4 = 97.97 + 0.7314 \times 1019.78 = 843.84$ Btu/lbm

$\Rightarrow\ w_{T,S} = 1204.06 - 843.84 = 360.22$ Btu/lbm

$w_{T,AC} = h_3 - h_{4AC} = 360.22 \times 0.85 =$ **306.2**

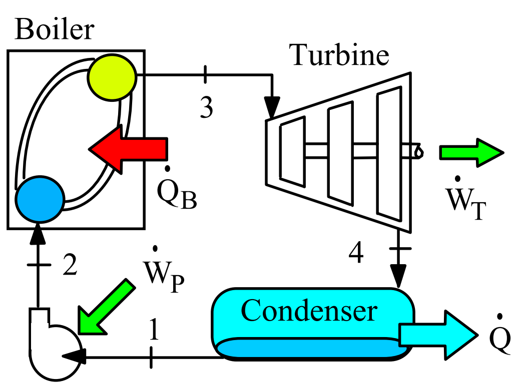
$\Rightarrow\ h_{4AC} = 897.86$ Btu/lbm (still two-phase)

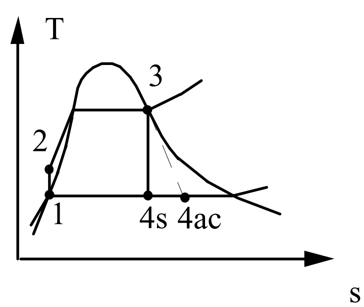
CV Boiler: $q_H = h_3 - h_2 = 1204.06 - 100.2 =$ **1103.9 Btu/lbm**

$q_L = h_{4AC} - h_1 = 897.86 - 97.97 =$ **799.9 Btu/lbm**

$\eta_{CYCLE} = (w_T - w_P)/q_H = (306.2 - 2.245)/1103.9 =$ **0.275**

Compared to (360.22-1.8)/1104.3 = 0.325 in the ideal case.

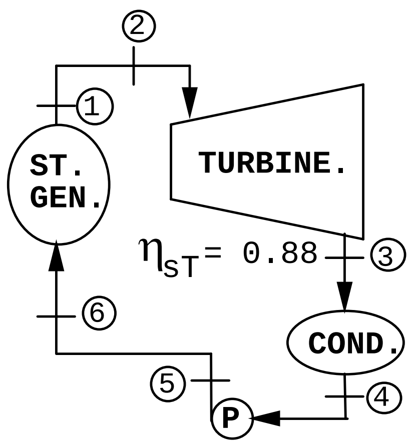


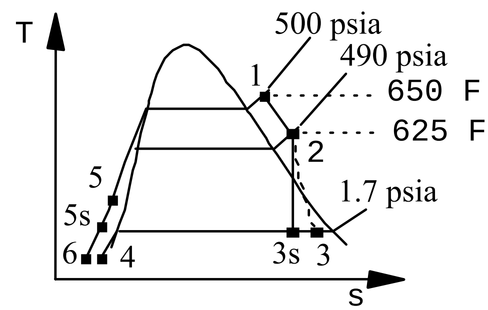


state 2s and 2ac nearly the same

11.180E

Steam leaves a power plant steam generator at 500 lbf/in.^{2}, 650 F, and enters the turbine at 490 lbf/in.^{2}, 625 F. The isentropic turbine efficiency is 88%, and the turbine exhaust pressure is 1.7 lbf/in.^{2}. Condensate leaves the condenser and enters the pump at 110 F, 1.7 lbf/in.^{2}. The isentropic pump efficiency is 80%, and the discharge pressure is 520 lbf/in.^{2}. The feedwater enters the steam generator at 510 lbf/in.^{2}, 100 F. Calculate the thermal efficiency of the cycle and the entropy generation of the flow in the line between the steam generator exit and the turbine inlet, assuming an ambient temperature of 77 F.





$\eta_{ST} = 0.88, \quad \eta_{SP} = 0.80$

$h_1 = 1328.0, \quad h_2 = 1314.0$ Btu/lbm

$s_{3S} = s_2 = 1.5752 = 0.16483 + x_{3S}\times1.7686 \quad => \quad x_{3S} = 0.79745$

$h_{3S} = 88.1 + 0.797\ 45\times1025.4 = 905.8$ Btu/lbm

$w_{ST} = h_2 - h_{3S} = 1314.0 - 905.8 = 408.2$ Btu/lbm

$w_T = \eta_{ST}w_{ST} = 0.88\times408.2 = 359.2$ Btu/lbm

$h_3 = h_2 - w_T = 1314.0 - 359.2 = 954.8$ Btu/lbm

$$w_{SP} = 0.016166(520-1.7)\frac{144}{778} = 1.55 \text{ Btu/lbm}$$

$w_p = w_{SP}/\eta_{SP} = 1.55/0.80 = 1.94$ Btu/lbm

$q_H = h_1 - h_6 = 1328.0 - 68.1 = 1259.9$ Btu/lbm

$\eta_{TH} = w_{NET}/q_H = (359.2 - 1.94)/1259.9 =$ **0.284**

C.V. Line from 1 to 2: $w = \emptyset$,

Energy Eq.: $q = h_2 - h_1 = 1314 - 1328 = -14$ Btu/lbm

Entropy Eq.: $s_1 + s_{gen} + q/T_0 = s_2 \quad =>$

$s_{gen} = s_2 - s_1 - q/T_0 = 1.5752 - 1.586 - (-14/536.7) =$ **0.0153 Btu/lbm R**

11.181E

A boiler delivers steam at 1500 lbf/in.^{2}, 1000 F to a two-stage turbine as shown in Fig. 11.17. After the first stage, 25% of the steam is extracted at 200 lbf/in.^{2} for a process application and returned at 150 lbf/in.^{2}, 190 F to the feedwater line. The remainder of the steam continues through the low-pressure turbine stage, which exhausts to the condenser at 2.225 lbf/in.^{2}. One pump brings the feedwater to 150 lbf/in.^{2} and a second pump brings it to 1500 lbf/in.^{2}. Assume the first and second stages in the steam turbine have isentropic efficiencies of 85% and 80% and that both pumps are ideal. If the process application requires 5000 Btu/s of power, how much power can then be cogenerated by the turbine?

3: $h_3 = 1490.32$, $s_3 = 1.6001$ Btu/lbmR

C.V. Turbine T1

4s: Rev and adiabatic $s_{4S} = s_3 \Rightarrow$

Table F.7.2 Sup. vapor

$h_{4S} = 1246.6$ Btu/lbm

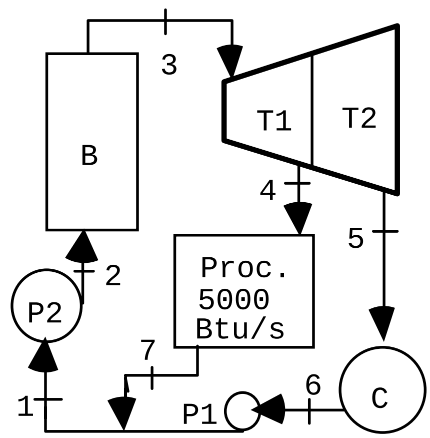
$w_{T1,S} = h_3 - h_{4S} = 243.7$ Btu/lbm

$\Rightarrow w_{T1,AC} = 207.15$ Btu/lbm

$h_{4AC} = h_3 - w_{T1,AC} = 1283.16$

4ac: P_4, h_{4AC}

$\Rightarrow s_{4AC} = 1.6384$ Btu/lbm R



5s: $s_{5S} = s_{4AC} \Rightarrow x_{5S} = \frac{1.6384 - 0.1817}{1.7292} = 0.8424$

$h_{5S} = 97.97 + x_{5S}\, 1019.78 = 957.03$ Btu/lbm

$w_{T2,S} = h_{4AC} - h_{5S} = 326.13$ Btu/lbm

$w_{T2,AC} = 260.9 = h_{4AC} - h_{5AC} \Rightarrow h_{5AC} = 1022.3$ Btu/lbm

7: Compressed liquid use sat. liq. same T: $h_7 = 158.02$ Btu/lbm;

C.V. process unit. Assume no work only heat out.

$q_{PROC} = h_{4AC} - h_7 = 1125.1$ Btu/lbm

$\dot{m}_4 = \dot{Q}/q_{PROC} = 5000/1125.1 = 4.444$ lbm/s $= 0.25\, \dot{m}_{TOT}$

$\Rightarrow \dot{m}_{TOT} = \dot{m}_3 = 17.776$ lbm/s, $\dot{m}_5 = \dot{m}_3 - \dot{m}_4 = 13.332$ lbm/s

C.V. Total turbine

$\dot{W}_T = \dot{m}_3 h_3 - \dot{m}_4 h_{4AC} - \dot{m}_5 h_{5AC} =$ **7160 Btu/s**

Brayton Cycles

11.182E

A large stationary Brayton cycle gas-turbine power plant delivers a power output of 100 000 hp to an electric generator. The minimum temperature in the cycle is 540 R, and the maximum temperature is 2900 R. The minimum pressure in the cycle is 1 atm, and the compressor pressure ratio is 14 to 1. Calculate the power output of the turbine, the fraction of the turbine output required to drive the compressor and the thermal efficiency of the cycle?

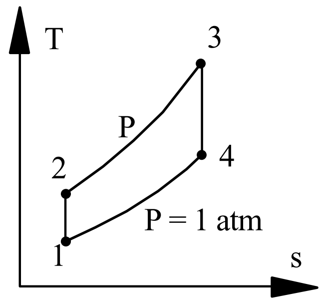
Brayton:

$\dot{w}_{NET} = 100\ 000$ hp

$P_1 = 1$ atm, $T_1 = 540$ R

$P_2/P_1 = 14$, $T_3 = 2900$ R

Solve using constant C_{P0}:



Compression in compressor: $s_2 = s_1 \Rightarrow$ Implemented in Eq.8.32

$$\rightarrow T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = 540(14)^{0.286} = 1148.6 \text{ R}$$

$$w_C = h_2 - h_1 = C_{P0}(T_2 - T_1) = 0.24\ (1148.6 - 540) = 146.1 \text{ Btu/lbm}$$

Expansion in turbine: $s_4 = s_3 \Rightarrow$ Implemented in Eq.8.32

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}} = 2900\left(\frac{1}{14}\right)^{0.286} = 1363.3 \text{ R}$$

$$w_T = h_3 - h_4 = C_{P0}(T_3 - T_4) = 0.24(2900 - 1363.3) = 368.8 \text{ Btu/lbm}$$

$$w_{NET} = w_T - w_C = 368.8 - 146.1 = 222.7 \text{ Btu/lbm}$$

$$\dot{m} = \dot{W}_{NET}/w_{NET} = 100\ 000 \times 2544/222.7 = 1\ 142\ 344 \text{ lbm/h}$$

$\dot{W}_T = \dot{m}w_T =$ **165 600 hp,** $\quad\quad w_C/w_T =$ **0.396**

Energy input is from the combustor

$$q_H = C_{P0}(T_3 - T_2) = 0.24(2900 - 1148.6) = 420.3 \text{ Btu/lbm}$$

$\eta_{TH} = w_{NET}/q_H = 222.7/420.3 =$ **0.530**

11.183E

A Brayton cycle produces 14 000 Btu/s with an inlet state of 60 F, 14.7 psia, and a compression ratio of 16:1. The heat added in the combustion is 400 Btu/lbm. What are the highest temperature and the mass flow rate of air, assuming cold air properties?

Solution:

Efficiency is from Eq.11.8

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_H} = \frac{w_{net}}{q_H} = 1 - r_p^{-(k-1)/k} = 1 - 16^{-0.4/1.4} = \mathbf{0.547}$$

from the required power we can find the needed heat transfer

$$\dot{Q}_H = \dot{W}_{net} / \eta = \frac{14\ 000}{0.547} = 25\ 594 \text{ Btu/s}$$

$$\dot{m} = \dot{Q}_H / q_H = \frac{25\ 594 \text{ Btu/s}}{400 \text{ Btu/lbm}} = \mathbf{63.99\ lbm/s}$$

Temperature after compression is

$$T_2 = T_1\, r_p^{(k-1)/k} = 520 \times 16^{0.4/1.4} = 1148 \text{ R}$$

The highest temperature is after combustion

$$T_3 = T_2 + q_H/C_p = 1148 + \frac{400}{0.24} = \mathbf{2815\ R}$$

11.184E

Do the previous problem with properties from table F.5 instead of cold air properties.

Solution:

With the variable specific heat we must go through the processes one by one to get net work and the highest temperature T_3.

From F.5: $h_1 = 124.38$ btu/lbm, $s^o_{T1} = 1.63074$ Btu/lbm R

The compression is reversible and adiabatic so constant s. From Eq.8.28

$$s_2 = s_1 \Rightarrow s^o_{T2} = s^o_{T1} + R \ln\left(\frac{P_2}{P_1}\right) = 1.63074 + \frac{53.34}{778} \ln 16$$

$$= 1.82083 \text{ Btu/lbm R}$$

back interpolate in F.5 $\Rightarrow$ $T_2 = 1133.5$ R, $h_2 = 274.58$ Btu/lbm

Energy equation with compressor work in

$$w_C = -{}_1w_2 = h_2 - h_1 = 274.58 - 124.383 = 150.2 \text{ Btu/lbm}$$

Energy Eq. combustor: $h_3 = h_2 + q_H = 274.58 + 400 = 674.6$ Btu/lbm

State 3: (P, h): $\mathbf{T_3 = 2600\ R}$, $s^o_{T3} = 2.04523$ Btu/lbm R

The expansion is reversible and adiabatic so constant s. From Eq.8.28

$$s_4 = s_3 \Rightarrow s^o_{T4} = s^o_{T3} + R\ln(P_4/P_3) = 2.04523 + \frac{53.34}{778} \ln(1/16) = 1.85514$$

$\Rightarrow$ $T_4 = 1297$ R, $h_4 = 316.21$ Btu/lbm

Energy equation with turbine work out

$$w_T = h_3 - h_4 = 674.6 - 316.21 = 358.4 \text{ Btu/lbm}$$

Now the net work is

$$w_{net} = w_T - w_C = 358.4 – 150.2 = 208.2 \text{ Btu/lbm}$$

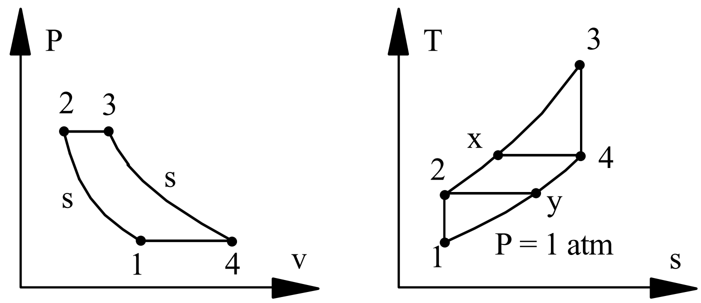
The total required power requires a mass flow rate as

$$\dot{m} = \frac{\dot{W}_{net}}{w_{net}} = \frac{14\,000}{208.2} \frac{\text{Btu/s}}{\text{Btu/lbm}} = \mathbf{67.2\ lbm/s}$$

11.185E

An ideal regenerator is incorporated into the ideal air-standard Brayton cycle of Problem 11.182. Calculate the cycle thermal efficiency with this modification.

Solution:



Compression ratio

$$\frac{P_2}{P_1} = 14$$

Max temperature

$T_3 = 2900$ R

The compression is reversible and adiabatic so constant s. From Eq.8.32

$$\rightarrow T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = 540(14)^{0.286} = 1148.6 \text{ R}$$

$$w_C = h_2 - h_1 = C_{P0}(T_2 - T_1) = 0.24\,(1148.6 - 540) = 146.1 \text{ Btu/lbm}$$

Expansion in turbine: $s_4 = s_3$ ⇒ Implemented in Eq.8.32

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}} = 2900\left(\frac{1}{14}\right)^{0.286} = 1363.3 \text{ R}$$

$$w_T = h_3 - h_4 = C_{P0}(T_3 - T_4) = 0.24(2900 - 1363.3) = 368.8 \text{ Btu/lbm}$$

$$w_{NET} = w_T - w_C = 368.8 - 146.1 = 222.7 \text{ Btu/lbm}$$

Ideal regenerator: $T_X = T_4 = 1363.3$ R

$$q_H = h_3 - h_X = 0.24(2900 - 1363.3) = 368.8 \text{ Btu/lbm} = w_T$$

$$\eta_{TH} = w_{NET}/q_H = 222.7/368.8 = \mathbf{0.604}$$

11.186E

An air-standard Ericsson cycle has an ideal regenerator as shown in Fig. P11.62. Heat is supplied at 1800 F and heat is rejected at 68 F. Pressure at the beginning of the isothermal compression process is 10 lbf/in.^{2}. The heat added is 275 Btu/lbm. Find the compressor work, the turbine work, and the cycle efficiency.

Identify the states

Heat supplied at high temperature $T_4 = T_3 = 1800$ F = 2349.7 R

Heat rejected at low temperature $T_1 = T_2 = 68$ F = 527.7 R

Beginning of the compression: $P_1 = 10$ lbf/in^2

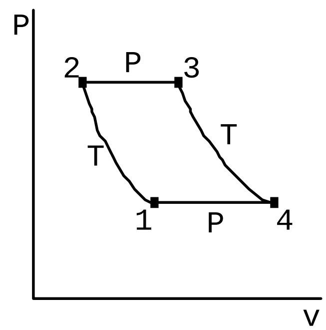
Ideal regenerator: ${}_2q_3 = -{}_4q_1 \;\Rightarrow\; q_H = {}_3q_4 \;\Rightarrow$

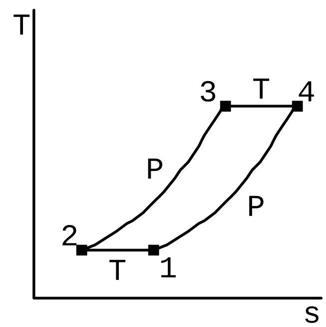
$w_T = q_H =$ **275 Btu/lbm**

$\eta_{TH} = \eta_{CARNOT\ TH.} = 1 - T_L/T_H = 1 - 527.7/2349.7 =$ **0.775**

$w_{net} = \eta_{TH}\, q_H = 0.775 \times 275 = 213.13$ Btu/lbm

$q_L = -w_C = 275 - 213.13 =$ **61.88 Btu/lbm**





11.187E

The turbine in a jet engine receives air at 2200 R, 220 lbf/in.^{2}. It exhausts to a nozzle at 35 lbf/in.^{2}, which in turn exhausts to the atmosphere at 14.7 lbf/in.^{2}. The isentropic efficiency of the turbine is 85% and the nozzle efficiency is 95%. Find the nozzle inlet temperature and the nozzle exit velocity. Assume negligible kinetic energy out of the turbine.

Solution:

C.V. Turbine: $h_i = 560.588$ Btu/lbm, $s^o_{Ti} = 1.99765$ Btu/lbm R, $s_{es} = s_i$

Then from Eq.8.28

$$\Rightarrow\ s^o_{Tes} = s^o_{Ti} + R\ \ln(P_e/P_i) = 1.99765 + \frac{53.34}{778} \ln(35/220) = 1.8716\ \frac{\text{Btu}}{\text{lbm R}}$$

Table F.5 $T_{es} = 1382$ R, $h_{es} = 338.27$ Btu/lbm,

Energy eq.: $w_{T,s} = h_i - h_{es} = 560.588 - 338.27 = 222.3$ Btu/lbm

Eq.9.27: $w_{T,AC} = w_{T,s} \times \eta_T = 188.96 = h_i - h_{e,AC} \quad \Rightarrow\ h_{e,AC} = 371.6$

Table F.5 $\Rightarrow$ $\mathbf{T_{e,AC} = 1509\ R}$, $s^o_{Te} = 1.8947$ Btu/lbm R

C.V. Nozzle: $h_i = 371.6$ Btu/lbm, $s^o_{Ti} = 1.8947$ Btu/lbm R, $s_{es} = s_i$

Then from Eq.8.28

$$\Rightarrow\ s^o_{Tes} = s^o_{Ti} + R\ \ln(P_e/P_i) = 1.8947 + \frac{53.34}{778} \ln\left(\frac{14.7}{35}\right) = 1.8352\ \frac{\text{Btu}}{\text{lbm R}}$$

Table F.5 $\Rightarrow$ $T_{e,s} = 1199.6$ R, $h_{e,s} = 291.3$ Btu/lbm

Energy Eq.: $(1/2)\mathbf{V}^2_{e,s} = h_i - h_{e,s} = 371.6 - 291.3 = 80.3$ Btu/lbm

Eq.9.30: $(1/2)\mathbf{V}^2_{e,AC} = (1/2)\mathbf{V}^2_{e,s} \times \eta_{NOZ} = 76.29$ Btu/lbm

$$\mathbf{V}_{e,AC} = \sqrt{2 \times 25037 \times 76.29} = \mathbf{1954\ ft/s}$$

Recall 1 Btu/lbm = 25 037 ft^2/s^2

Otto, Diesel, Stirling and Carnot Cycles

11.188E

Air flows into a gasoline engine at 14 lbf/in.^{2}, 540 R. The air is then compressed with a volumetric compression ratio of 8:1. In the combustion process 560 Btu/lbm of energy is released as the fuel burns. Find the temperature and pressure after combustion.

Solution:

Solve the problem with constant heat capacity.

Compression 1 to 2: $s_2 = s_1 \Rightarrow$ From Eq.8.33 and Eq.8.34

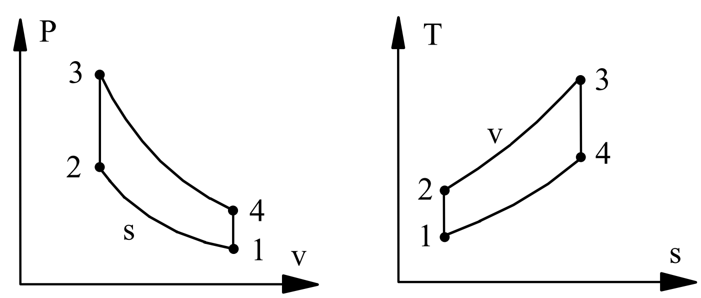
$$T_2 = T_1 (v_1/v_2)^{k-1} = 540 \times 8^{0.4} = 1240.6 \text{ R}$$

$$P_2 = P_1 \times (v_1/v_2)^k = 14 \times 8^{1.4} = 257.3 \text{ lbf/in}^2$$

Combustion 2 to 3 at constant volume: $u_3 = u_2 + q_H$

$$T_3 = T_2 + q_H/C_v = 1240.6 + 560/0.171 = \mathbf{4515 \text{ R}}$$

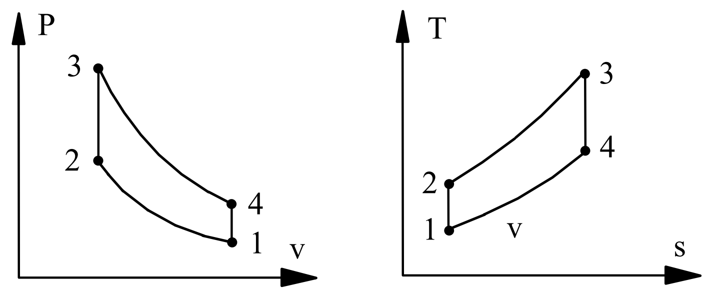
$$P_3 = P_2 \times (T_3/T_2) = 257.3\ (4515 / 1240.6) = \mathbf{936 \text{ lbf/in}^2}$$



11.189E

To approximate an actual spark-ignition engine consider an air-standard Otto cycle that has a heat addition of 800 Btu/lbm of air, a compression ratio of 7, and a pressure and temperature at the beginning of the compression process of 13 lbf/in.^{2}, 50 F. Assuming constant specific heat, with the value from Table F.4, determine the maximum pressure and temperature of the cycle, the thermal efficiency of the cycle and the mean effective pressure.

Solution:



State 1: $v_1 = RT_1/P_1 = \frac{53.34 \times 510}{13 \times 144} = 14.532$ ft^3/lbm, $v_2 = v_1/7 = 2.076$ ft^3/lbm

The compression process, reversible adiabatic so then isentropic. The constant s is implemented with Eq.8.25 leading to Eqs.8.34 and 8.32

$$P_2 = P_1(v_1/v_2)^k = 13(7)^{1.4} = 198.2 \text{ lbf/in}^2$$

$$T_2 = T_1(v_1/v_2)^{k-1} = 510(7)^{0.4} = 1110.7 \text{ R}$$

The combustion process with constant volume, $q_H = 800$ Btu/lbm

$$T_3 = T_2 + q_H/C_{V0} = 1110.7 + 800/0.171 = \mathbf{5789\ R}$$

$$P_3 = P_2T_3/T_2 = 198.2 \times 5789/1110.7 = \mathbf{1033\ lbf/in^2}$$

Cycle efficiency from the ideal cycle as in Eq.11.18

$$\eta_{TH} = 1 - (T_1/T_2) = 1 - 510/1110.7 = \mathbf{0.541}$$

To get the mean effective pressure we need the net work

$$w_{NET} = \eta_{TH} \times q_H = 0.541 \times 800 = 432.8 \text{ Btu/lbm}$$

$$P_{m\,eff} = \frac{w_{NET}}{v_1 - v_2} = \frac{432.8 \times 778}{(14.532 - 2.076) \times 144} = \mathbf{188\ lbf/in^2}$$

11.190E

A gasoline engine has a volumetric compression ratio of 10 and before compression has air at 520 R, 12.2 psia in the cylinder. The combustion peak pressure is 900 psia. Assume cold air properties. What is the highest temperature in the cycle? Find the temperature at the beginning of the exhaust (heat rejection) and the overall cycle efficiency.

Solution:

Compression. Isentropic so we use Eqs.8.33-8.34

$$P_2 = P_1(v_1/v_2)^k = 12.2\ (10)^{1.4} = 306.45 \text{ psia}$$

$$T_2 = T_1(v_1/v_2)^{k-1} = 520\ (10)^{0.4} = 1306.2 \text{ R}$$

Combustion. Constant volume

$$T_3 = T_2\ (P_3/P_2) = 1306.2 \times 900/306.45 = \mathbf{3836\ R}$$

Exhaust. Isentropic expansion so from Eq.8.33

$$T_4 = T_3\ /\ (v_1/v_2)^{k-1} = T_3\ /\ 10^{0.4} = 3836\ /\ 2.5119 = \mathbf{1527\ R}$$

Overall cycle efficiency is from Eq.11.18, $r_v = v_1/v_2$

$$\eta = 1 - r_v^{1-k} = 1 - 10^{-0.4} = \mathbf{0.602}$$

Comment: No actual gasoline engine has an efficiency that high, maybe 35%.

11.191E

A for stroke gasoline engine has a compression ratio of 10:1 with 4 cylinders of total displacement 75 in^3. the inlet state is 500 R, 10 psia and the engine is running at 2100 RPM with the fuel adding 750 Btu/lbm in the combustion process. What is the net work in the cycle and how much power is produced?

Solution:

Overall cycle efficiency is from Eq.11.18, $r_v = v_1/v_2$

$$\eta_{TH} = 1 - r_v^{1-k} = 1 - 10^{-0.4} = \mathbf{0.602}$$

$$w_{net} = \eta_{TH} \times q_H = 0.602 \times 750 = 451.5 \text{ Btu/lbm}$$

We also need specific volume to evaluate Eqs.11.15 to 11.17

$$v_1 = RT_1 / P_1 = 53.34 \times 500 / (10 \times 144) = 18.52 \text{ ft}^3\text{/lbm}$$

$$P_{meff} = \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1 (1 - \frac{1}{r_v})} = \frac{451.5}{18.52 \times 0.9} \frac{778}{144} = 146.3 \text{ psia}$$

Now we can find the power from Eq.11.17

$$\dot{W} = P_{meff} V_{displ} \frac{RPM}{60} \frac{1}{2} = 146.3 \times \frac{75}{12} \times \frac{2100}{60} \times \frac{1}{2} = \mathbf{16\ 002\ lbf\text{-}ft/s}$$

$$= \mathbf{29\ hp}$$

Recall 1 hp = 550 lbf-ft/s.

11.192E

It is found experimentally that the power stroke expansion in an internal combustion engine can be approximated with a polytropic process with a value of the polytropic exponent *n* somewhat larger than the specific heat ratio *k*. Repeat Problem 11.189 but assume the expansion process is reversible and polytropic (instead of the isentropic expansion in the Otto cycle) with *n* equal to 1.50.

First find states 2 and 3. based on the inlet state we get

$v_4 = v_1 = RT_1/P_1 = 53.34 \times 510/13 \times 144 = 14.532 \text{ ft}^3/\text{lbm}$

$v_3 = v_2 = v_1/7 = 2.076 \text{ ft}^3/\text{lbm}$

After compression we have constant s leads to Eq.8.34 and Eq.8.32

$P_2 = P_1(v_1/v_2)^k = 13(7)^{1.4} = 198.2 \text{ lbf/in}^2$

$T_2 = T_1(v_1/v_2)^{k-1} = 510(7)^{0.4} = 1110.7 \text{ R}$

Constant volume combustion

$T_3 = T_2 + q_H/C_{V0} = 1110.7 + 800/0.171 = 5789 \text{ R}$

$P_3 = P_2T_3/T_2 = 198.2 \times 5789/1110.7 = 1033 \text{ lbf/in}^2$

Process 3 to 4: $Pv^{1.5}$ = constant.

$P_4 = P_3(v_3/v_4)^{1.5} = 1033(1/7)^{1.5} = 55.78 \text{ lbf/in}^2$

$T_4 = T_3(v_3/v_4)^{0.5} = 5789(1/7)^{0.5} = 2188 \text{ R}$

For the mean effective pressure we need the net work and therefore the induvidual process work terms

$_1w_2 = \int P\, dv = R(T_2 - T_1)/(1 - 1.4)$

$= -53.34(1110.7 - 510)/(0.4 \times 778) = -102.96 \text{ Btu/lbm}$

$_3w_4 = \int P\, dv = R(T_4 - T_3)/(1 - 1.5)$

$= -53.34(2188 - 5789)/(0.5 \times 778) = 493.8 \text{ Btu/lbm}$

$w_{NET} = 493.8 - 102.96 = 390.84 \text{ Btu/lbm}$

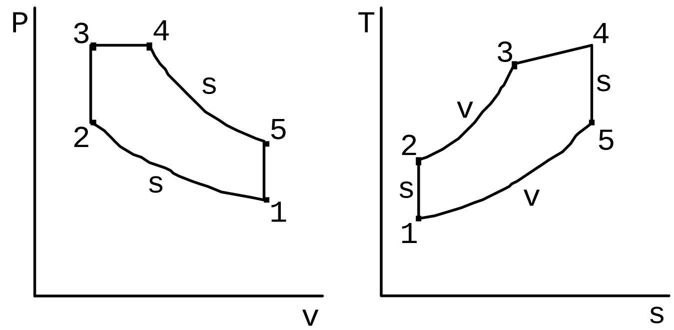
$\eta_{CYCLE} = w_{NET}/q_H = 390.84/700 = \mathbf{0.488}$

$P_{meff} = w_{NET}/(v_1 - v_2) = 390.84 \times 778/(14.532 - 2.076) = \mathbf{169.5\ lbf/in^2}$

Notice a smaller w_{NET}, η_{CYCLE}, P_{meff} compared to ideal cycle.

11.193E

In the Otto cycle all the heat transfer q_H occurs at constant volume. It is more realistic to assume that part of q_H occurs after the piston has started its downwards motion in the expansion stroke. Therefore consider a cycle identical to the Otto cycle, except that the first two-thirds of the total q_H occurs at constant volume and the last one-third occurs at constant pressure. Assume the total q_H is 700 Btu/lbm, that the state at the beginning of the compression process is 13 lbf/in.^{2}, 68 F, and that the compression ratio is 9. Calculate the maximum pressure and temperature and the thermal efficiency of this cycle. Compare the results with those of a conventional Otto cycle having the same given variables.



$P_1 = 13,\ T_1 = 527.67$ R

$r_V = v_1/v_2 = 7$

$q_{23} = \frac{2}{3} \times 700 = 466.7 \ \frac{\text{Btu}}{\text{lbm}}$

$q_{34} = \frac{1}{3} \times 700 = 233.3 \ \frac{\text{Btu}}{\text{lbm}}$

$P_2 = P_1(v_1/v_2)^k = 13(9)^{1.4} = 281.8 \text{ lbf/in}^2$

$T_2 = T_1(v_1/v_2)^{k-1} = 527.67(9)^{0.4} = 1270.7 \text{ R}$

$T_3 = T_2 + q_{23}/C_{V0} = 1270.7 + 466.7/0.171 = 4000 \text{ R}$

$P_3 = P_2(T_3/T_2) = 281.8 \times 4000/1270.7 = \mathbf{887.1\ lbf/in^2} = P_4$

$T_4 = T_3 + q_{34}/C_{P0} = 4000 + 233.3/0.24 = \mathbf{4972\ R}$

$\frac{v_5}{v_4} = \frac{v_1}{v_4} = (P_4/P_1) \times (T_1/T_4) = \frac{88.1}{13} \times \frac{527.67}{4972} = 7.242$

$T_5 = T_4(v_4/v_5)^{k-1} = 4972(1/7.242)^{0.4} = 2252 \text{ R}$

$q_L = C_{V0}(T_5 - T_1) = 0.171(2252 - 527.67) = 294.9 \text{ Btu/lbm}$

$\eta_{TH} = 1 - q_L/q_H = 1 - 294.9/700 = \mathbf{0.579}$

Standard Otto cycle: $\eta_{TH} = 1 - (9)^{-0.4} = \mathbf{0.585}$

11.194E

A diesel engine has a bore of 4 in., a stroke of 4.3 in. and a compression ratio of 19:1 running at 2000 RPM (revolutions per minute). Each cycle takes two revolutions and has a mean effective pressure of 200 lbf/in.^{2}. With a total of 6 cylinders find the engine power in Btu/s and horsepower, hp.

Solution:

Work from mean effective pressure.

$P_{meff} = w_{net} / (v_{max} - v_{min}) \quad -> \quad w_{net} = P_{meff} (v_{max} - v_{min})$

The displacement is

$\Delta V = \pi \text{Bore}^2 \times 0.25 \times S = \pi \times 4^2 \times 0.25 \times 4.3 = 54.035 \text{ in}^3$

Work per cylinder per power stroke

$W = P_{meff}(V_{max} - V_{min}) = 200 \times 54.035 / (12 \times 778) = 1.1575 \text{ Btu/cycle}$

Only every second revolution has a power stroke so we can find the power

$$\dot{W} = W \times N_{cyl} \times RPM \times 0.5 \left(\frac{\text{cycles}}{\text{min}}\right) \times \left(\frac{\text{min}}{60 \text{ s}}\right) \times \left(\frac{\text{Btu}}{\text{cycle}}\right)$$

$= 1.1575 \times 6 \times 2000 \times 0.5 \times (1/60) = 115.75 \text{ Btu/s}$

$= 115.75 \times 3600/2544.43 \text{ hp} =$ **164 hp**

11.195E

At the beginning of compression in a diesel cycle T = 540 R, P = 30 lbf/in.^{2} and the state after combustion (heat addition) is 2600 R and 1000 lbf/in.^{2}. Find the compression ratio, the thermal efficiency and the mean effective pressure.

Solution:

Compression process (isentropic) from Eqs.8.33-8.34

$P_2 = P_3 = 1000$ lbf/in^2 => $v_1/v_2 = (P_2/P_1)^{1/k} = (1000/30)^{0.7143} = \mathbf{12.24}$

$T_2 = T_1(P_2/P_1)^{(k-1)/k} = 540(1000/30)^{0.2857} = 1470.6$ R

Expansion process (isentropic) first get the volume ratios

$v_3/v_2 = T_3/T_2 = 2600/1470.6 = 1.768$

$v_4/v_3 = v_1/v_3 = (v_1/v_2)(v_2/v_3) = 12.24/1.768 = 6.923$

The exhaust temperature follows from Eq.8.33

$T_4 = T_3(v_3/v_4)^{k-1} = 2600*6.923^{-0.4} = 1199$ R

$q_L = C_V(T_4 - T_1) = 0.171(1199-540) = 112.7$ Btu/lbm

$q_H = h_3 - h_2 = C_P(T_3 - T_2) = 0.24(2600 - 1470.6) = 271.1$ Btu/lbm

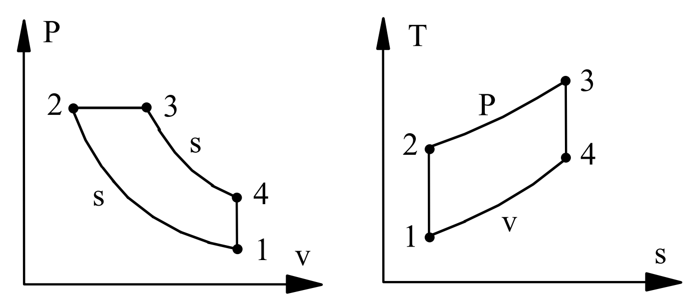
$\eta = 1 - q_L/q_H = 1 - 112.7 / 271.1 = \mathbf{0.5843}$

$w_{net} = q_{net} = 271.1 - 112.7 = 158.4$ Btu/lbm

$v_{max} = v_1 = RT_1/P_1 = 53.34 \times 540/(30 \times 144) = 6.6675$ ft^3/lbm

$v_{min} = v_{max}(v_1/v_2) = 6.6675 / 12.24 = 0.545$ ft^3/lbm

$P_{meff} = [158.4/(6.6675 - 0.545)] \times (778/144) = \mathbf{139.8\ lbf/in^2}$



Remark: This is a too low compression ratio for a practical diesel cycle.

11.196E

Consider an ideal air-standard diesel cycle where the state before the compression process is 14 lbf/in.^{2}, 63 F and the compression ratio is 20. Find the maximum temperature(by iteration) in the cycle to have a thermal efficiency of 60%.

Diesel cycle: $P_1 = 14$, $T_1 = 522.67$ R, $v_1/v_2 = 20$, $\eta_{TH} = 0.60$

From the inlet state and the compression we get

$$T_2 = T_1(v_1/v_2)^{k-1} = 522.67(20)^{0.4} = 1732.4 \text{ R}$$

$$v_1 = \frac{53.34\times522.67}{14\times144} = 13.829 \text{ ft}^3/\text{lbm}, \qquad v_2 = \frac{13.829}{20} = 0.6915 \text{ ft}^3/\text{lbm}$$

Constant pressure combustion relates v_3 and T_3

$$v_3 = v_2\times T_3/T_2 = 0.6915\times T_3/1732.4 = 0.000399\, T_3$$

The expansion then gives T_4 interms of T_3

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{k-1} = \left(\frac{13.829}{0.000399\, T_3}\right)^{0.4} \rightarrow \qquad T_4 = 0.0153\, T_3^{1.4}$$

Now these T's relate to the given efficiency

$$\eta_{TH} = 0.60 = 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} = 1 - \frac{0.0153\, T_3^{1.4} - 522.67}{1.4(T_3 - 1732.4)}$$

$$\Rightarrow \quad 0.0153\, T_3^{1.4} - 0.56\, T_3 + 447.5 = 0$$

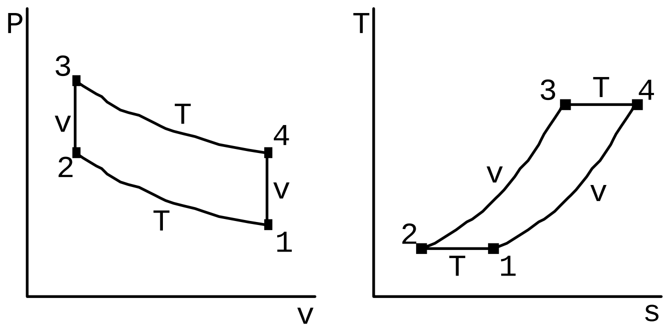
Trial and error on this non-linear equation

5100 R: LHS = -35.54, 5500 R: LHS = 5.04, 5450 R: LHS = -0.5

Linear interpolation, T_3 = **5455 R**

11.197E

Consider an ideal Stirling-cycle engine in which the pressure and temperature at the beginning of the isothermal compression process are 14.7 $lbf/in.^2$, 80 F, the compression ratio is 6, and the maximum temperature in the cycle is 2000 F. Calculate the maximum pressure in the cycle and the thermal efficiency of the cycle with and without regenerators.



Ideal Stirling cycle

$T_1 = T_2 = 80$ F

$P_1 = 14.7$ lbf/in^2

$\frac{v_1}{v_2} = 6$

$T_3 = T_4 = 2000$ F

$$T_1 = T_2 \rightarrow P_2 = P_1 \times v_1/v_2 = 14.7 \times 6 = 88.2$$

$$V_2 = V_3 \rightarrow P_3 = P_2 \times T_3/T_2 = 88.2 \times \frac{2460}{540} = \mathbf{401.8\ lbf/in^2}$$

$$w_{34} = q_{34} = RT_3 \ln(v_4/v_3)$$

$$= (53.34/778) \times 2460 \ln 6 = 302.2 \text{ Btu/lbm}$$

$$q_{23} = C_{V0}(T_3 - T_2) = 0.171(2460-540) = 328.3 \text{ Btu/lbm}$$

$$w_{12} = q_{12} = -RT_1 \ln\frac{v_1}{v_2} = -\frac{53.34}{778} \times 540 \ln 6 = -66.3 \text{ Btu/lbm}$$

$$w_{NET} = 302.2 - 66.3 = 235.9 \text{ Btu/lbm}$$

$$\eta_{NO\ REGEN} = \frac{235.9}{302.2+328.3} = \mathbf{0.374,}$$

$$\eta_{WITH\ REGEN} = \frac{235.9}{302.2} = \mathbf{0.781}$$

11.198E

An ideal air-standard Stirling cycle uses helium as working fluid. The isothermal compression brings the helium from 15 lbf/in.^{2}, 70 F to 90 lbf/in.^{2}. The exspansion takes place at 2100 R and there is no regenerator. Find the work and heat transfer in all four processes per lbm helium and the cycle efficiency.

Substance helium F.4: $R = 386$ ft-lbf/lbmR, $C_v = 0.753$ Btu/lbm R

$$v_4/v_3 = v_1/v_2 = P_2/P_1 = 90/15 = 6$$

1 -> 2: $-{}_1w_2 = -{}_1q_2 = \int P\, dV = RT \ln (v_1/v_2)$

$$= 386 \times 530 \times \ln(6)/778 = 471.15 \text{ Btu/lbm}$$

2 -> 3: ${}_2w_3 = 0;\quad {}_2q_3 = C_P(T_3 - T_2) = 0.753(2100 - 530) = 1182.2$

3 -> 4: ${}_3w_4 = {}_3q_4 = RT_3 \ln(v_4/v_3) = 386 \times 2100 \times \ln(6)/778$

$$= 1866.8 \text{ Btu/lbm}$$

4 -> 1: ${}_4w_1 = 0;\quad {}_4q_1 = C_P(T_4 - T_1) = -1182.2$ Btu/lbm

$$\eta_{Cycle} = w_{net}/\, q_H = \frac{-471.15 + 1866.0}{1182.2 + 1866.8} = \mathbf{0.458}$$

11.199E

The air-standard Carnot cycle was not shown in the text; show the *T–s* diagram for this cycle. In an air-standard Carnot cycle the low temperature is 500 R and the efficiency is 60%. If the pressure before compression and after heat rejection is 14.7 lbf/in.^{2}, find the high temperature and the pressure just before heat addition.

Solution:

Carnot cycle efficiency from Eq.7.5

$$\eta = 0.6 = 1 - T_H/T_L$$

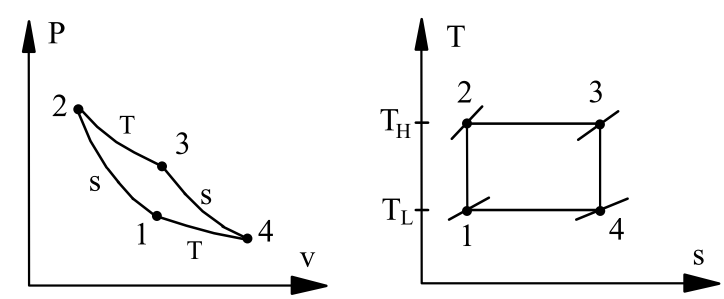
$$\Rightarrow\ T_H = T_L/0.4 = 500/0.4 = \mathbf{1250\ R}$$

Just before heat addition is state 2 and after heat rejection is state 1 so P_1 = 100 kPa and the isentropic compression is from Eq.8.32

$$P_2 = P_1(T_H/T_L)^{\frac{1}{k-1}} = 14.7\left(\frac{1250}{500}\right)^{3.5} = \mathbf{363.2\ lbf/in^2}$$

OR if we do not use constant specific heat, but use Table F.5 in Eq.8.28

$$P_2 = P_1 \exp[(s^o_{T2} - s^o_{T1})/R] = 14.7 \times \exp\left[\frac{1.84573 - 1.62115}{53.34 / 778}\right] = \mathbf{389\ lbf/in^2}$$



11.200E

Air in a piston/cylinder goes through a Carnot cycle in which T_L = 80.3 F and the total cycle efficiency is η = 2/3. Find T_H, the specific work and volume ratio in the adiabatic expansion for constant C_p, C_v.

Carnot cycle:

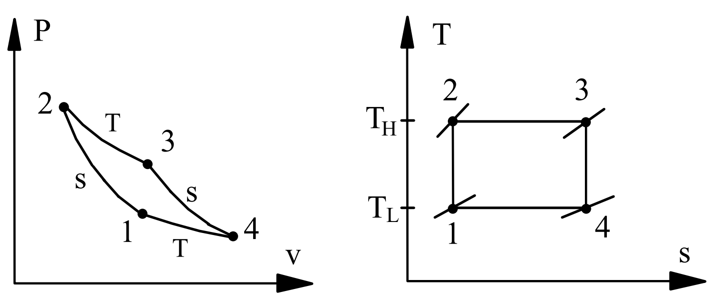
$\eta = 1 - T_L/T_H = 2/3 \;\Rightarrow\; T_H = 3 \times T_L = 3 \times 540 = \mathbf{1620\ R}$

Adiabatic expansion 3 to 4: Pv^k = constant

${}_3w_4 = (P_4v_4 - P_3v_3)/(1 - k) = [R/(1-k)](T_4 - T_3) = u_3 - u_4$

$= C_v(T_3 - T_4) = 0.171(1620 - 540) = \mathbf{184.68\ Btu/lbm}$

$v_4/v_3 = (T_3/T_4)^{1/(k - 1)} = 3^{2.5} = \mathbf{15.6}$



11.201E

Do the previous problem 11.200E using Table F.5.

Air in a piston/cylinder goes through a Carnot cycle in which T_L = 80.3 F and the total cycle efficiency is η = 2/3. Find T_H, the specific work and volume ratio in the adiabatic expansion for constant C_p, C_v.

Carnot cycle:

$$\eta = 1 - T_L/T_H = 2/3 \;\Rightarrow\; T_H = 3 \times T_L = 3 \times 540 = \mathbf{1620\ R}$$

$${}_3w_4 = u_3 - u_4 = 290.13 - 92.16 = \mathbf{197.97\ Btu/lbm}$$

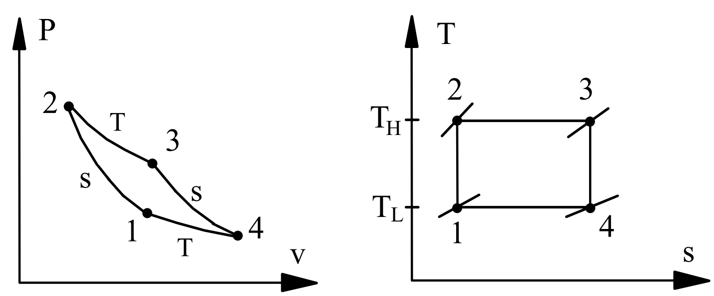
Adiabatic expansion 3 to 4: $s_4 = s_3 \;\Rightarrow$ Eq.8.28

$$s^o_{T4} = s^o_{T3} + R \ln \frac{P_4}{P_3} \quad \Rightarrow \quad \text{Table F.5 for standard entropy}$$

$$\frac{P_4}{P_3} = \exp\left[(s^o_{T4} - s^o_{T3})/R\right] = \exp\left[\frac{1.63979\text{-}1.91362}{53.34/778}\right] = 0.018426$$

Ideal gas law then gives

$$\frac{v_4}{v_3} = \frac{T_4}{T_3} \times \frac{P_3}{P_4} = \frac{540}{1620} \times \frac{1}{0.018426} = \mathbf{18.09}$$



Refrigeration Cycles

11.202E

A car air-conditioner (refrigerator) in 70 F ambient uses R-134a and I want to have cold air at 20 F produced. What is the minimum high P and the maximum low P it can use?

Since the R-134a must give heat transfer out to the ambient at 70 F, it must at least be that hot at state 3.

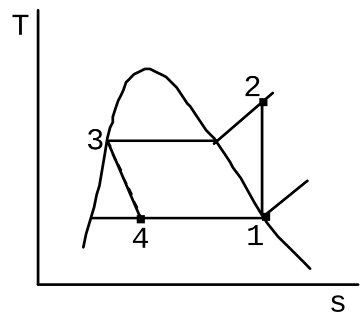
From Table F.10.1: $P_3 = P_2 = P_{sat} =$ **85.95 psia** is minimum high P.

Since the R-134a must absorb heat transfer at the cold air 20 F, it must at least be that cold at state 4.

From Table F.10.1: $P_1 = P_4 = P_{sat} =$ **33.29 psia** is maximum low P.

Ideal Ref. Cycle
$T_{cond} = 70\text{ F} = T_3$
$T_{evap} = 20\text{ F}$

Use Table F.10 for R-134a

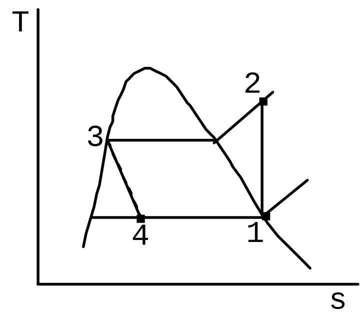


11.203E

Consider an ideal refrigeration cycle that has a condenser temperature of 110 F and an evaporator temperature of 5 F. Determine the coefficient of performance of this refrigerator for the working fluids R-12 and R-22.

Ideal Ref. Cycle
T_{cond} = 110 F = T_3
T_{evap} = 5 F

Use Table F.9 for R-22
Use computer table for R-12



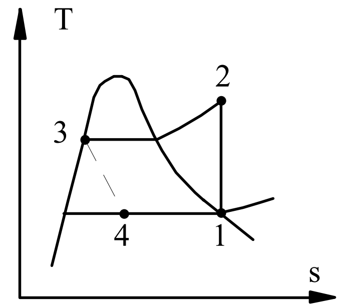
	R-12	R-22
h_1, Btu/lbm	77.803	104.954
$s_2 = s_1$	0.16843	0.22705
P_2, lbf/in^2	151.11	241.04
T_2, F	127.29	161.87
h_2, Btu/lbm	91.107	123.904
h_3=h_4, Btu/lbm	33.531	42.446
$-w_C = h_2-h_1$	13.3	18.95
$q_L = h_1-h_4$	44.27	62.51
$\beta = q_L/(-w_C)$	3.33	3.30

11.204E

The environmentally safe refrigerant R-134a is one of the replacements for R-12 in refrigeration systems. Repeat Problem 11.203 using R-134a and compare the result with that for R-12.

Ideal refrigeration cycle
$T_{cond} = 110$ F $= T_3$
$T_{evap} = 5$ F

Use Table F.10 for R-134a or computer table



C.V. Compressor: Adiabatic and reversible so constant s

State 1: Table F.10.1 $h_1 = 167.32$ Btu/lbm, $s_1 = 0.4145$ Btu/lbm R

State 2: $s_2 = s_1$ and $P_2 = 161.1$ psia $= P_3 = P_{sat\ 110\ F}$

Interpolate => $h_2 = 184.36$ Btu/lbm and $T_2 = 121.8$ F

Energy eq.: $w_C = h_2 - h_1 = 184.36 - 167.32 = 17.04$ Btu/lbm

Expansion valve: $h_3 = h_4 = 112.46$ Btu/lbm

Evaporator: $q_L = h_1 - h_4 = 167.32 - 112.46 = 54.86$ Btu/lbm

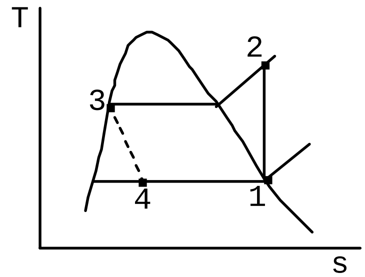
Overall performance, COP

$\beta = q_L/w_C = 54.86 / 17.04 =$ **3.22**

11.205E

Consider an ideal heat pump that has a condenser temperature of 120 F and an evaporator temperature of 30 F. Determine the coefficient of performance of this heat pump for the working fluids R-12, R-22, and ammonia.

Ideal Heat Pump
T_{cond} = 120 F
T_{evap} = 30 F
Use Table F.8 for NH_3
Use Table F.9 for R-22
Use computer table for R-12



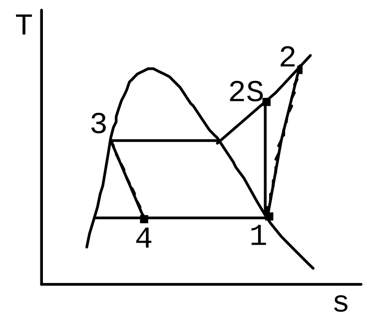
	R-12	R-22	NH_3
h_1, Btu/lbm	80.42	107.28	619.58
$s_2 = s_1$	0.1665	0.2218	1.2769
P_2, lbf/in^2	172.3	274.6	286.5
T_2, F	132.2	160.4	239.4
h_2, Btu/lbm	91.0	122.17	719.5
h_3=h_4, Btu/lbm	36.011	45.71	178.83
$-w_C = h_2-h_1$	10.58	14.89	99.92
$q_H = h_2-h_3$	54.995	76.46	540.67
$\beta' = q_H/(-w_C)$	5.198	5.135	5.411

11.206E

The refrigerant R-22 is used as the working fluid in a conventional heat pump cycle. Saturated vapor enters the compressor of this unit at 50 F; its exit temperature from the compressor is measured and found to be 185 F. If the compressor exit is 300 psia, what is the isentropic efficiency of the compressor and the coefficient of performance of the heat pump?

R-22 heat pump: T_2 = 185 F

T_{EVAP} = 50 F



State 1: Table F.9.1

h_1 = 108.95 Btu/lbm,

s_1 = 0.2180 Btu/lbm R

State 2: h_2 = 126.525 Btu/lbm

Compressor work: $w_C = h_2 - h_1 = 126.525 - 108.95 = 17.575$ Btu/lbm

Isentropic compressor: $s_{2S} = s_1 = 0.2180$ Btu/lbm R

State 2s: (P_2, s) T_{2S} = 160 F, h_{2S} = 120.82 Btu/lbm

Ideal compressor work: $w_{C\,s} = h_{2S} - h_1 = 120.82 - 108.95 = 11.87$ Btu/lbm

The efficiency is the ratio of the two work terms

$$\eta_{S\ COMP} = \frac{w_{C\,s}}{w_C} = \frac{11.87}{17.575} = \mathbf{0.675}$$

The condenser has heat transfer as ($h_3 = h_f$ at 300 psia)

$$q_H = h_2 - h_3 = 126.525 - 48.02 = 78.505 \text{ Btu/lbm}$$

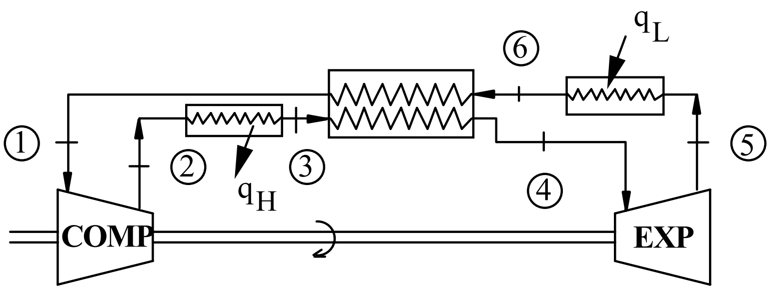
and a coefficient of performance of

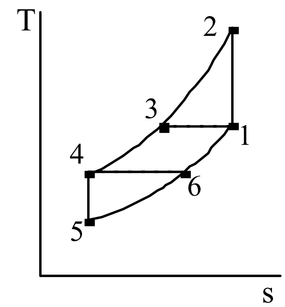
$$\beta' = q_H/w_C = \mathbf{4.47}$$

11.207E

Consider an air standard refrigeration cycle that has a heat exchanger included as shown in Fig. P11.137. The low pressure is 14.7 psia and the high pressure is 200 psia. The temperature into the compressor is 60 F which is T_1 and T_3 in Fig.11.38, and $T_4 = T_6$ = -60 F. Determine the coefficienct of performance of this cycle.

Solution:





Standard air refrigeration cycle with

$T_1 = T_3 = 60 \text{ F} = 519.67 \text{ R}, \quad P_1 = 14.7 \text{ psia}, \quad P_2 = 200 \text{ psia}$

$T_4 = T_6 = -60 \text{ F} = 399.67 \text{ R}$

We will solve the problem with cold air properties.

Compressor, isentropic $s_2 = s_1$ so from Eq.8.32

$$\Rightarrow T_2 = T_1(P_2/P_1)^{\frac{k-1}{k}} = 519.67\ (200/14.7)^{0.2857} = 1095.5 \text{ R}$$

$$w_C = -w_{12} = C_{P0}(T_2 - T_1) = 0.24\ (1095.5 - 519.67) = 138.2 \text{ Btu/lbm}$$

Expansion in expander (turbine)

$$s_5 = s_4 \Rightarrow T_5 = T_4\ (P_5/P_4)^{\frac{k-1}{k}} = 399.67\ (14.7/200)^{0.2857} = 189.58 \text{ R}$$

$$w_E = C_{P0}(T_4 - T_5) = 0.24\ (399.67 - 189.58) = 50.42 \text{ Btu/lbm}$$

Net cycle work

$$w_{NET} = 50.42 - 138.2 = -87.78 \text{ kJ/kg}$$

$$q_L = C_{P0}(T_6 - T_5) = w_E = 50.42 \text{ Btu/lbm}$$

Overall cycle performance, COP

$$\beta = q_L/(-w_{NET}) = 50.42 / 87.78 = \mathbf{0.574}$$

Availability and Combined Cycles

11.208E

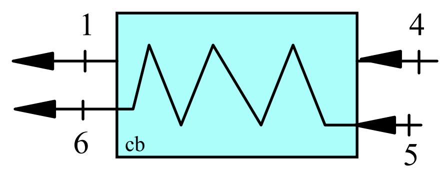
Find the flows and fluxes of exergy in the condenser of Problem 11.172E. Use those to determine the 2nd law efficiency.

A smaller power plant produces 50 lbm/s steam at 400 psia, 1100 F, in the boiler. It cools the condenser with ocean water coming in at 55 F and returned at 60 F so that the condenser exit is at 110 F. Find the net power output and the required mass flow rate of the ocean water.

Solution:

Take the reference state at the ocean temperature 55 F = 514.7 R

The states properties from Tables F.7.1 and F.7.2. Ref. state 14.7 lbf/in^2, 55 F,
h_0 = 23.06 Btu/lbm,
s_0 = 0.0458 Btu/lbm R



State 1: 110 F, x = 0: h_1 = 78.01 Btu/lbm, s_1 = 0.1473 Btu/lbm R,

State 3: 400 psia, 1100 F: h_3 = 1577.44 Btu/lbm, s_3 = 1.7989 Btu/lbm R

C.V. Turbine : $w_T = h_3 - h_4$; $s_4 = s_3$

$$s_4 = s_3 = 1.7989 = 0.1473 + x_4 (1.8101) \quad \Rightarrow \quad x_4 = 0.9124$$

$$\Rightarrow h_4 = 78.01 + 0.9124 (1031.28) = 1018.95 \text{ Btu/lbm}$$

C.V. Condenser : $q_L = h_4 - h_1 = 1018.95 - 78.01 = 940.94$ Btu/lbm

$$\dot{Q}_L = \dot{m}q_L = 50 \times 940.94 = 47\,047 \text{ Btu/s} = \dot{m}_{ocean} C_p \Delta T$$

$$\dot{m}_{ocean} = \dot{Q}_L / C_p \Delta T = 47\,047 / (1.0 \times \bullet 5) = 9409 \text{ lbm/s}$$

The specific flow exergy for the two states are from Eq.10.24 neglecting kinetic and potential energy

$$\psi_4 = h_4 - h_0 - T_0(s_4 - s_0), \quad \psi_1 = h_1 - h_0 - T_0(s_1 - s_0)$$

The net drop in exergy of the water is

$$\dot{\Phi}_{water} = \dot{m}_{water} [h_4 - h_1 - T_o(s_4 - s_1)]$$

$$= 50 [\, 1018.95 - 78.01 - 514.7 (1.7989 - 0.1473)]$$

$$= 47\,047 - 42\,504 = \mathbf{4543 \text{ Btu/s}}$$

The net gain in exergy of the ocean water is

$$\dot{\Phi}_{ocean} = \dot{m}_{ocean}[h_6 - h_5 - T_o(s_6 - s_5)]$$

$$= \dot{m}_{ocean}[C_p(T_6 - T_5) - T_oC_p \ln\left(\frac{T_6}{T_5}\right)]$$

$$= 9409 \text{ [} 1.0 \text{ } (60 - 55) - 514.7 \times 1.0 \ln \frac{459.7 + 60}{459.7 + 55} \text{]}$$

$$= 47\,047 - 46\,818 = \mathbf{229\ Btu/s}$$

The second law efficiency is

$$\eta_{II} = \dot{\Phi}_{ocean} / \dot{\Phi}_{water} = \frac{229}{4543} = \mathbf{0.05}$$

In reality all the exergy in the ocean water is destroyed as the 60 F water mixes with the ocean water at 55 F after it flows back out into the ocean and the efficiency does not have any significance. Notice the small rate of exergy relative to the large rates of energy being transferred.

11.209E

(Adv.) Find the availability of the water at all four states in the Rankine cycle described in Problem 11.173. Assume the high-temperature source is 900 F and the low-temperature reservoir is at 65 F. Determine the flow of availability in or out of the reservoirs per pound-mass of steam flowing in the cycle. What is the overall cycle second law efficiency?

Ref. state 14.7 lbf/in^2, 77°F, h_0 = 45.08 Btu/lbm, s_0 = 0.08774 Btu/lbm R

For this cycle from Table F.7

State 3: Superheated vapor h_3 = 1350.62 Btu/lbm, s_3 = 1.5871 Btu/lbm R,

State 1: Saturated liquid h_1 = 97.97 Btu/lbm, v_1 = 0.01625 ft^3/lbm

C.V. Pump: Adiabatic and reversible. Use incompressible fluid so

$$w_P = \int v\, dP = v_1(P_2 - P_1) = 0.01625(600 - 2.2)\frac{144}{778} = \mathbf{1.8\ Btu/lbm}$$

$$h_2 = h_1 + w_P = 95.81 \text{ Btu/lbm}$$

C.V. Boiler: $q_H = h_3 - h_2 = 1350.62 - 97.97 = \mathbf{1252.65\ Btu/lbm}$

C.V. Tubine: $w_T = h_3 - h_4$, $s_4 = s_3$

$$s_4 = s_3 = 1.5871 \text{ Btu/lbm R} = 0.1817 + x_4\, 1.7292 \Rightarrow x_4 = 0.8127,$$

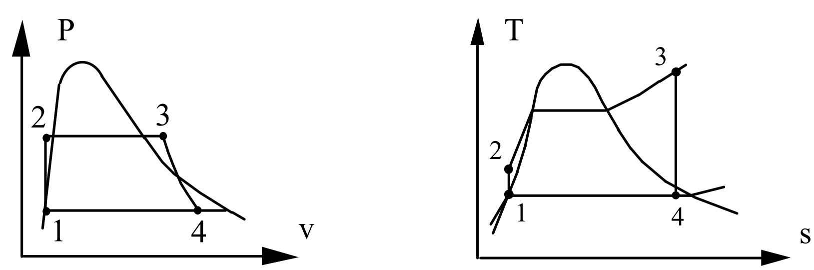
$$h_4 = 97.97 + 0.8127 \times 1019.78 = 926.75 \text{ Btu/lbm}$$

$$w_T = 1350.62 - 926.75 = \mathbf{423.87\ Btu/lbm}$$

$$\eta_{CYCLE} = (w_T - w_P)/q_H = (423.87 - 1.8)/1252.65 = \mathbf{0.337}$$

C.V. Condenser:

$$q_L = h_4 - h_1 = 926.75 - 97.97 = \mathbf{828.8\ Btu/lbm}$$



From solution to 11.121:

s_1 = 0.17497, s_2 = 0.175 = s_1, s_4 = s_3 = 1.5871 Btu/lbm R

h_1 = 94.01, h_2 = 95.81, h_3 = 1350.6, h_4 = 921.23 Btu/lbm

$\psi = h - h_0 - T_0(s - s_0)$

$\psi_1 = 94.01 - 45.08 - 536.67(0.17497 - 0.08774) =$ **2.116 Btu/lbm**

$\psi_2 =$ **3.92**, $\psi_3 =$ **500.86**, $\psi_4 =$ **71.49 Btu/lbm**

$\Delta\psi_H = (1 - T_0/T_H)q_H = 0.6054 \times 1254.79 =$ **759.65 Btu/lbm**

$\Delta\psi_L = (1 - T_0/T_0)q_C =$ **Ø**

$\eta_{II} = w_{NET}/\Delta\psi_H = (429.37 - 1.8)/759.65 =$ **0.563**

Notice $T_H > T_3$, $T_L < T_4 = T_1$, so cycle is externally irreversible. Both q_H and q_C over finite ΔT.

11.210E

Find the flows of exergy into and out of the feedwater heater in Problem 11.176E.

State 1: $x_1 = 0$, $h_1 = 97.97$ Btu/lbm, $v_1 = 0.01625$ ft^3/lbm, $s = 0.17497$

State 3: $x_3 = 0$, $h_3 = 330.67$ Btu/lbm, $s_3 = 0.49199$ Btu/lbm R

State 5: $h_5 = 1350.52$ Btu/lbm, $s_5 = 1.5871$ Btu/lbm R

State 6: $s_6 = s_5 = 1.5871$ Btu/lbm R => $h_6 = 1208.93$ Btu/lbm

C.V Pump P1

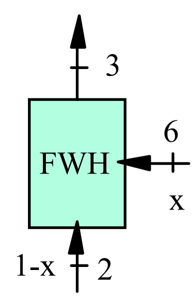
$$w_{P1} = h_2 - h_1 = v_1(P_2 - P_1) = 0.01625(150 - 2.225)\frac{144}{778} = 0.44 \text{ Btu/lbm}$$

$$\Rightarrow h_2 = h_1 + w_{P1} = 97.97 + 0.4439 = 98.41 \text{ Btu/lbm}$$

$$s_2 = s_1 = 0.17497 \text{ Btu/lbm R}$$

C.V. Feedwater heater: Call $\dot{m}_6 / \dot{m}_{tot} = x$ (the extraction fraction)

Energy Eq.: $(1 - x)\, h_2 + x\, h_6 = 1\, h_3$



$$x = \frac{h_3 - h_2}{h_6 - h_2} = \frac{330.67 - 98.41}{1208.93 - 98.41} = 0.2091$$

Ref. State: 14.7 psia, 77 F, $s_o = 0.08774$ Btu/lbm R, $h_o = 45.08$ Btu/lbm

$$\psi_2 = h_2 - h_o - T_o(s_2 - s_o)$$

$$= 98.41 - 45.08 - 536.67(0.17497 - 0.08774) = 6.52 \text{ Btu/lbm}$$

$$\psi_6 = 1208.93 - 45.08 - 536.67(1.5871 - 0.08774) = 359.2 \text{ Btu/lbm}$$

$$\psi_3 = 330.67 - 45.08 - 536.67(0.49199 - 0.08774) = 68.64 \text{ Btu/lbm}$$

The rate of exergy flow scaled with maximum flow rate is then

$\dot{\Phi}_2/\dot{m}_3 = (1 - x)\, \psi_2 = 0.7909 \times 6.52 =$ **5.157 Btu/lbm**

$\dot{\Phi}_6/\dot{m}_3 = x\psi_6 = 0.2091 \times 359.2 =$ **75.109 Btu/lbm**

$\dot{\Phi}_3/\dot{m}_3 = \psi_3 =$ **68.64 Btu/lbm**

The mixing is destroying 5.157 + 75.109 – 68.64 = 11.6 Btu/lbm of exergy

11.211E

Consider the Brayton cycle in problem 11.183E. Find all the flows and fluxes of exergy and find the overall cycle second-law efficiency. Assume the heat transfers are internally reversible processes and we then neglect any external irreversibility.

Solution:

Efficiency is from Eq.11.8

$$\eta = \dot{W}_{NET}/\dot{Q}_H = \frac{w_{net}}{q_H} = 1 - r_p^{-(k-1)/k} = 1 - 16^{-0.4/1.4} = \mathbf{0.547}$$

from the required power we can find the needed heat transfer

$$\dot{Q}_H = \dot{W}_{net} / \eta = 14\ 000 / 0.547 = 25\ 594 \text{ Btu/s}$$

$$\dot{m} = \dot{Q}_H / q_H = 25\ 594 \text{ (Btu/s)} / 400 \text{ Btu/lbm} = \mathbf{63.99\ lbm/s}$$

Temperature after compression is

$$T_2 = T_1\, r_p^{(k-1)/k} = 519.67 \times 16^{0.4/1.4} = 1148 \text{ R}$$

The highest temperature is after combustion

$$T_3 = T_2 + q_H/C_p = 1148 + \frac{400}{0.24} = \mathbf{2815\ R}$$

For the exit flow I need the exhaust temperature

$$T_4 = T_3\ \ r_p^{-(k-1)/k} = 2815 \times 16^{-0.2857} = 1274.8 \text{ R}$$

The high T exergy input from combustion is

$$\dot{\Phi}_H = \dot{m}(\psi_3 - \psi_2) = \dot{m}[h_3 - h_2 - T(s_3 - s_2)]$$

$$= 63.99\ [400 - 536.67 \times 0.24 \ln (\frac{2815}{1148})] = \mathbf{17\ 895\ Btu/s}$$

Since the low T exergy flow out is lost the second law efficiency is

$$\eta_{II} = \dot{W}_{NET}/\dot{\Phi}_H = 14\ 000 / 17\ 895 = \mathbf{0.782}$$

$$\dot{\Phi}_{flow\ out} = \dot{m}(\psi_4 - \psi_o) = \dot{m}[h_4 - h_o - T(s_4 - s_o)]$$

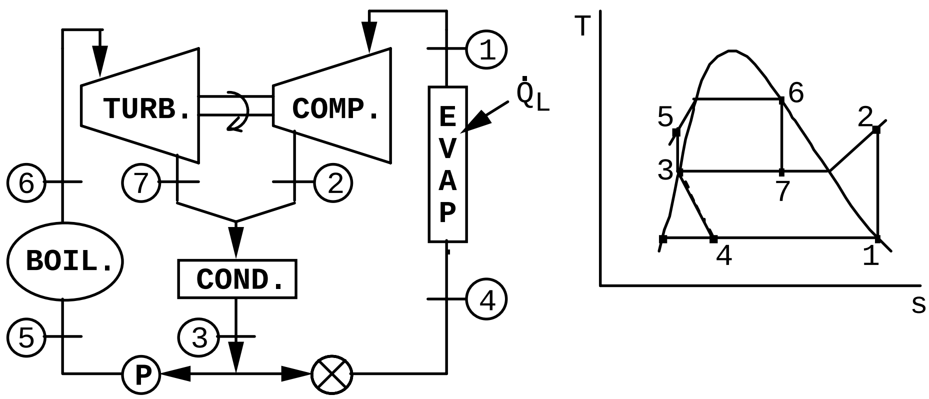
$$= 63.99\ [\ 0.24(1274.8 - 536.7) - 536.7 \times 0.24 \ln (\frac{1274.8}{536.7})\ ] = \mathbf{4205\ Btu/s}$$

$$\dot{\Phi}_{flow\ in} = \dot{m}(\psi_1 - \psi_o) = \dot{m}[h_1 - h_o - T(s_1 - s_o)]$$

$$= 63.99\ [\ 0.24(60 - 77) - 536.7 \times 0.24 \ln (\frac{519.7}{536.7})\ ] = \mathbf{4.2\ Btu/s}$$

11.212E

Consider an ideal dual-loop heat-powered refrigeration cycle using R-12 as the working fluid, as shown in Fig. P11.144. Saturated vapor at 220 F leaves the boiler and expands in the turbine to the condenser pressure. Saturated vapor at 0 F leaves the evaporator and is compressed to the condenser pressure. The ratio of the flows through the two loops is such that the turbine produces just enough power to drive the compressor. The two exiting streams mix together and enter the condenser. Saturated liquid leaving the condenser at 110 F is then separated into two streams in the necessary proportions. Determine the ratio of mass flow rate through the power loop to that through the refrigeration loop. Find also the performance of the cycle, in terms of the ratio Q_L/Q_H.



	T	P	h	s
	F	lbf/in^2	Btu/lbm	Btu/lbm R
1	0	23.849	77.271	168.88
2	-	151.11		168.88
3	110	151.11	33.531	0.067 45
4	0	23.849	33.531	
5	-	524.43		0.067 45
6	220	524.43	89.036	0.151 49
7	110	151.11		0.151 49

Computer tables for properties.

$P_2=P_3=P_{SAT}$ at 110 F

$P_5=P_6=P_{SAT}$ at 220 F

$s_2=s_1=0.168\,88$

$h_2=91.277$

Pump work:

$-w_P = h_5-h_3$

$\approx v_5(P_5-P_3)$

$$-w_P = 0.0129(524.4 - 151.1)\frac{144}{778} = 0.894$$

$$h_5 = 33.531 + 0.894 = 34.425 \text{ Btu/lbm}$$

$$(1-x_7) = \frac{0.162\,79 - 0.151\,49}{0.095\,34} = \frac{0.011\,30}{0.095\,34} = 0.1187$$

$$h_7 = 87.844 - 0.1187(54.313) = 81.397 \text{ Btu/lbm}$$

CV: turbine + compressor

Continuity Eq.: $\dot{m}_1 = \dot{m}_2, \quad \dot{m}_6 = \dot{m}_7$

Energy Eq.: $\dot{m}_1 h_1 + \dot{m}_6 h_6 = \dot{m}_2 h_2 + \dot{m}_7 h_7$

$$\dot{m}_1/\dot{m}_6 = \frac{89.036\text{-}81.397}{91.277\text{-}77.271} = \frac{7.639}{14.006} = 0.545, \qquad \dot{m}_6/\dot{m}_1 = \mathbf{1.833}$$

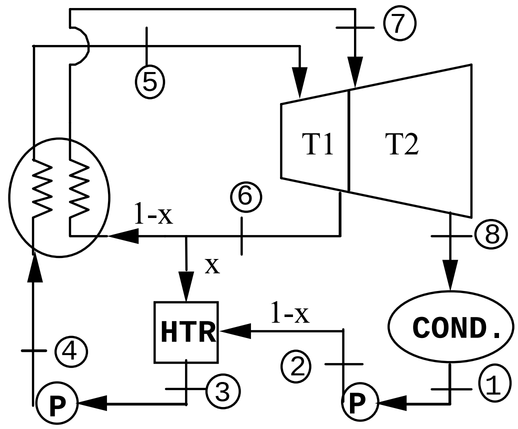
CV: pump: $-w_P = v_3(P_5 - P_3), \quad h_5 = h_3 - w_P$

CV evaporator: $\dot{Q}_L = \dot{m}_1(h_1 - h_4)$, CV boiler: $\dot{Q}_H = \dot{m}_6(h_6 - h_5)$

$$\Rightarrow \beta = \dot{Q}_L/\dot{Q}_H = \frac{\dot{m}_1(h_1 - h_4)}{\dot{m}_6(h_6 - h_5)} = \frac{77.271\text{-}33.531}{1.833(89.036\text{-}34.425)} = \mathbf{0.436}$$

11.213E

Consider an ideal combined reheat and regenerative cycle in which steam enters the high-pressure turbine at 500 lbf/in.^{2}, 700 F, and is extracted to an open feedwater heater at 120 lbf/in.^{2} with exit as saturated liquid. The remainder of the steam is reheated to 700 F at this pressure, 120 lbf/in.^{2}, and is fed to the low-pressure turbine. The condenser pressure is 2 lbf/in.^{2}. Calculate the thermal efficiency of the cycle and the net work per pound-mass of steam.



5: $h_5 = 1356.66, \quad s_5 = 1.6112$

7: $h_7 = 1378.17, \quad s_7 = 1.7825$

3: $h_3 = h_f = 312.59, v_3 = 0.01788$

C.V. T1

$s_5 = s_6 \quad => h_6 = 1209.76$

$w_{T1} = h_5 - h_6 = 1356.66 - 1209.76$

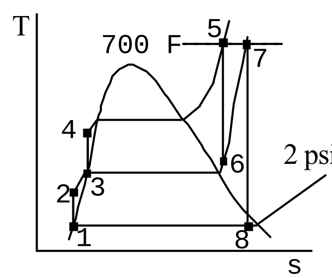
$= 146.9$ Btu/lbm

C.V. Pump 1

$-w_{P1} = h_2 - h_1 = v_1(P_2 - P_1)$

$= 0.01623(120 - 2) = 0.354$

$=> \quad h_2 = h_1 - w_{P1} = 93.73 + 0.354 = 94.08$ Btu/lbm



C.V. FWH

$x\, h_6 + (1 - x)\, h_2 = h_3$

$$x = \frac{h_3 - h_2}{h_6 - h_2} = \frac{312.59 - 94.08}{1209.76 - 94.08} = 0.1958$$

C.V. Pump 2

$-w_{P2} = h_4 - h_3 = v_3(P_4 - P_3) = 0.01788(500 - 120)(144/778) = 1.26$ Btu/lbm

$=> \quad h_4 = h_3 - w_{P2} = 312.59 + 1.26 = 313.85$ Btu/lbm

$q_H = h_5 - h_4 + (1 - x)(h_7 - h_6) = 1042.81 + 135.43 = 1178.2$ Btu/lbm

C.V. Turbine 2

$s_7 = s_8 \quad => x_8 = (1.7825 - 0.1744)/1.746 = 0.921$

$h_8 = h_f + x_8\, h_{fg} = 93.73 + 0.921 \times 1022.2 = 1035.2$

$w_{T2} = h_7 - h_8 = 1378.17 - 1035.2 = 342.97$

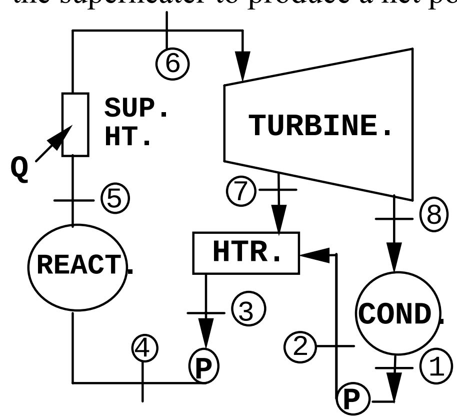
$w_{net} = w_{T1} + (1 - x)\, w_{T2} + (1 - x)\, w_{P1} + w_{P2}$

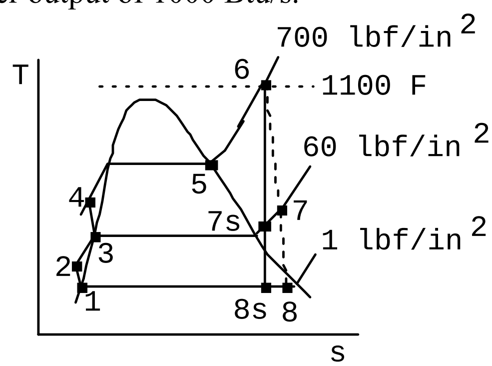
$= 146.9 + 275.8 - 0.285 - 1.26 =$ **421.15 kJ/kg**

$\eta_{cycle} = w_{net} / q_H = 421.15 / 1178.2 =$ **0.357**

11.214E

In one type of nuclear power plant, heat is transferred in the nuclear reactor to liquid sodium. The liquid sodium is then pumped through a heat exchanger where heat is transferred to boiling water. Saturated vapor steam at 700 lbf/in.^{2} exits this heat exchanger and is then superheated to 1100 F in an external gas-fired superheater. The steam enters the turbine, which has one (open-type) feedwater extraction at 60 lbf/in.^{2}. The isentropic turbine efficiency is 87%, and the condenser pressure is 1 lbf/in.^{2}. Determine the heat transfer in the reactor and in the superheater to produce a net power output of 1000 Btu/s.





$\dot{W}_{NET} = 1000$ Btu/s, $\eta_{ST} = 0.87$

$-w_{P12} = 0.016136(60 - 1)144/778 = 0.18$ Btu/lbm

$h_2 = h_1 - w_{P12} = 69.73 + 0.18 = 69.91$ Btu/lbm

$-w_{P34} = 0.017378(700 - 60)144/778 = 2.06$ Btu/lbm

$h_4 = h_3 - w_{P34} = 262.24 + 2.06 = 264.3$ Btu/lbm

$s_{7S}=s_6 = 1.7682$, P_7 => $T_{7S} = 500.8$ F, $h_{7S} = 1283.4$

$h_7 = h_6 - \eta_{ST}(h_6 - h_{7S}) = 1625.8 - 0.87(1625.8 - 1283.4) = 1327.9$

$s_{8S} = s_6 = 1.7682 = 0.13264 + x_{8S} \times 1.8453$ => $x_{8S} = 0.8863$

$h_{8S} = 69.73 + 0.8863 \times 1036 = 987.9$ Btu/lbm

$h_8 = h_6 - \eta_{ST}(h_6 - h_{8S}) = 1625.8 - 0.87(1625.8 - 987.9) = 1070.8$

CV: heater: cont: $m_2 + m_7 = m_3 = 1.0$ lbm, 1st law: $m_2h_2 + m_7h_7 = m_3h_3$

$m_7 = (262.24\text{-}69.91) / (1327.9\text{-}69.91) = 0.1529$

CV: turbine: $w_T = (h_6 - h_7) + (1 - m_7)(h_7 - h_8)$

$= 1625.8\text{-}1327.9 + 0.8471(1327.9\text{-}1070.8) = 515.7$ Btu/lbm

CV pumps: $w_P = m_1w_{P12} + m_3w_{P34} = -(0.8471\times0.18 + 1\times2.06) = -2.2$ Btu/lbm

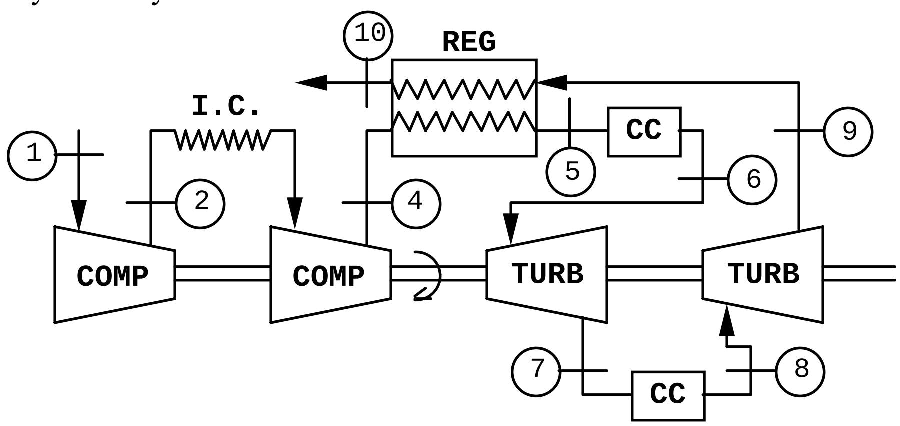
$w_{NET} = 515.7 - 2.2 = 513.5$ Btu/lbm => $\dot{m} = 1000/513.5 = 1.947$ lbm/s

CV: reactor $\dot{Q}_{REACT} = \dot{m}(h_5 - h_4) = 1.947(1202 - 264.3) =$ **1825.7 Btu/s**

CV: superheater $\dot{Q}_{SUP} = \dot{m}(h_6 - h_5) = 1.947(1625.8 - 1202) =$ **825 Btu/s**

11.215E

Consider an ideal gas-turbine cycle with two stages of compression and two stages of expansion. The pressure ratio across each compressor stage and each turbine stage is 8 to 1. The pressure at the entrance to the first compressor is 14 lbf/in.^{2}, the temperature entering each compressor is 70 F, and the temperature entering each turbine is 2000 F. An ideal regenerator is also incorporated into the cycle. Determine the compressor work, the turbine work, and the thermal efficiency of the cycle.



$P_2/P_1 = P_4/P_3 = P_6/P_7 = P_8/P_9 = 8.0$

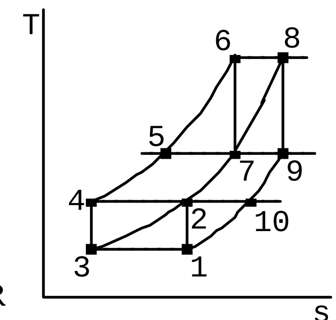
$P_1 = 14 \text{ lbf/in}^2$

$T_1 = T_3 = 70 \text{ F}, \quad T_6 = T_8 = 2000 \text{ F}$

Assume const. specific heat

$s_2 = s_1$ and $s_4 = s_3$

$$T_4 = T_2 = T_1(P_2/P_1)^{\frac{k-1}{k}} = 529.67(8)^{0.2857} = 959.4 \text{ R}$$



Total compressor work

$$-w_C = 2 \times (-w_{12}) = 2C_{P0}(T_2 - T_1) = 2 \times 0.24(959.4 - 529.67) = \mathbf{206.3 \text{ Btu/lbm}}$$

Also $s_6 = s_7$ and $s_8 = s_9$

$$\Rightarrow \; T_7 = T_9 = T_6\left(\frac{P_7}{P_6}\right)^{\frac{k-1}{k}} = 2459.67\left(\frac{1}{8}\right)^{0.2857} = 1357.9 \text{ R}$$

Total turbine work

$$w_T = 2\times w_{67} = 2C_{P0}(T_6 - T_7) = 2 \times 0.24(2459.67 - 1357.9) = 528.85 \text{ Btu/lbm}$$

$$w_{NET} = 528.85 - 206.3 = 322.55 \text{ Btu/lbm}$$

Ideal regenerator: $T_5 = T_9, \;\; T_{10} = T_4$

$\Rightarrow q_H = (h_6 - h_5) + (h_8 - h_7) = 2C_{P0}(T_6 - T_5)$

$= 2 \times 0.24(2459.67 - 1357.9) = w_T = 528.85$ Btu/lbm

$\eta_{TH} = w_{NET}/q_H = 322.55/528.85 =$ **0.61**

11.216E

Repeat Problem 11.215, but assume that each compressor stage and each turbine stage has an isentropic efficiency of 85%. Also assume that the regenerator has an efficiency of 70%.

$T_{4S} = T_{2S} = 959.4$ R, $-w_{CS} = 206.3$

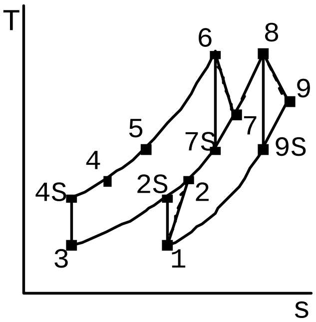
$T_{7S} = T_{9S} = 1357.9$ R, $w_{TS} = 528.85$

$\Rightarrow \quad -w_C = -w_{SC}/\eta_{SC} = 242.7$ Btu/lbm

$-w_{12} = -w_{34} = 242.7/2 = 121.35$ Btu/lbm

$T_2 = T_4 = T_1 + (-w_{12}/C_{P0})$

$= 529.67 + 121.35/0.24 = 1035.3$ R



$w_T = \eta_T \, w_{TS} = 449.5$ Btu/lbm

$T_7 = T_9 = T_6 - (+w_{67}/C_{P0}) = 2459.67 - 449.5/2\times0.24 = 1523$ R

$$\eta_{REG} = \frac{h_5 - h_4}{h_9 - h_4} = \frac{T_5 - T_4}{T_9 - T_4} = \frac{T_5 - 1035.3}{1523 - 1035.3} = 0.7 \quad \Rightarrow \; T_5 = 1376.7 \text{ R}$$

$q_H = C_{P0}(T_6 - T_5) + C_{P0}(T_8 - T_7)$

$= 0.24(2459.67 - 1376.7) + 0.24(2459.67 - 1523) = 484.7$ Btu/lbm

$w_{NET} = w_T + w_C = 449.5 - 242.7 = 206.8$ Btu/lbm

$\eta_{TH} = w_{NET}/q_H = 206.8/484.7 =$ **0.427**

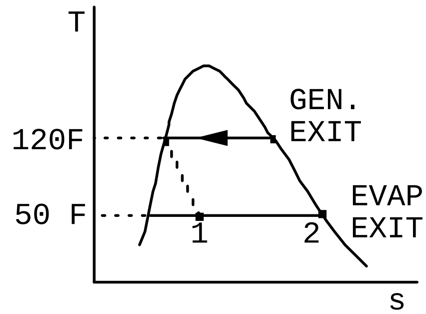
11.217E

Consider a small ammonia absorption refrigeration cycle that is powered by solar energy and is to be used as an air conditioner. Saturated vapor ammonia leaves the generator at 120 F, and saturated vapor leaves the evaporator at 50 F. If 3000 Btu of heat is required in the generator (solar collector) per pound-mass of ammonia vapor generated, determine the overall performance of this system.

NH_3 absorption cycle:

sat. vapor at 120 F exits the generator.
Sat. vapor at 50 F exits the evaporator

$q_H = q_{GEN} = 3000$ Btu/lbm NH_3 out of generator.



$q_L = h_2 - h_1 = h_{G\ 50\ F} - h_{F\ 120\ F} = 624.28 - 178.79$

$= 445.49$ Btu/lbm $\Rightarrow$ $q_L/q_H = 445.49/3000 =$ **0.1485**