ME 221
APPLIED
THERMO

DRALLMEIER

ME 221 - Applied Thermodynamics
Course Structure - Fall 2010

Instructor: Professor Jim Drallmeier doesn't use blackboard
127 Toomey Hall
341-4710; drallmei@mst.edu
Office Hours: At any time by appointment (i.e. to ensure I am there)

Schedule: Lecture: 1:00 - 1:50 MWF

Prerequisite: Thermodynamics (ME 219, Grade of "C" or better)

Grading: 4 exams 20% each, 80% total

Approximate timing:

- Exam 1- week of 9/20
- Exam 2 - week of 10/18
- Exam 3 - week of 11/15
- Exam 4 - week of 12/13 (final exam week)

Homework 20% total

Final grades are curved. Approximate breakdown:
B-C cutoff at class mean score.
A: above 1 standard deviation from mean
D: below 1 standard deviation from mean
F: below 2 standard deviations from mean

I will provide you with your standing and current class statistics at any time in the semester upon request.

Exam dates are chosen by the class. Makeup exams will rarely be given. Vacations or lack of preparation are not valid reasons. If you are very ill, contact me **before** the exam so we can discuss options. If you contact me **after** the exam, you will receive a zero for that exam. If you require extra accommodations for exams, the exam must still be taken on the same day beginning at the scheduled class time. If you feel you have a justifiable conflict with the exam date chosen by the class (e.g. interview trip for a co-op), visit with me at least one week in advance to discuss options. No extra credit is available.

Homework:

- Homework will be assigned each class (typically 2 problems), due at the beginning of the next class.
- Typically, only one of the assigned problems will be collected.
- No late homework is accepted. However, the two lowest homework scores are dropped.
- Solutions to all problems will be posted, after homework is collected, outside of room ME 170. Solutions to my problems will be put on electronic reserve before exams (http://web.mst.edu/~lib-circ/). My solutions to book problems can only be posted in paper form.
- A problem session will be scheduled. Attendance is voluntary.
- Strategy for success: Review your solution with the posted solution immediately…not just before exam.

Class Notes:

- Given in class is an outline format.
- Essential to success on exams.
- Strategy for success: Review the problem examples provided in class before doing HW.

Text: Thermodynamics – Cengel and Boles, Sixth Edition

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Test Notes -- Hour Exam II

Ideal Gas Relations:

$$Pv = RT,\ R = \overline{R} / MW\ (\overline{R} = 8.314\ kJ/kmol\text{-}K,\ MW = \text{molecular weight})$$

$$C_{P,AIR} = 1.0035 \frac{kJ}{kg\ K} \quad ; R_{AIR} = 0.287 \frac{kJ}{kg\ K}$$

$$C_{V,AIR} = 0.7165 \frac{kJ}{kg\ K}$$

h, u, c_p, c_v = functions of temperature only

$$du(T) = c_v(T)\,dT,\ dh(T) = c_p(T)\,dT \quad (\text{any process})$$

$$s_2 - s_1 = s^o_{T2} - s^o_{T1} - R\ln\left(\frac{P_2}{P_1}\right)$$

$$s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) \quad (\text{constant specific heats})$$

$$= c_v \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{V_2}{V_1}\right) \quad (\text{constant specific heats})$$

Isentropic Processes (constant specific heats):

$$Pv^k = \text{Constant}$$

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

Polytropic Processes:

$$Pv^n = \text{constant} \qquad {}_1w_2 = \left(\frac{R}{1-n}\right)(T_2 - T_1)$$

	✓	✓	✓	✓	✓	✓	✓	✓		✓
13	14	15	16	17	18	19	20	21	22	23
		energy			algebraically	only T-s diagram				

Fixed Mass Systems:

$$\delta Q - \delta W = dE \qquad {}_1Q_2 - {}_1W_2 = E_2 - E_1$$

$$E = 1/2\, m\, v^2 + m\, g\, z + U \qquad W = \int P\, dV$$

$$S_2 - S_1 = \int_1^2 \frac{\delta Q}{T} + S_{GEN}$$

$$x = (u - u_o) + P_o(v - v_o) - T_o(s - s_o)$$

$$W_{REV} = m(x_1 - x_2)$$

Open Systems:

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}_{in}\left(h + \frac{V^2}{2} + g\,z\right)_{in} - \dot{m}_{out}\left(h + \frac{V^2}{2} + g\,z\right)_{out}$$

if isentropic: $W = \int_o^i v\, dP$

$$\frac{dS_{cv}}{dt} = \sum_{in} \dot{m}_{in}\, s_{in} - \sum_{out} \dot{m}_{out}\, s_{out} + \sum \frac{\dot{Q}_i}{T_i} + \dot{S}_{GEN}$$

$$\psi = (h - h_o) - T_o(s - s_o) + 1/2V^2$$

$$\dot{W}_{REV} = \sum \dot{m}_i \psi_i - \sum \dot{m}_e \psi_e$$

Brayton Cycle:

$$\eta_{th} = 1 - \left(\frac{1}{P_2/P_1}\right)^{(k-1)/k}$$ (w/o regeneration, constant specific heats, isentropic devices)

Efficiencies:

$$\eta_{th} = \frac{W_{NET}}{Q_H}$$

$$\eta_T = \frac{W_{ACT}}{W_S} = \frac{(h_1 - h_2)_{ACT}}{(h_1 - h_2)_S} \qquad \eta_p = \eta_c = \frac{W_S}{W_{ACT}} = \frac{(h_1 - h_2)_S}{(h_1 - h_2)_{ACT}}$$

$COP_R = \frac{q_L}{w}$; $COP_{HP} = \frac{q_H}{w}$; 1 ton = 3.52 kW

$$\psi = (h - h_o) - T_o(s - s_o) + 1/2V^2$$

$$\dot{W}_{REV} = \Sigma \dot{m}_i \psi_i - \Sigma \dot{m}_e \psi_e$$

Ideal Gas Mixtures:

$$mf_i = \frac{m_i}{m} \quad ; \quad y_i = \frac{n_i}{n} \quad ; \quad MW = \sum_i y_i MW_i \quad ; \quad mf_i = y_i \frac{MW_i}{MW}$$

$$\bar{u} = \sum_i y_i\, u_i \quad ; \quad \bar{h} = \sum_i y_i\, h_i$$

$$u = \sum_i c_i u_i \quad ; \quad h = \sum_i c_i\, h_i$$

$$\bar{C}_v = \sum_i y_i\, \bar{C}_{v,i} \quad ; \quad \bar{C}_p = \sum_i y_i\, \bar{C}_{p,i}$$

$$\bar{s} = \sum_i y_i\, s_i \quad \text{where } s_i = s\,(T,\ P_i)$$

$$\Delta\,\bar{u} = \sum_i \left[y_2\, \bar{u}\,(T_2) - y_1\, \bar{u}\,(T_1)\right]_i \quad ; \quad \Delta\bar{h} = \sum_i \left[y_2\, \bar{h}(T_2) - y_1\, \bar{h}\,(T_1)\right]_i$$

$$\Delta\bar{s} = \sum_i \left[y_2\, \bar{s}(T_2, P_{i2}) - y_1\, \bar{s}\,(T_1, P_{i1})\right]_i$$

Psychrometrics:

$$\omega = \frac{m_v}{m_a} \cong 0.622 \frac{P_v}{P - P_v}$$

$$\phi = \left.\frac{P_v}{P_g}\right|_{T,P}$$

$$h\big|_{SYST} = h_a + \omega h_v \qquad \text{(kJ /kg dry air)}$$

Combustion:

$$\phi = \frac{\left(\frac{F}{A}\right)_{actual}}{\left(\frac{F}{A}\right)_{stoichiometric}}$$

1st law for Open Reacting Systems (Control Volume)

$$\bar{Q} - \bar{W} = \sum_P n_i(\bar{h}_f^o + \Delta\bar{h})_i - \sum_R n_i(\bar{h}_f^o + \Delta\bar{h})_i$$

1st Law for Closed Reacting Systems (Fixed Mass)

$$\bar{Q} - \bar{W} = \sum_P n_i(\bar{h}_f^o + \Delta\bar{h} - \bar{R}T)_i - \sum_R n_i(\bar{h}_f^o + \Delta\bar{h} - \bar{R}T)_i$$

Heating Value

$$HV = -\,\bar{Q} = \sum_R n_i(\bar{h}_f^o)_i - \sum_P n_i(\bar{h}_f^o)_i$$

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Test Notes -- Hour Exam IV

Ideal Gas Relations:

$$Pv = RT, R = \overline{R} / MW \quad (\overline{R} = 8.314\ kJ/kmol - K, MW = \text{molecular weight})$$

$$s_2 - s_1 = s^o_{T2} - s^o_{T1} - R\ln\left(\frac{P_2}{P_1}\right)$$

$$s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) \quad \text{(constant specific heats)}$$

$$= c_v \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{V_2}{V_1}\right) \quad \text{(constant specific heats)}$$

Isentropic Processes (constant specific heats):

$$Pv^k = \text{Constant}$$

$$\frac{P_1}{P_2} = \left(\frac{v_2}{v_1}\right)^k = \left(\frac{T_1}{T_2}\right)^{\frac{k}{k-1}}, \quad \frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}, \quad \frac{v_1}{v_2} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{k-1}} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{k}}$$

Fixed Mass Systems:

$$\delta Q - \delta W = dE \qquad {}_1Q_2 - {}_1W_2 = E_2 - E_1$$

$$E = 1/2\, m v^2 + mgz + U \qquad W = \int P\, dV$$

$$S_2 - S_1 = \int_1^2 \frac{\delta Q}{T} + S_{GEN}$$

$$x = (u - u_o) + P_o(v - v_o) - T_o(s - s_o)$$

$$W_{REV} = m(x_1 - x_2)$$

Open Systems:

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}_{in}\left(h + \frac{V^2}{2} + gz\right)_{in} - \dot{m}_{out}\left(h + \frac{V^2}{2} + gz\right)_{out}$$

if isentropic: $W = \int_o^i v\, dP$

$$\frac{dS_{cv}}{dt} = \sum_{in} \dot{m}_{in}\, s_{in} - \sum_{out} \dot{m}_{out}\, s_{out} + \sum \frac{\dot{Q}_i}{T_i} + \dot{S}_{GEN}$$

Exams

NAME: David Malawey

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Good Job.

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Exam III
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(20 pts) 1. A 5m x 5m x 3m room contains air at 20°C and 100 kPa. The relative humidity is 75%. Find:

(5 pts) a) The partial pressure of dry air (kPa)

$V = 75\,m^3 \qquad \phi = .75 = \frac{P_v}{P_g} \qquad P_g = 2.3392 \qquad \Rightarrow \boxed{P_v = 1.754\ kPa}$ ✓

(5 pts) b) The specific humidity

$\omega = .622\left(\frac{P_v}{P - P_v}\right) = .622\left(\frac{1.754}{100 - 1.754}\right)$

$\boxed{\omega = .0111}$ ✓

pts) c) The masses of dry air and water vapor in the room (kg)

$m_a = \frac{P_a \forall}{R_a T} = \frac{(100 - 1.754)(75)}{(.2870)(293)} = \boxed{87.62\ kg}$ ✓

$m_v = \frac{P_v \forall}{R_v T} = \frac{(1.754)(75)}{(.4615)(293)} = \boxed{.9729\ kg}$ ✓

(5 pts) d) The window temperature at which moisture in the air will start to condense on the inner window surface.

$T_{dp}: \quad P_v = P_g$ @ interpolate $\left(\frac{1.754 - 1.706}{2.339 - 1.706}\right)(5) + 15 = \boxed{15.38^\circ C}$ ✓

20

(30 pts) 2. Two insulated (adiabatic) tanks, A and B, are connected by a valve. Tank A has a volume of 1 m^3 and initially contains O_2 at 300 kPa and 20°C. Tank B has a volume of 2 m^3 and initially contains CO_2 at 250 kPa and 50°C. The valve is opened and remains open until the resulting gas mixture comes to a uniformly mixed state.

For this problem assume constant specific heats.

(5 pts) a) Find the number of kmoles of O_2 and the number of kmoles of CO_2.

$$m_{O_2} = \frac{PV}{RT} = \frac{300(1)}{(.2598)(293)} = 3.941 \text{ kg}$$

$$m_{CO_2} = \frac{PV}{RT} = \frac{250(2)}{.1889(323)} = 4.097 \text{ kg}$$

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$$n_i = \frac{MW_i}{m_i} = \qquad O_2 : \frac{3.941 \text{ kg}}{32 \text{ kg/kmol}} \Rightarrow \boxed{.1232 \text{ Kmol } O_2}$$

$$CO_2 : \frac{4.097}{44} \Rightarrow \boxed{.0931 \text{ kmol } CO_2}$$

(5 pts) b) Find the average molecular weight of the final mixture.

$$MW = \Sigma y_i \, mw_i = .5696(32) + .4304(44) = \boxed{37.16 \text{ kg/kmol}}$$

$$y_{O_2} = \frac{.1232}{(.1232 + .0931)} = .5696$$

$$y_{CO_2} = \frac{.0931}{(\qquad)} = .4304$$

OK 5

(5 pts) c) Find the mass fractions of O_2 and CO_2 in the final mixture.

$$mf_{O_2} = \frac{3.941}{(3.941 + 4.097)} = \boxed{.4903}$$

$$mf_{CO_2} = \frac{4.097}{(\qquad)} = \boxed{.5097}$$

OK 5

(10 pts) d) Find the final mixture temperature.

$$\Delta E = 0$$

$$m_{O_2} u_{O_2,1} + m_{CO_2} u_{CO_2,1} = m_{O_2} u_{O_2,2} + m_{CO_2} u_{CO_2,2}$$

$$3.941 \text{ kg} (.658)(T_2 - 293) + 4.097(.657)(T_2 - 323) = 0$$

$$\Rightarrow T_2 = 308.3 \text{ K} \qquad \boxed{T_2 = 35.3\,^\circ\text{C}}$$

OK ~~8~~ 10

(5 pts) e) Find the final pressure (kPa).

$$P_2 = \Sigma P_{i_2} = \left(\frac{mRT}{V}\right)_{O_2} + \left(\frac{mRT}{V}\right)_{CO_2} = \left(3.941(.2598) + 4.097(.1889)\right)\left(\frac{308.3}{3 \text{ m}^3}\right)$$

$$\boxed{P_2 = 184.8 \text{ kPa}}$$

OK 5

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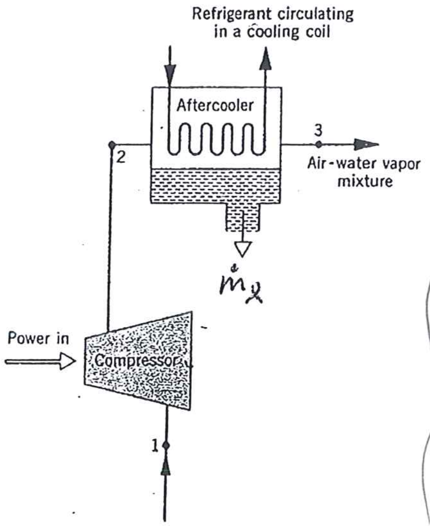
(50 pts) 3. The figure below shows a compressor followed by an aftercooler. Atmospheric air at 100 kPa, 30°C and a relative humidity of 75% enters the compressor with a volumetric flow rate of $\boxed{0.0472\ \text{m}^3/\text{s}\ \textbf{of dry air}}$. The moist air exiting the compressor at 680 kPa, 200°C flows through the aftercooler, where it is cooled at constant pressure, exiting at 40°C. Condensate also exits the aftercooler at 40 °C. Consider the compressor to be adiabatic.



$T_3 = 40°C$
$P_3 = P_2$

$T_{condensate} = 40°C$

$T_2 = 200°C$
$P_2 = 680\ \text{kPa}$

a)

$T_1 = 30°C$
$P_1 = 100\text{kPa}$
$\phi_1 = 75\%$

$\omega_2 = \omega_1 = .622\left(\frac{3.185}{100-3.185}\right) = \boxed{.02046}$ 10

(10 pts) a) Find the humidity ratio after the compressor, ω_2

$\omega = .622\left(\frac{P_v}{P-P_v}\right)$ $\phi_1 = .75$ $P_g < 30°C > = 4.247$ $P_{v_1} = (.75)(4.247) = 3.185$ kPa

$\dot{m}_{v_1} = ?$ $\frac{\dot{m}_{v_1}}{\dot{m}_a} = .622\left(\frac{3.185}{100-3.185}\right) = .02046$

$\dot{m}_{v_1} = \dot{m}_a(.02046) = .001076$ kg/s

(10 pts) b) Find the mass flow rate of dry air through the system (kg/s)

$Pv = RT$

$v = \frac{RT_i}{P_1} = \frac{.2870(303)}{(100\text{kPa}-3.185)}$ $.8982\ \text{m}^3/\text{kg}$

$\dot{m}_a = \left(\frac{.0472\ \text{m}^3/\text{s}}{.8982\ \text{m}^3/\text{kg}}\right) = \boxed{.0526\ \text{kg/s dry air}}$ 10

(10 pts) c) Find the compressor work required (kW)

~~$Q = (\Delta u + W) = h_2 - h_1$~~ $= \dot{m}_a c_{pa}(T_2 - T_1) + \dot{m}_v(h_2 - h_1)$

~~${}_1W_2 = {}_1Q_2 - \Delta u$~~ $= .0526(1.005)(200-30) + (.001076)(2792-2556)$

$\dot{Q} - \dot{W} = \dot{m}(h_2 - h_1)$ $\boxed{\dot{W} = 9.241\ \text{kJ/s}}$ (into system)

~~$\dot{m}(u_2 - u_1) = .0526(.718)(170) + .02046(2594 - 2416)$~~

~~$= 10.06\ \text{kJ/s}$~~

10

30

(10 pts) d) Find the mass flow of liquid water condensate out of the aftercooler (g/s).

$P_{v3} = P_{sat}(40°C) = 7.385$

$\dot{m}_l = \dot{m}_{v2} - \dot{m}_{v3}$

$\dot{m}_{v2} = \dot{m}_{v1} = .001076 \ kg/s$

$\dot{m}_{v3} = \omega_3 \dot{m}_a = (.0526)\frac{kg}{s}(.622)\left(\frac{7.385}{680-7.385}\right) = .000359 \ kg/s$

$\dot{m}_l = .001076 - .000359 = \boxed{.000717 \ kg/s}$

10

(10 pts) e) Find the rate of heat transfer from the moist air to the refrigerant in the aftercooler (kW).

$Q - \cancel{W}^{0} = \dot{m}_a (c_p)(T_3 - T_2) + \dot{m}_{v_3} h_{v_3} + \dot{m}_l h_l - \dot{m}_{v_2} h_{v_2}$

$= (.0526)(1.005)(40-200) + (.000359)(2574) + 167.53(.000717) - (.001076)(2792)$

$\boxed{\dot{Q} = -10.42 \ \frac{kJ}{s}}$ or kW out of moist air

10

20

TABLE A – 1

Molar mass, gas constant, and critical-point properties

Substance	Formula	Molar mass, M kg/kmol	Gas constant, R kJ/kg · K*	Critical-point properties		
				Temperature, K	Pressure, MPa	Volume, m³/kmol
Air	—	28.97	0.2870	132.5	3.77	0.0883
Ammonia	NH_3	17.03	0.4882	405.5	11.28	0.0724
Argon	Ar	39.948	0.2081	151	4.86	0.0749
Benzene	C_6H_6	78.115	0.1064	562	4.92	0.2603
Bromine	Br_2	159.808	0.0520	584	10.34	0.1355
n-Butane	C_4H_{10}	58.124	0.1430	425.2	3.80	0.2547
Carbon dioxide	CO_2	44.01	0.1889	304.2	7.39	0.0943
Carbon monoxide	CO	28.011	0.2968	133	3.50	0.0930
Carbon tetrachloride	CCl_4	153.82	0.05405	556.4	4.56	0.2759
Chlorine	Cl_2	70.906	0.1173	417	7.71	0.1242
Chloroform	$CHCl_3$	119.38	0.06964	536.6	5.47	0.2403
Dichlorodifluoromethane (R-12)	CCl_2F_2	120.91	0.06876	384.7	4.01	0.2179
Dichlorofluoromethane (R-21)	$CHCl_2F$	102.92	0.08078	451.7	5.17	0.1973
Ethane	C_2H_6	30.070	0.2765	305.5	4.48	0.1480
Ethyl alcohol	C_2H_5OH	46.07	0.1805	516	6.38	0.1673
Ethylene	C_2H_4	28.054	0.2964	282.4	5.12	0.1242
Helium	He	4.003	2.0769	5.3	0.23	0.0578
n-Hexane	C_6H_{14}	86.179	0.09647	507.9	3.03	0.3677
Hydrogen (normal)	H_2	2.016	4.1240	33.3	1.30	0.0649
Krypton	Kr	83.80	0.09921	209.4	5.50	0.0924
Methane	CH_4	16.043	0.5182	191.1	4.64	0.0993
Methyl alcohol	CH_3OH	32.042	0.2595	513.2	7.95	0.1180
Methyl chloride	CH_3Cl	50.488	0.1647	416.3	6.68	0.1430
Neon	Ne	20.183	0.4119	44.5	2.73	0.0417
Nitrogen	N_2	28.013	0.2968	126.2	3.39	0.0899
Nitrous oxide	N_2O	44.013	0.1889	309.7	7.27	0.0961
Oxygen	O_2	31.999	0.2598	154.8	5.08	0.0780
Propane	C_3H_8	44.097	0.1885	370	4.26	0.1998
Propylene	C_3H_6	42.081	0.1976	365	4.62	0.1810
Sulfur dioxide	SO_2	64.063	0.1298	430.7	7.88	0.1217
Tetrafluoroethane (R-134a)	CF_3CH_2F	102.03	0.08149	374.2	4.059	0.1993
Trichlorofluoromethane (R-11)	CCl_3F	137.37	0.06052	471.2	4.38	0.2478
Water	H_2O	18.015	0.4615	647.1	22.06	0.0560
Xenon	Xe	131.30	0.06332	289.8	5.88	0.1186

*The unit kJ/kg · K is equivalent to kPa · m³/kg · K. The gas constant is calculated from $R = R_u/M$, where R_u = 8.31447 kJ/kmol · K and M is the molar mass.

Source: K. A. Kobe and R. E. Lynn, Jr., *Chemical Review* 52 (1953), pp. 117–236; and ASHRAE, *Handbook of Fundamentals* (Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1993), pp. 16.4 and 36.1.

TABLE A–2

Ideal-gas specific heats of various common gases

(a) At 300 K

Gas	Formula	Gas constant, R kJ/kg · K	c_p kJ/kg · K	c_v kJ/kg · K	k
Air	—	0.2870	1.005	0.718	1.400
Argon	Ar	0.2081	0.5203	0.3122	1.667
Butane	C_4H_{10}	0.1433	1.7164	1.5734	1.091
Carbon dioxide	CO_2	0.1889	0.846	0.657	1.289
Carbon monoxide	CO	0.2968	1.040	0.744	1.400
Ethane	C_2H_6	0.2765	1.7662	1.4897	1.186
Ethylene	C_2H_4	0.2964	1.5482	1.2518	1.237
Helium	He	2.0769	5.1926	3.1156	1.667
Hydrogen	H_2	4.1240	14.307	10.183	1.405
Methane	CH_4	0.5182	2.2537	1.7354	1.299
Neon	Ne	0.4119	1.0299	0.6179	1.667
Nitrogen	N_2	0.2968	1.039	0.743	1.400
Octane	C_8H_{18}	0.0729	1.7113	1.6385	1.044
Oxygen	O_2	0.2598	0.918	0.658	1.395
Propane	C_3H_8	0.1885	1.6794	1.4909	1.126
Steam	H_2O	0.4615	1.8723	1.4108	1.327

Note: The unit kJ/kg · K is equivalent to kJ/kg · °C.

Source: Chemical and Process Thermodynamics 3/E by Kyle, B. G., © 2000. Adapted by permission of Pearson Education, Inc., Upper Saddle River, NJ.

1a) $P_v = \phi P_g$; $P_g = 2.3392\ kPa$ AT 20°C

$P_v = (0.75)(2.3392\ kPa) = 1.754\ kPa$; $P_a = 100 - 1.754 = 98.25\ kPa$

b) $\omega = 0.622 \frac{P_v}{P-P_v} = 0.622\left(\frac{1.754}{100-1.754}\right) = 0.0111$ [illegible]

c) $m_v = \frac{P_v \forall}{R_v T} = \frac{(1.754\ kPa)(5 \times 5 \times 3)}{(0.4615\ kJ/kgK)(293\ K)} = 0.9729\ kg$

$m_a = \frac{98.25(5 \times 5 \times 3)}{(0.287)(293)} = 87.62\ kg$

d) $T_{DP} = T$ WHERE $P_v = P_g$; $P_v = 1.754\ kPa$

$\Rightarrow \frac{20 - T_{DP}}{20 - 15} = \frac{2.3392 - 1.754}{2.3392 - 1.7057} \Rightarrow T_{DP} = 15.38°C$

2 a) $n_{O_2} = \frac{P_{O_2} \forall_A}{R_u T_A} = \frac{(300)(1\ m^3)}{(8.314\ kJ/kmol\ K)(293\ K)} = 0.1231\ kmol\ O_2$

$n_{CO_2} = \frac{P_{CO_2} \forall_B}{R_u T_B} = \frac{(250)(2\ m^3)}{(8.314\ kJ/kmol\ K)(323\ K)} = 0.1862\ kmol\ CO_2$

b) $MW = \sum y_i MW_i$; $y_{O_2} = \frac{0.1231}{0.1231 + 0.1862} = 0.398$; $y_{CO_2} = 0.602$

$MW = (0.398)(32\ kg/kmol) + (0.602)(44\ kg/kmol) = 39.22\ kg/kmol$

c) $mf_i = y_i \frac{MW_i}{MW}$; $mf_{O_2} = 0.398\left(\frac{32}{39.22}\right) = 0.325$

$mf_{CO_2} = 0.602\left(\frac{44}{39.22}\right) = 0.675$

d) $\cancel{Q - W} = U_2 - U_1 = n_{O_2} \bar{C}_{V O_2}(T_2 - T_{1,O_2}) + n_{CO_2} \bar{C}_{V CO_2}(T_2 - T_{1,CO_2})$

$\bar{C}_{V,O_2} = C_{V O_2} \cdot MW_{O_2} = (0.658\ kJ/kg\ K)(32\ kg/kmol) = 21.06\ kJ/kmol\ K$

$\bar{C}_{V,CO_2} = C_{V,CO_2} \cdot MW_{CO_2} = (0.657\ kJ/kg\ K)(44\ kg/kmol) = 28.91\ kJ/kmol\ K$

$0 = (0.1231\ kmol\ O_2)(21.06\ \frac{kJ}{kmol\ K})(T_2 - 293) + (0.1862)(28.91)(T_2 - 323)$

$\Rightarrow T_2 = 313.2\ K$

e) $P_f = \frac{n_T R_u T_2}{\forall} = \frac{(0.1862 + 0.1231)(8.314)(313.2)}{1\ m^3 + 2\ m^3} = 268.6\ kPa$

3)a) $\omega_2 = \omega_1 = 0.622\left(\frac{P_{v_1}}{P - P_{v_1}}\right)$; $P_{v_1} = \phi_1 P_g(T_1)$

$P_g(T_1) = 4.2469\ kPa \Rightarrow P_{v_1} = (0.75)(4.2469) = 3.185\ kPa$

$\omega_2 = 0.622\left(\frac{3.185}{100 - 3.185}\right) = 0.0205$

b) $\dot{m}_a = \rho_{a_1} \dot{\forall}_{a_1}$; $\rho_{a_1} = \frac{P_{a_1}}{R_a T_1} = \frac{P_1 - P_{v_1}}{R_a T_1}$

$\rho_{a_1} = \frac{(100 - 3.185)}{(0.287)(303)} = 1.113$

$\dot{m}_a = (1.113)(0.0472\ m^3/s) = 0.0525\ kg/s$

NAME: David Malawey

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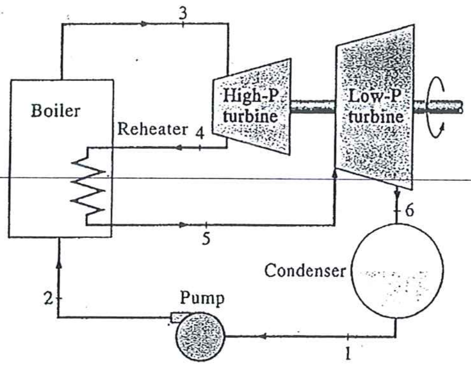
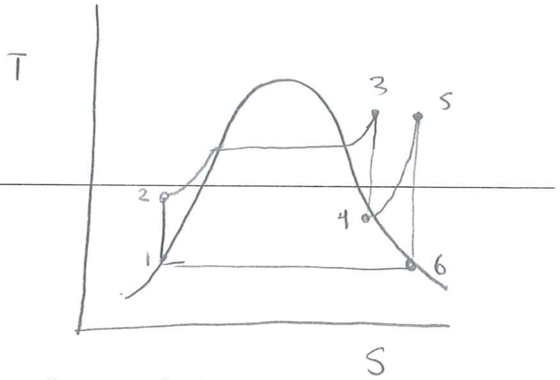
ME221
Exam II
Spring 2010
Drallmeier

(25 pts) 1. A steam power system operates using a Rankine cycle with reheat. The steam leaves the boiler at 4 MPa, 400 °C and expands isentropically to 0.4 MPa where it is reheated at constant pressure to 400 °C. The steam is further expanded to 10 kPa at which point it enters the condenser. Assume a isentropic pump and turbine and saturated liquid entering the pump. You may also assume a constant pressure boiler and condenser.





(10 pts) a) Find the quality after **both** the high pressure **and** low pressure turbines (i.e. x_4 and x_6).

1	2	3	4	5	6
sat liq P= 10kPa	P=4MPa	P=4MPa T=400C S=6.7714	P=.4MPa h=2686	P=.4MPa T=400C	P=10kPa h=2504

$x_4 = ?$ $S_4 = S_3 = 6.7714 = 1.7765 + x_4(5.1191)$ $\Rightarrow \boxed{x_4 = .9757}$

$h_4 = 604.7 + (.9757)2133 = 2686$

x_6: $S_6 = S_5 = 7.9003 = .6492 + x_6(7.4996)$ $\Rightarrow \boxed{x_6 = \boxed{.9668}}$

$h_6 = 191.81 + (.9668)2392 = 2504$

$h_3 = 3215$
$h_5 = 3274$

(5 pts) b) Find the specific pump work (kJ/kg)

$_1w_2 = h_1 - h_2 = v(P_1 - P_2)$
$= .001010(10 - 4000)$

$$\boxed{W_p = -4.030 \text{ kJ/kg}}$$

15

c)

(10 pts) ~~d)~~ Find the net specific work from **the system** (kJ/kg)

$$w_{net} = {}_1w_2 + {}_3w_4 + {}_5w_6$$

$${}_3w_4 = h_3 - h_4$$

$${}_5w_6 = h_5 - h_6$$

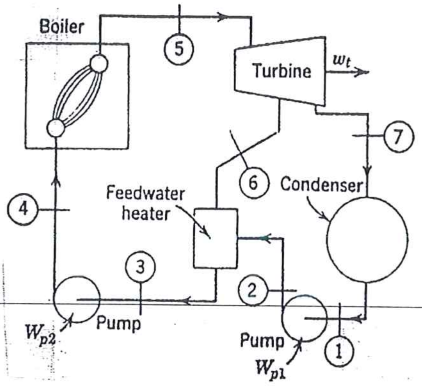
$$w_{net} = \overset{\underline{12}}{-4.03} + \overset{3}{3215} - \overset{4}{2686} + \overset{5}{3274} - \overset{6}{2504}$$

$$\boxed{w_{net} = 1295 \text{ kJ/kg}}$$

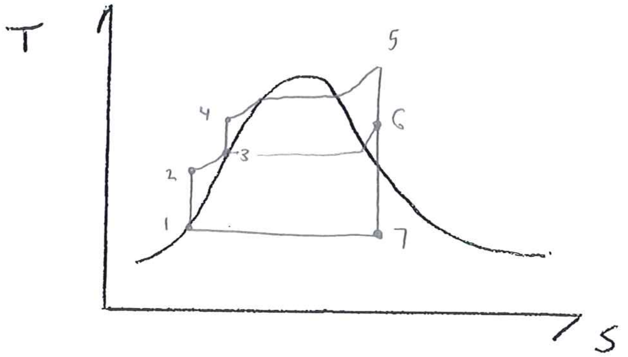
10

(30 pts) 2. A power plant operates on a Rankine cycle with a boiler output of 20 kg/s of steam at 2 MPa. The condenser operates at a temperature of 50 °C. As shown below there is one open feed water heater with extraction steam from the turbine (i.e. state 6) at 600 kPa and 400 °C. The exit of the FWH, state 3, is saturated. Assume an adiabatic feedwater heater and the specific work into pump 1 (between states 1 and 2) is 0.595 kJ/kg. Also assume a saturated liquid leaving the condenser (i.e. $x_1 = 0$).

3



(5 pts) a) **Sketch the T-S diagram**, labeling the states. Points will be given based on the clarity of your sketch.



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(15 pts) b) Find the fraction of steam extracted for the FWH relative to the steam in the boiler (i.e. $\frac{\dot{m}_6}{\dot{m}_5}$).

1 (sat)	2	3	4	5	6 (SHV)	7
P = 12.352 T = 50°C S = .7038 h = 209.3	P = 600kPa S = .7038 h = 209.9	sat liq. P = 600kPa h = 670.4		P = 2MPa	P = 600kPa T = 400°C h = 3271 s = 7.710	T = 50°C s = 7.710 h = 2490

$h_2 = h_1 - {}_1w_2$

${}_1w_2 = .001012(12.35 - 600) = -.5947$

$h_2 = 209.9$

15

$$\frac{\dot{m}_6}{\dot{m}_5} = \frac{670.4 - 209.9}{3271 - 209.9}$$

$$\boxed{\frac{\dot{m}_6}{\dot{m}_5} = .1504}$$

20

(10 pts) c) The condenser dumps energy into a river. If the river water should not be heated more than 5 °C, how much water should be pumped from the river to cool the condenser? (kg/s) (Note that the specific heat of liquid water is roughly 4.18 kJ/kgK).

$q_{out} = {}_7q_1 = h_1 - h_7 = 209.3 - 2490$

$= -2281$ kJ/kg

3

$\dot{m}_7 = ?$

$5°C \Rightarrow (5\,K)\,c_p = \Delta h\, \dot{m}_7$

5

$\wedge$

$\dot{m}_R$

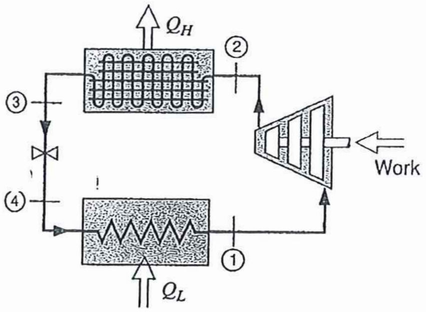
$\left(\frac{5\,kg\,K}{s}\right)\left(\frac{4\,kJ}{kg\,K}\right)\left(2281\ kJ/kg\right) = 2728\ kg/s$

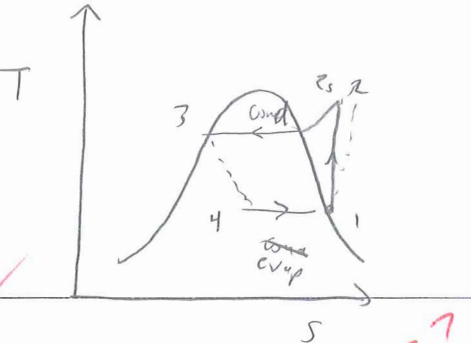
$\uparrow$

4.18

8

(15 pts) 3. Refrigerant 134a is used in a non-ideal vapor compression refrigeration cycle. The refrigerant enters the compressor at 100 kPa, -10 °C and leaves at 1.2 MPa, and 100 °C. Note that the compressor is non-isentropic compressor but can be assumed adiabatic. The refrigerant leaves the condenser at 1.19 MPa and 44 °C. The expansion valve can be assumed to be adiabatic.





(5 pts) a) Find the specific compressor work. (kJ/kg)

$${}_1w_2 = h_1 - h_2 = 247.49 - 332.7 = \underline{-115.21}$$

MATH? UNITS?

4

(5 pts) b) Find the specific heat into the evaporator. You may need to make an approximation with the tables given the measured states. Clear state that approximation. (kJ/kg)

$${}_4q_1 = h_1 - h_4 \quad (h_4 = h_3)$$

$$h_1 - h_3 \quad \text{approx } h_3 \text{ @ } h < 1.2\text{ MPa}, \overset{50°C}{\cancel{40°C}})$$

$$h_3 = 278.3$$

$${}_4q_1 = 247.49 - 278.3$$

$$= -30.81$$ ← this should actually be heat in

3

(5 pts) c) Find the coefficient of performance

$$COP_R = \frac{q_L}{W} = \frac{-30.81}{-115.21} = \boxed{.2674} \leftarrow ???$$

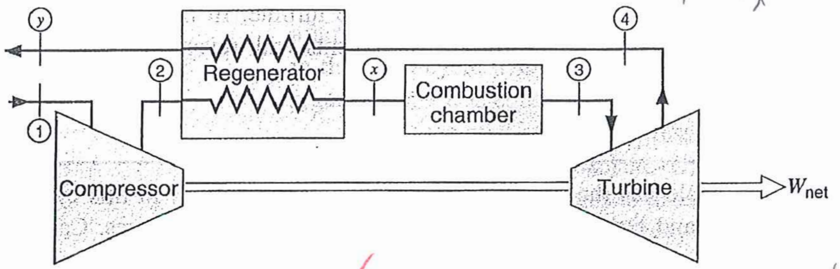
OK

5

12

(30 pts) 4. Air enters the compressor of an ideal air-standard Brayton cycle at 100 kPa and 288 K. The compressor pressure ratio is 10. The turbine inlet temperature is 1373 K. Assume isentropic compressor and turbine, the regenerator is ideal and that $P_1 = P_4 = P_y$. Assume constant specific heats (i.e. $C_v = 0.717$ kJ/kgK, $C_p = 1.004$ kJ/kgK, $k = 1.4$).

$T_4 = T_x$



(5 pts) a) Find the temperature after the compressor (K)

$P_1 = 100\text{kPa}$

$T_1 = 288\text{K}$

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

(10)

✓ ✓ ✓

4

(5 pts) b) Find the specific compressor work (kJ/kg)

$${}_1w_2 = \cancel{h_2 - h_1}\ h_1 - h_2 = C_p(T_1 - T_2)$$

✓ ✓

4

(5 pts) c) Find the temperature after the turbine (K) $T_4 = ? = T_x$

$P_4 = $ ✓

$P_3 = P_2$ ✓ $T_3 = $ given 1 ( ) isentropic relation gives T_4 ✓

WHICH?

+3

(5 pts) d) Find the **net** specific work from the system (kJ/kg)

$$W = {}_1W_2 + {}_3W_4 = C_p(T_1 - T_2 + T_3 - T_4)$$

✓ ✓ ✓ ✓

4

$T_3 = ?$

(10 pts) e) Find the thermal efficiency of the system

$$\eta_{th} = \frac{W_{\text{net}}}{q_{in}} = \frac{✓}{C_p(T_3 - T_x)}$$

✓ ↑ T_4 ✓

7

20

~~19~~

7) David Malawey

#2 b) $\frac{\dot m_6}{\dot m_5}$:

$$\dot m_6 h_6 + \dot m_2 h_2 = \dot m_3 h_3$$

$$\dot m_6 h_6 + (\dot m_5 - \dot m_6) h_2 = \dot m_5 h_3$$

$$\dot m_6 h_6 + \dot m_5 h_2 - \dot m_6 h_2 = \dot m_5 h_3$$

$$\dot m_6 (h_6 - h_2) = \dot m_5 (h_3 - h_2)$$

$$\frac{\dot m_6}{\dot m_5} = \frac{(h_3 - h_2)}{(h_6 - h_2)}$$

$s_7 = 7.710 = .7038 + x_7\, 7.371 \Rightarrow x_7 = .9575$

$h_7 = 209.3 + x_7 (2382) \Rightarrow h_7 = 2490$

#3

1 ~~Sat~~ SHV	2	3	4
100kPa	P=1.2MPa	P=1.19 MPa	P=100kPa
-10°C	T=100 C	T=44°C	T= -10°C
h= 247.49	h= 3327		

2.5

1.2 MPa

1a) STATE 3 : $P_3 = 4\,MPA, T_3 = 400\,^\circ C \Rightarrow h_3 = 3214.5\,^{kJ}/_{kg}\ ;\ s_3 = 6.7714\,^{kJ}/$

STATE 4 : $P_4 = 0.4\,MPA,\ s_4 = s_3 \Rightarrow s_4 = s_f + x_4(s_{fg})$

$1.7765 + x_4(5.1191) = 6.7714 \Rightarrow x_4 = 0.9757$ ◁

STATE 5 : $P_5 = 0.4\,MPA,\ T_5 = 400\,^\circ C \Rightarrow h_5 = 3273.9\ ;\ s_5 = 7.9003$

STATE 6 : $P_6 = 10\,kPA,\ s_6 = s_5 \Rightarrow s_6 = s_f + x_6 s_{fg}$

$7.9003 = 0.6492 + x_6(7.4996) \Rightarrow x_6 = 0.9669$ ◁

b) ${}_1w_2 = v_1(P_1 - P_2)\ ;$ STATE 1 : $x_1 = 0,\ P_1 = 10\,kPA$

$\Rightarrow v_1 = 0.001010\ ^{m^3}/_{kg}$

${}_1w_2 = 0.001010\ ^{m^3}/_{kg}\,(10 - 4000) = -4.03\ ^{kJ}/_{kg}$ ◁

c) $w_{NET} = w_{HP} + w_{LP} + w_P$

$w_{HP} = {}_3w_4 = h_3 - h_4\ ;\ h_4 = h_f + x_4 h_{fg}$

$h_4 = 604.66 + (0.9757)(2133.4) = 2686.2$

$w_{HP} = 3214.5 - 2686.2 = 528.3\ ^{kJ}/_{kg}$

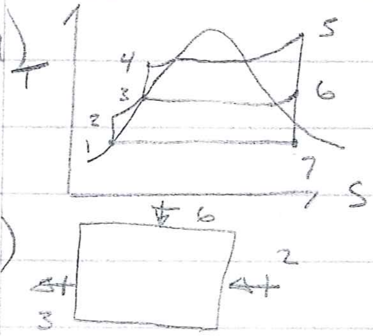
$w_{LP} = {}_5w_6 = h_5 - h_6$

$h_6 = h_f + x_6 h_{fg} = 191.81 + 0.9669(2392.1) = 2504.7$

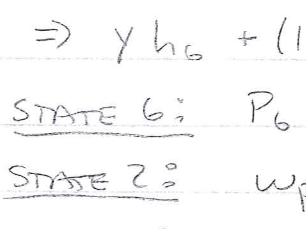
$w_{LP} = 3273.9 - 2504.7 = 769.2\ ^{kJ}/_{kg}$

$w_{NET} = 528.3 + 769.2 - 4.03 = 1293.5\ ^{kJ}/_{kg}$ ◁

2a)



b)



$\dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3\ ;\ \dot{m}_5 = \dot{m}_3$

$\frac{\dot{m}_6}{\dot{m}_5} h_6 + \frac{\dot{m}_2}{\dot{m}_5} h_2 = h_3\ ;\ \dot{m}_2 = \dot{m}_3 - \dot{m}_6$

$\Rightarrow y h_6 + (1-y) h_2 = h_3 \Rightarrow y = \frac{h_3 - h_2}{h_6 - h_2}$

<u>STATE 6</u>: $P_6 = 600\,kPA,\ T_6 = 400\,^\circ C \Rightarrow h_6 = 3270.8\ ^{kJ}/_{kg}$

<u>STATE 2</u>: $w_P = h_1 - h_2 = -0.595\ ^{kJ}/_{kg}$

<u>STATE 1</u>: $x_1 = 0,\ T_1 = 50\,^\circ C \Rightarrow h_1 = 209.34\ ^{kJ}/_{kg}$

$\Rightarrow h_2 = 209.34 + 0.595 = 209.9\ ^{kJ}/_{kg}$

STATE 3: $x_3 = 0$, $P_3 = P_6 = 600\,kPa \Rightarrow h_3 = 670.38\ kJ/kg$

$y = \frac{670.38 - 209.9}{3270.8 - 209.9} = 0.1504$ ◁

b)

$\dot{Q}_L = \dot{m}_7 (h_7 - h_1) = \dot{m}_R C_{H_2O} \Delta T$

$\dot{m}_7 = (1-y)\dot{m}_5 = (1 - 0.1504)(20\ kg/s) = 16.99\ kg/s$

STATE 7: $T_7 = 50°C$, $S_7 = S_6 = 7.7097 = S_f + x_7 S_{fg}$

$7.7097 = 0.7038 + x_7 (7.3710) \Rightarrow x_7 = 0.9505 \Rightarrow h_7 = 2473$

$\Rightarrow (16.99\ kg/s)(209.34 - 2473) = \dot{m}_R (4.18\ kJ/kg\,K)(5°)$

$\dot{m}_R = 1840\ kg/s$ ◁

3.) a) $w_c = h_1 - h_2$ STATE 1: $P_1 = 100\,kPa$, $T_1 = -10°C \Rightarrow h_1 = 247.$

STATE 2: $P_2 = 1.2\,MPa$, $T_2 = 100°C \Rightarrow h_2 = 332.73$

$w_c = 247.4 - 332.73 = -85.24\ kJ/kg$ △

b) $q_L = h_1 - h_4$; $h_3 = h_4$; STATE 3: $P_3 = 1.19\,MPa$, $T_3 = 44$

$\Rightarrow$ SUBCOOLED $\Rightarrow h_3 \cong h_g(T_3) = 114.28$

$\Rightarrow q_L = 247.49 - 114.28 = 133.21\ kJ/kg$

c) $\beta = \frac{q_L}{w_c} = \frac{133.21}{85.24} = 1.567$

4) a) $T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{k-1/k} = 288(10)^{0.4/1.4} = 556K$.

b) $w_c = C_p (T_1 - T_2) = 1.004(288 - 556) = -269.1\ kJ/kg$

c) $T_4 = T_3 \left(\frac{P_4}{P_3}\right)^{k-1/k} = 1373\left(\frac{1}{10}\right)^{0.4/1.4} = 711.1\ kJ/kg$

d) $w_{NET} = w_T + w_C$

$w_T = h_3 - h_4 = C_p (T_3 - T_4) = 1.004(1373 - 711.1) = 664.5\ \frac{kJ}{kg}$

$w_{NET} = 664.5 - 269.1 = 395.4\ kJ/kg$.

e) $\eta_{th} = \frac{w_{NET}}{q_{IN}}$ $q_{IN} = h_3 - h_x = C_p (T_3 - T_x)$

FOR IDEAL REGENERATOR, $T_x = T_4$

$q_{IN} = 1.004(1373 - 711.1) = 664.5\ kJ/kg = w_T$

$\eta_{th} = \frac{395.4}{664.5} = 0.595 \Rightarrow 59.5\%$

NAME: David Malawey

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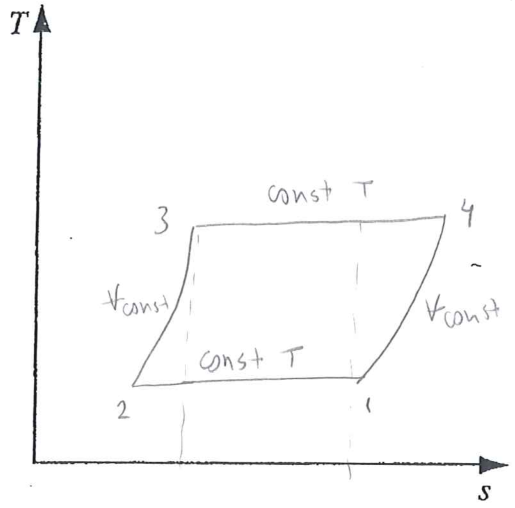
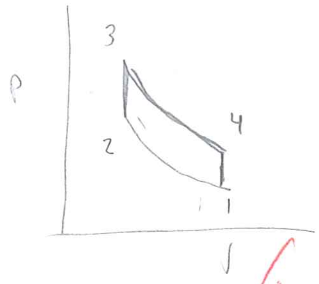
Sol'ns attached

ME221
Exam 1
Fall 2010
Drallmeier

(20 pts ttl) 1. An ideal Stirling cycle uses air as the working fluid. The isothermal compression brings air from 100 kPa, 25° C to 600 kPa. The expansion takes place at 1200 K. For the calculations below **assume constant specific heats.**

(5 pts) a) Sketch the cycle on a T-S diagram. Give the assumptions for each processes.





(1 - 2) isothermal compression ✓

(2 - 3) const ∀ q_{in} ✓

(3 - 4) isoth. expand ✓

(4 - 1) const ∀ q_{out} ✓

① 100kPa
T = 298K

② P = 600kPa
T = 298K

③

5/5

(12 pts) b) Find the work and the heat transfer during compression. (kJ/kg)

$${}_1q_2 - {}_1w_2 = u_2 - u_1 = 0$$

$${}_1q_2 = h_2 - h_1 = c_p(T_2 - T_1) =$$

$T_2 = T_1$

$${}_1q_2 = {}_1w_2 = \int P dV$$

$$= P_{avg} \ln\left(\frac{V_2}{V_1}\right) \qquad v_2 = \frac{RT_2}{P_2} \quad \ldots$$

$RT \ln\left(\frac{v_2}{v_1}\right)$

5/

(3 pts) c) When does the efficiency of an ideal Stirling cycle equal that of the Carnot cycle?

when the regenerator is ~~100%~~ efficient

3/

13

(15 pts) 2) Heated, high pressure air is the result of an industrial process. It is proposed to use this air to drive a steady flow turbine to extract power. This air, with a mass flow rate of 2kg/s, enters the turbine at 1200 K, 3000 kPa and exits at 800 K, 100 kPa. The ambient temperature is 298 K. What is the maximum power that could be extracted out of this turbine? (kW). Assume ideal gas behavior but **do not assume constant specific heats** (i.e. use variable specific heats). Note: $R_{air} = 0.287$ kJ/kgK

$\dot{m}_1 = 2kg/s$ ① 1200K, 3000 kPa ② T=800K, P=100kPa amb. T=298K

$P_{max} = \dot{W}_{max} = \dot{W}_{rev}$

$(\psi_1 - \psi_2)$ NOT $(\psi_2 - \psi_1)$

$$\dot{m}\psi = \dot{m}\left[h_2 - h_1 - T_0(s_2 - s_1)\right] + \cancel{\tfrac{1}{2}v^2}^{\,0}$$

$$s_2 - s_1 = s^\circ_{T_2} - s^\circ_{T_1} - R\ln\left(\frac{P_2}{P_1}\right)$$

$$= 2.718 - 3.179 - .287\left(\frac{100}{3000}\right) = -.4706$$

~~MATH~~

12

$$\dot{m}\psi = \frac{2kg}{s}\left[821.95 - 1278 - 298(-.4706)\right]$$

$$\dot{W}_{max} = -631.6\ kJ/s = \boxed{\dot{W}_{max} = -631.6\ kW}$$

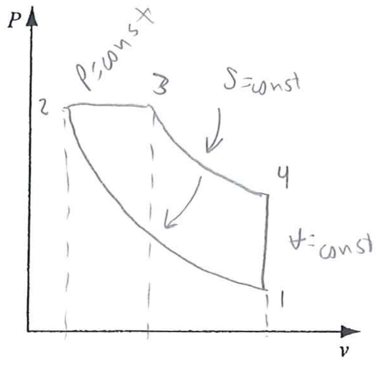
12

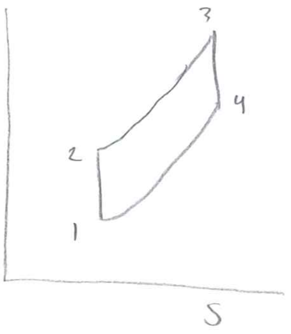
1

(35 pts ttl) 3.) An engine is to be modeled using a diesel cycle with air as the working fluid. At the beginning of compression, T = 300 K, P = 200 kPa. After combustion (heat addition) is complete the temperature and pressure are determined to be T = 2000 K and P = 7.0 MPa, respectively. Assume ideal gas behavior and **constant specific heats.**

3

(5 pts) a) Sketch the P-V diagram for this cycle. Give the assumptions for each process.





① T = 300

(1-2) isentropic compression

(2-3) const P q in

(3-4) isentropic exp.

(4,-1) const V q out

5

(5 pts) b) Find the compression ratio, r.

$r = \frac{V_1}{V_2}$

$P_2 = P_3$

$r = \frac{V_1}{V_2} = \frac{RT_1}{P_1}\left(\frac{P_2}{RT_2}\right)$

$\frac{V_1}{V_2} = \frac{T_1}{T_2}\frac{P_3}{P_1}$

2

(1-2) isent. q in $\overset{0}{\cancel{q}} - {}_1w_2 = c_v(T_2 - T_1)$

${}_1w_2 = c_v(T_1 - T_2)$

(2-3) ${}_2q_3 - {}_2w_3 = c_v(T_3 - T_2)$

${}_2q_3 = c_p(T_3 - T_2)$

$V_3 = \frac{RT_3}{P_3} = .082$

$P_3V_3 = RT_3$

$P_2 = \frac{RT_3}{V_3} = .287\left(\frac{2000}{.082}\right)$

(10 pts) c) Find the temperature after expansion. (K)

$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1}$ 3

↑

$\frac{V_3}{V_4} = \frac{V_3}{V_2}\cdot\frac{V_2}{V_4}$ where $\frac{V_2}{V_4} = \frac{1}{r}$ 2

12

(5 pts) d) Find the energy input into the cycle (kJ/kg)

$q_{in} = {}_2q_3 \longrightarrow {}_2q_3 - {}_2W_3 = U_3 - U_2$ ✓

~~$= C_v(T_3 - T_2)$~~ $\downarrow$

${}_2q_3 = h_3 - h_2$ ✓

${}_2q_3 = C_p(T_3 - T_2)$ ✓

4

(10 pts) e) Find the thermal efficiency

$$= \frac{W_{net}}{q_{in}} = \frac{\boxed{\overset{C_v \Delta T}{{}_1q_2}} + {}_2q_3 + \boxed{\overset{C_v \Delta T}{{}_3q_4}} + {}_4q_1}{{}_2q_3} ✓$$

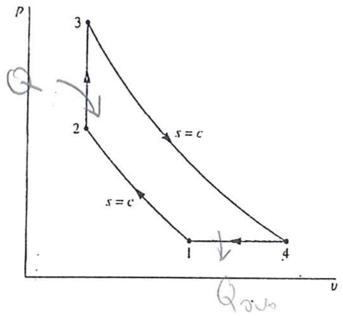
${}_1q_2 = 0$

${}_3q_4 = 0$

~~2~~3

7

(30 pts ttl) 4.) A cycle has been suggested that would extend the expansion process with hopes of gaining more work out. As shown in the P-V diagram below, the only difference of this proposed cycle with an Otto cycle is the constant pressure heat rejection process to extend the length of the expansion stroke. For this problem, assume $T_1 = 300K$ and $P_1 = 100$ kPa and the working fluid is air. The compression ratio is 9 (i.e. V_1/V_2) and the heat addition is 1500 kJ/kg. Assume a fixed mass cycle, air with ideal gas behavior and **constant specific heats**.



(5 pts) a) Find T_2. (K)

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = 300(9)^{0.4} \quad \boxed{T_2 = 722.467\ K}$$

5

(5 pts) b) Find T_3. (K)

$${}_2q_3 - \cancel{{}_2w_3}^{0} = c_v(T_3 - T_2) \qquad \frac{1500}{.718} + 722.5 = \boxed{T_3 = 2812K}$$

5

(10 pts) c) Find T_4. (K)

$$T_4 = T_3\left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} \qquad P_3 = \frac{RT_3}{v_3} \text{ where } v_3 = v_2$$

5

(10 pts) d) Find the heat rejected for the cycle (kJ/kg) (Note: if you we unable to find T_4, use 800 K) ✓

$$\cancel{{}_3q_4}^{0} - {}_3w_4 = C_v(T_4 - T_3) \qquad q_{net} = {}_2q_3 + {}_4q_1$$

$$q_{out} = {}_4q_1 = C_v(T_1 - T_4)$$

9 $\boxed{\text{heat rejected} = 521.8\ KJ/Kg}$

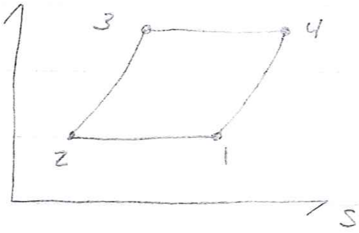
$${}_4q_1 - {}_4w_1 = C_v(T_1 - T_4)$$

$${}_4q_1 = u_1 - u_4 + {}_4w_1 \quad \overset{\text{Pconst}}{\Rightarrow} \quad {}_4q_1 = h_1 - h_4 = 300.2 - 821.95$$

CONST SPEC. HEATS

24

1 a)



(1-2) ISOTHERMAL COMPRESSION

(2-3) CONSTANT $\forall$ HEAT ADDITION

(3-4) ISOTHERMAL EXPANSION

(4-1) CONSTANT $\forall$ HEAT REJECTION

b) $${}_1q_2 - {}_1w_2 = \cancel{(u_2 - u_1)}^{\to 0,\ T_1 = T_2} \Rightarrow {}_1q_2 = {}_1w_2 = \int_1^2 P\,dv \ ;\ P = \frac{RT}{v}$$

$$\Rightarrow {}_1q_2 = {}_1w_2 = RT_1 \ln\left(\frac{v_2}{v_1}\right) \ ;\ \frac{v_2}{v_1} = \left(\frac{\cancel{R}\cancel{T_2}}{\cancel{R}\cancel{T_1}}\right)\left(\frac{P_1}{P_2}\right) = \frac{100}{600} = \frac{1}{6}$$

$${}_1q_2 = {}_1w_2 = (0.287)(298)\ln\left(\tfrac{1}{6}\right) \doteq -153.2\ \text{kJ/kg} \quad \triangleleft$$

c) WHEN AN IDEAL REGENERATOR IS USED TO CAPTURE ${}_4q_1$ AND REINTRODUCE IT AS ${}_2q_3$

2. $$\dot{W}_{MAX} = \dot{m}(\psi_1 - \psi_2) \ ;\ \psi = (h - h_0) - T_0(s - s_0)$$

$$\dot{W}_{MAX} = \dot{m}\left[(h_1 - h_2) - T_0(s_1 - s_2)\right]$$

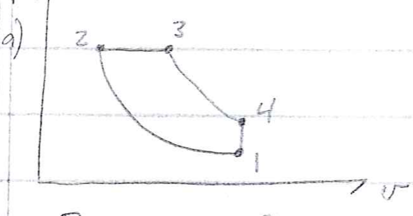
$$h_1 = h(T_1) = 1277.79 \ ;\ h_2 = h(T_2) = 821.95$$

$$s_1 - s_2 = s^\circ_{T_1} - s^\circ_{T_2} - R\ln\left(\frac{P_1}{P_2}\right) \ ;\ s^\circ_{T_2} = 2.71787 \ ;\ s^\circ_{T_1} = 3.17888$$

$$s_1 - s_2 = 3.17888 - 2.71787 - (0.287)\ln\left(\frac{3000}{100}\right) = -0.515\ \text{kJ/kg K}$$

$$\dot{W}_{MAX} = 2\ \text{kg/s}\left[(1277.79 - 821.95) - (298)(-0.515)\right] = 1219\ \text{kW} \triangleleft$$

3. a)



(1-2) ISENTROPIC COMPRESSION

(2-3) CONST. P HEAT ADDITION

(3-4) ISENTROPIC EXPANSION

(4-1) CONSTANT $\forall$ HEAT REJECTION

b) $P_1 = 200\ \text{kPa}$, $P_3 = P_2 = 7\ \text{MPa}$

$$\frac{\forall_1}{\forall_2} = \left(\frac{P_2}{P_1}\right)^{1/k} = \left(\frac{7000}{200}\right)^{1/1.4} = 12.67 = r \quad \triangleleft$$

c) $$T_4 = T_3\left(\frac{\forall_3}{\forall_4}\right)^{k-1} \ ;\ \frac{\forall_3}{\forall_4} = \frac{\forall_3}{\forall_2} \cdot \frac{\forall_2}{\forall_4} = \frac{T_3}{T_2} \cdot \frac{1}{r}$$

$$T_2 = T_1\left(\frac{\forall_1}{\forall_2}\right)^{k-1} = 300(12.67)^{0.4} = 828.5\ \text{K}$$

$$T_4 = 2000\text{K}\left\{\left(\frac{2000}{828.5}\right)\left(\frac{1}{12.67}\right)\right\}^{1.4-1} = 1030\ \text{K} \quad \triangleleft$$

d) $${}_2q_3 - {}_2w_3 = u_3 - u_2 \ ;\ {}_2w_3 = \int P\,dv = P_2(v_3 - v_2) \Rightarrow {}_2q_3 = h_3 - h_2$$

$${}_2q_3 = C_p(T_3 - T_2) = \left(1.005\ \frac{\text{kJ}}{\text{kg K}}\right)(2000 - 828.5) = 1177\ \text{kJ/kg} \quad \triangleleft$$

e) $$\eta_{th} = \frac{w_{NET}}{q_{IN}} = \frac{q_{NET}}{q_{IN}} = \frac{{}_2q_3 + {}_4q_1}{{}_2q_3} \ ;\ {}_4q_1 = C_v(T_1 - T_4)$$

$${}_4q_1 = (0.718\ \text{kJ/kg K})(300 - 1030) = -524.1\ \text{kJ/kg}$$

$$\Rightarrow \eta_{th} = \frac{1177 - 524.1}{1177} = 0.55 \Rightarrow 55\% \quad \triangleleft$$

4) a) $T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = 300(9)^{0.4} = 722.5K$

b) ${}_2q_3 - \cancel{{}_2w_3}^{\,0} = u_3 - u_2 \Rightarrow {}_2q_3 = C_v(T_3 - T_2)$

$1500\ kJ/kg = (0.718\ kJ/kgK)(T_3 - 722.5) \Rightarrow T_3 = 2811.6K$

c) $T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{P_4}{P_3}\right)^{k-1/k}$; $P_4 = P_1 = 100kPA$

$P_3 = P_2\left(\frac{T_3}{T_2}\right)$ (CONST V, $V_2 = V_3$)

$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{k} = 100kPA(9)^{1.4} = 2167kPA$

$\Rightarrow P_3 = (2167\ kPa)\left(\frac{2811.6}{722.5}\right) = 8434\ kPA$

$\Rightarrow T_4 = 2811.6k\left(\frac{100}{8434}\right)^{0.4/1.4} = 791.9K$

d) ${}_4q_1 - {}_4w_1 = u_1 - u_4$; ${}_4w_1 = \int_4^1 P\,dv = P_1(v_1 - v_4)$

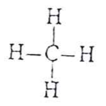
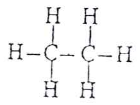
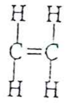
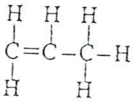
$\Rightarrow {}_4q_1 = h_1 - h_4 = C_p(T_1 - T_4) = (1.005)(300 - 791.9)$

${}_4q_1 = -494.3\ kJ/kg$

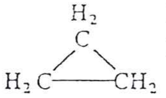
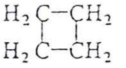
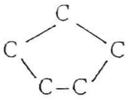
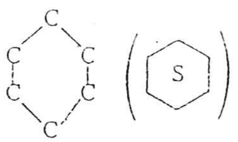
Handouts

Table 2.1 Structure and general properties of lighter members of hydrocarbon series

ALIPHATIC or ACYCLIC

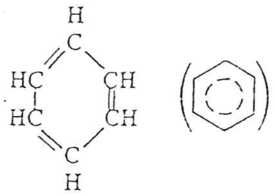
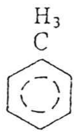
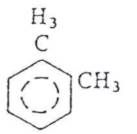
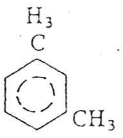
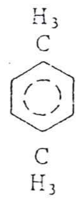
Carbon number	1	2	3	4	5	6
Paraffins (alkanes) C_nH_{2n+2}	 $\mathrm{H}-\overset{\mathrm{H}}{\underset{\mathrm{H}}{\mathrm{C}}}-\mathrm{H}$ methane	 $\mathrm{H}-\overset{\mathrm{H}}{\underset{\mathrm{H}}{\mathrm{C}}}-\overset{\mathrm{H}}{\underset{\mathrm{H}}{\mathrm{C}}}-\mathrm{H}$ ethane	C–C–C propane	C–C–C–C *n*-butane	C–C–C–C–C *n*-pentane	C–C–C–C–C–C → etc. *n*-hexane
	Saturated (single C–C bonds); hence stable in storage. High H content; hence high calorific value. Poor spark-knock resistance unless transformed to branched-chain isomers.					
Olefins (alkenes) C_nH_{2n}	–	 $\overset{\mathrm{H}}{\underset{\mathrm{H}}{\mathrm{C}}}=\overset{\mathrm{H}}{\underset{\mathrm{H}}{\mathrm{C}}}$ ethene (ethylene)	 $\overset{\mathrm{H}}{\underset{\mathrm{H}}{\mathrm{C}}}=\overset{\mathrm{H}}{\mathrm{C}}-\overset{\mathrm{H}}{\underset{\mathrm{H}}{\mathrm{C}}}-\mathrm{H}$ propene	C=C–C–C but-1-ene or → etc. C–C=C–C but-2-ene		
	Unsaturated (C=C bonds) hence unstable ('oil-forming'). Not naturally occurring; formed during refining.					
Alkynes (acetylenes) C_nH_{2n-2}	–	HC≡CH acetylene (ethyne)	HC≡C–CH propyne	C≡C–C–C but-1-yne or → etc. C–C≡C–C but-2-yne		
	Unsaturated (triple C≡C bonds) hence highly unstable. Stored in acetone solution. High combustion temperature and flame speed.					

ALICYCLIC

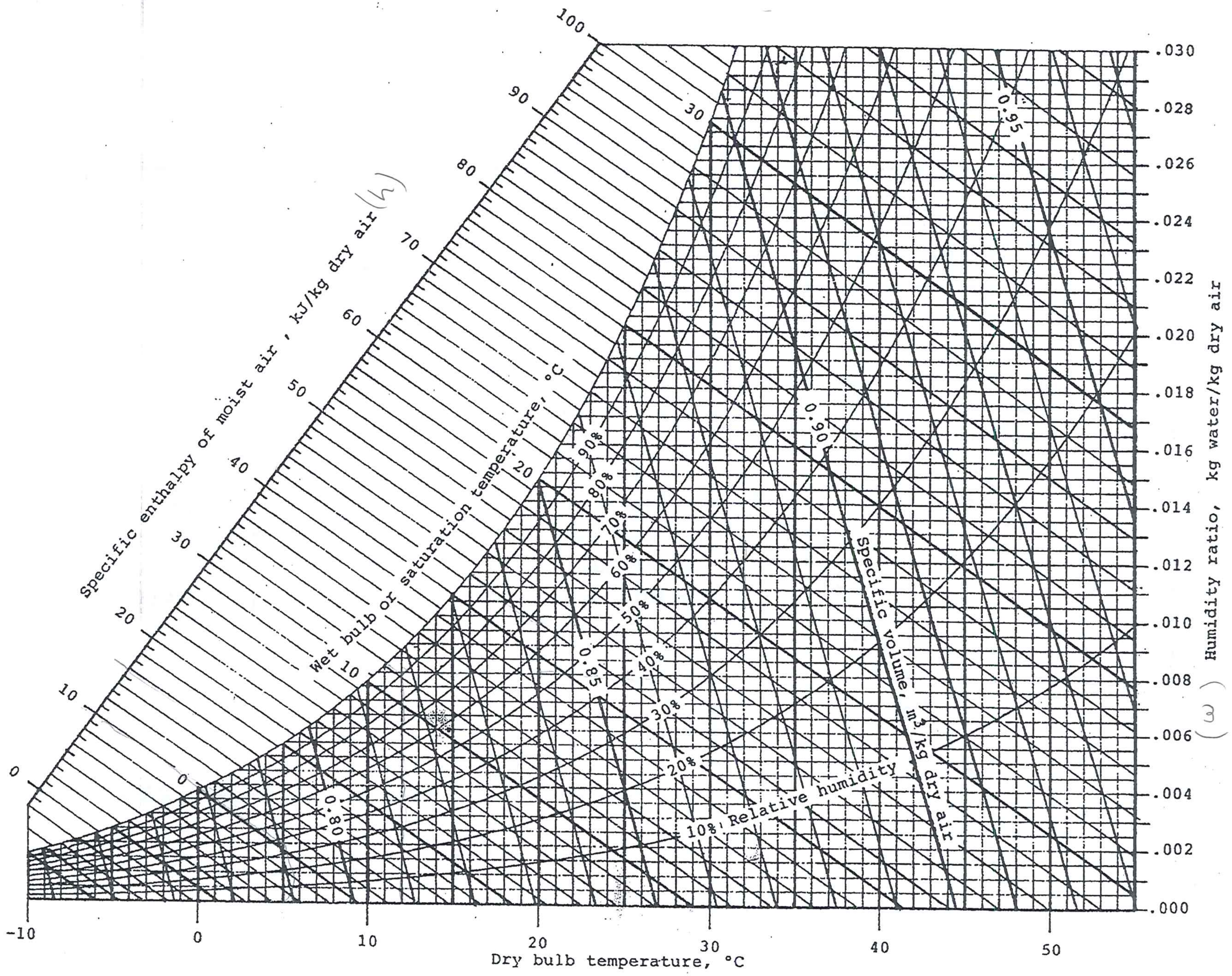
	1	2	3	4	5	6
Naphthenes (cyclanes) C_nH_{2n}	–	–	 cyclopropane	 cyclobutane	 cyclopentane	 cyclohexane
	Saturated (hence stable), but some ring strain in first two members.					

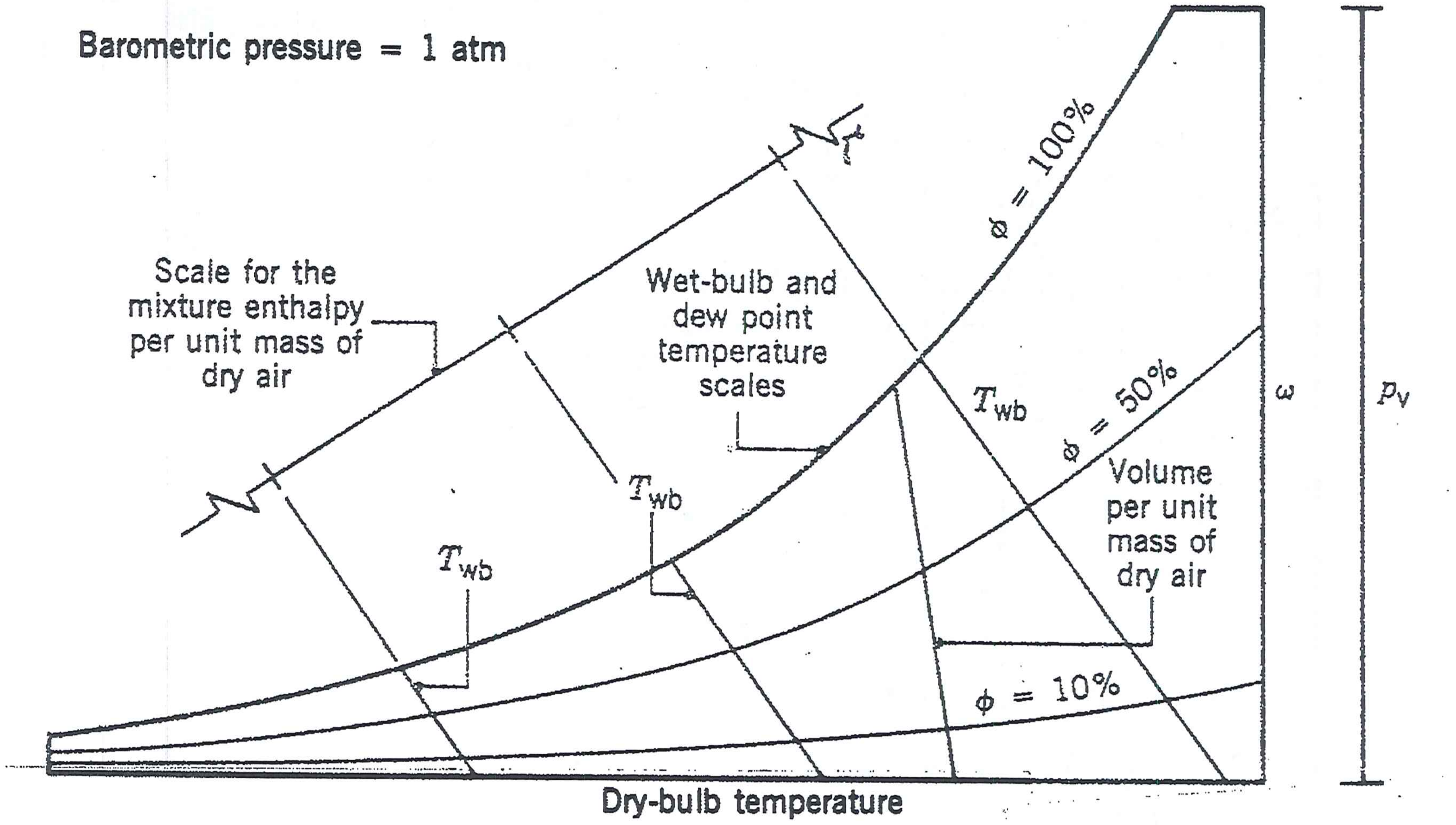
CARBOCYCLIC

ENZENOID

Carbon number	6	7	8		
Aromatics C_nH_{2n-6}	 benzene	 toluene	 *ortho*	 *meta*	 *para*
			xylenes		
	Stable, but high C content; hence smoke and low calorific value. High freezing points. Good spark-knock resistance.				

Note: In general, carbon skeletons only shown from third member onwards.


Specific enthalpy of moist air , kJ/kg dry air (h)
0
10
20
30
40
50
60
70
80
90
100
Wet bulb or saturation temperature, °C
0
10
20
30
Humidity ratio, kg water/kg dry air (ω)
.000
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020
.022
.024
.026
.028
.030
Dry bulb temperature, °C
-10
0
10
20
30
40
50
Specific volume, m3/kg dry air
0.80
0.85
0.90
0.95
Relative humidity
10%
20%
30%
40%
50%
60%
70%
80%
90%


Barometric pressure = 1 atm
Scale for the mixture enthalpy per unit mass of dry air
Wet-bulb and dew point temperature scales
T_{wb}
T_{wb}
T_{wb}
$\phi = 100\%$
$\phi = 50\%$
$\phi = 10\%$
Volume per unit mass of dry air
ω
p_v
Dry-bulb temperature

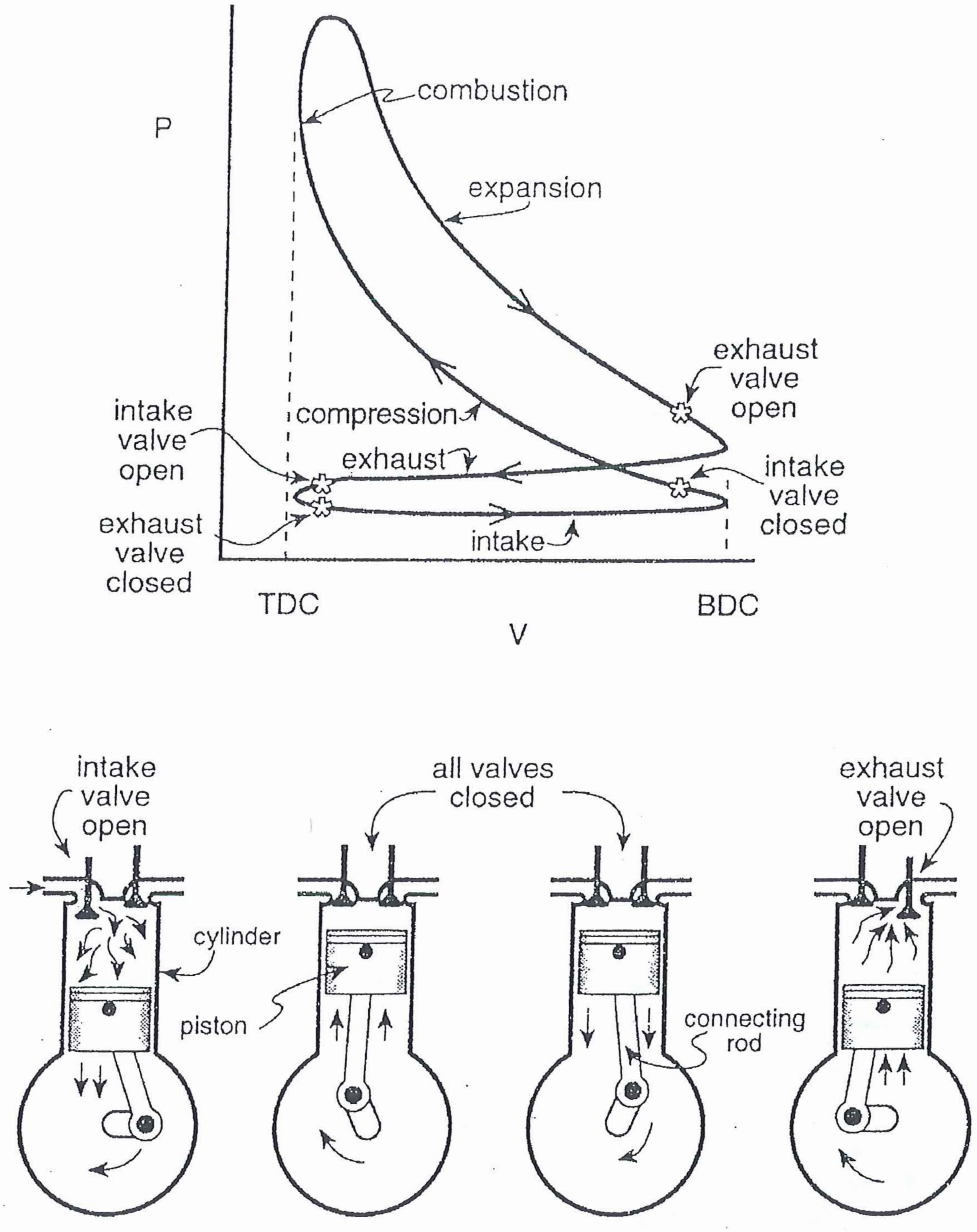


Figure 9.3 Four-stroke IC engine: (1) intake stroke; (2) compression stroke; (3) power stroke; and (4) exhaust stroke.

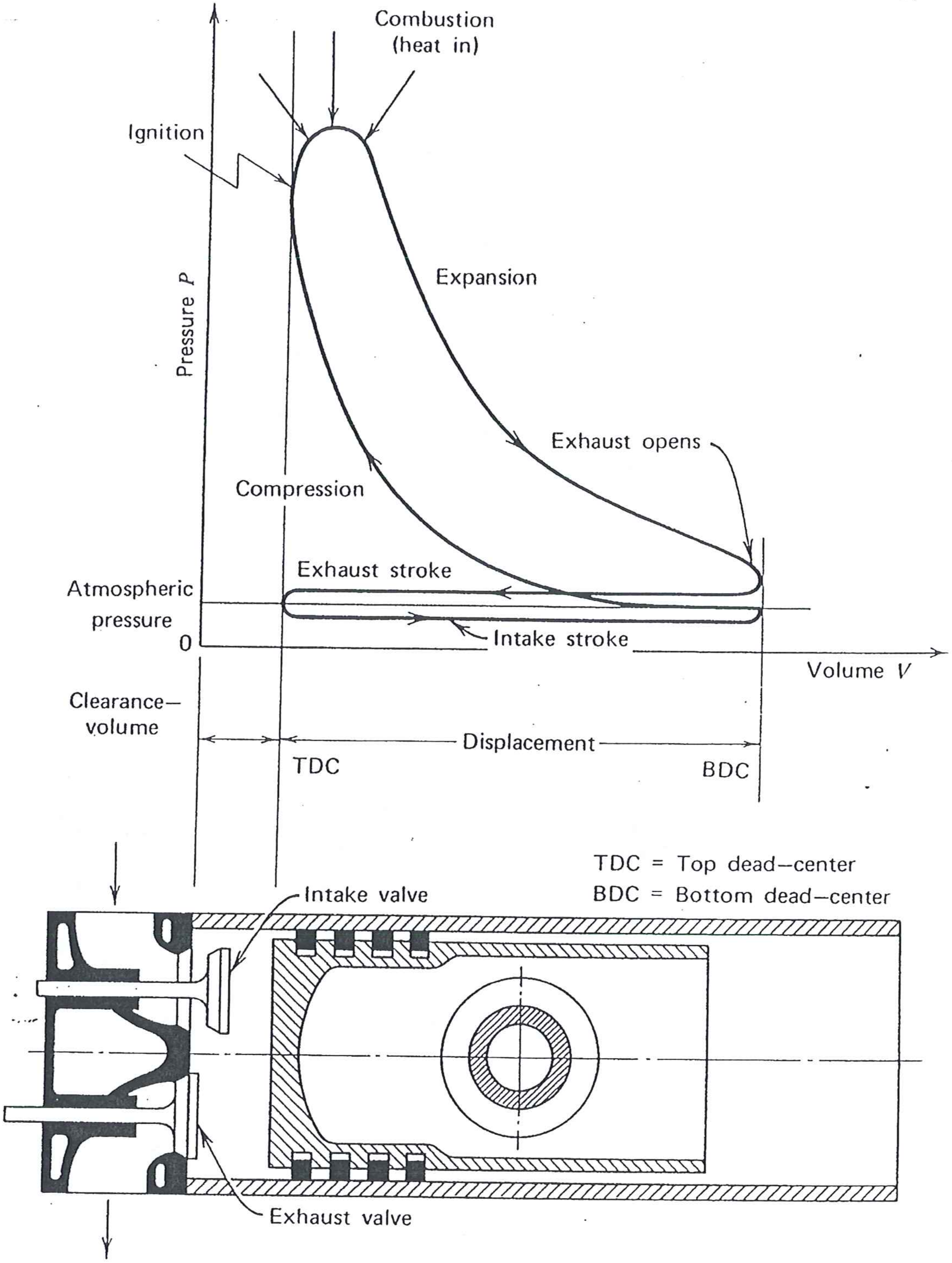


FIGURE 3.3 Actual P-V diagram showing the deviation from the ideal cycle and the loss due to pumping for a four-stroke-cycle engine.

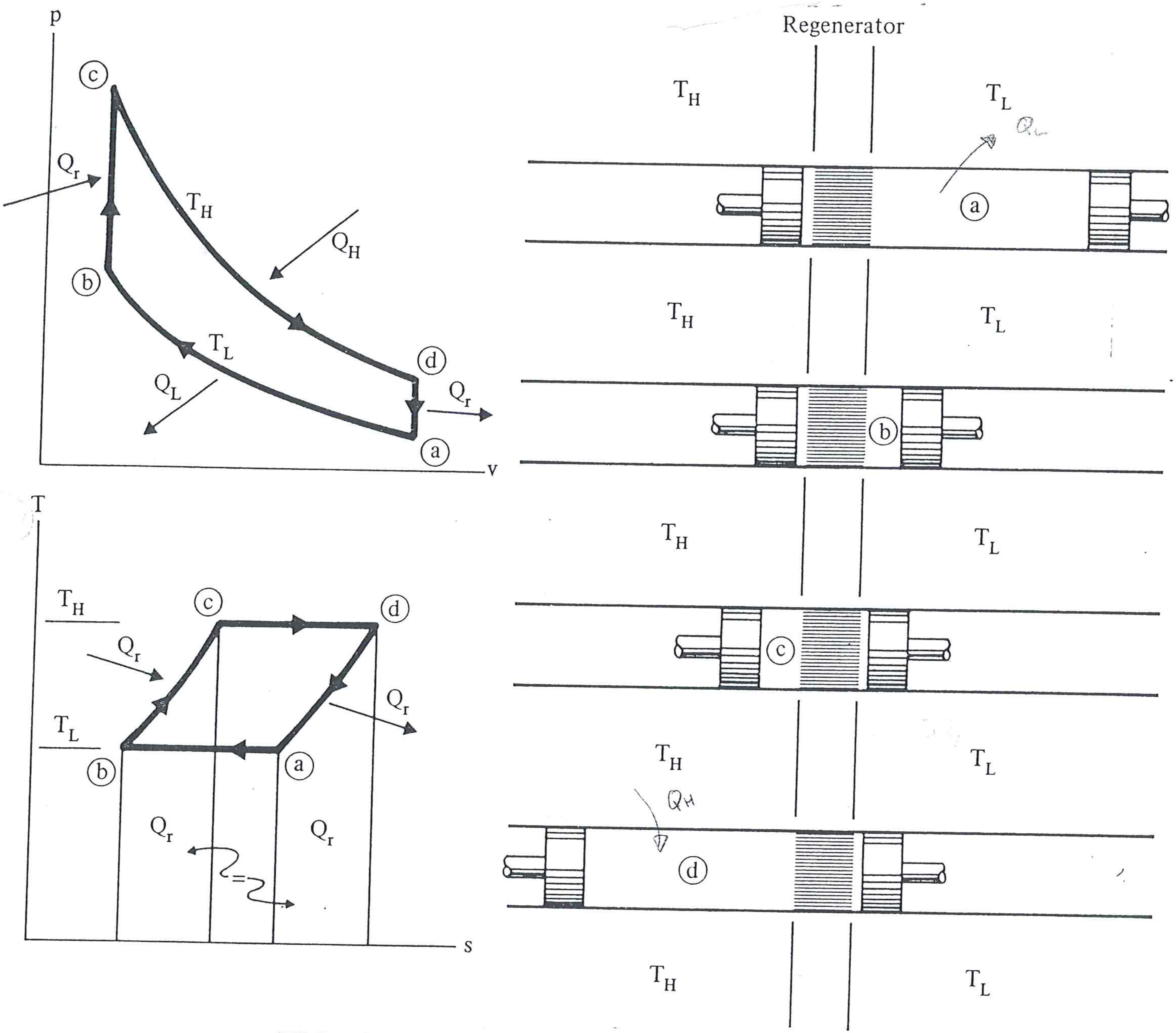
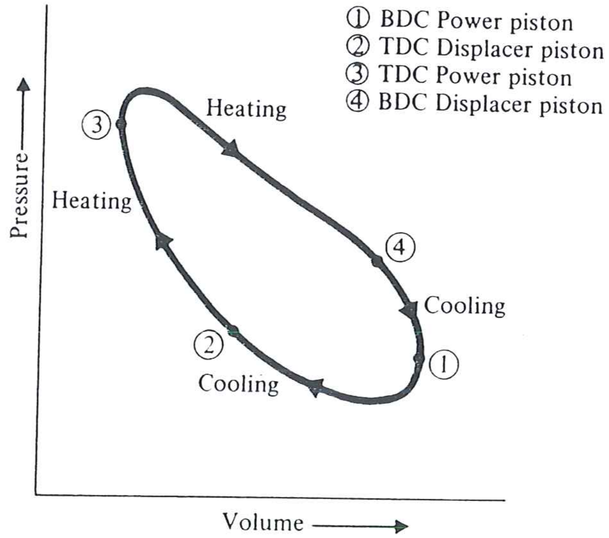
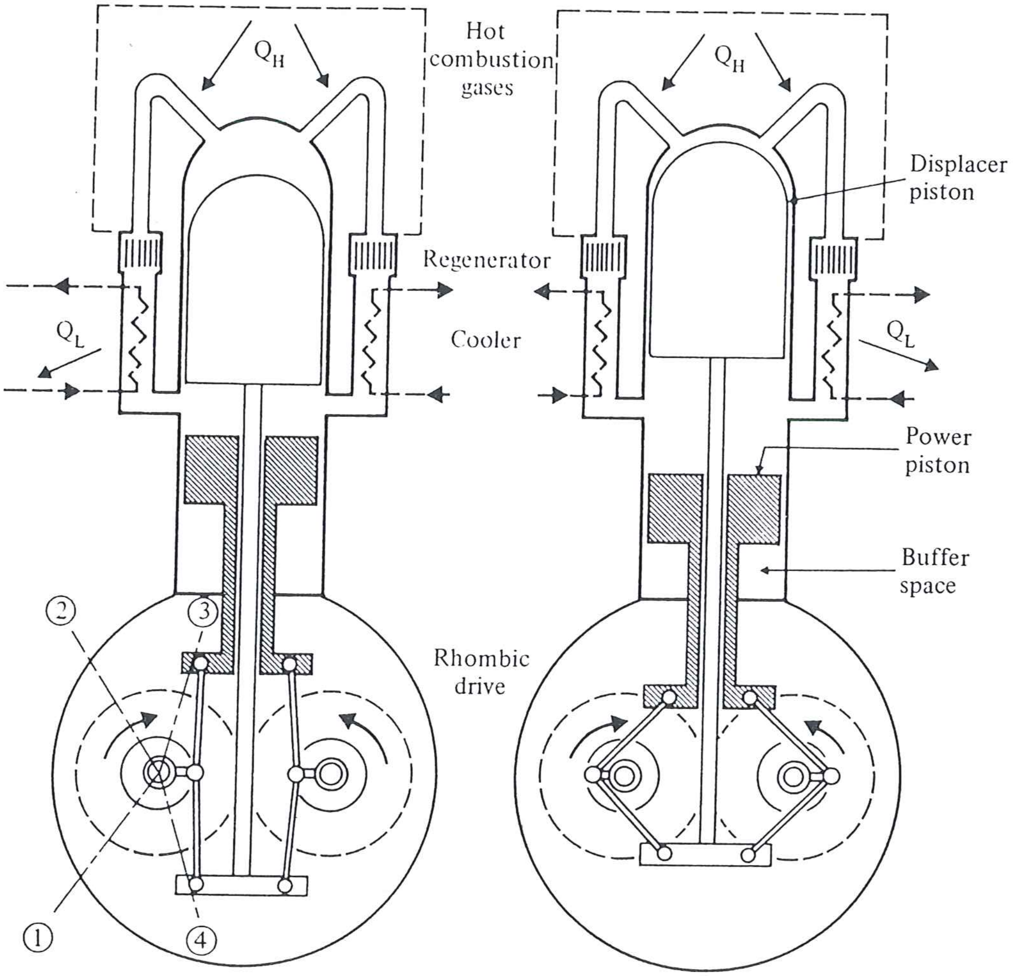


FIG. 3.27 An idealized Stirling cycle engine.



Actual p-v diagram



Two pistons closest together

Two pistons farthest apart

FIG. 3.28 A practical Stirling cycle engine.

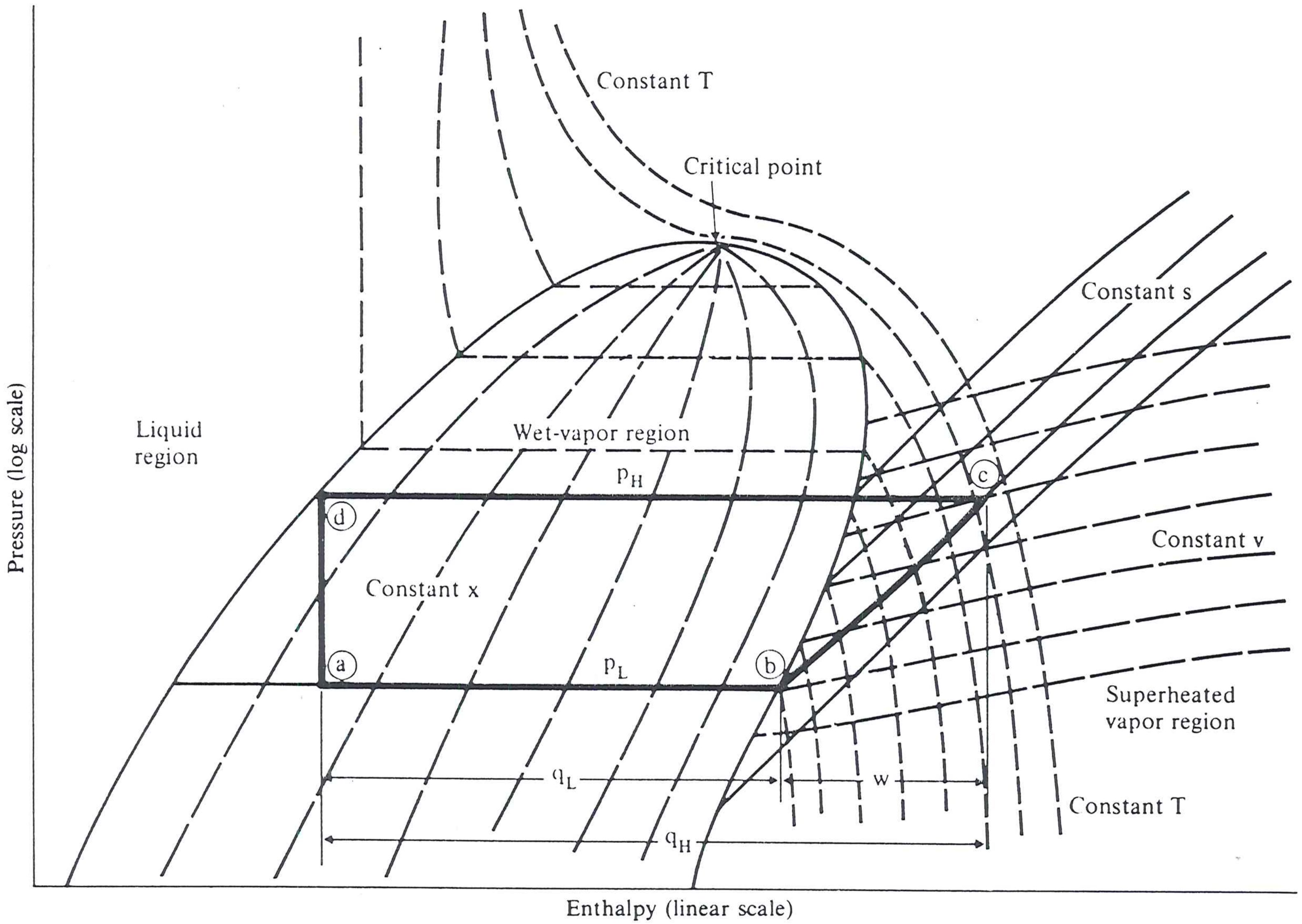
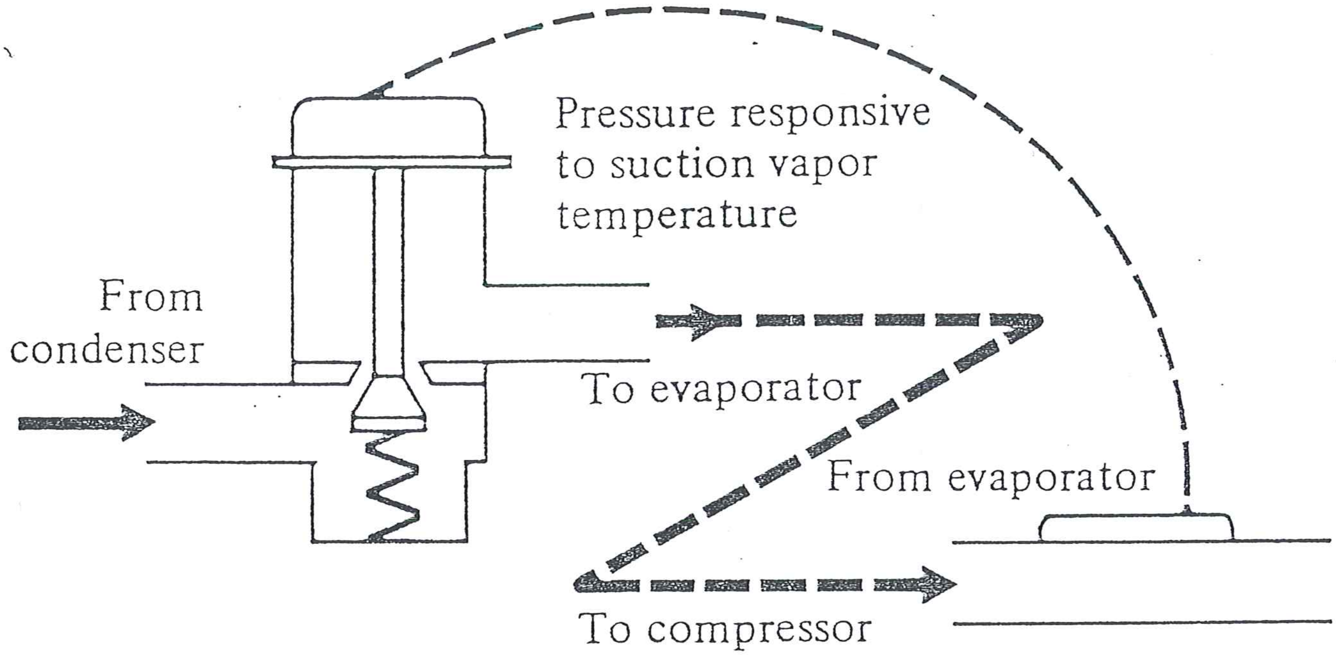


FIG. 4.6 A typical *p-h* diagram for a refrigerant with the ideal vapor-compression cycle shown.



Thermostatic expansion valve

Hill & Peterson

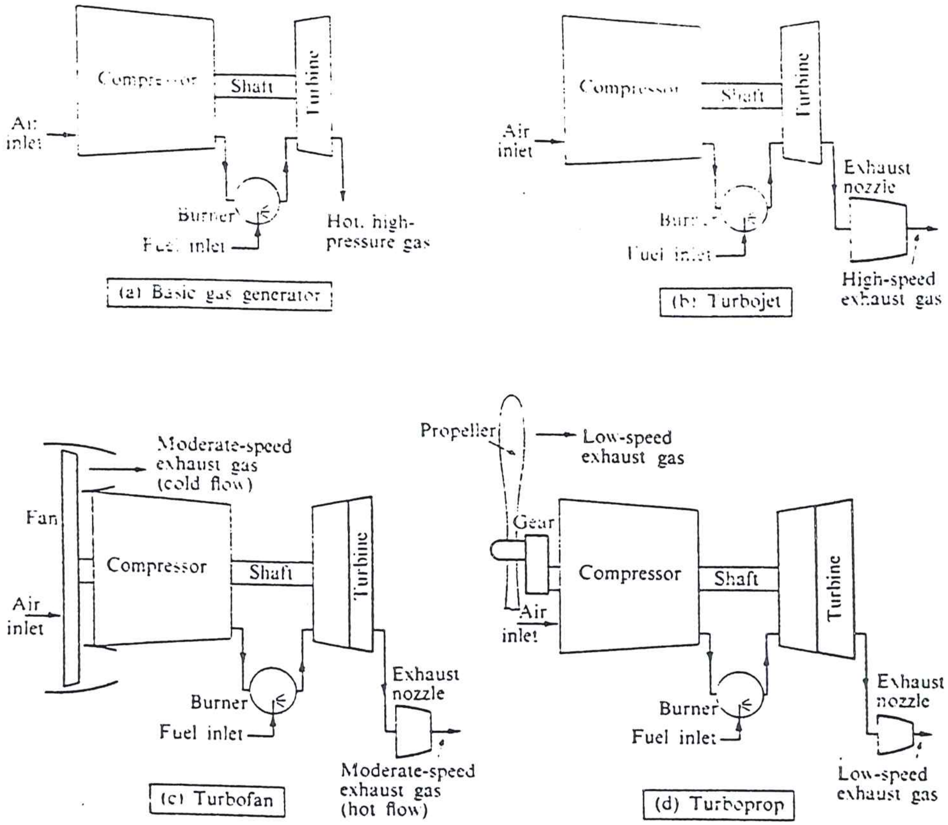
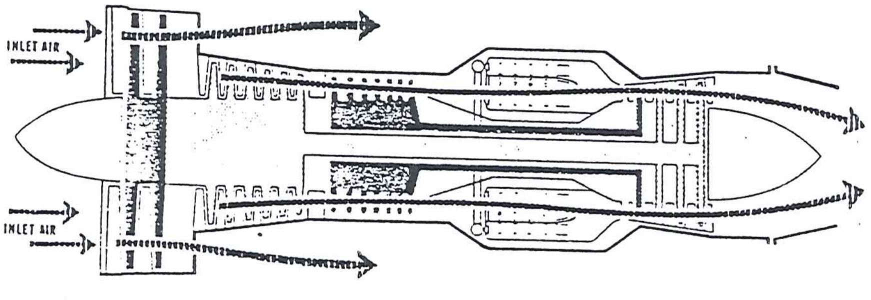


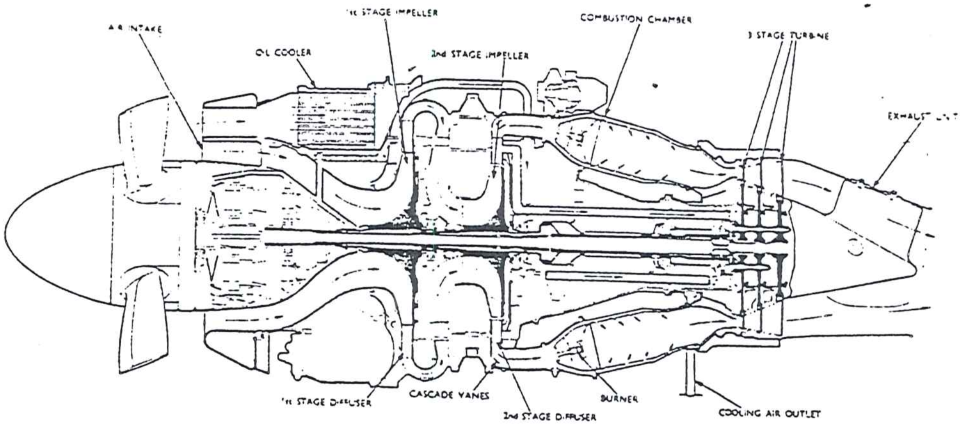
FIG. 6–2. Development of turbine engines from the basic gas generator.



(a) Pratt & Whitney JT3D

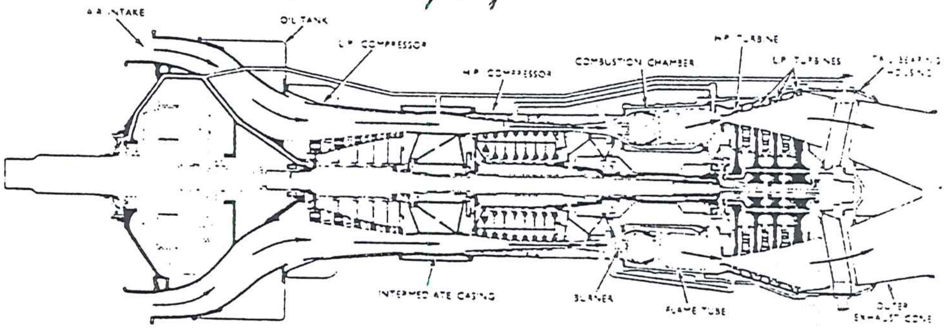
turbofan

Handout 9-24-10



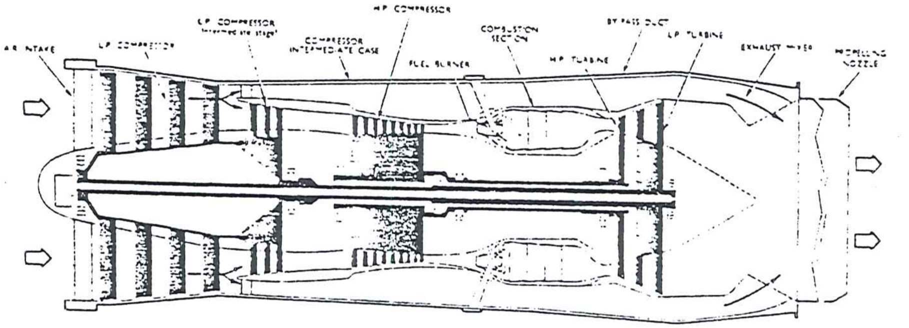
(a) Rolls-Royce Dart RDa-7

turboprop



(b) Rolls-Royce Tyne

turboprop



(b) Rolls-Royce Conway R.Co.42

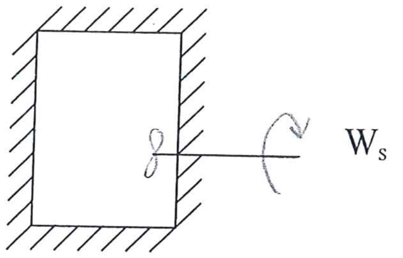
turbo fan

II. Exergy Analysis

D) Examples

Given:

Air in an insulated tank with a stirrer.



$T_{ambient} = 21°C$

$m_{air} = 0.91 kg$

$T_1 = 21°C\ (294\ K)$
$P_1 = 138\ kPa$

$T_2 = 54°C\ (327\ K)$

Note that this is a fixed mass system.

a) Calculate the actual W_s into the gas:

1st law:
$$\overset{0}{\cancel{Q}} - W = U_2 - U_1$$
$$Q = O,\ \text{adiabatic}$$
$$-W = mC_v(T_2 - T_1) \qquad \text{(Ideal Gas)}$$

$$\begin{aligned} W &= mC_v(T_1 - T_2) \\ &= (0.91kg)(0.718\ kJ/kgK)(294 - 327) \\ &= \underset{\substack{\uparrow \\ \text{"in"}}}{-}21.56\ kJ \end{aligned}$$

b) Calculate the “minimum” work needed into system:

The minimum work needed to move the system from state 1 to state 2 would correspond to the change in exergy.

By definition of exergy:

$$\boxed{W_{min} = X_1 - X_2}$$

$$X = (U-U_o) + P(V-V_o) - T_o(S-S_o)$$
$$x = u-u_o + \overset{0}{\cancel{P(v-v_o)}} - T_o(s-s_o)$$
$$\therefore\ W_{min} = m(x_1-x_2)$$
$$= mC_v(T_1-T_2) - mT_o(s_1-s_2)$$

For an ideal gas:

$$s_2 - s_1 = C_v \ln\frac{T_2}{T_1} + R\ln\cancel{\frac{V_2}{V_1}}$$

0, same V and m

Can I use this on D.1, part d

$$W_{min} = (0.91\ \text{kg})(0.718\ \text{kJ/kgK})(294\ \text{K}-327\ \text{K})$$
$$-\ (0.91\ \text{kg})(294\ \text{K})\left((0.718\ \text{kJ/kgK})(\ln\frac{294}{327})\right)$$
$$W_{min} = -21.56\ \text{kJ} - (-20.44\ \text{kJ}) = -1.12\ \text{kJ}$$

c) Find the exergy destroyed:

From earlier , $X_{DESTROYED} = T_o S_{GEN}$

2nd Law:

$$\int_1^2 \cancel{\frac{\delta Q}{T}} + S_{GEN} = S_2 - S_1$$

0, adiabatic

$$S_2 - S_1 = m(s_2 - s_1)$$

$\therefore$ $X_{DESTROYED} = mT_o(s_2 - s_1) = 20.44$ kJ from above calculation

Does this make sense with our exergy balance?

Recall for a fixed mass system:

$$\cancelto{0}{\int_1^2 (1-\frac{To}{T})\delta Q} - [W - P_o(V_2 - V_1)] - T_o S_{GEN} = X_2 - X_1$$

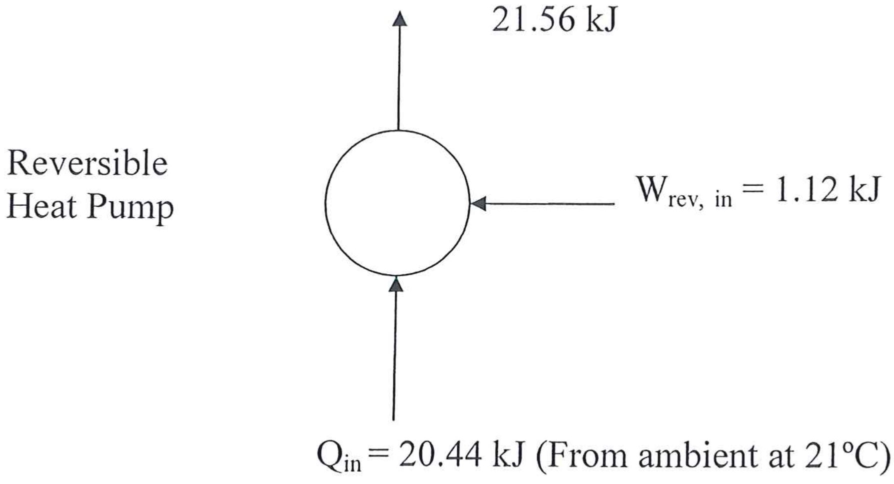
$$T_o S_{GEN} = 20.44 \text{ kJ}$$

$$X_2 - X_1 = 1.12 \text{ kJ}$$

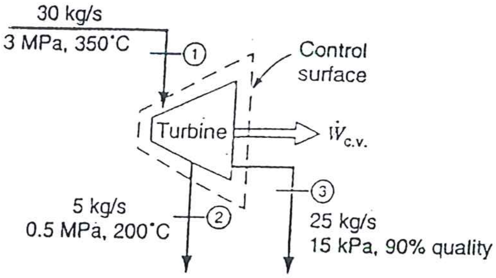
$$\Rightarrow \quad -W = 1.12 \text{ kJ} + 20.44 \text{ kJ}$$

$$W = -21.56 \text{ kJ}$$

NOTE: If could put in 21.56 kJ into system reversibly, would require less work-in, e.g.:



Given: An insulated steam turbine receives 30 kg of steam per second at 3 MPa, 350°C. At the point in the turbine where the pressure is 0.5 MPa, steam is bled off for processing equipment at the rate of 5 kg/s. The temperature of this steam is 200°C. The balance of the steam leaves the turbine at 15 kPa, 90% quality. Determine the exergy per kilogram of the steam entering and at both points at which steam leaves the turbine, the isentropic efficiency and the second-law efficiency for this process.



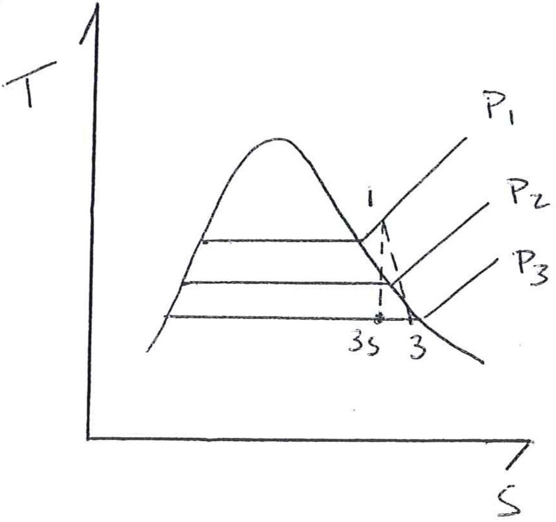
Solution:

The exergy at any point for the steam entering or leaving the turbine is given by:

$$x = \psi = (h - h_0) - T_0(s - s_0) + \frac{v^2}{2} + g(z - z_0)$$

Neglecting changes in kinetic and potential energy

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



For the ideal isentropic turbine, this redefines the exit state

↓ adiabatic, so Q = 0

$$s_{e_s} - s_i = 0 \qquad s_{3_s} = s_{2_s} = s_1$$

$$\dot{W}_s = \dot{m}(h_1 - h_{es}) = \dot{m}_1 h_1 - \dot{m}_2 h_{2_s} - \dot{m}_3 h_{3_s}$$

For the actual turbine,

↳ insulated, Q = 0

$$\dot{W} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$

At the pressure and temperature of the surroundings, 0.1 Mpa, 25°C, the water is a slightly compressed liquid, and the properties of the water are essentially equal to those for saturated liquid at 25°C.

$$h_0 = 104.9\, kJ/kg \qquad s_0 = 0.3674\, kJ/kg\, K$$

Therefore,

$$\psi_1 = (3115.3 - 104.9) - 298.15(6.7428 - 0.3674) = 1109.6\ kJ/kg$$

$$\psi_2 = (2855.4 - 104.9) - 298.15(7.0592 - 0.3674) = 755.3\ kJ/kg$$

$$\psi_3 = (2361.8 - 104.9) - 298.15(7.2831 - 0.3674) = 195.0\ kJ/kg$$

Change in ψ: (which is equal to the $\dot{W}_{\max}$)

$$\dot{m}_1\psi_1 - \dot{m}_2\psi_2 - \dot{m}_3\psi_3 = 30(1109.6) - 5(755.3) - 25(195.0) = 24,\ 637\ kW$$

For the ideal isentropic turbine,

$$s_{2_s} = 6.7428 = 1.8606 + x_{2_s}\ x\ 4.9606 \quad (at\ 0.5\ MPa)$$

$$x_{2_s} = 0.9842$$

$$h_{2_s} = 640.21 + 0.9842\ x\ 2108.47 = 2715.4$$

$$s_{3_s} = 6.7428 = 0.7549 + x_{3_s}\ x\ 7.2536$$

$$x_{3_s} = 0.8255$$

$$h_{3_s} = 225.9 + 0.8255\ x\ 2373.1 = 2184.9$$

$$\dot{W}_s = 30(3115.3) - 5(2715.4) - 25(2184.9) = 25260\ kW$$

For the actual turbine,

$$\dot{W}_{ACT} = 30(3115.3) - 5(2855.4) - 25(2361.8) = 20137\ kW$$

The isentropic efficiency is

$$\eta_s = \frac{\dot{W}_{ACT}}{\dot{W}_s} = \frac{20137}{25260} = 0.797$$

and the second-law efficiency is

$$\eta_{2nd\ law} = \frac{\dot{W}_{ACT}}{\dot{W}_{Max}} = \frac{\dot{W}_{ACT}}{\dot{m}_i\psi_i - \dot{m}_0\psi_0} = \frac{20137}{24637} = 0.817$$

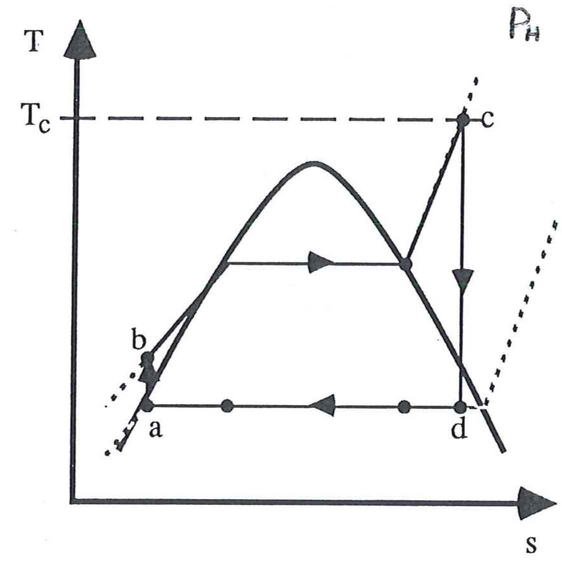
Note that isentropic efficiency is lower, but not all the isentropic work shown is available.

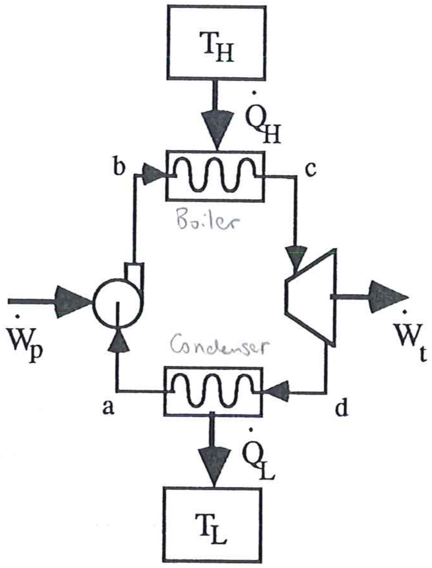
Note that a typical 1st law efficiency that would be applied here is:

$$\eta_{1st\ Law} = \frac{\dot{W}_{ACT}}{Energy\ Flow\ In}$$

$$= \frac{\dot{W}_{ACT}}{\dot{m}_1 h_1} = \frac{20,137}{30(3115)} = 21.5\%$$

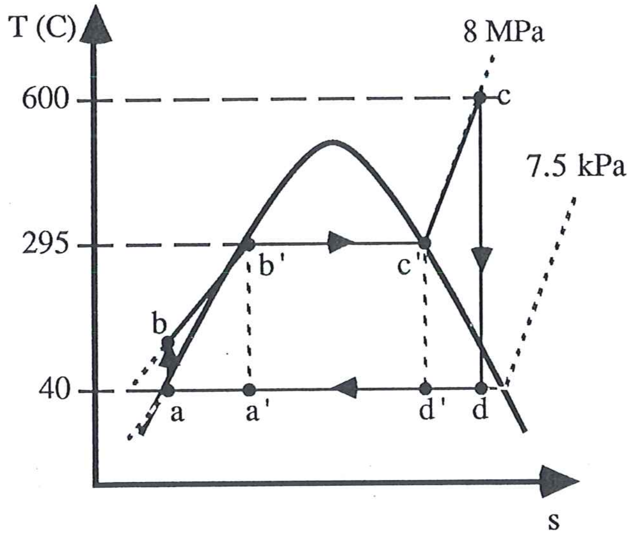
Ideal Steam Rankine Cycle

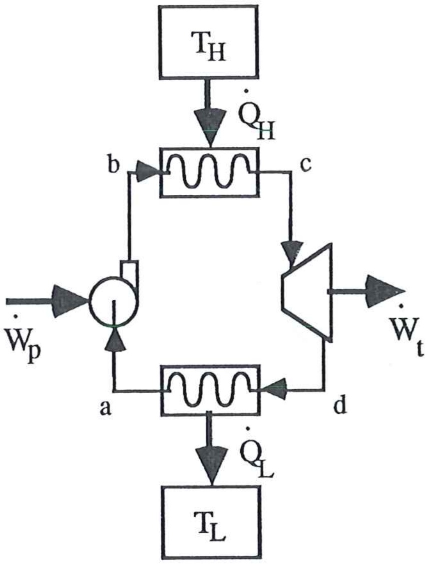




BOILER P			CONDENSER P			SUPERHEAT		
$P_L = 10$ kPa, $T_c = 350$ C			$P_H = 3.5$ MPa, $T_c = 350$ C			$P_H = 3.5$ MPa, $P_L = 10$ kPa		
P_H (MPa)	η_{th} (%)	x_d (%)	P_L (kPa)	η_{th} (%)	x_d (%)	T_c (C)	η_{th} (%)	x_d (%)
1.0	28	89	5	36	78	243	33	73
3.5	34	80	10	34	80	350	34	80
6.0	36	76	50	29	86	500	36	87
10.0	38	71	100	26	88	800	41	97

Steam Power Cycles





Cycle	Carnot a'b'c'd'a'	Rankine w/o superheat abc'd'a	Carnot w/superheat a'b'cda'	Rankine w/superheat abcda
w_t (kJ/kg)	970	970	1452	1452
w_p	323	8	323	8
w_{net}	646	961	1129	1444
q_H	1441	2581	2325	3465
q_L	795	1620	1197	2021
T_L	40	40	40	40
T_H	295	295	600	600
η_{th} (%)	45	37	49	42
η_{MAX}	45	45	64	64

$$\eta_{th} = \frac{W_{NET}}{q_H}$$

$$\eta_{MAX} = 1 - \frac{T_L}{T_H}$$

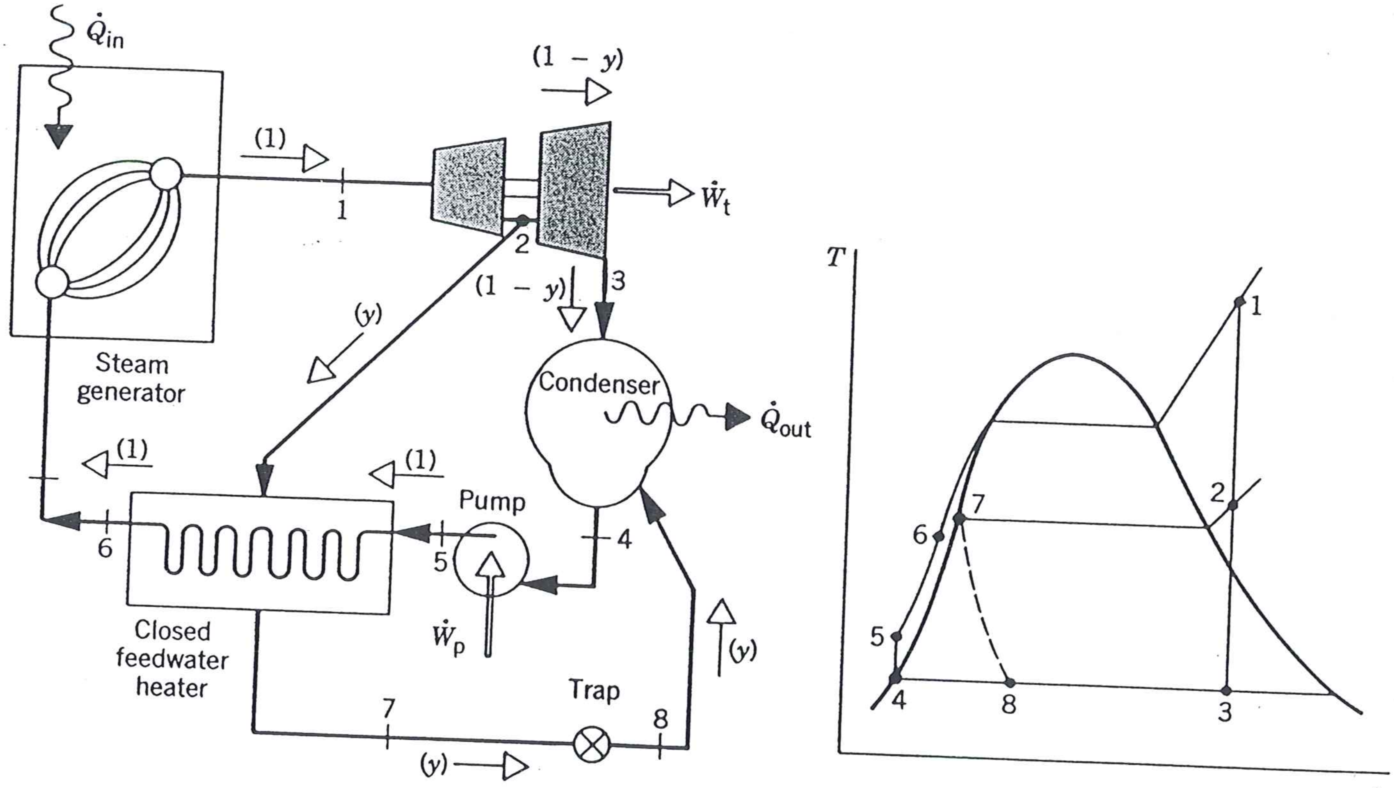


FIGURE 8.11 Regenerative vapor power cycle with one closed feedwater heater.

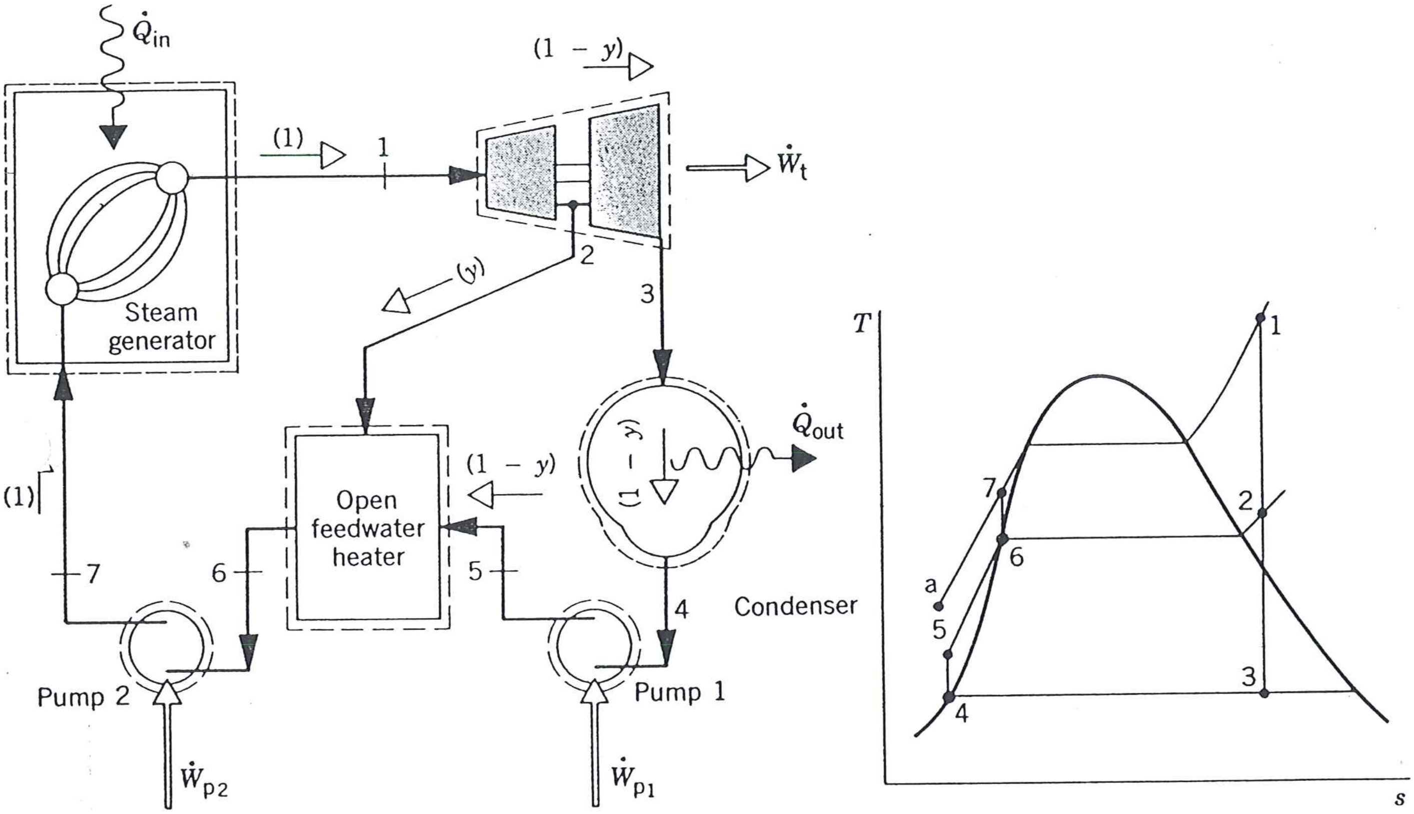


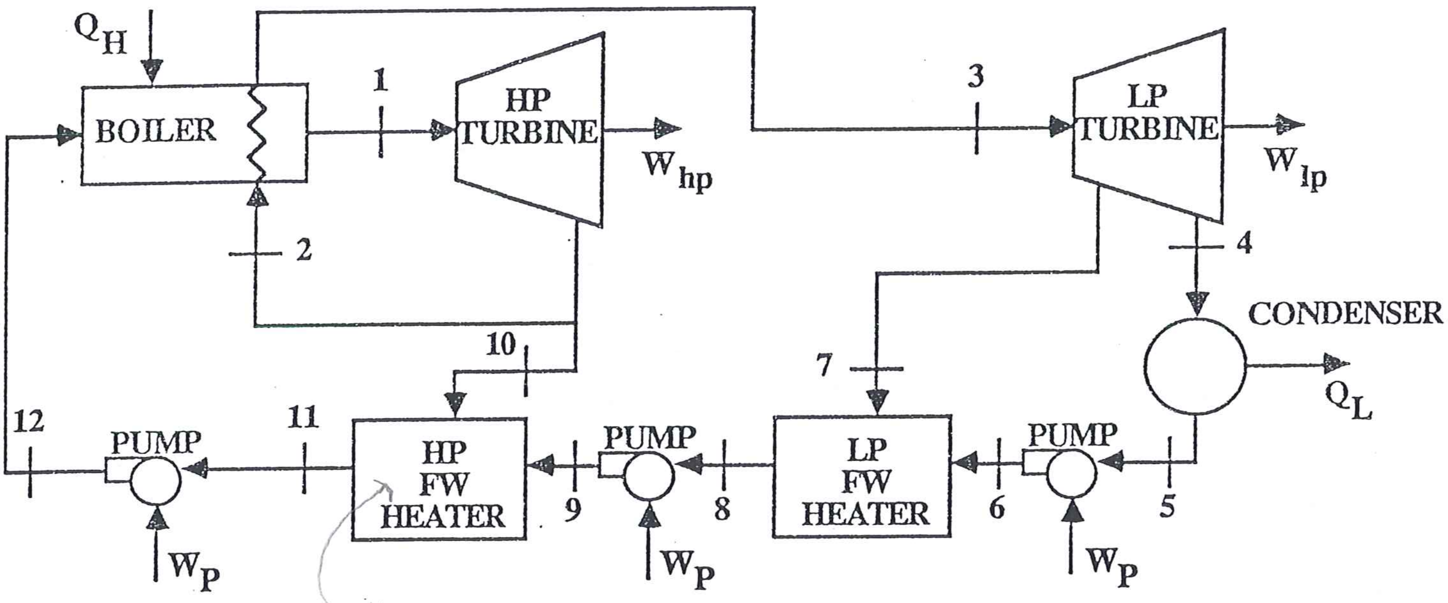
FIGURE 8.9 Regenerative vapor power cycle with one open feedwater heater.

RANKINE CYCLE EXAMPLE

D. Example

Given:

Consider a combined reheat and regenerative cycle utilizing steam as the working fluid. Steam enters the high pressure turbine at 3.5 MPa, 350° C, and is extracted for purposes of feedwater heating at 0.8 MPa. The remainder of the steam is reheated at this pressure to 350°C and fed to the low pressure trubine. Steam is extracted from the low pressure turbine at 0.2 MPa for feedwater heating. The condenser pressure is 10 kPa and both feedwater heaters are open heaters. Calculate the cycle thermal efficiency and the net work per kilogram of steam.



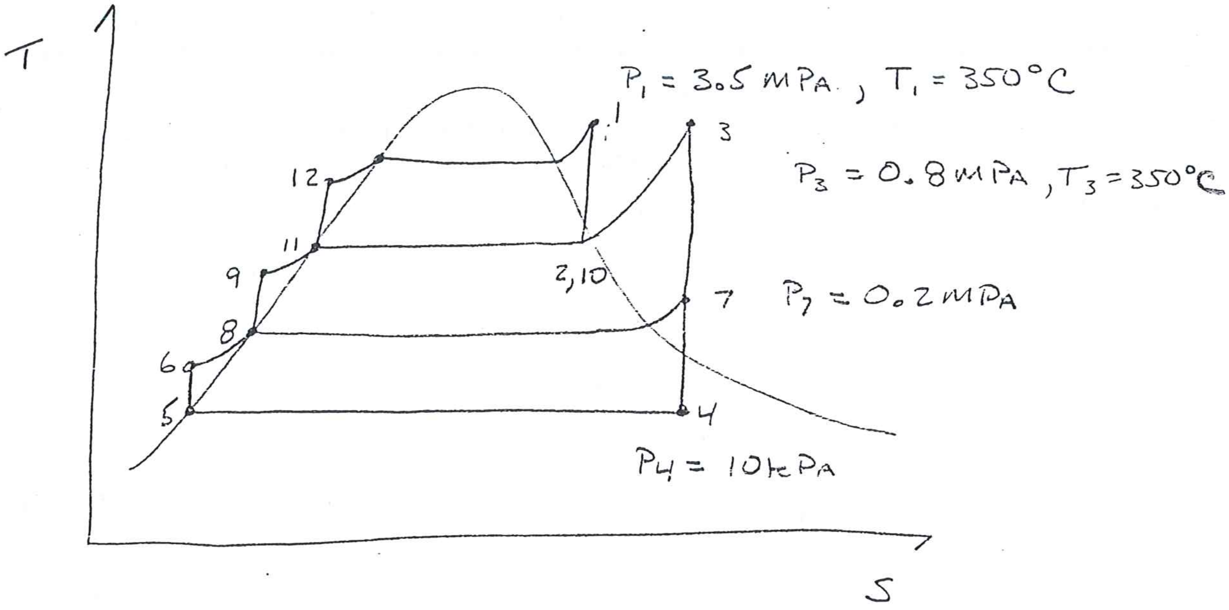
Horcepower
FenWay
Heater

Find: η_{th}, W_{TOTAL} (kJ/kg)

Assumptions: Isentropic Pumps & Turbines

SOLN|

Draw T-S Diagram:



Note: Actual position of points 2, 10, 7 and 4 with respect to dome are unknown.

- Need mass flow distribution through system. Lets assume $\dot{m}_1 = 1$ kg/s and find final answer on a per kg basis.

HP Turbine $\dot{m}_1 = \dot{m}_2 + \dot{m}_{10} = 1 \text{ kg/s}$

$$\dot{W}_{Hpt} = \dot{m}_1 h_1 - \dot{m}_{10} h_{10} - \dot{m}_2 h_2 \ ; \ h_2 = h_{10} \ ; \ \dot{Q}=0$$

$$\dot{W}_{Hpt} = \dot{m}_1 (h_1 - h_2)$$

$h_1 = h_1(P_1, T_1) = 3104 \text{ kJ/kg}$

Isentropic $\quad \therefore s_1 = s_2 = s_{10}$

$s_1 = s_1 (P_1, T_1) = 6.6578 \text{ kJ/kg K}$

$s_2 = s_{f_2} + x_2 (s_{fg_2})$

with $\quad P_2 = 0.8 \text{ MPa} ; \quad s_{f_2} = 2.0461 , \; s_{fg_2} = 4.6166$

$\therefore x_2 = 0.9989$ (just inside dome)

$h_2 = h_{f_2} + x_2 (h_{fg_2})$

$= 721.1 + (0.9989)(2048) = 2766.8 \text{ kJ/kg}$

$\therefore \dot{W}_{HPt} = 1 \text{ kg/s } (3104 - 2767) = 337 \text{ kJ/s}$

LP Turbine $\qquad \dot{m}_2 = \dot{m}_3 = \dot{m}_7 + \dot{m}_4$

$\dot{W}_{LPt} = \dot{m}_3 h_3 - \dot{m}_7 h_7 - \dot{m}_4 h_4 \; ; \; \dot{Q} = 0$

$h_3 = h_3 (T_3, P_3) = 3162 \text{ kJ/kg}$

Isentropic $\Rightarrow \; s_3 = s_7 = s_4$

$s_3 = s_3 (T_3, P_3) = 7.4089$

$s_4 = s_{f_4} + x_4 (s_{fg_4})$

with $\quad P_4 = 10 \text{ kPa}, \; s_{f_4} = 0.6492 , \; s_{fg_4} = 7.5010$

$\Rightarrow x_4 = 0.901 \Rightarrow h_4 = h_{f_4} + x_4 (h_{fg_4}) = 2348 \text{ kJ/kg}$

$s_7 = s_4 = s_3 = 7.4089$

with $P_7 = 0.2\text{MPa} \Rightarrow$ must be superheated

$(s_g(@P = 0.2\text{ MPa}) = 7.1271)$

$\therefore$ with P_7 & s_7 : $T_7 = 178.5°\text{C}$, $h_7 = 2827\text{ kJ/kg}$

but need $\dot{m}_3$, $\dot{m}_7$ & $\dot{m}_4$

Pump 5-6: $\dot{m}_5 = \dot{m}_6$

$\dot{W}_p = \dot{m}_5 (h_5 - h_6)$

But for a steady, adiabatic pump:

$$W = \int_0^i v dp$$

$v_5 \sim v_6 = 0.001010\text{ m}^3\text{/kg}$ at 10 kPa

$\therefore \dot{m}_5(h_5 - h_6) = \dot{m}_5(v_5(P_5 - P_6))$

$h_5 = h_5(P_5, x_5 = 0) = 191.83\text{kJ/kg}$

$\Rightarrow h_6 = 191.83 + 0.001010(200 - 10)$

$h_6 = 192\text{ kJ/kg}$

Still need $\dot{m}_5$ for $\dot{W}_p$

LP Feedwater: $\dot{m}_8 = \dot{m}_7 + \dot{m}_6$

$0 = \dot{m}_6 h_6 + \dot{m}_7 h_7 - \dot{m}_8 h_8$; $\dot{W} = 0$; $\dot{Q} = 0$

Know h_6, h_7

$h_8 = h(P_8, x_8 = 0) = 504.7\text{ kJ/kg}$

Pump 8-9: $\dot{m}_8 = \dot{m}_9$

$$\dot{W}_p = \dot{m}_8 (h_8 - h_9) = \dot{m}_8 v_8 (P_8 - P_9)$$

$$v_8 = 0.001061 \text{ m}^3/\text{kg} @ 200 \text{ kPa}$$

$$\Rightarrow h_9 = v_8 (P_9 - P_8) + h_8 = 505.3 \text{ kJ/kg}$$

HP Feedwater: $\dot{m}_9 + \dot{m}_{10} = \dot{m}_{11} = 1 \text{ kg/s}$

$$\dot{W} = 0 \; ; \; \dot{Q} = 0 \; ; \; 0 = \dot{m}_9 h_9 + \dot{m}_{10} h_{10} - \dot{m}_{11} h_{11}$$

$$h_{10} = h_2 = 2766.9 \text{ kJ/kg}$$

$$h_{11} = h(P_{11}, x = 0) = 721.11 \text{kJ/kg}$$

$\therefore$

$$(1 \text{kg/s})(721.11 \text{ kJ/kg}) = \dot{m}_9 (505.3) + \dot{m}_{10} (2766.9)$$

along with $\dot{m}_9 + \dot{m}_{10} = 1 \text{ kg/s}$

Gives: $\dot{m}_{10} = 0.0955 \text{ kg/s} \; ; \; \dot{m}_9 = 0.9045 \text{ kg/s} = \dot{m}_8$

Then with; $\dot{m}_8 h_8 = \dot{m}_6 h_6 + \dot{m}_7 h_7 \;\&\; \dot{m}_8 = \dot{m}_6 + \dot{m}_7$

Gives:

$$\dot{m}_7 = 0.1074 \text{ kg/s}$$

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_6 = 0.7972 \text{ kg/s}$$

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_8 = 0.\ 9045 \text{ kg/s}$$

$$\dot{m}_{11} = \dot{m}_{12} = 1 \text{ kg/s}$$

Also:

$$\dot{W}_{HPt} = 337 \text{ kJ/s} \cdot \dot{W}_{LPt} = 684 \text{ kJ/s}$$

$$\dot{W}_{p5-6} = -0.136 \text{ kJ/s} \; ; \; \dot{W}_{p9-8} = -0.543 \text{ kJ/s} \; ; \; \dot{W}_{p11-12} = -3.01 \text{ kJ/s}$$

$$\dot{W}_{NET} = 337 + 684 - 0.136 - 0.543 - 3.01 = 1017 \text{ kJ/s}$$

Condenser: $\dot{W} = 0 \;;\; -\dot{Q}_L = \dot{m}_4 h_4 - \dot{m}_5 h_5$

$$\dot{Q}_L = \dot{m}_4 (h_5 - h_4) = -1719 \text{ kJ/s}$$

$$\Rightarrow \eta_{th} = \frac{W_{NET}}{Q_H} = \frac{W_{NET}}{Q_L + W_{NET}} = \frac{1017 \text{ kJ/s}}{1719 \text{ kJ/s} + 1017 \text{ kJ/s}}$$

$$\eta_{th} = 0.372 = 37.2 \%$$

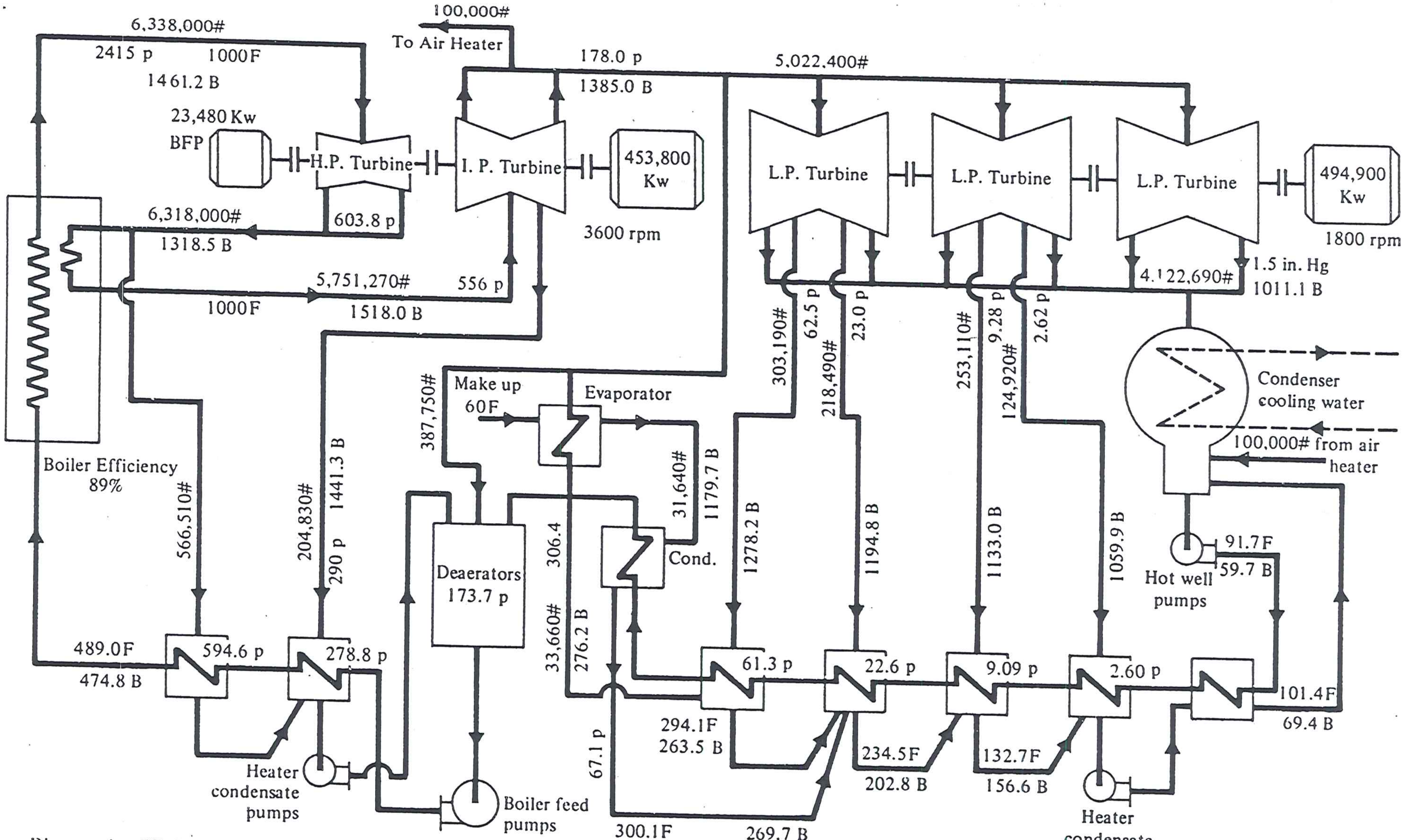


Diagram simplified:
Shaft seal and other minor extractions and returns not shown; thus mass rates do not balance exactly.

#	-Pounds per hour,	F	-Fahrenheit.
p	-psia,	B	-Btu per pound,

FIG. 1.21 Heat balance. Ravenswood No. 3 Steam Power Plant. Nominal rating: 1000 MW. Data from Consolidated Edison Co. of New York [A. 4].

Exam practice

HIGH : 100
LOW : 45
MEAN : 69.7
ST DEV: 14.9

NAME: ZACH BECKMANN

78

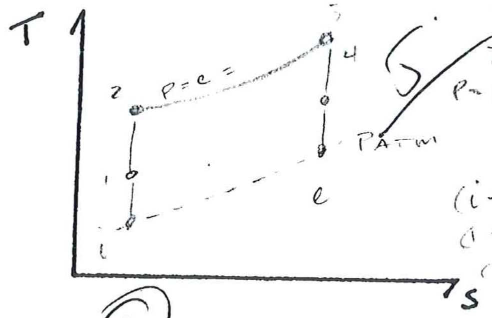
ME221
Exam II
Fall 2008
Drallmeier

(25 pts) 1. In a turboprop engine, the turbine drives a propeller as well as the compressor. Air enters the diffuser with a mass flow rate of 45 kg/s at 40 kPa, 240 K and a velocity of 180 m/s. The compressor pressure ratio is 10 and assume the compressor is isentropic. The turbine inlet temperature is 1140 K and assume the turbine is isentropic. The turbine exit pressure is 50 kPa. Neglect kinetic energy except at the diffuser inlet and nozzle exit. Assume constant specific heats. (i.e. $C_v = 0.717$ kJ/kgK, $C_p = 1.004$ kJ/kgK).

(5 pts) a) Sketch the T-s diagram for the cycle. Label the states entering the diffuser, compressor, combustor, turbine and nozzle.

STATE 1
$P = 50$ kPa

STATE 2
$P = 500$ kPa



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STATE 3
$T = 1140$
$P = 500$ kPa
$h =$

STATE 4
$P = 50$ kPa

STATE e
$P = 40$ kPa

STATE i
$\dot{m} = 45$ kg/s
$P = 40$ kPa
$T = 240$
$V = 180$ m/s

(i-1) DIFFUSER
(1-2) ISENTROPIC EXPANSION (COMPRESSOR)
(2-3) CONST 'P' HEAT ADD (COMBUSTOR)
(3-4) ISENTROP COMP (TURBINE)
(4-e) NOZZLE

(5 pts) b) Find the compressor inlet pressure (kPa)

$P_i = 40$ kPa $T_i = 240$ K $V_i = 45$ kg/s $\underline{P_1 = ?}$

$$\frac{P_2}{P_1} = 10 \qquad \frac{P_2}{50} = 10 \qquad \boxed{P_2 = \cancel{500} \text{ kPa}}$$

P_1 IS INLET.

1

(10 pts) c) Find the power delivered to the propeller (kW)

$${}_3\dot{W}_4 = \dot{m}(h_3 - h_4) = \dot{m}c_p(T_3 - T_4) \qquad -W_c = W_p \Rightarrow \dot{m}c_p(T_1 - T_2) = \dot{m}c_p(T_3 -$$

$$T_1 = T_i\left(\frac{P_1}{P_i}\right)^{\frac{k-1}{k}} \Rightarrow T_1 = 240\left(\frac{50}{40}\right)^{\frac{.4}{1.4}} \qquad \dot{W} = \dot{m}c_p(T_1 - T_2)$$

$T_1 = 255.8$ K

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \Rightarrow T_2 = 255.8(10)^{.4/1.4}$$

$T_2 = 493.87$ K

$$\dot{W}_p = 45(1.004)(493.87 - 255.8)$$

$$\boxed{\dot{W}_p = 10{,}756 \text{ kW}}$$

COMP. WORK ONLY. 1

6

(5 pts) d) Find the velocity at the nozzle exit (m/s)

$$T_4 = T_3\left(\frac{1}{10}\right)^{\frac{k-1}{k}}$$

$T_4 = 540.46$

$$T_e = T_4\left(\frac{P_e}{P_4}\right)^{\frac{.4}{1.4}}$$

$$T_e = 540.46\left(\frac{40}{50}\right)^{.4/1.4}$$

$T_e = 553.99$ K

$$\cancel{\dot{Q}} - \cancel{\dot{W}} + \Sigma\dot{m}\left(h + \tfrac{1}{2}V^2\right) - \Sigma\dot{m}\left(h + \tfrac{1}{2}V^2\right) = \frac{dE_{cv}}{dt}$$

$$\cancel{\dot{m}}\left(h_4 + \tfrac{1}{2}\cancel{V_4^2}\right) = \cancel{\dot{m}}\left(h_e + \tfrac{1}{2}V_e^2\right)$$

$$V_e = \sqrt{2(1.004)(1000)(T_4 - T_e)}$$

$$V_e = \sqrt{2(1004)(540.46 - 553.99)}$$

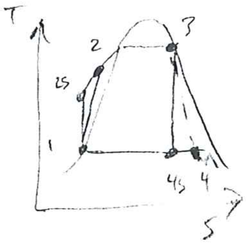
$$\boxed{V_e = 270.61 \text{ m/s}}$$

5

17

(20 pts) 2. A steam power generation system operates using a Rankine cycle. The water leaves the boiler as a saturated vapor at 3.5 MPa. The working fluid leaves the turbine and enters the condenser at 10 kPa. The pump has an isentropic efficiency of 80% and the turbine has an isentropic efficiency of 85%. Both can be considered adiabatic. You may assume a constant pressure boiler and condenser.

(5 pts) a) Find the **actual** specific pump work. (kJ/kg)



STATE 1
P=10kPa
h=191.81
V=.00101

STATE 2S
P=3500kPa

STATE 2

$_1w_{pump_s} = {}_1w_{2s} = \int VdP = .00101(10-3500) = -3.52$ kJ/kg $= {}_1w_{2s}$

5

$_1w_{2A} = \frac{_1w_{2s}}{\eta_P} = \frac{-3.52}{.80} = \boxed{-4.41 \text{ kJ/kg} = {}_1w_{2A} \text{ pump}}$

(10 pts)b) Find the **actual** specific turbine work (kJ/kg)

STATE 3
P=3.5 MPa
S=6.1244
h=2802.7

STATE 4s
P=10kPa
$x = \frac{6.1244-.6492}{7.4996}$
x=.73
$h_s = 191.81 + .73(2392.1)$
$h_s = 1938.2$

$_3w_{4s} = h_3 - h_{4s} = 2802.7 - 1938.2 = 846.5$ kJ/kg

10

$_3w_{4A} = (\eta_T)({}_3w_{4s}) = .85(846.5)$

$\boxed{_3w_{4A} = 734.825 \text{ kJ/kg} = w_{TURBINE}}$

(5 pts) c) Find the **actual** specific heat rejected from the condenser (kJ/kg)

$_3w_{4A} = h_3 - h_4$ $\quad h_4 = h_3 - {}_3w_{4A}$ $\quad h_4 = 2802.7 - (734.825)$

$h_4 = 2067.87$

$_4q_1 = h_1 - h_4 = 191.81 - 2067.87$

5

$\boxed{_4q_1 = -1876.06 \text{ kJ/kg} = \text{HEAT REJECTED FROM CONDENSER}}$

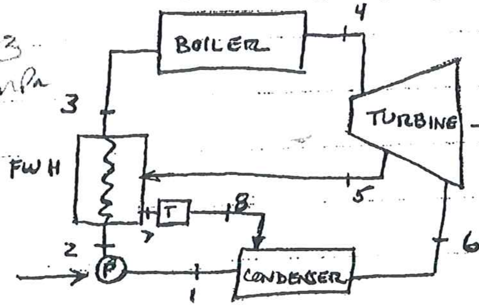
20

(35 pts) 3. An ideal (i.e. isentropic pump and turbine) steam regenerative cycle has steam entering the turbine at 3.5 MPa and 400 °C and entering the condenser at 10 kPa. A portion of the steam is extracted from the turbine at 0.8 MPa for use in a closed feedwater heater. Assume an ideal feedwater heater where the feedwater leaving to enter the boiler is equal in temperature to the saturated liquid leaving the heater to enter the condenser (i.e. $T_3 = T_7$) Condensate from the feedwater heater is drained through the trap to the condenser. Assume saturated liquid entering the pump.

STATE 1
P = 10 MPa
V = .00101

STATE 2
P = 3.5 MPa

STATE 3
P = 3.5 MPa



STATE 4
P = 3.5 MPa
W_T T = 400 °C
h = 3223.2
S =

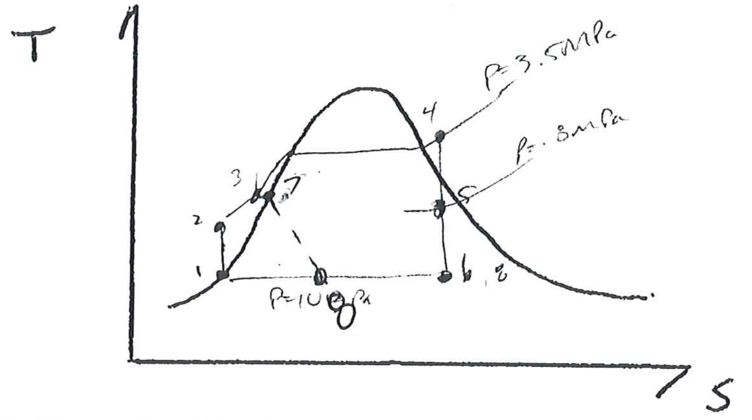
STATE 5
P = .8 MPa

STATE 6
P = 10 kPa

STATE 7
P = 3.5 MPa

STATE 8
P = 10 kPa

(5 pts) a) Sketch the T-S diagram, labeling the states.



3

(5 pts) b) Find the quality of the flow which leaves the turbine and enters the condenser (x_6).

$S_4 = S_6 = 6.8428$

$$x_6 = \frac{6.8428 - .6492}{7.4996}$$

$$\boxed{x_6 = .8259}$$

5

(5 pts) c) Find the temperature of the flow which leaves the FWH and enters the boiler (T_3). (K)

$w_p = \int v\,dP = .00101(10-3500) = -3.5249$

T_{SAT} (0.8 MPa)

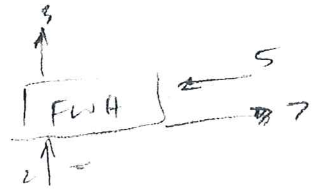
STATE 7
P = 3.5 MPa
$T_{SAT} = 242.56$

$\boxed{T_7 = T_3 = 242.56\ ^\circ C = 515.56\ K}$

1

9

(10 pts) d) Find the ratio of the mass flow extracted from the turbine for the FWH to the mass flow through the pump. (i.e. $\frac{\dot{m}_5}{\dot{m}_3}$). Do NOT give a numerical answer. Leave the answer as a ratio of enthalpies but clearly show at what state the enthalpies are evaluated. Show work.



$$\cancel{\dot{Q}} - \cancel{\dot{W}} + \sum \dot{m}_i(h) - \sum \dot{m}_e(h) = \cancel{\frac{dE_{cv}}{dt}}$$

$$\dot{m}_2 h_2 + \dot{m}_5 h_5 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$

$$\frac{\dot{m}_2 h_2}{\dot{m}_3} + \frac{\dot{m}_5}{\dot{m}_3} h_5 = \frac{\dot{m}_3}{\dot{m}_3} h_3 + \frac{\dot{m}_7}{\dot{m}_3} h_7$$

$$h_2 + y h_5 = h_3 + y h_7$$

$$y h_5 - y h_7 = h_3 - h_2$$

$$\boxed{y = \frac{h_3 - h_2}{h_5 - h_7}}$$

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

$\dot{m}_5 = \dot{m}_7$

$\dot{m}_2 + \dot{m}_5 = \dot{m}_3 + \dot{m}_7$

$\frac{\dot{m}_2}{\dot{m}_3} + \frac{\dot{m}_5}{\dot{m}_3} = 1 + \frac{\dot{m}_7}{\dot{m}_3}$. $y = \frac{\dot{m}_5}{\dot{m}_3}$

$\cancel{\frac{\dot{m}_2}{\dot{m}_3}}^{1} + y = 1 + \frac{\dot{m}_7}{\dot{m}_3}$

$y = \frac{\dot{m}_7}{\dot{m}_3}$

10

(10 pts) e) Find the specific turbine work for the entire turbine (i.e. $\frac{\dot{W}_T}{\dot{m}_3}$). Leave answer only as an expression which includes the flow ratio of part d) and enthalpies. Show work and clearly indicate the state at which the enthalpies should be calculated.

$${}_4\cancel{\dot{Q}}_6 - {}_4\dot{W}_6 + \sum \dot{m}_i(h)_{in} - \sum \dot{m}_e(h)_{out} = \cancel{\frac{dE_{cv}}{dt}}$$

$$-{}_4\dot{W}_6 + \dot{m}_4 h_4 - \dot{m}_5 h_5 - \dot{m}_6 h_6$$

4

ALSO

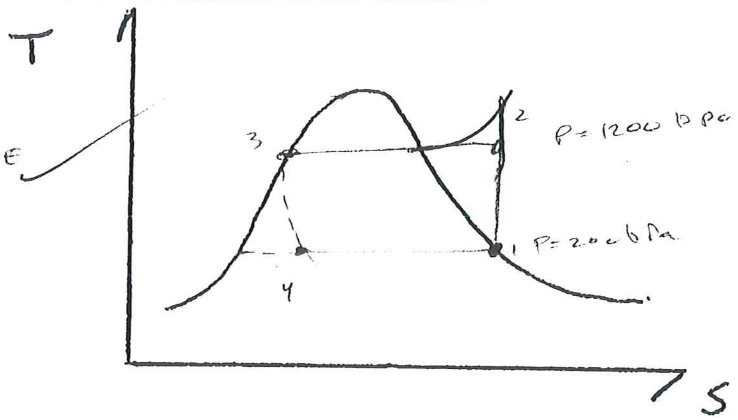
$\dot{m}_4 = \dot{m}_5 + \dot{m}_6$

14

(20 pts) 4. The air conditioner in your car is modeled with a vapor compression refrigeration cycle using R-134a. Refrigerant enters the compressor at 200 kPa and is compressed to 1200 kPa, 60°C. Note that the compressor can be considered adiabatic but not reversible. The required compressor input is 1.5 kW. Assume saturated vapor entering the compressor and a saturated liquid leaving the condenser.

(4 pts) a) On the axis below, sketch the T-S diagram for this cycle, labeling the states and giving the assumptions for each of the four processes.

(1-2) ISENTROPIC COMP.
2-3) CONST. P HEAT REJ
(3-4) ADIABATIC THROTTLE
(4-1) CONST P HEAT ADD.



$T_2 = 60°C$
$P_2 = 1200$ kPa

4

(8 pts) b) Find the mass flow rate of the refrigerant. (kJ/kg)

STATE 1
P= 200 kPa
h = 244.46

STATE 2
P= 1200 kPa
T = 60°C
h = 289.64

$_1q_2 - {}_1w_2 = h_2 - h_1$

$_1w_2 = h_1 - h_2 = 244.46 - 289.64$

$_1w_2 = -45.15$ kJ/kg

$\frac{\dot{W}}{w} = \dot{m}$

UNITS

W

$\dot{m} = \frac{1500}{45.15}$ = $\boxed{0.0332 \text{ kg/s} \quad \dot{m}}$

kJ/kg

6

(8 pts) c) Find the capacity of the system (i.e. the heat transfer rate through the evaporator) and the coefficient of performance. (kW)

STATE 4
P= 200 kPa
$h_4 \approx h_3 =$

STATE 3
P= 1200 kPa
h = 117.77

$_4q_1 = h_1 - h_4$

$_4q_1 = 244.46 - 117.77$

$_4q_1 = 126.69$

$\dot{Q} = 126.69 (33.2) = \boxed{4206.125 \text{ kW} = \dot{Q}}$

ok 8

$COP = \frac{q_L}{w} = \frac{126.69}{45.15} = \boxed{2.806 = COP_R}$

18

Exam I review

D7.

① T = 333 K ② T = ③ ④

P = 90 kPa P =

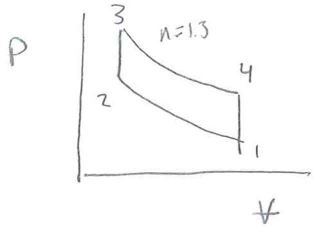
r = 8

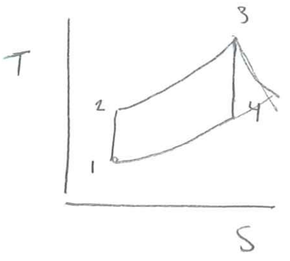
$q_{in} = 2000\ kJ/kg$

Otto cycle

polytropic expansion n = 1.3

const. spec. heats





a) (1-2) Isen comp. $T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{k-1} = 333\,k\,(8)^{1.4-1}$ $\boxed{T_2 = 765.0\,K}$

$P_2 = P_1 \left(\frac{V_1}{V_2}\right)^k$

$= 90(8)^{1.4}$

$\boxed{P_2 = 1654\ kPa}$

b) √ Peak temp: $(T_3) =$

$q_{in} = \cancel{c_p(T_3 - T_2)}\ C_v(T_3 - T_2)$ (V const) $\boxed{T_3 = \ldots}$

c) √ T after expansion

$P_3 V_3^n = P_4 V_4^n$ $P_4 = \frac{RT_4}{V_4}$

$P_3 V_3^n = RT_4 V_4^n$

$T_4 = \frac{RT_3}{V_3 R}\left(\frac{V_3}{V_4}\right)^n = T_3 \left(\frac{V_3}{V_4}\right)^{n-1} = 3554\left(\frac{1}{8}\right)^{0.3} = 1905\,k$

d) √ exp. work

$_3w_4 = \left(\frac{R}{1-n}\right)(T_4 - T_3)$ √

e) net q = ? $= {_1W_2} + {_3w_4}$ $_1q_2 - {_1w_2} = u_2 - u_1$ (no flow) $_1w_2 = u_1 - u_2 = C_v(T_1 - T_2)$

$C_v(T_1 - T_2) + \left(\frac{R}{1-n}\right)(T_4 - T_3)$

D.4 ① 100kPa 298k ② 1MPa 540k $Q_{out} = 50\ kJ/kg$ $T_{amb} = 298k$ Const $C_{p,v}$

a) $${}_1q_2 - {}_1w_2 = c_p(T_2 - T_1)$$

$${}_1w_2 = c_p(T_1 - T_2) - 50kJ/kg \qquad \boxed{{}_1w_2 = -293.2\ kJ/kg}$$

b) min. work = $W_{rev} = \dot{m}\psi_1 - \dot{m}\psi_2$

$$\psi \quad (h_1 - h_2) - T_0(s_1 - s_2) + \cancel{\frac{V^2}{2}}^{0} \qquad (s_1 - s_2) = \left[c_p \ln\left(\frac{T_1}{T_2}\right) - R\ln\left(\frac{P_1}{P_2}\right)\right]$$

$\boxed{-262.1}$ kJ/kg

c)

$$= \frac{293.2}{26} \qquad \frac{262.1}{293.2} = 89.4\%$$

d) isentropic ~~$P_2 = P_1\left(\frac{v_1}{v_2}\right)^k$~~ $T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$ $T_2 = \underline{575.3}$

e)

$$W_{isen} = c_p(T_1 - T_2')$$

$$\eta = \frac{W_{act}}{W_{isen}} \quad \checkmark$$

Exam #1 Review

D 3)

adiabatic turbine

5kg/s 550 kPa 425 k outlet 100kPa 325 k ambient 298k 100kPa

const c_p, 300k

a) exergy flow entering = ?

$V_1 = \frac{RT_1}{P_1}$

~~$x = u - u_0 + P_0(v - v_0) - T_0(s - s_0)$~~

~~$= c_v(T_1 - T_0) + P_0\left(\frac{RT_1}{P_1} - \frac{RT_0}{P_0}\right) - T_0$~~

$$\dot{m}\,\psi = (h_1 - h_0) - T_0(s - s_0) + \cancel{\frac{V^2}{2}}^{\,0}$$

$$= c_p(T_1 - T_0) - T_0\left[c_p \ln\frac{T_1}{T_0} - R\ln\frac{P_1}{P_0}\right]$$

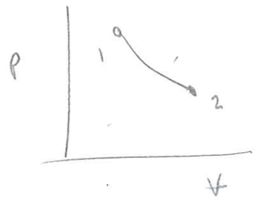
b) actual power

$\underset{0,\text{isent.}}{\cancel{\dot{Q}}} - \dot{W} = \dot{h}_2 - \dot{h}_1$

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

$\dot{W} = \dot{m}\,c_p(T_1 - T_2)$

c) max power $= \dot{m}\Delta\psi = \dot{m}(\psi_2 - \psi_1)$



d) $\frac{\dot{W}_{net}}{\dot{q}_{in}} = \frac{h_1 - h_2 \text{ actual}}{h_1 - h_2 \text{ isentropic}}$

$q_{in} = \dot{m}(h_{in})$

isentropic $\Delta S = 0$

Cycles, Thermo II

	(1–2)	(2–3)	(3–4)	(4–1)
otto	isentropic comp	Const $\forall$ q_{in}	isentropic expand	const $\forall$ q_{out}
carnot	isent. comp.	isothermal q_{in} (reversible)	isentropic expand	Isothermal q_{out} (reversible)
diesel	isent. comp.	const. p. q_{in}	isent. expand	const $\forall$ q_{out}
stirling	isotherm. comp	const $\forall$. q_{in}	isotherm. expand	const $\forall$ q_{out}

Ch. Reviews

The energy content of the universe is constant, just as its mass content is. Yet at times of crisis we are bombarded with speeches and articles on how to "conserve" energy. As engineers, we know that energy is already conserved. What is not conserved is *exergy,* which is the useful work potential of the energy. Once the exergy is wasted, it can never be recovered. When we use energy (to heat our homes for example), we are not destroying any energy; we are merely converting it to a less useful form, a form of less exergy.

The useful work potential of a system at the specified state is called *exergy.* Exergy is a property and is associated with the state of the system and the environment. A system that is in equilibrium with its surroundings has zero exergy and is said to be at the *dead state.* The exergy of heat supplied by thermal energy reservoirs is equivalent to the work output of a Carnot heat engine operating between the reservoir and the environment.

Reversible work W_{rev} is defined as the maximum amount of useful work that can be produced (or the minimum work that needs to be supplied) as a system undergoes a process between the specified initial and final states. This is the useful work output (or input) obtained when the process between the initial and final states is executed in a totally reversible manner. The difference between the reversible work W_{rev} and the useful work W_u is due to the irreversibilities present during the process and is called the *irreversibility I.* It is equivalent to the *exergy destroyed* and is expressed as

$$I = X_{\text{destroyed}} = T_0 S_{\text{gen}} = W_{\text{rev,out}} - W_{u,\text{out}} = W_{u,\text{in}} - W_{\text{rev,in}}$$

where S_{gen} is the entropy generated during the process. For a totally reversible process, the useful and reversible work terms are identical and thus exergy destruction is zero. Exergy destroyed represents the lost work potential and is also called the *wasted work* or *lost work.*

The *second-law efficiency* is a measure of the performance of a device relative to the performance under reversible conditions for the same end states and is given by

$$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\text{th,rev}}} = \frac{W_u}{W_{\text{rev}}}$$

for heat engines and other work-producing devices and

$$\eta_{\text{II}} = \frac{\text{COP}}{\text{COP}_{\text{rev}}} = \frac{W_{\text{rev}}}{W_u}$$

for refrigerators, heat pumps, and other work-consuming devices. In general, the second-law efficiency is expressed as

$$\eta_{\text{II}} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy destroyed}}{\text{exergy supplied}}$$

The exergies of a fixed mass (nonflow exergy) and of a flow stream are expressed as

Nonflow exergy: $\phi = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$

$$= (e - e_0) + P_0(v - v_0) - T_0(s - s_0)$$

Flow exergy: $\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$

Then the *exergy change* of a fixed mass or fluid stream as it undergoes a process from state 1 to state 2 is given by

$$\begin{aligned}\Delta X = X_2 - X_1 &= m(\phi_2 - \phi_1)\\ &= (E_2 - E_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)\\ &= (U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)\\ &\quad + m\frac{V_2^2 - V_1^2}{2} + mg(z_2 - z_1)\end{aligned}$$

$$\begin{aligned}\Delta\psi = \psi_2 - \psi_1 &= (h_2 - h_1) - T_0(s_2 - s_1)\\ &\quad + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\end{aligned}$$

Exergy can be transferred by heat, work, and mass flow, and exergy transfer accompanied by heat, work, and mass transfer are given by

Exergy transfer by heat: $X_{\text{heat}} = \left(1 - \frac{T_0}{T}\right)Q$

Exergy transfer by work: $X_{\text{work}} = \begin{cases} W - W_{\text{surr}} & \text{(for boundary work)}\\ W & \text{(for other forms of work)}\end{cases}$

Exergy transfer by mass: $X_{\text{mass}} = m\psi$

The exergy of an isolated system during a process always decreases or, in the limiting case of a reversible process, remains constant. This is known as the *decrease of exergy principle* and is expressed as

$$\Delta X_{\text{isolated}} = (X_2 - X_1)_{\text{isolated}} \leq 0$$

Exergy balance for *any system* undergoing *any process* can be expressed as

General: $\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}}$

General, rate form: $\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{dX_{\text{system}}/dt}_{\substack{\text{Rate of change}\\ \text{in exergy}}}$

General, unit-mass basis: $(x_{\text{in}} - x_{\text{out}}) - x_{\text{destroyed}} = \Delta x_{\text{system}}$

where

$$\dot{X}_{\text{heat}} = (1 - T_0/T)\dot{Q}$$

$$\dot{X}_{\text{work}} = \dot{W}_{\text{useful}}$$

$$\dot{X}_{\text{mass}} = \dot{m}\psi$$

For a *reversible process,* the exergy destruction term $X_{\text{destroyed}}$ drops out. Taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the general exergy balance relations can be expressed more explicitly as

$$\sum\left(1 - \frac{T_0}{T_k}\right)Q_k - [W - P_0(V_2 - V_1)]$$

$$+ \sum_{\text{in}} m\psi - \sum_{\text{out}} m\psi - X_{\text{destroyed}} = X_2 - X_1$$

$$\sum\left(1 - \frac{T_0}{T_k}\right)\dot{Q}_k - \left(\dot{W} - P_0\frac{dV_{\text{CV}}}{dt}\right)$$

$$+ \sum_{\text{in}} \dot{m}\psi - \sum_{\text{out}} \dot{m}\psi - \dot{X}_{\text{destroyed}} = \frac{dX_{\text{CV}}}{dt}$$

Entropy

The second law of thermodynamics leads to the definition of a new property called *entropy*, which is a quantitative measure of microscopic disorder for a system. Any quantity whose cyclic integral is zero is a property, and entropy is defined as

$$dS = \left(\frac{\delta Q}{T}\right)_{\text{int rev}}$$

For the special case of an internally reversible, isothermal process, it gives

$$\Delta S = \frac{Q}{T_0}$$

The inequality part of the Clausius inequality combined with the definition of entropy yields an inequality known as the *increase of entropy principle*, expressed as

$$S_{\text{gen}} \geq 0$$

where S_{gen} is the *entropy generated* during the process. Entropy change is caused by heat transfer, mass flow, and irreversibilities. Heat transfer to a system increases the entropy, and heat transfer from a system decreases it. The effect of irreversibilities is always to increase the entropy.

The *entropy-change* and *isentropic relations* for a process can be summarized as follows:

1. *Pure substances:*

Any process: $\Delta s = s_2 - s_1$

Isentropic process: $s_2 = s_1$

2. *Incompressible substances:*

Any process: $s_2 - s_1 = c_{\text{avg}} \ln \frac{T_2}{T_1}$ (7-29)

Isentropic process: $T_2 = T_1$

3. *Ideal gases:*

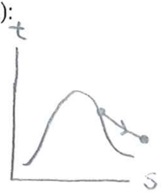
a. Constant specific heats (approximate treatment):

Any process:

no

$$s_2 - s_1 = c_{v,\text{avg}} \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1}$$

$$s_2 - s_1 = c_{p,\text{avg}} \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}$$



$Q_{\text{int rev}} = \int_1^2 T\,ds$
$= T_{\text{avg}}(s_2 - s_1)$

Fusion/evap
$s = \dot{m}\, s_{fg}$

Isentropic process:

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

$$\left(\frac{T_2}{T_1}\right)_{s=\text{const.}} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\left(\frac{T_2}{T_1}\right)_{s=\text{const.}} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$\left(\frac{P_2}{P_1}\right)_{s=\text{const.}} = \left(\frac{v_1}{v_2}\right)^{k}$$

~~*b.* Variable specific heats (exact treatment):~~

~~Any process:~~

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

Isentropic process:

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

~~$\left(\frac{P_2}{P_1}\right)_{s=\text{const.}} = \frac{P_{r2}}{P_{r1}}$~~

~~$\left(\frac{v_2}{v_1}\right)_{s=\text{const.}} = \frac{v_{r2}}{v_{r1}}$~~

where P_r is the *relative pressure* and v_r is the *relative specific volume*. The function s° depends on temperature only.

The *steady-flow work* for a reversible process can be expressed in terms of the fluid properties as

$$w_{\text{rev}} = -\int_1^2 v\,dP - \Delta\text{ke} - \Delta\text{pe}$$

For incompressible substances (v = constant) it simplifies to

$$w_{\text{rev}} = -v(P_2 - P_1) - \Delta\text{ke} - \Delta\text{pe}$$

The work done during a steady-flow process is proportional to the specific volume. Therefore, v should be kept as small as possible during a compression process to minimize the work input and as large as possible during an expansion process to maximize the work output.

The reversible work inputs to a compressor compressing an ideal gas from T_1, P_1 to P_2 in an isentropic (Pv^k = constant), polytropic (Pv^n = constant), or isothermal (Pv = constant) manner, are determined by integration for each case with the following results:

Isentropic: $w_{\text{comp,in}} = \frac{kR(T_2 - T_1)}{k-1} = \frac{kRT_1}{k-1}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]$

Polytropic: $w_{\text{comp,in}} = \frac{nR(T_2 - T_1)}{n-1} = \frac{nRT_1}{n-1}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right]$

ex (7-13) p 374

Isothermal: $w_{\text{comp,in}} = RT \ln \frac{P_2}{}$

The work input to a compressor can be reduced by using multistage compression with intercooling. For maximum savings from the work input, the pressure ratio across each stage of the compressor must be the same.

Most steady-flow devices operate under adiabatic conditions, and the ideal process for these devices is the isentropic process. The parameter that describes how efficiently a device approximates a corresponding isentropic device is called *isentropic* or *adiabatic efficiency*. It is expressed for turbines, compressors, and nozzles as follows:

$$\eta_T = \frac{\text{Actual turbine work}}{\text{Isentropic turbine work}} = \frac{w_a}{w_s} \cong \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$$

$$\eta_C = \frac{\text{Isentropic compressor work}}{\text{Actual compressor work}} = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

$$\eta_N = \frac{\text{Actual KE at nozzle exit}}{\text{Isentropic KE at nozzle exit}} = \frac{V_{2a}^2}{V_{2s}^2} \cong \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$$

In the relations above, h_{2a} and h_{2s} are the enthalpy values at the exit state for actual and isentropic *processes*, respectively.

The entropy balance for any system undergoing any process can be expressed in the general form as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

or, in the *rate form*, as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy} \\ \text{transfer by} \\ \text{heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{dS_{\text{system}}/dt}_{\substack{\text{Rate of change} \\ \text{in entropy}}}$$

For a general *steady-flow process* it simplifies to

$$\dot{S}_{\text{gen}} = \sum \dot{m}_e s_e - \sum \dot{m}_i s_i - \sum \frac{\dot{Q}_k}{T_k}$$

2nd Law, heat engines, heat Pumps

The *second law of thermodynamics* states that processes occur in a certain direction, not in any direction. A process does not occur unless it satisfies both the first and the second laws of thermodynamics. Bodies that can absorb or reject finite amounts of heat isothermally are called *thermal energy reservoirs* or *heat reservoirs*.

Work can be converted to heat directly, but heat can be converted to work only by some devices called *heat engines*. The *thermal efficiency* of a heat engine is defined as

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_H} = 1 - \frac{Q_L}{Q_H}$$

where $W_{\text{net,out}}$ is the net work output of the heat engine, Q_H is the amount of heat supplied to the engine, and Q_L is the amount of heat rejected by the engine.

Refrigerators and heat pumps are devices that absorb heat from low-temperature media and reject it to higher-temperature ones. The performance of a refrigerator or a heat pump is expressed in terms of the *coefficient of performance*, which is defined as

$$\text{COP}_{\text{R}} = \frac{Q_L}{W_{\text{net,in}}} = \frac{1}{Q_H/Q_L - 1}$$

$$\text{COP}_{\text{HP}} = \frac{Q_H}{W_{\text{net,in}}} = \frac{1}{1 - Q_L/Q_H}$$

The *Kelvin–Planck statement* of the second law of thermodynamics states that no heat engine can produce a net amount of work while exchanging heat with a single reservoir only. The *Clausius statement* of the second law states that no device can transfer heat from a cooler body to a warmer one without leaving an effect on the surroundings.

Any device that violates the first or the second law of thermodynamics is called a *perpetual-motion machine*.

A process is said to be *reversible* if both the system and the surroundings can be restored to their original conditions. Any other process is *irreversible*. The effects such as friction, non-quasi-equilibrium expansion or compression, and heat transfer through a finite temperature difference render a process irreversible and are called *irreversibilities*.

The *Carnot cycle* is a reversible cycle that is composed of four reversible processes, two isothermal and two adiabatic. The *Carnot principles* state that the thermal efficiencies of all reversible heat engines operating between the same two reservoirs are the same, and that no heat engine is more efficient than a reversible one operating between the same two re voirs. These statements form the basis for establishing a *the modynamic temperature scale* related to the heat transfe between a reversible device and the high- and low-temperatu reservoirs by

$$\left(\frac{Q_H}{Q_L}\right)_{\text{rev}} = \frac{T_H}{T_L}$$

Therefore, the Q_H/Q_L ratio can be replaced by T_H/T_L fo reversible devices, where T_H and T_L are the absolute tempera tures of the high- and low-temperature reservoirs, respective

A heat engine that operates on the reversible Carnot cycle called a *Carnot heat engine*. The thermal efficiency of Carnot heat engine, as well as all other reversible heat engin is given by

$$\eta_{\text{th,rev}} = 1 - \frac{T_L}{T_H}$$

This is the maximum efficiency a heat engine operatin between two reservoirs at temperatures T_H and T_L can have

The COPs of reversible refrigerators and heat pumps a given in a similar manner as

$$\text{COP}_{\text{R,rev}} = \frac{1}{T_H/T_L - 1}$$

and

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L/T_H}$$

Again, these are the highest COPs a refrigerator or a heat pum operating between the temperature limits of T_H and T_L can hav

$$W_{rev,in} = \frac{\dot{Q}_H}{COP_{HP}}$$

$$1\,hp = 2544.5\,\frac{Btu}{h} = .7068\,\frac{Btu}{s}$$

Mass & Energy, Control Volumes

The *conservation of mass principle* states that the net mass transfer to or from a system during a process is equal to the net change (increase or decrease) in the total mass of the system during that process, and is expressed as

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \text{and} \quad \dot{m}_{\text{in}} - \dot{m}_{\text{out}} = dm_{\text{system}}/dt$$

where $\Delta m_{\text{system}} = m_{\text{final}} - m_{\text{initial}}$ is the change in the mass of the system during the process, $\dot{m}_{\text{in}}$ and $\dot{m}_{\text{out}}$ are the total rates of mass flow into and out of the system, and dm_{system}/dt is the rate of change of mass within the system boundaries. The relations above are also referred to as the *mass balance* and are applicable to any system undergoing any kind of process.

The amount of mass flowing through a cross section per unit time is called the *mass flow rate*, and is expressed as

$$\dot{m} = \rho V A$$

where ρ = density of fluid, V = average fluid velocity normal to A, and A = cross-sectional area normal to flow direction. The volume of the fluid flowing through a cross section per unit time is called the *volume flow rate* and is expressed as

$$\dot{V} = VA = \dot{m}/\rho$$

The work required to push a unit mass of fluid into or out of a control volume is called *flow work* or *flow energy*, and is expressed as $w_{\text{flow}} = Pv$. In the analysis of control volumes, it is convenient to combine the flow energy and internal energy into *enthalpy*. Then the total energy of a flowing fluid is expressed as

$$\theta = h + \text{ke} + \text{pe} = h + \frac{V^2}{2} + gz$$

The total energy transported by a flowing fluid of mass m with uniform properties is $m\theta$. The rate of energy transport by a fluid with a mass flow rate of $\dot{m}$ is $\dot{m}\theta$. When the kinetic and potential energies of a fluid stream are negligible, the amount and rate of energy transport become $E_{\text{mass}} = mh$ and $\dot{E}_{\text{mass}} = \dot{m}h$, respectively.

The *first law of thermodynamics* is essentially an expression of the conservation of energy principle, also called the *energy balance*. The general mass and energy balances for *any system* undergoing *any process* can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Changes in internal, kinetic,}\\ \text{potential, etc., energies}}}$$

It can also be expressed in the *rate form* as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc., energies}}}$$

Thermodynamic processes involving control volumes can be considered in two groups: steady-flow processes and unsteady-flow processes. During a *steady-flow process*, the fluid flows through the control volume steadily, experiencing no change with time at a fixed position. The mass and energy content of the control volume remain constant during a steady-flow process. Taking heat transfer *to* the system and work done *by* the system to be positive quantities, the conservation of mass and energy equations for steady-flow processes are expressed as

$$\sum_{\text{in}} \dot{m} = \sum_{\text{out}} \dot{m}$$

$$\dot{Q} - \dot{W} = \underbrace{\sum_{\text{out}} \dot{m}\left(h + \frac{V^2}{2} + gz\right)}_{\text{for each exit}} - \underbrace{\sum_{\text{in}} \dot{m}\left(h + \frac{V^2}{2} + gz\right)}_{\text{for each inlet}}$$

These are the most general forms of the equations for steady-flow processes. For single-stream (one-inlet–one-exit) systems such as nozzles, diffusers, turbines, compressors, and pumps, they simplify to

$$\dot{m}_1 = \dot{m}_2 \rightarrow \frac{1}{v_1} V_1 A_1 = \frac{1}{v_2} V_2 A_2$$

$$\dot{Q} - \dot{W} = \dot{m}\left[h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right]$$

In these relations, subscripts 1 and 2 denote the inlet and exit states, respectively.

Most unsteady-flow processes can be modeled as a *uniform-flow process*, which requires that the fluid flow at any inlet or exit is uniform and steady, and thus the fluid properties do not change with time or position over the cross section of an inlet or exit. If they do, they are averaged and treated as constants for the entire process. When kinetic and potential energy changes associated with the control volume and the fluid streams are negligible, the mass and energy balance relations for a uniform-flow system are expressed as

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}}$$

$$Q - W = \sum_{\text{out}} mh - \sum_{\text{in}} mh + (m_2 u_2 - m_1 u_1)_{\text{system}}$$

where $Q = Q_{\text{net,in}} = Q_{\text{in}} - Q_{\text{out}}$ is the net heat input and $W = W_{\text{net,out}} = W_{\text{out}} - W_{\text{in}}$ is the net work output.

When solving thermodynamic problems, it is recommended that the general form of the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ be used for all problems, and simplify it for the particular problem instead of using the specific relations given here for different processes.

Potential energy!

Kinetic energy:

$$\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2} \qquad \frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}$$

Homeworks, ~85 pages (see ME221_applied_thermo_binder2.pdf)

Energy, Closed Systems

Chapter 4

Work is the energy transferred as a force acts on a system through a distance. The most common form of mechanical work is the *boundary work,* which is the work associated with the expansion and compression of substances. On a P-V diagram, the area under the process curve represents the boundary work for a quasi-equilibrium process. Various forms of boundary work are expressed as follows:

(1) General $\quad W_b = \int_1^2 P\,dV$

(2) Isobaric process (surroundings)

$$W_b = P_0(V_2 - V_1) \quad (P_1 = P_2 = P_0 = \text{constant})$$

(3) Polytropic process

$$W_b = \frac{P_2V_2 - P_1V_1}{1-n} \quad (n \neq 1) \quad (PV^n = \text{constant})$$

(4) Isothermal process of an ideal gas (T const)

$$W_b = P_1V_1 \ln\frac{V_2}{V_1} = mRT_0 \ln\frac{V_2}{V_1} \quad (PV = mRT_0 = \text{constant})^{\text{or } \frac{P_1}{P_2}}$$

The first law of thermodynamics is essentially an expression of the conservation of energy principle, also called the energy balance. The general energy balances for *any system* undergoing *any process* can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc., energies}}} \quad (\text{kJ})$$

It can also be expressed in the *rate form* as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt}_{\substack{\text{Rate of change in internal,}\\ \text{kinetic, potential, etc., energies}}} \quad (\text{kW})$$

Taking heat transfer *to* the system and work done *by* the system to be positive quantities, the energy balance for a closed system can also be expressed as

$$Q - W = \Delta U + \Delta \text{KE} + \Delta \text{PE} \quad (\text{kJ})$$

where

$$W = W_{\text{other}} + W_b$$
$$\Delta U = m(u_2 - u_1)$$
$$\Delta \text{KE} = \tfrac{1}{2}m(V_2^2 - V_1^2)$$
$$\Delta \text{PE} = mg(z_2 - z_1)$$

For a *constant-pressure process,* $W_b + \Delta U = \Delta H$. Thus,

$$Q - W_{\text{other}} = \Delta H + \Delta \text{KE} + \Delta \text{PE} \quad (\text{kJ})$$

The amount of energy needed to raise the temperature of a unit mass of a substance by one degree is called the *specific heat at constant volume* c_v for a constant-volume process and the *specific heat at constant pressure* c_p for a constant-pressure process. They are defined as

$$c_v = \left(\frac{\partial u}{\partial T}\right)_v \quad \text{and} \quad c_p = \left(\frac{\partial h}{\partial T}\right)_p$$

For ideal gases u, h, c_v, and c_p are functions of temperature alone. The Δu and Δh of ideal gases are expressed as

$$\Delta u = u_2 - u_1 = \int_1^2 c_v(T)\,dT \cong c_{v,\text{avg}}(T_2 - T_1)$$

$$\Delta h = h_2 - h_1 = \int_1^2 c_p(T)\,dT \cong c_{p,\text{avg}}(T_2 - T_1)$$

For ideal gases, c_v and c_p are related by

$$c_p = c_v + R \quad (\text{kJ/kg}\cdot\text{K})$$

where R is the gas constant. The *specific heat ratio* k is defined as

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

For *incompressible substances* (liquids and solids), both the constant-pressure and constant-volume specific heats are identical and denoted by c:

$$c_p = c_v = c \quad (\text{kJ/kg}\cdot\text{K})$$

The Δu and Δh of imcompressible substances are given by

$$\Delta u = \int_1^2 c(T)\,dT \cong c_{\text{avg}}(T_2 - T_1) \quad (\text{kJ/kg})$$

$$\Delta h = \Delta u + v\Delta P \quad (\text{kJ/kg})$$

Isothermal = T-const.

adiabatic = no heat transf. w/surroundings

Air turbine Isentropic

$\dot{E}_{in} = \dot{E}_{out}$

$m(h_1 + \frac{V^2}{2}) = m(h_2 + \frac{V_2^2}{2}) + W_{out}$

ΔQ Water, liquid only Temp change

$= m\Delta h = mc_p(\Delta T)$

Compressed water:

$h = h_f\langle T\rangle + v\langle T\rangle(P - P_{sat}\langle T\rangle)$