Problems for Chapter 8: Power Cycles

A. The Rankine cycle

8.A-1 The purpose of this problem is to determine the optimum pressure at which to reheat steam in the two-stage non-ideal Rankine cycle with reheat, shown in Figure 8-13(a). The cycle is operating with water. The hot reservoir temperature is T_H = 825 K and the cold reservoir is T_C = 310 K. The boiler pressure is P_b = 6.11 MPa. The pressure drop in the heat exchangers can be ignored. However, the condenser has an approach temperature difference of ΔT_{cond} = 5 K, the boiler has an approach temperature difference of ΔT_b = 25 K, and the reheater has an approach temperature difference of ΔT_{rh} = 20 K. The efficiency of the pump is η_p = 0.5. The efficiencies of the high and low pressure turbines are $\eta_{HPt} = \eta_{LPt}$ = 0.9.

a.) Initially, assume that the reheat pressure is P_{rh} = 1 MPa. Determine the thermodynamic states at all points in the cycle and the efficiency of the power plant.

b.) Plot the cycle efficiency as a function of the reheat pressure in order to identify the optimum reheat pressure.

8.A-14 Figure 8.A-14 shows a Rankine steam cycle in which a double-extraction turbine is used to supply both power and process steam for an industrial plant. The energy source for the boiler is natural gas which has a heating value of 50,000 kJ/kg. Thermodynamic data for all points in the cycle are provided in Table 8.A-14. The process steam is supplied at state 4 at 1250 kPa, 280°C at 45 kg/s. All of the process steam is returned as condensate at state 9 and mixed with the condensate from the condenser before the combine flow enters the feedwater heater. The surroundings are at 25°C, 1 atm.

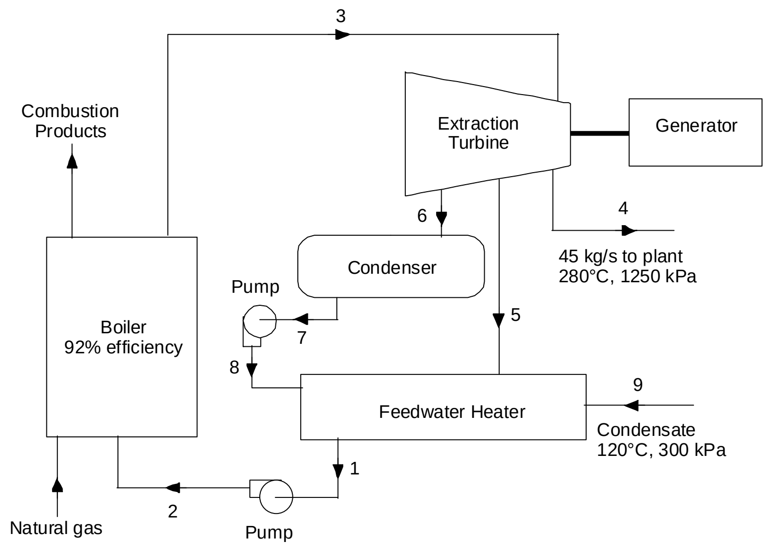

Figure 8.A-14 Steam plant with a double-extraction turbine

Table 8.A-14: Property information at each state

	$\dot{m}$ kg/s	T °C	P kPa	h kJ/kg	s kJ/kg-K	v m^3/kg
1	96.4	130	300	546.4	1.635	0.00107
2	96.4	132.7	3020	559.8	1.66	0.001071
3	96.4	400	3000	3231	6.921	0.09935
4	45	280	1250	3000	6.932	0.1968
5	11.4	160	300	2782	7.127	0.6506
6	40	36.2	6	2567	8.329	23.74
7	40	30	6	125.7	0.4365	0.001004
8	40	32	300	134.3	0.4639	0.001005
9	45	120	300	503.8	1.528	0.00106

a.) Determine the power supplied to the generator, assuming the turbine to be adiabatic.
b.) Determine the efficiency of the main boiler pump, located between states 1 and 2.
c.) Determine the rate of heat transfer between the feedwater heater and the surroundings.
d.) What is the rate at which gas is provided if the boiler efficiency is 0.92?
e.) Determine the Second-Law efficiency of this plant. Assume that the exergy of the natural gas is equal to its heating value.
f.) Determine the rate of exergy destruction in the boiler
g.) Determine the rate of exergy destruction in the feedwater heater.

8.A-2 Figure 8.A-2 illustrates a Rankine cycle with an open feedwater heater that uses water as the working fluid.

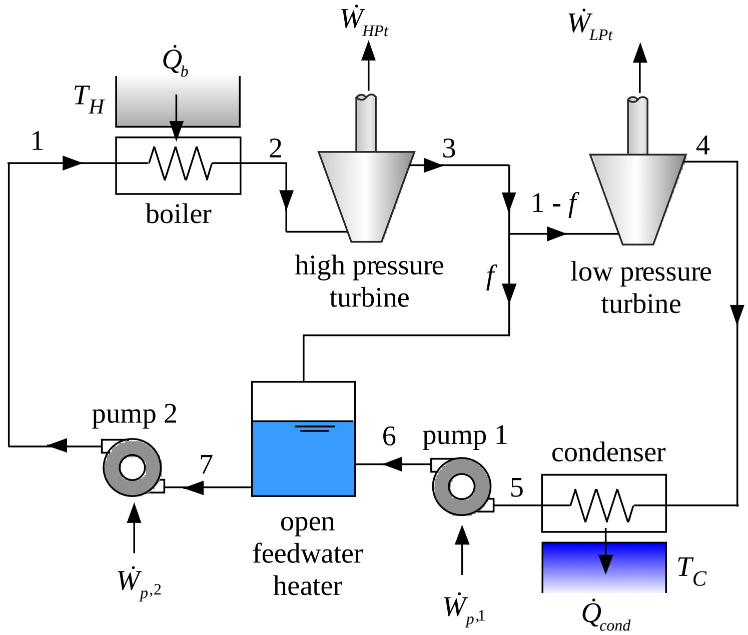

Figure 8.A-2: Rankine cycle with reheat.

The boiler receives heat at a rate of $\dot{Q}_b$ = 100 kW from a source at T_H = 800 K and the condenser rejects heat to a sink at T_C = 325 K. The thermodynamic states associated with the cycle are listed in Table 8.A-6.

Table 8.A-6: Thermodynamic states associated with the cycle.

State	Temperature (K)	Pressure (Pa)	Enthalpy (J/kg)	Entropy (J/kg-K)	Quality
1	453.3	3.000E+06	765052	2139	-
2	800	3.000E+06	3.517E+06	7311	-
3	626.1	1000000	3.164E+06	7311	-
4	325	13523	2.356E+06	7311	0.8999
5	325	13523	217064	727.6	0
6	325.4	1000000	219530	732.1	-
7	453	1000000	762798	2139	0

a.) What is the net power produced by the cycle?
b.) What is the efficiency of the cycle?
c.) What is the rate at which power is being lost due to entropy generation in this cycle (W)?
d.) What is the efficiency of the high pressure turbine?
e.) Estimate the efficiency of pump 1. For your calculation, assume that liquid water is incompressible with specific volume v_w = 0.001 m^3/kg and specific heat capacity c_w = 4200 J/kg-K.

8.A-3 Geothermal-based electrical power production is one alternative energy source that is attracting some attention. Cool water is pumped into the ground (several kilometers deep) where it is heated by geothermal energy and then returned to the surface. The hot water returning from the ground is used as the heat source to power some type of power plant. The down-side of geothermal energy is that heat is provided at low temperature (relative to burning coal or other, more conventional sources of energy). Even in the western U.S., where the geothermal resource is the best, the temperature of the ground does not exceed 250°C. In this problem you will try to identify an optimal working fluid for use in a low-temperature Rankine cycle that utilizes geothermal heat. In order to quickly change working fluids, define a string variable, `F$`, to be the name of the fluid; for example, if you want to use water then assign `F$='Water'`. Using this technique, each time you want to evaluate a property, use `F$` to specify the substance. For example: `h[1]=Enthalpy(F$,T=T[1],P=P[1])`. Figure 8-4 illustrates a simple Rankine cycle. Assume that the components are ideal (i.e., reversible turbine and compressor and zero approach temperature difference for the boiler and condenser). Further, assume that the boiler pressure is selected so that the quality of the fluid leaving the turbine is sufficiently high so as to prevent damage, x_3 = 0.9.

a.) Develop a model of the ideal Rankine cycle using water assuming T_C = 30°C and T_H = 200°C. What is the efficiency of the cycle?

b.) Plot the efficiency of the cycle as a function of T_H for the range of temperatures of interest for geothermal systems, 150°C < T_H < 250°C. Overlay on your plot the efficiency for other potential working fluids (e.g., toluene, propylene, and ammonia).

8.A-4 Figure 8.A-4(a) illustrates a Rankine cycle with reheat. Water leaves the condenser as saturated liquid at $T_6 = T_C = 30°C$. The water is compressed in a reversible pump to $P_1 = P_b = 8.0$ MPa. The boiler heats the water at constant pressure to $T_2 = T_H = 600°C$. The water leaves the boiler and passes through the high pressure turbine where it is expanded to $P_3 = P_{reheat} = 0.7$ MPa. The efficiency of the high pressure turbine is $\eta_{HPT} = 0.87$. The water is reheated at constant pressure to $T_4 = T_H = 600°C$ and then expanded in the low pressure turbine to the condenser pressure, $P_5 = P_6$. The efficiency of the low pressure turbine is $\eta_{LPT} = 0.89$.

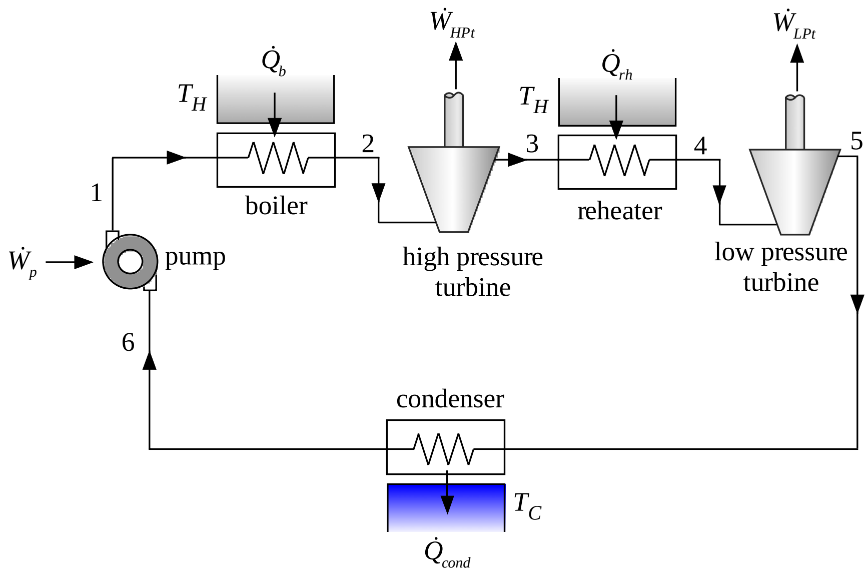

Figure 8.A-4(a): Rankine cycle with reheat.

a.) Create an arrays table that includes the pressure, temperature, enthalpy, and entropy of each of the states 1 through 6 in Figure 8.A-4. Determine the energy transfer per unit of mass flow rate for each of the components.
b.) Determine the efficiency of the power plant.
c.) If the net power output of the power plant is $\dot{W}_{net}$ = 500 MW then what is the mass flow rate passing through each component?
d.) Create a plot showing the power plant efficiency as a function of P_{reheat}.
e.) What is the optimal reheat pressure? Use the Min/Max capability in EES to find the precise value of the reheat pressure.

8.A-5 A basic Rankine cycle operates at steady-state with water as the working fluid. A schematic of the cycle is shown in Figure 8-4 of the text, which identifies the state points. At the design point, the pressure and temperature at the boiler outlet (state 2) are P_2 = 100 bar, T_2 = 520°C and the mass flow rate of the water is $\dot{m}$ = 376 kg/s. The pressure at the turbine outlet is P_3 = 0.08 bar. The water leaving the condenser at state 4 is saturated liquid. The isentropic efficiencies of the turbine and pump are η_t = 0.84 and η_p = 0.47, respectively. The heat input to the boiler is from a source at T_H = 800°. Cooling water at 1 atm enters the condenser at $T_{c,in}$ = 20°C and exits at $T_{c,out}$ = 35°C. Assume that the pressure drops through boiler, condenser, and the piping are negligible. Also assume that heat losses from the jackets of the equipment and the piping are negligible.

a.) Calculate the following quantities for steady-state operation at the design point:

- the mass flow rate of the cooling water (in kg/s),
- the pump power (in kW),
- the turbine power (in kW),
- the thermal efficiency of the cycle,
- the rate of entropy production in the boiler, turbine, condenser and pump (in kW/K).

b.) Plot the cycle on a temperature-entropy diagram in EES and identify states 1 through 4.

c.) The plant management is interested in possible ways to increase the efficiency of this plant. To help them with their analyses, please plot the net power produced by the plant (turbine less pump) as a function of the boiler pressure for boiler pressures between 80 and 120 bar and the net power as a function of boiler outlet temperatures ranging between 460°C and 580°C. Based on these plots, what recommendations can you offer the plant management?

8.A-6 In a geothermal system, water is pumped into the ground (several kilometers deep) where it is heated by geothermal energy and then returned to the surface to provide energy for a power plant. In 2005, the U.S. had 2851 MW of installed geothermal power production; most of this capacity is in California and Nevada. This problem examines the use of geothermal energy for power production with the Rankine cycle shown in Figure 8.A-6. The cycle employs both reheat and regeneration with two open feedwater heaters.

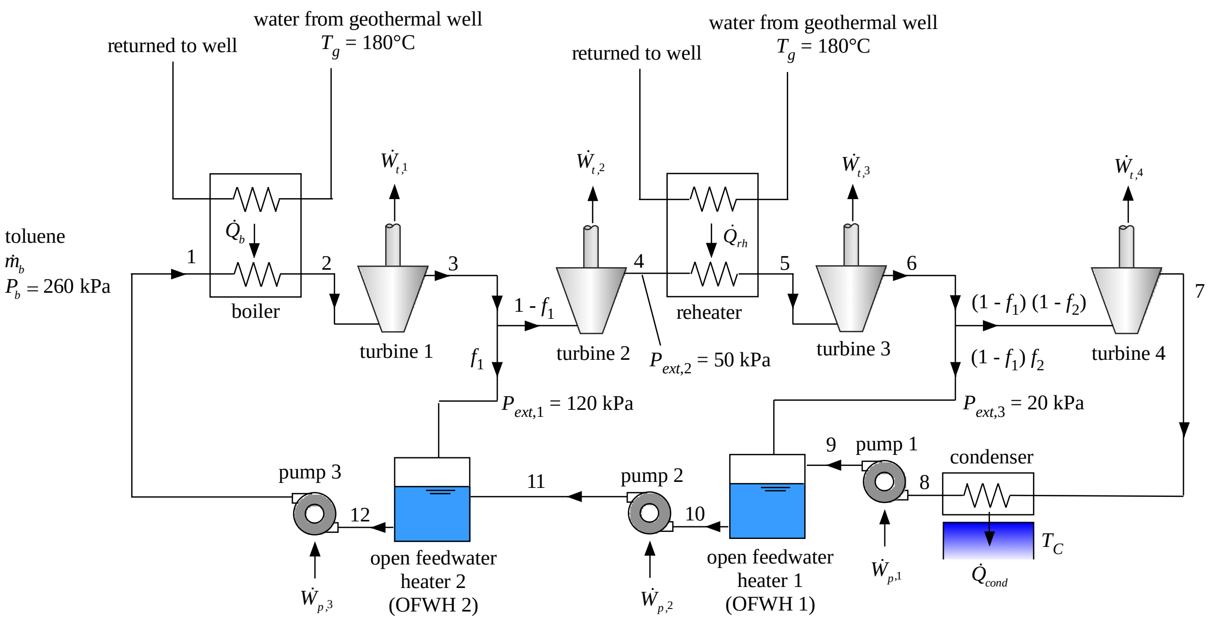

Figure 8.A-6: Schematic of the power plant cycle consisting of a Rankine cycle with reheat and regeneration using two open feedwater heaters.

The Rankine cycle uses toluene as the working fluid due to the low temperature of the heat source. The boiler pressure is P_b = 260 kPa. The water returning from the geothermal well is at T_g = 180°C. The plant rejects heat to a temperature reservoir at T_C = 30°C. The fluid is extracted at state 3 from turbine 1 at $P_{ext,1}$ = 120 kPa and a fraction of the flow f_1 is fed to the open feedwater heater 1. The fluid is subsequently extracted at state 4 from turbine 2 at $P_{ext,2}$ = 50 kPa and reheated. The fluid is finally extracted at state 6 from turbine 3 at $P_{ext,3}$ = 20 kPa and the fraction of the flow f_2 is fed to open feedwater heater 2. The remainder of the flow passes through turbine 4 to the condenser. The efficiency of the four turbines are $\eta_{t,1}$ = 0.85, $\eta_{t,2}$ = 0.86, $\eta_{t,3}$ = 0.88, and $\eta_{t,4}$ = 0.89. The flow leaving the condenser is pumped to $P_{ext,3}$ with pump 1 having efficiency $\eta_{p,1}$ = 0.65. Saturated liquid is pulled from open feedwater heater 1 and pumped to the first extraction pressure, $P_{ext,1}$, with pump 2 having efficiency $\eta_{p,2}$ = 0.67. Saturated liquid is pulled from open feedwater heater 2 and pumped to the boiler pressure with pump 3 having efficiency $\eta_{p,3}$ = 0.69. The approach temperature differences associated with the boiler and the reheater are ΔT_b = 15 K and ΔT_{rh} = 10 K, respectively. The pinch points for both of these heat exchangers occur at their warm end. Therefore, water leaves the boiler at $T_2 = T_{hf,in} - \Delta T_b$ and it leaves the reheater at $T_5 = T_{hf,in} - \Delta T_{rh}$. The approach temperature difference associated with the condenser is ΔT_{cond} = 5 K. Neglect pressure loss for all of the heat exchangers.

a.) Develop two procedures to facilitate the analysis. One procedure should be capable of analyzing any of the four turbines in the system and the other should be applicable to any of the three pumps in the system.

b.) Determine each of the states associated with the cycle. Print out an Arrays table that includes at least the entropy, enthalpy, temperature, and pressure for each state. Plot your states on a *T-s* diagram for Toluene. Label each of your states.
c.) Determine the efficiency of the cycle.
d.) The total rate at which heat that can be extracted from the geothermal source is $\dot{Q}_g$ = 2.5 MW (this is the sum of the heat transfer to the boiler and reheater). Determine the mass flow rate of toluene passing through the boiler and the net power produced by this power plant.
e.) Determine the value of the electricity produced by the plant over a *time* = 10 year period. Assume that you can sell the electricity to the power company at a rate of *ec* = 0.055$/kW-hr and neglect the time value of money.
f.) Determine the effectiveness of the boiler and the condenser.

The surface area required for the boiler and condenser are directly related to their effectiveneness. Your company has thoroughly tested heat exchangers operating with toluene and developed the following correlation for the surface area (A) as a function of the effectiveness (ε) and toluene mass flow rate ($\dot{m}$):

$$A = K_{HX}\ \dot{m}\left[-\ln\left(1-\varepsilon\right)\right]$$

where K_{HX} = 170 m^2-s/kg is an empirical constant that is appropriate for both the boiler and condenser. The cost of these heat exchangers scales linearly with their surface area according to:

$$Cost = C_{HX}\ A$$

where C_{HX} = 50 \$/m^2 is the cost coefficient that is also appropriate for both the boiler and condenser.

g.) Determine the size (surface area) and cost of the boiler and the condenser.
h.) You have estimated that the capital cost of the balance of the plant (the turbines, pumps, reheater, etc.) is $Cost_{mech}$ = 500x10^3 \$. What is the net profit that you make over 10 years by building and operating the plant? Neglect the time value of money.
i.) Plot the profit as a function of the approach temperature difference for the boiler for 2 K < ΔT_b < 30 K. You should see that there is an optimal value of the boiler approach temperature difference. Explain why this is true.
j.) Use the Min/Max capability in EES to determine the optimal (from an economic standpoint) value of the condenser and boiler approach temperature differences. Note that you will need to provide reasonable bounds for these values (e.g., 2 K to 30 K).

8.A-7 Cogeneration is important in the paper industry because paper manufacturing requires large amounts of hot water, low pressure steam, and electricity. In a particular case, 50,000 lb_m/hr of steam at 170 psia and 9000 kW of electrical power are needed in addition to a hot water supply at 180°F as shown in Figure 8.A-7. These needs can be supplied by the system proposed in the figure.

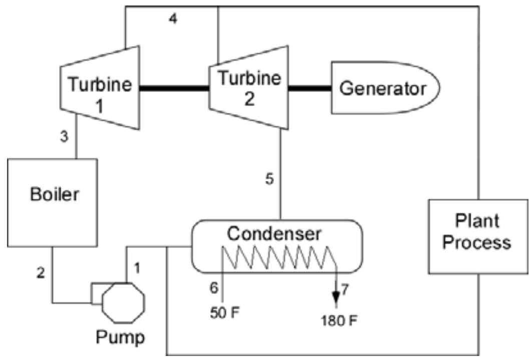

Figure 8.A-7: Cogeneration plant for the paper industry

In the proposed system water is heated in a boiler to 700°F, 600 psia (state 3) and then expanded in a first stage turbine. 50,000 lb_m/hr of steam that exits the first stage turbine is tapped for use in the plant process. The remainder is expanded in the second stage turbine to 20 psia. This relatively high condenser pressure allows water to be heated from 50°F to 180°F in the condenser. State 1 is saturated liquid at 20 psia. The water is returned from the plant process as saturated liquid at 20 psia. Pressure losses in the piping and heat exchange equipment is negligible. Other operational data are as follows:

Isentropic efficiency of the turbines	0.78
Isentropic efficiency of the pump	0.60
Generator efficiency	0.94
Boiler efficiency (based on HHV)	0.92
Higher heating value (HHV) of fuel	14,090 Btu/lb_m
Exergy of the fuel	14,110 Btu/lb_m

a.) Determine the steady state mass flow rate of steam through the boiler and the state properties at all 7 locations.
b.) Determine the mass flow rate of the 180°F water at state 7.
c.) Determine the required mass flow rate of fuel
d.) Determine the (First-Law) efficiency of this plant
e.) Determine the Second-Law efficiency of this plant.
f.) The electricity produced by the plant has a value of 0.10/kW-hr. What are the values of the hot water and the steam provided to the plant process in \$/1000 lb_m.

8.A-8 The purpose of this problem is to investigate the advantages/disadvantages of the regeneration modification in Figure 8.A-8 for a particular case in which combustion gas having an average specific heat capacity of c_p=1050 J/kg-K is supplied at 925 K, 1 atm (state 10) and used to superheat steam to 700 K, 4100 kPa in the boiler (state 1). The boiler is sized so that there is a minimum of 10 K difference between the combustion gas and water temperatures at any point in the boiler. This pinch point temperature difference occurs between state 11 for the combustion gas and state 9 for the steam where state 9 is saturated liquid at the boiler pressure. After leaving the boiler, the combustion gas is discharged to the environment at state 12. The turbine efficiency is 0.84 (for both stages) and pump efficiencies are 0.56. The condenser pressure is maintained at 7 kPa. The system produces 8 MW of power with a generator efficiency of 0.94. Neglect pressure losses in the heat exchangers and piping.

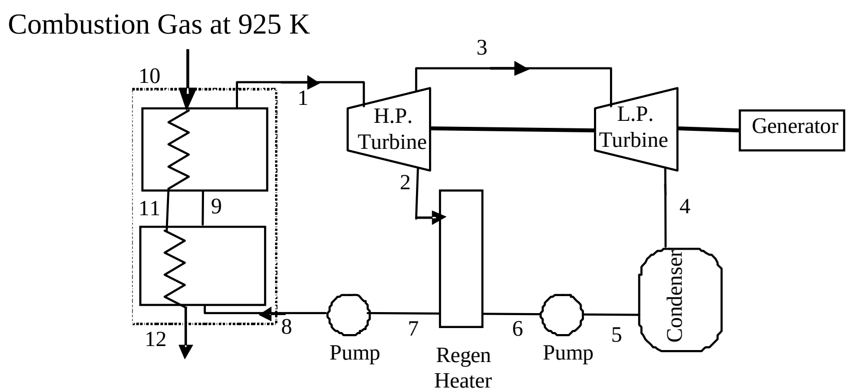

Figure 8.A-8: Power plant using one feedwater heater and fueled by combustion gas

a.) Prepare a plot of the cycle efficiency as a function of the extraction pressure at states 2 and 3. Note that the cycle efficiency is defined as the net power divided by the thermal energy provided to the cycle from the combustion gas, which is the product of the mass flow rate of combustion gas and the enthalpy difference between states 10 and 12. Also plot the temperature of the exiting combustion gas at state 12.

b.) Since the combustion gas at state 12 is discharged to the environment, a global efficiency can be defined as the ratio of the net power divided by the energy provided in the combustion gas. Note that the energy provided in the combustion gas is the product of the mass flow rate of combustion gas, its specific heat and the difference in temperature between state 12 and the ambient temperature. Plot the global efficiency versus extraction pressure on the same plot axes used in part a for the cycle efficiency plot.

c.) What can you conclude from these plots?

8.A-9 A hot gas stream is available at $\dot{m}_{gas}$ = 10 kg/s and $T_{gas,in}$ = 350°C at ambient pressure. This gas has constant pressure specific heat capacity of $c_{P,gas}$ = 1.1 kJ/kg-K. It has been proposed to use this gas to provide power with a simple Rankine cycle (Figure 8-4) that uses toluene as the working fluid. Assume that the turbine and pump efficiencies are 100%. Preliminary estimates indicate that the rate of heat transfer in the boiler can be computed according to:

$$\dot{Q}_B = \varepsilon_B\, \dot{m}_{gas}\, c_{P,gas} \left(T_{gas,in} - T_2\right)$$

where ε_B = 0.45 is the effectiveness of the boiler and T_2 is the saturation temperature at the boiler exit. Cooling water is available at $\dot{m}_w$ = 20 kg/s at $T_{w,in}$ = 10°C and P_{atm} = 1 atm. The condenser heat transfer rate is:

$$\dot{Q}_C = \varepsilon_C\, \dot{m}_w\, c_w \left(T_4 - T_{w,in}\right)$$

where ε_C = 0.70 is the effectiveness of the condenser and T_4 is the saturation temperature at the condenser exit.

a.) Prepare a plot of power versus efficiency for this Rankine cycle by varying the boiler pressure between 100 kPa and 4 MPa. Determine the maximum power that could be obtained by this plant and the corresponding efficiency from the plot.

b.) Plot the optimum cycle on a *T*-*s* diagram for toluene.

8.A-10 A hog slaughtering plant requires 60 kW of electricity (for refrigeration, lights, and other equipment) and 2.0 lb_m/s of water at 175°F heated from 50°F for cleaning purposes. These energy quantities are currently provided by the equipment shown in Figure 8.A-10a. The boiler operates at 550 psia and is fueled with natural gas which is combusted at an efficiency of 85% based on the higher heating value of methane (23,868 Btu/lb_m) The temperature exiting the boiler is 650°F. The generator efficiency is 0.84. The turbine adiabatic efficiency is 0.72 and the pump efficiency is 0.46. The temperature of water exiting the condenser is 90°F. An alternative system to provide the electrical and hot water needs is shown in Figure 8.A-10b. In this case, the equipment is the same except that the condenser pressure is raised so as to allow the condenser to provide the necessary hot water.

a.) Determine the required rate of methane use for plant A.
b.) Indicate what pressure the condenser needs to operate at in plant B if there is a minimum pinch point temperature difference of 12°F between the saturated steam and the water in the condenser.
c.) Determine the required rate of methane use for plant B.
d.) Which plant is the better choice and why?

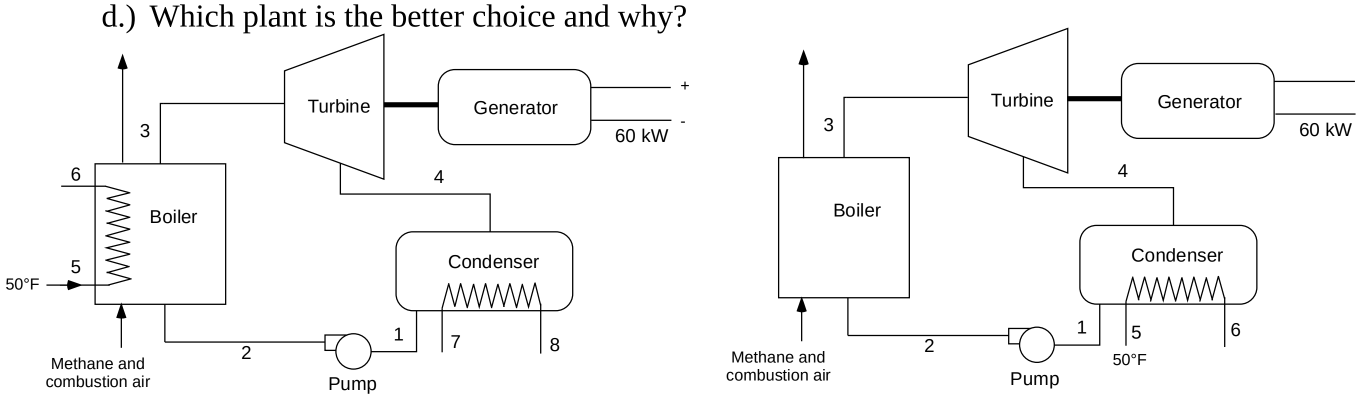

Figure 8.A-10a: Current system (a) and proposed system (b) for producing hot water and power

8.A-11 Figure 8.A-11 illustrates a basic Rankine cycle using water as the working fluid.

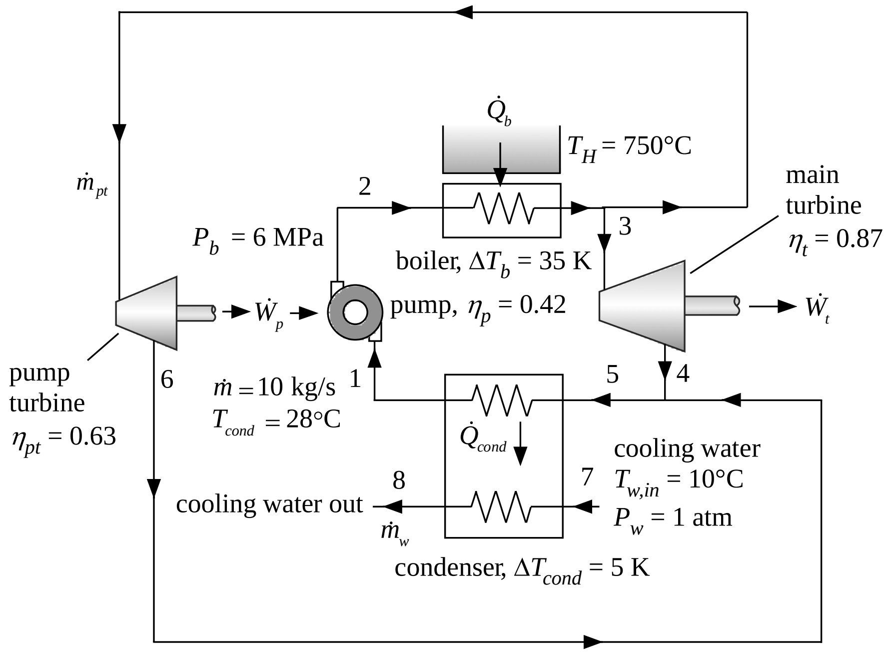

Figure 8.A-11: Rankine cycle with steam driven pump.

Saturated liquid at T_{cond} = 28ºC leaves the condenser and enters the pump at state 1. The mass flow rate of water passing through the pump is $\dot{m}$ = 10 kg/s and the pump efficiency is η_p = 0.42. The pressure of the water leaving the pump is P_b = 6 MPa. The high pressure liquid leaving the pump at state 2 enters the boiler. The boiler provides heat from a thermal reservoir at T_H = 750ºC to the fluid and the boiler approach temperature difference is ΔT_b = 35 K. Neglect any pressure loss in the boiler. The high pressure steam leaving the boiler at state 3 is split into two streams. A small fraction of the total mass flow rate is used to drive the pump turbine that is used to provide power to the pump. The mass flow rate that drives the pump turbine is $\dot{m}_{pt}$. The pump turbine efficiency is η_{pt} = 0.63. The remainder of the flow passes through the main power turbine which has efficiency η_t = 0.87. The two streams (the stream leaving the pump turbine at state 6 and the main turbine at state 4) are mixed upstream of the condenser so that water at state 5 enters the condenser. The condenser has an approach temperature difference of ΔT_{cond} = 5 K. Neglect any pressure loss in the condenser or in the mixing process. Cooling water is used to provide the heat rejection for the power plant. The cooling water enters the condenser with temperature $T_{w,in}$ = 10ºC and pressure P_w = 1 atm.

a.) Determine the power required by the pump, $\dot{W}_p$.
b.) Determine the rate of heat transfer provided by the boiler, $\dot{Q}_b$.
c.) Determine the mass flow rate required by the pump turbine, $\dot{m}_{pt}$.
d.) Determine the rate of power produced by the main turbine, $\dot{W}_t$.
e.) Determine the rate of heat transfer transferred in the condenser, $\dot{Q}_{cond}$.
f.) Determine the mass flow rate of cooling water passing through the condenser, $\dot{m}_w$.
g.) Determine the effectiveness of the condenser.
h.) Check your answers by carrying out a total energy balance on the system.
i.) Determine the efficiency of the Rankine power cycle.

8.A-12 A solar trough power plant is a Rankine cycle that uses solar energy as its heat input. Solar energy is focused by parabolic trough receivers onto a pipe that carries a heat transfer fluid, as shown in Example 8.2-1, Figure 1. The heat transfer fluid is heated as it flows through the field and then returns to the power plant. The fluid transfers heat to the working fluid of the power plant in order to provide the thermal energy that drives the power cycle, which is shown in Figure 8.A-12.

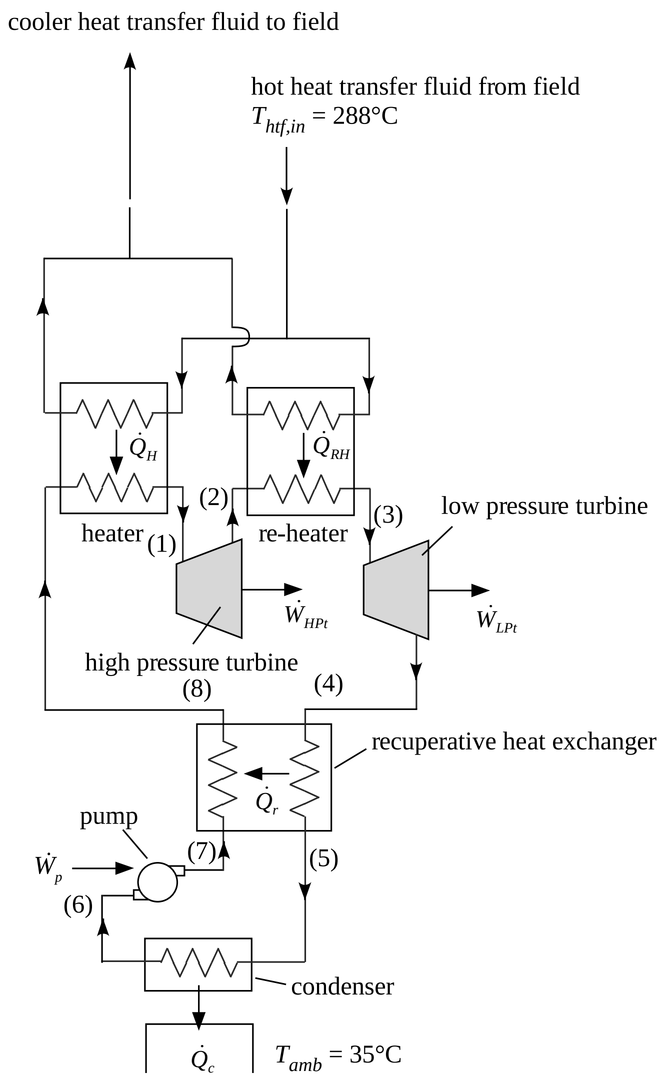

Figure 8.A-12: Rankine power cycle.

The heat transfer fluid leaves the field and enters the power plant at $T_{htf,in}$ = 288°C. Some of the fluid enters the heater where it heats the working fluid for the cycle. Because the working temperature for the cycle is so low, water is not a very efficient working fluid. Instead, toluene is used in the cycle. The toluene leaves the heater at:

$$T_1 = T_{htf,in} - \Delta T_H$$

where ΔT_H = 20 K is the heater approach temperature difference. The toluene enters the high pressure turbine at $P_1 = P_{high}$ = 1034 kPa and is expanded to $P_2 = P_{reheat}$ = 250 kPa. The efficiency of the high pressure turbine is η_{HPt} = 0.81. The remainder of the heat transfer fluid enters the re-heater where it re-heats the toluene leaving high pressure turbine to:

$$T_3 = T_{htf,in} - \Delta T_{RH}$$

where ΔT_{RH} = 20 K is the re-heater approach temperature difference. The toluene leaving the re-heaters passes through the low pressure turbine which has an efficiency of η_{LPt} = 0.78. The condensing pressure is set so that the toluene leaving the condenser is saturated liquid at:

$$T_6 = T_{amb} + \Delta T_c$$

where T_{amb} = 35°C is the ambient temperature and ΔT_c = 15 K is the condenser approach temperature difference. The efficiency of the pump is η_p = 0.6. The toluene leaving the low pressure turbine is used to pre-heat the toluene before it reaches the heater using a recuperative heat exchanger. The temperature of the toluene leaving the recuperative heat exchanger and entering the condenser has temperature:

$$T_5 = T_7 + \Delta T_r$$

where ΔT_r = 20 K is the recuperator approach temperature difference. Neglect the pressure drop through the heater, re-heater, recuperative heat exchanger, and condenser.

a.) Set the pressures at each of the states shown in Figure 8.A-12.

b.) Analyze the low pressure turbine. Determine the power per unit of mass flow rate of toluene obtained from the low pressure turbine ($\dot{W}_{LPt} / \dot{m}$) and the rate of entropy generation in the turbine per unit of mass flow rate ($\dot{S}_{gen,LPt} / \dot{m}$).

c.) Analyze the pump. Determine the power per unit of mass flow rate of toluene required by the pump ($\dot{W}_p / \dot{m}$) and the rate of entropy generation in the pump per unit of mass flow rate ($\dot{S}_{gen,p} / \dot{m}$).

d.) Analyze the recuperative heat exchanger. Determine the rate of heat transfer from the low pressure stream to the high pressure stream per unit of mass flow rate of toluene ($\dot{Q}_r / \dot{m}$) and the rate of entropy generation in the recuperator per unit of mass flow rate ($\dot{S}_{gen,r} / \dot{m}$).

e.) Analyze the high pressure turbine. Determine the power per unit of mass flow rate of toluene obtained from the high pressure turbine ($\dot{W}_{HPt} / \dot{m}$) and the rate of entropy generation in the turbine per unit of mass flow rate ($\dot{S}_{gen,HPt} / \dot{m}$).

f.) Determine the rate of heat transfer in the heater, re-heater, and condenser per unit of mass flow rate of toluene ($\dot{Q}_H / \dot{m}$, $\dot{Q}_{RH} / \dot{m}$, and $\dot{Q}_c / \dot{m}$, respectively).

g.) Check your solution by drawing a system boundary that encompasses the cycle and showing that energy balances for the system.

h.) Determine the thermal efficiency of the power plant.

i.) Prepare a *T-s* plot for the cycle - overlay your states (labeled) onto a *T-s* diagram for toluene. Make sure each process illustrated on the *T-s* diagram makes sense to you.

j.) Plot the plant efficiency as a function of the reheat pressure, P_{reheat}. You should see that an optimal reheat pressure exists; use the Min/Max feature in EES to identify the optimal value of P_{reheat} and set the reheat pressure to its optimal value for the remainder of the problem.

k.) Plot the plant efficiency as a function of the heat transfer fluid inlet temperature, $T_{htf,in}$ (i.e., the temperature of the heat transfer fluid returning from the solar field), for 250ºC < $T_{htf,in}$ < 440ºC.

8.A-13 A solar-driven Rankine cycle power plant uses RC318 (octafluorocyclobutane) as the working fluid. At design conditions, the turbine inlet condition is saturated vapor at 220°F. The air-cooled condenser operates at a saturation pressure of 60 psia. The pump and turbine efficiencies are 0.55 and 0.78, respectively. The system is expected to produce 100 hp. The efficiency of the solar collector field, defined as the ratio of the rate at which thermal energy is collected to the rate of incident solar energy, is:

$$\eta = 0.416 - \frac{0.17[\text{Btu/hr-ft}^2\text{-F}]\left(T_{coll} - T_{amb}\right)}{G_{solar}}$$

where G_{solar} is the incident solar radiation in Btu/hr-ft^2, T_{coll} is the temperature of the collector absorber which is 220°F for this application and T_{amb} is the outdoor temperature, which is assumed to be 75°F.

a.) Determine the efficiency of the Rankine cycle at the design conditions.

b.) Prepare a plot of the efficiency of the entire cycle, including the solar collector field, as a function of solar radiation for a range between 150 to 325 Btu/hr-ft^2.

c.) Calculate and plot the required area of the collector field needed to produce 100 hp of net power as a function of solar radiation for a range between 150 to 325 Btu/hr-ft^2.

B: Gas Turbine Cycles

8.B-1 Space borne power systems often utilize radioisotope thermoelectric generators (RTGs) to produce power. RTGs release heat due to the radioactive decay of an isotope like plutonium-238. The half-life of plutonium-238 is about 90 years; therefore, the RTG produces heat at approximately a constant rate during the duration of the mission. The heat from an RTG is typically converted to electricity using a thermoelectric power production system because it has no moving parts and is therefore very reliable. However, the efficiency of such systems is very low (3-7%). Therefore, you are examining the alternative power production system shown in Figure 8.B-1.

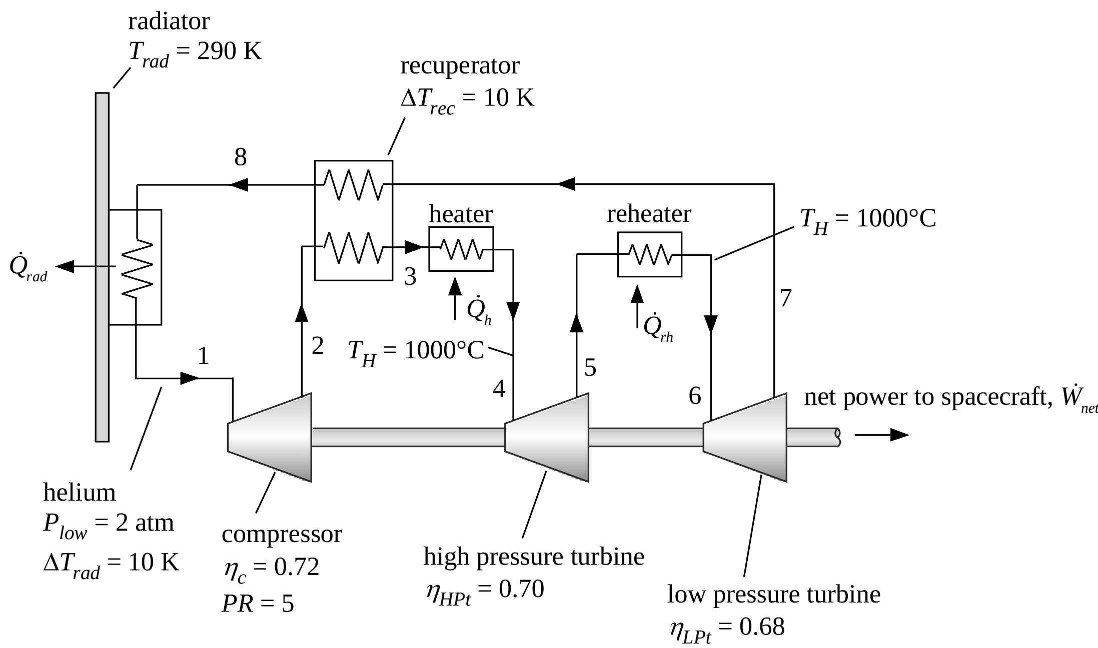

Figure 8.B-1: Gas turbine engine power by RTGs.

The RTGs are used to provide heat to a reheated and recuperated gas turbine engine. The gas turbine engine is a closed cycle system as opposed to the typical air-breathing gas turbine engines used on earth. The working fluid is helium and the system rejects heat to space using a radiator that is at T_{rad} = 290 K. Helium enters the compressor at state 1 with pressure P_{low} = 2 atm and temperature $T_1 = T_{rad} + \Delta T_{rad}$ where ΔT_{rad} = 10 K is the approach temperature difference associated with the heat exchanger than interfaces the radiator with the helium. The compressor pressure ratio is PR = 5 and its efficiency is η_c = 0.72. The helium leaves the compressor at state 2 and is preheated in a recuperative heat exchanger to state 3. The helium at state 3 is heated by an RTG and leaves the heater at $T_4 = T_H$ = 1000°C. The high pressure turbine has efficiency η_{HPt} = 0.70. The helium leaving the high pressure turbine at state 5 enters a reheater where it is reheated by another RTG to $T_6 = T_H$ = 1000°C. The low pressure turbine has efficiency η_{LPt} = 0.68. The pressure ratio across the two turbines is the same. The helium is then cooled in the recuperative heat exchanger to state 8 and finally cooled by the radiator back to state 1. The pinch point for the recuperative heat exchanger occurs at the hot end and the approach temperature difference is ΔT_{rec} = 10 K. You may neglect pressure drop in all of the heat exchangers. Model helium as an ideal gas (i.e., using the substance 'He' in EES).

a.) Determine each of the state points in the cycle. Print out an Arrays table that contains at least the temperature, pressure, entropy, and enthalpy of each state point.

b.) Use EES to generate a T-s diagram for helium. Overlay on this plot each of state points and label them.
c.) Determine the efficiency of the cycle.
d.) The spacecraft requires power at a rate of $\dot{W}_{net}$ = 250 W. Determine the mass flow rate of helium required ($\dot{m}$) and the size of the RTG system required (the rate that heat is added to the cycle).
e.) Determine the effectiveness of the recuperator (ε_{rec}).

The mass of the recuperator can be calculated according to:

$$m_{rec} = \frac{K_{HX}\, \dot{m}\, \varepsilon_{rec}}{\left(1-\varepsilon_{rec}\right)}$$

where K_{HX} = 200 s is an empirical constant for the particular heat exchanger design being considered.
f.) Determine the mass of the recuperator required.

The area of the radiator required can be calculated according to:

$$\dot{Q}_{rad} = \sigma\, A_{rad}\, T_{rad}^4$$

where σ = 5.67x10^{-8} W/m^2-K^4 is the Stefan-Boltzmann constant and $\dot{Q}_{rad}$ is the rate of heat transfer from the helium to the radiator (and therefore from the radiator to space). The mass of the radiator panel is calculated from:

$$m_{rad} = K_{rad}\, A_{rad}$$

where K_{rad} = 1.85 kg/m^2 is the mass of the panel per unit area.
g.) Determine the area and mass of the radiator panel required.

The radiator and recuperator are the most massive parts of the system. The mass of the remainder of the system is relatively fixed and equal to m_{misc} = 1.8 kg.
h.) Determine the total mass of the system.

System mass is the most important parameter for a space borne power system. There are three free parameters that you, as a system designer, can vary in order to minimize the system mass: the recuperator performance (ΔT_{rec}), the radiator temperature (T_{rad}), and the pressure ratio (PR).
i.) Plot the system mass as a function of ΔT_{rec} for 10 K < ΔT_{rec} < 150 K (with T_{rad} and PR set to their nominal values). You should see that an optimal value of ΔT_{rec} exists. Explain why this is true. You may want to generate additional plots to support your explanation.
j.) Plot the system mass as a function of T_{rad} for 200 K < T_{rad} < 350 K (with ΔT_{rad} and PR set to their nominal values). You should see that an optimal value of T_{rad} exists. Explain why this is true.
k.) Use EES' multidimensional optimization capability to determine the optimal values of PR, T_{rad}, and ΔT_{rec} and the associated minimum possible system mass.

8.B-2 Figure 8.B-2 shows a gas turbine system with two-stage compression and intercooling and two-stage expansion with reheat. A recuperative heat exchanger is also provided to recover energy in the turbine exhaust. Air enters the compressor at T_1 = 25°C and atmospheric pressure, P_1 = 1 atm. Air exits the water-cooled intercooler at T_3 = 45°C. The fuel is methane, which has a heat of combustion HC = 50,000 kJ/kg. The turbine inlet temperature for both the high and low pressure turbines is T_6 = T_8 = 1100°C. The recuperator effectiveness is ε = 0.40 and the recuperator pinch point is at the warm end. The overall pressure ratio of the two compressors is $P_4 / P_1 = 10$. Pressure losses in the heat exchange equipment are negligible.

Compressor and turbine performance is often expressed in terms of the isentropic efficiency. However, the isentropic efficiency varies with pressure ratio. A better representation of the performance is provided by the *polytropic efficiency*. The manufacturer has indicated that the polytropic efficiency of the compressors and turbines in the gas turbine engine are $\eta_{c,poly} = \eta_{t,poly}$ = 0.76. The manufacturer indicates that the isentropic efficiency of the compressors, $\eta_{c,isen}$, can be related to the polytropic efficiency, $\eta_{c,poly}$, according to:

$$\eta_{c,isen} = \frac{\left(P_{out} / P_{in}\right)^{e} - 1}{\left(P_{out} / P_{in}\right)^{e/\eta_{c,poly}} - 1}$$

where P_{in} and P_{out} are the inlet and outlet pressures of the compressor and e is a property of the air that is defined in terms of the constant pressure and constant volume specific heat capacities:

$$e = 1 - \frac{c_v}{c_p} = \frac{k-1}{k}$$

where k is the ratio c_p/c_v. The isentropic efficiency of the turbines, $\eta_{t,isen}$ is related to the polytropic efficiency, $\eta_{t,poly}$, according to:

$$\eta_{t,isen} = \frac{1 - \left(P_{out} / P_{in}\right)^{e\eta_{t,poly}}}{1 - \left(P_{out} / P_{in}\right)^{e}}$$

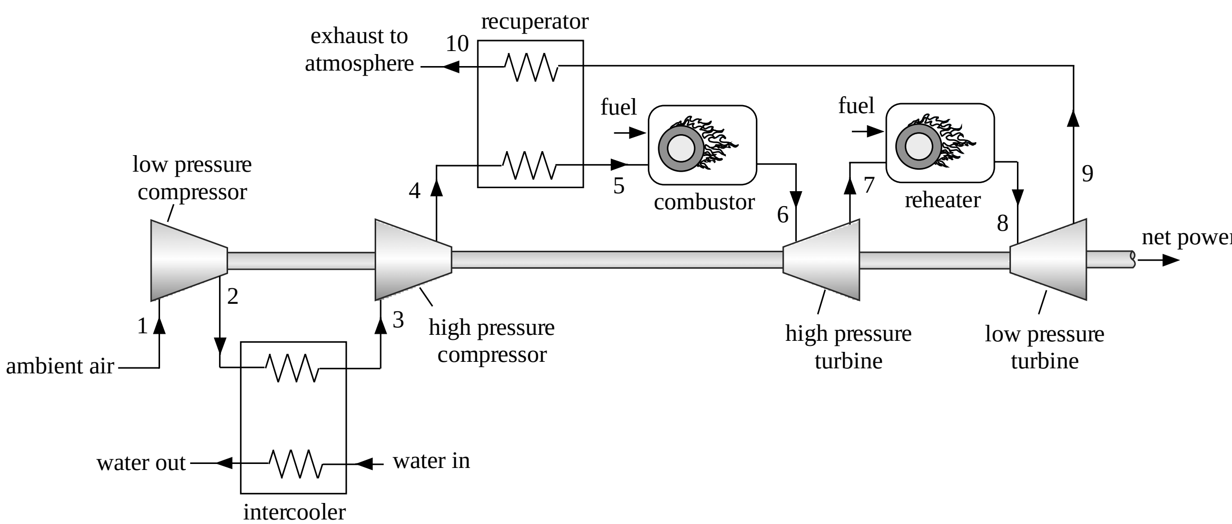

Figure 8.B-2: Gas turbine system with staged compression and expansion and regeneration.

Model air as an ideal gas and assume that the combustion products can be modeled as being air.

a.) Assume that the intercooling and reheating pressure are both equal to $\sqrt{P_1 P_4}$. Determine the net work per unit mass of inlet air, the combustor and reheater air fuel ratios, and the cycle efficiency. Print out the Arrays table that contains the properties (at least *h*, *s*, *P*, and *T*) at each state and develop a *T*-*s* diagram that shows the states.

b.) Determine the intercooling and reheat pressures (i.e., P_2 and P_7) that maximize the net work. How well do these pressures compare with the ideal value determined assuming constant specific heat capacity, $\sqrt{P_1 P_4}$?

8.B-3 A combined cycle power plant is shown in Figure 8.B-3. The turbine inlet temperature that can be tolerated for a gas turbine is much higher than for a steam turbine. Therefore, the Brayton cycle can operate at very high temperature. However, this leads to very hot gas leaving the gas turbine. Rather than waste this high temperature gas, it is first sent through a boiler in order to transfer heat to a Rankine cycle (which has higher efficiency than a Brayton cycle, but must operate at lower temperature). Finally, the low temperature gas leaving the boiler is used to produce steam for heating purposes. This cycle is an example of a combined heating and power (CHP) system.

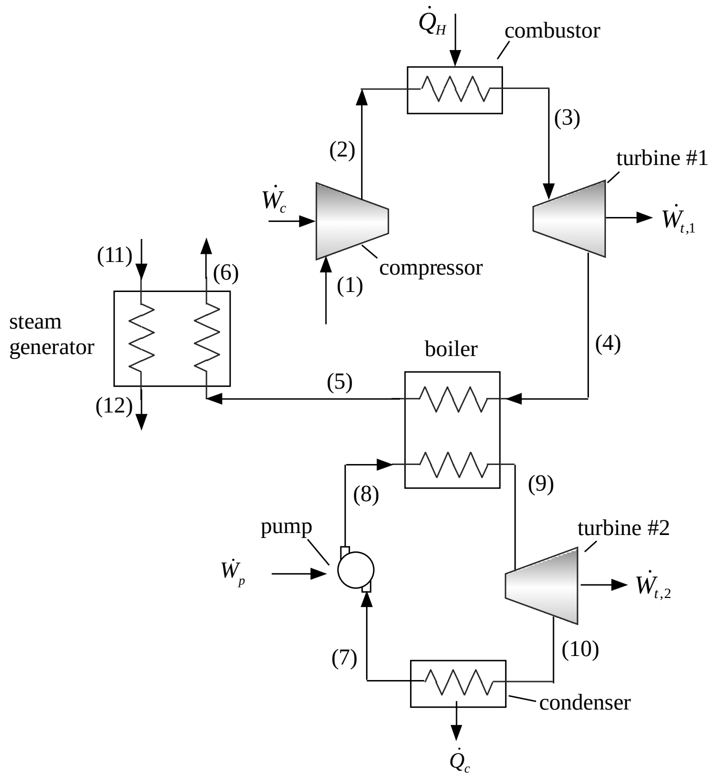

Figure 8.B-3: Combined cycle power plant.

Model the Brayton cycle assuming that the working fluid is air (i.e., use the substance 'Air' in EES). The mass flow rate of air is $\dot{m}_a$ = 73 kg/s. Air enters the compressor at state 1 with P_{atm} = 1 atm and T_{amb} = 20°C. The compressor has a pressure ratio PR = 7.5 and an efficiency η_c = 0.85. Air enters the combustor and is heated to a temperature T_H = 1250°C. The turbine has an efficiency of $\eta_{t,1}$ = 0.87. The air leaving the turbine enters the boiler at state 4 where it transfers heat to the water in the Rankine cycle. The ratio of the mass flow rate of water in the Rankine cycle to air in the Brayton cycle is $\dot{m}_w / \dot{m}_a$ = 0.1. The boiler has an approach temperature difference of ΔT_b = 15 K; therefore, the water leaving the boiler achieves temperature $T_9 = T_4 - \Delta T_b$. (Note that T_5 is not equal to T_8). The air enters the steam generator at state 5. The air is used to heat water from the inlet state of $P_{11} = P_s$ = 5 atm, $T_{11} = T_{w,in}$ = 20°C to saturated vapor at $P_{12} = P_{11}$. The temperature of the air leaving the steam generator is $T_6 = T_{a,out}$ = 180°C. In the Rankine cycle, the water enters the pump at state 7 as saturated liquid. The pump efficiency is η_p = 0.65 and the exit pressure of the pump is P_{boiler} = 8.0 MPa. The water enters the heat exchanger at state 8 where it is converted to steam by the heat transfer from the air. The water enters the steam turbine at state 9. The steam turbine has an efficiency of $\eta_{t,2}$ = 0.84. The water enters the condenser at state 10. The condenser transfers heat from the water to a cooling water at T_{amb} =

20°C. The condenser approach temperature is ΔT_c = 5 K; therefore, the water leaves the condenser at $T_c + \Delta T_c$. Neglect the pressure drop in all heat exchangers. Hint: analyze the compressor, combustor, and turbine #1. Then analyze the pump and boiler. Finally, analyze the steam generator.

a.) Determine all of the state points for both cycles. You should have an array table with at least the pressure, temperature, entropy and enthalpy of states 1 through 11.

b.) Determine the mass flow rate of steam produced in the steam generator.

c.) Check your solution by carrying out an overall energy balance on the entire cycle. (There are a several ways to do this calculation.)

d.) Determine the efficiency of the Brayton cycle, the efficiency of the Rankine cycle, and the efficiency of the combined cycle (relative to producing power - do not include the value of the steam that is produced).

e.) What is the net power produced by the cycle?

f.) If natural gas is used to energize the cycle and the cost of natural gas is NGc = 8$/million Btu then determine the yearly fuel cost required to run the plant.

g.) Determine the cost associated with producing the same amount of power using a conventional natural gas fired plant with efficiency η_{conv} = 0.34 and the same amount of steam using a natural gas fired boiler. What is the savings per year associated with using the combined cycle?

8.B-4 A schematic of a turbo-fan engine is shown in Figure 8.B-4.

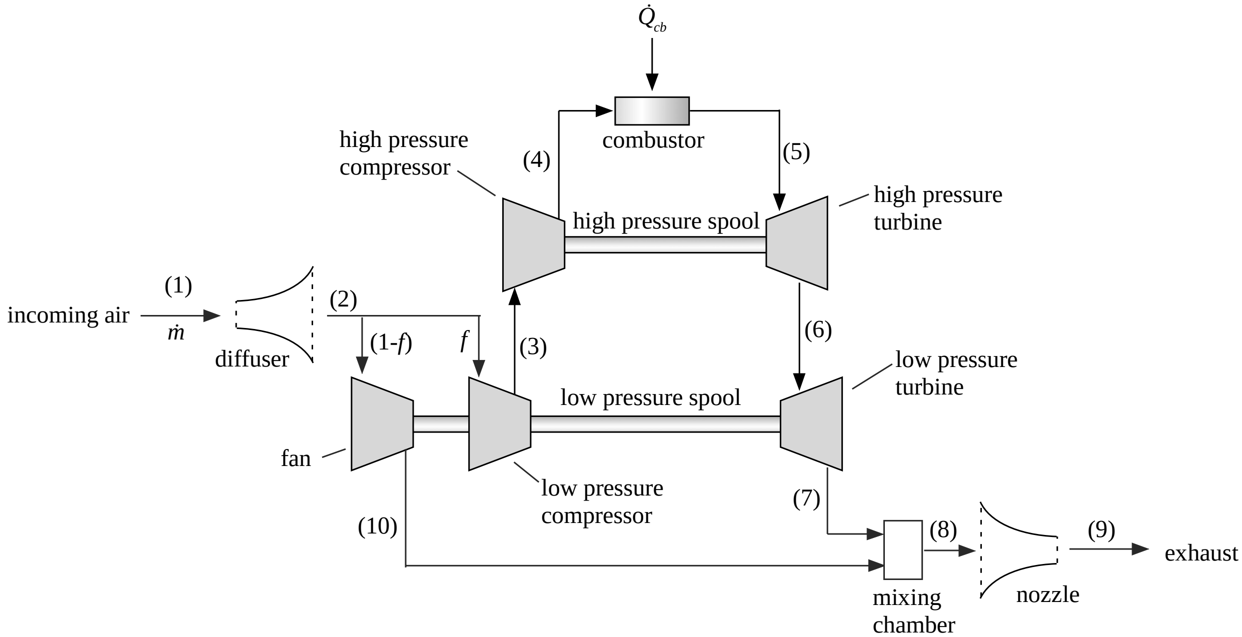

Figure 8.B-4: Turbo-fan engine.

Incoming air enters the engine with mass flow rate $\dot{m}$ = 200 kg/s and passes through a diffuser. After the air leaves the diffuser, it splits into two streams. A fraction of the air (f) passes through the low pressure compressor (2-3) and then through the high pressure compressor (3-4) where it is mixed with fuel and combusted (4-5). The air then passes through the high pressure turbine (5-6) and low pressure turbine (6-7). The remaining air (1-f) passes through the fan (2-10) and then mixes with the air leaving the low pressure turbine. All of the air leaves through the nozzle (8-9). The high pressure compressor and turbine are collocated on the high pressure spool and the fan, low pressure compressor and low pressure turbine are collocated on the low pressure spool. Most of the properties of the air at each of the states are summarized in Table 8.B-4.

Table 8.B-4: Properties.

State	Pressure (kPa)	Temperature (K)	Velocity (m/s)	Enthalpy (J/kg)	Entropy (J/kg-K)
1	68	283.2		283507	5758
2	74.9	292.7	0	293110	5764
3	165	379.4	0	380462	5799
4	1037	677.7	0	689654	5870
5	1037	1773	0	1.970E+06	6981
6	475	1521	0	1.661E+06	7017
7	93.6	1101	0	1.162E+06	7100
8	93.6		0		
9	68				
10	93.6	317.4	0	317901	5781

a.) Determine the velocity of the air entering the diffuser, $\tilde{V}_1$.

b.) Determine the fraction of air that passes through the combustor, f.

For the remainder of this problem you may assume that f = 0.057. (This may or may not be the correct answer to (b)).

c.) Determine the enthalpy of the air entering the nozzle, h_8.

For the remainder of this problem you may assume that the temperature of the air entering the nozzle is T_8 = 365 K. (This may or may not be consistent with the correct answer to (c)).

d.) Determine the velocity of the air leaving the nozzle, $\tilde{V}_9$. The nozzle efficiency is η_n = 0.92. For this part of the problem, you may assume that air can be modeled as an ideal gas with constant specific heat capacity c_P = 1004 J/kg-K and R = 287 J/kg-K.

e.) Determine the thrust force produced by the engine.

8.B-5 Many electric utilities rely on gas turbine generators to help supply their peak electrical demand during the summer months. Gas turbines operate less efficiently than the Rankine power cycle and this is one reason that electrical rates are higher during the summer than they are during the winter in many locations. If a utility is unable to supply their peak demand, they must purchase electricity from other utilities at a relatively high price. One way to ensure that the peak demand is met is to install extra gas turbine cycle generators. Another way, which may be more economical, is to cool the compressor inlet air using an ice storage unit, as shown in Figure 8.B-5. Cooling the inlet increases both the capacity and efficiency of the gas turbine cycle. For the specific case under consideration, the ice is produced by an electrically-driven vapor compression refrigeration cycle with *COP* = 2.8. The vapor compression cycle operates during off-peak times and consumes electricity that has a utility cost of *ec* = 0.04 \$/kW-hr. The effective cost of the fuel (including a consideration for combustion efficiency) that is used to run the gas turbine is *fc* = 5.5 \$/GJ.

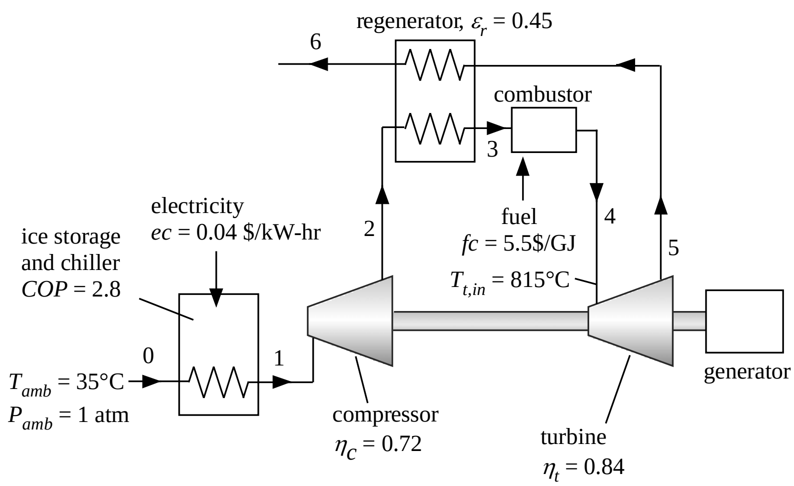

Figure 8.B-5: Gas turbine power system with inlet air cooled by ice storage system.

Assume the isentropic efficiencies of the compressor and turbine to be η_c = 0.72 and η_t = 0.84, respectively. The temperature at state 4 is $T_{t,in}$ = 815°C. Pressure losses in the combustor and in each heat exchange operation are approximately *PL* = 2% of the inlet pressure to the component. The regenerator effectiveness is ε_r = 0.45 and the pinch point is at the hot end. With ice storage, the inlet air can be cooled to T_{is} = 5°C. Consider at peak day in which the ambient air is at T_{amb} = 35°C, P_{amb} = 1 atm. Assume the working fluid to be pure air. You may assume that the mass flow rate throughout the cycle is the same (i.e., neglect the small change in the mass flow rate associated with the addition of the fuel). State any other assumptions you employ.

a) Determine the pressure ratio that maximizes the efficiency of the gas turbine cycle assuming that there is no ice storage unit.

b.) Determine the pressure ratio that maximizes the efficiency of the gas turbine cycle assuming that there is an ice storage unit that is sufficient to cool state 1 to 5°C. Compare the optimized efficiency with the result in part (a).

c.) Estimate the utility cost (in \$/kW-hr) of the electricity produced by the gas turbine system without and with the ice storage unit.

d.) What is your assessment of the ice storage concept based on these calculations?

8.B-6 An air-standard gas turbine with regeneration is shown in Figure 8.B-6. The net electrical output is 20 MW. The electric generator has an efficiency of 96%. The conditions at various points in the cycle are given in Table 8.B-5. Assume the air is an ideal gas with specific heats dependent upon temperature.

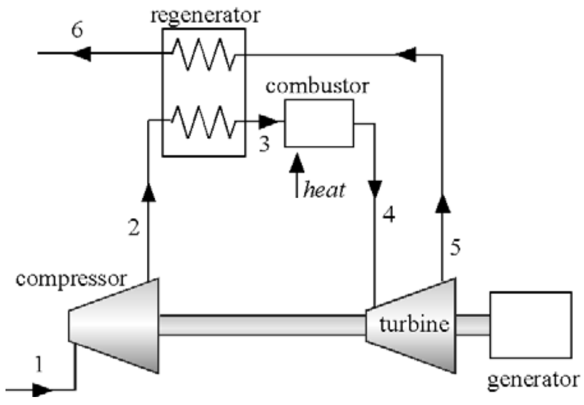

Figure 8.B-5: Gas turbine system

Table 8.B-6: Property Information

	T [K]	***P*** [kPa]	***h*** [kJ/kg]
1	300.0	100.0	300.4
2	628.9	1000.0	637.5
3	1181.8	950.0	1256.3
4	2000.0	940.0	2251.1
5	1356.2	150.0	1462.5
6	819.7	100.0	843.7

a.) The mass flow rate of the air in the gas turbine is ________________kg/s.

b.) The overall efficiency from thermal to electrical energy is _______________.

c.) The effectiveness of the regenerator is ______________.

d.) Write an EES program to determine the efficiency of the compressor.

8.B-7 Supercritical carbon dioxide gas turbine cycles have been receiving a lot of attention as a possible replacement for the Rankine cycle in both nuclear and solar-power cycles. This problem will analyze the performance of a supercritical carbon dioxide cycle in a manner that includes the effects of the heat exchangers. The cycle is shown in Figure 8.B-7. Carbon dioxide steadily enters the compressor at state 4 with a volumetric flow rate of $\dot{V}$ = 0.10 m^3/s and pressure of P_4 = 7.5 MPa after exiting the pre-cooler (the low temperature heat exchanger). The carbon dioxide is compressed adiabatically with a pressure ratio *PR* to state 1 where *PR* is defined as P_1/P_4. At this point, the carbon dioxide is heated in the primary heat exchanger by a hot fluid that has an entering temperature of $T_{H,in}$ = 600°C and capacitance rate of $\dot{C}_H$ = 18,000 W/K. The heat transfer effectiveness of the primary heat exchanger is ε_{phx} = 0.90. The heated carbon dioxide is expanded in an adiabatic turbine to state 3. Energy is rejected in the pre-cooler to a cooling stream that enters the heat exchanger at $T_{C,in}$ = 25°C with a capacitance rate of $\dot{C}_L$ = 78,000 W/K. The heat exchanger effectiveness of the pre-cooler is ε_{pc} = 0.95. Assume that the pressure losses in the heat exchangers are negligible. Also, in analyzing the heat exchangers, assume that the specific heat capacity of the carbon dioxide is constant throughout the heat exchanger and equal to the value that it has at the heat exchanger inlet. Because of the high pressures, carbon dioxide will not obey the ideal gas law. Therefore, use the fluid 'CarbonDioxide' rather than 'CO2' in your EES program.

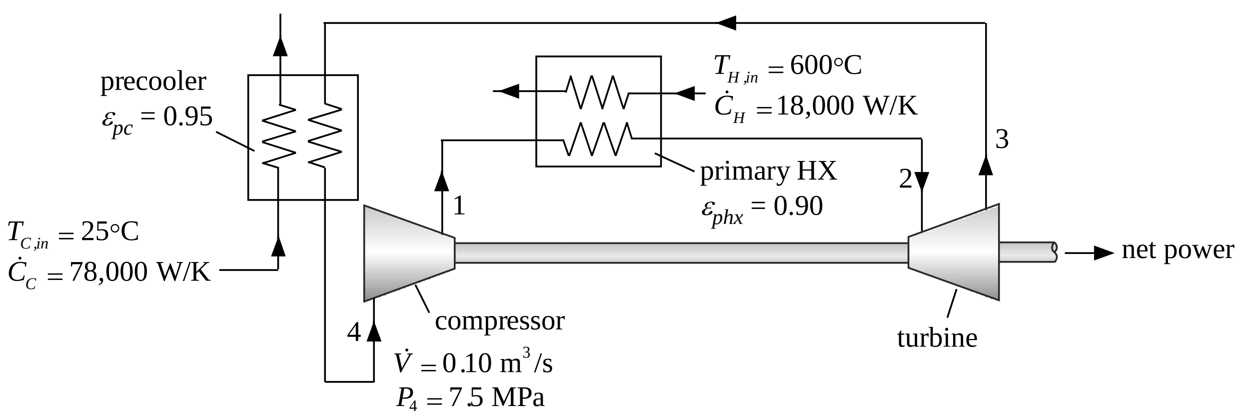

Figure 8.B-7: Supercritical carbon dioxide gas turbine cycle.

a.) Assume that the compressor and turbine operate adiabatically and reversibly. Vary the pressure ratio between 1 and 25 and plot the net power output versus the pressure ratio, the cycle efficiency versus the pressure ratio, and the net power output versus the cycle efficiency
b.) Repeat part (a), but assume that the turbine and compressor both have isentropic efficiencies of $\eta_t = \eta_c = 0.90$.
c.) Summarize the results from your plots. What conclusions can you draw from these results?

8.B-8 A more detailed analysis of the supercritical carbon dioxide gas turbine considered in Problem 8.B-7 is needed. The cooling water is discharged to the environment (at T_{amb} = 25°C and P_{atm} = 1 atm) after leaving the pre-cooler. The hot fluid is returned to a storage unit after leaving the primary heat exchanger. Assume that the isentropic efficiencies of the compressor and turbine are both $\eta_c = \eta_t = 0.90$ and that both components operate adiabatically.

a.) Identify the pressure ratio that provides the maximum power for this cycle. What is the maximum power?

b.) Determine the specific exergy of the carbon dioxide at all states for the cycle determined in (a).

c). Determine the exergy destruction rate occurring in each system component for this cycle. Please put this information in a table showing a rank ordering of the destruction rates.

d.) Determine the First and Second Law efficiencies for this cycle.

e.) Based on your results, please provide a suggestion that will improve the performance of this cycle. Model your suggestion and determine the increase in the 2nd Law efficiency.

8.C-1 A simplified analysis of the processes occurring in a spark-ignition internal combustion engine is described by the air-standard Otto cycle. The processes in the compression ignition engine are described by the Diesel cycle. The purpose of this problem is to compare the Otto and Diesel cycle as a function of compression ratio.

a.) Calculate and plot the thermal efficiency and mean effective pressure for an air standard Otto cycle for compression ratios between 5 and 35 with a maximum cycle temperature of 2,500 K. Assume that the state of the air before compression is 300 K, 1 bar.

b.) Calculate and plot the thermal efficiency and mean effective pressure for an air standard Diesel cycle for compression ratios between 6 and 35 with a maximum cycle temperature of 2,500 K. Assume that the state of the air before compression is 300 K, 1 bar.

c.) What conclusions can you draw from your plots?

8.C-2 A four cylinder four-stroke internal combustion engine has a bore of 3.5 in and a stroke of 3.5 in. The compression ratio of this engine is 8.4. The engine is designed to operate with an air-fuel ratio of 16 with a fuel that has an energy content of 45,000 kJ/kg and a molar mass of 114 kg/kmol. Assume that the air and fuel mixture behave as an ideal gas mixture with the same properties as pure air. The pressure and temperature at the start of the Otto cycle processes are 98 kPa, 32°C.

a.) What is the total engine displacement for all cylinders in liters?
b.) Determine the mass of fuel and the mass of air in one cylinder for each engine cycle.
c.) What is the maximum temperature occurring in the cycle?
d.) Determine the engine efficiency.
e.) Calculate the engine power and fuel flow rate at 3000 rpm.

8.C-3 A supercharged engine uses a compressor to raise the pressure of the gas that is in the cylinder at the start of the cycle. The air-fuel ratio of an internal combustion engine is approximately constant. Therefore, increasing the amount of air in the cylinder increases the amount of fuel and thus the engine power. This purpose of this problem is to determine the power and efficiency behavior of a supercharged engine. The engine considered in this problem is the one described in Example 8.4-1 in which a polytropic model is used to represent the compression and expansion processes and the residual gas in the cylinder is considered. The polytropic exponents for compression and expansion are n_c = 1.35 and n_e = 1.40, respectively. The engine has N_{cyl} = 4 cylinders and operates at N = 3600 rev/min with a compression ratio of CR = 8.3 and an air-fuel ratio of AF = 16. The heat of combustion of the fuel is HC = 44 MJ/kg. The ambient air conditions are T_{amb} = 32°C and P_{atm} = 100 kPa. The supercharger is a compressor that is driven by the engine. It operates adiabatically with an isentropic efficiency of η_c = 0.74. The engine exhausts to atmospheric pressure. Assume that an aftercooler is not employed so that the air entering the engine through the intake valves is at the same pressure and temperature as the air exiting the compressor.

a.) Prepare a plot of the power and efficiency of the supercharged engine as a function of the supercharger pressure ratio (defined as the ratio of the outlet to inlet pressures of the supercharger) for a range between PR = 1 (no supercharger) to PR = 5.

8.C-4 The air-fuel ratio of an internal combustion engine is approximately constant. Increasing the amount of air in the cylinder at the start increases the amount of fuel and thus the engine power. A turbocharged engine uses a compressor to raise the pressure of the gas in the cylinder at the start of the cycle. Unlike a supercharged engine (see Problem 8.C-3), the compressor in a turbocharged engine is driven by a gas turbine operating off the exhaust gas rather than being driven by the engine. This purpose of this problem is to determine the power and efficiency behavior of a turbocharged engine. The engine considered in this problem is the one described in Example 8.4-1 in which polytropic model is used to represent the compression and expansion processes and the residual gas in the cylinder is considered. The polytropic exponents for compression and expansion are 1.35 and 1.40, respectively. The engine has 4 cylinders and operates at 3600 rev/min with a compression ratio of 8.3 and an air-fuel ratio of 16. The heat of combustion of the fuel is 44 MJ/kg. The ambient air conditions are 32°C and 100 kPa.

The compressor in the turbocharger operates adiabatically with an isentropic efficiency of 0.74. It is driven by the turbine, which operates adiabatically using the exhaust gas with an isentropic efficiency of 0.78. However, not all of the exhaust gas passes through the turbine as this would result in more power output from the turbine than required to drive the compressor. A portion of the exhaust gas passes through a gate valve, by passing the turbine. Assume that the pressure in the exhaust manifold is equal to the pressure in the inlet manifold and the temperature of the air in the exhaust manifold is the average of the temperature at the conclusion of the expansion process and the temperature at the end of the exhaust stroke. Prepare a plot the power and efficiency of the turbocharged engine as a function of the compressor pressure ratio (defined as the ratio of the compressor outlet to ambient pressure) for a range between 1 (no turbocharger) and 2.5. Assume that an aftercooler is not used, so that the pressure and temperature of air entering the engine through the intake valve are the pressure and temperature at the exit of the compressor or the turbocharger.

8.C-5 Approximating the combustion process in spark-ignition (Otto) or compression-ignition (Diesel) internal combustion engine cycles as a constant-volume or constant-pressure heat-addition process, respectively, is overly simplistic. A better (but slightly more complex) approach would be to model the combustion process in both the Otto and Diesel engines as a combination of two heat-transfer processes, one a constant volume and the other at constant pressure. The ideal cycle based on this concept is called the *dual cycle*. The purpose of this problem is to investigate the power – efficiency tradeoffs for the dual cycle, which consists of the following five ideal processes with pure air as the working fluid:

1-2 adiabatic compression
2-3 constant volume heat addition
3-4 isobaric heat addition
4-5 adiabatic expansion
5-1 constant volume heat rejection

Another simplistic assumption in the traditional analyses of internal combustion engines is that the compression and expansion processes are isentropic. In your analyses, assume that the compression process (1-2) and expansion process (4-5) are adiabatic with isentropic efficiencies of $\eta_c = n_e = 0.80$. State 1 may be assumed to be air at T_{amb} = 300 K and P_{atm} = 100 kPa. The maximum temperature in the cycle is T_{max} = 2350 K. The compression ratio is defined as $CR = V_1/V_2$ and the pressure ratio is defined as $PR = P_3/P_2$. The maximum practical compression ratio is CR_{max} = 30. Develop a thermodynamic analysis of this cycle. You may ignore combustion processes and the mass of fuel in your cycle, but please consider the variation of thermodynamics properties of air with temperature.

a.) Determine the pressure ratio that will produce the maximum work per cycle for compression ratios ranging between 5 and 30.
b.) The maximum work per cycle as a function of compression ratio.
c.) The optimum pressure ratio as a function of compression ratio.
d.) The maximum work per cycle as a function of the cycle thermal efficiency.

8.C-6 The air-standard Otto cycle provides a simplistic description of the processes occurring in a spark-ignition internal combustion engine. The analysis provides reasonable trends but it significantly overestimates engine efficiency. One reason for the overestimate is that the compression and expansion processes are assumed to be isentropic. In addition heat transfer between the working fluid (assumed here to be pure air) and the cylinder walls is ignored. The purpose of this problem is to investigate the effect of compression ratio and the relationship between power and efficiency for a more realistic analysis in which these non-ideal processes are considered.

In the actual cycle, energy is supplied to the working fluid by combusting fuel. In this analysis, assume that the working fluid is pure air. The combustion step is simulated by assuming fuel having an effective energy content of 40,000 kJ/kg is provided to the air with an air-fuel ratio of 16.

a.) Calculate and plot the thermal efficiency and net specific work for compression ratios between 5 and 15 and ideal compression and expansion processes. Assume that the state of the air before compression is 25ºC, 1 atm.

b.) Repeat the calculations for part a) but in this case assume that the isentropic efficiency for the compression and expansion processes is 0.80.

c.) Some of the energy provided with the fuel is transferred to the 'cold' engine walls which are maintained at 110°C. Assume that the rate of heat transfer to the engine walls is: $q_{wall} = K\left(\overline{T} - T_{wall}\right)$ where K=0.5 kJ/K-kg (air) and $\overline{T}$ is the average temperature occurring during the combustion process, e.g, $(T_2+T_3)/2$. Repeat part b) including the wall heat transfer consideration.

d.) What conclusions can you draw from your plots?

8.D-1 A geothermal plant is being considered to generate 5000 kW of electrical power with a regenerative Rankine cycle having one closed feedwater heater as shown in Figure 8.D-1. The generator efficiency is 0.92. The working fluid in the power cycle is an organic fluid called R245fa. Geothermal brine (liquid with dissolved salts) is available at 115°C. The brine has a specific heat of 4.25 kJ/kg-K. The turbine efficiencies are 0.82 and 0.74 for the high and low pressure turbine, respectively. The pump efficiency is 0.43 for both the high and low pressure pumps. As shown in the figure, the boiler can be represented as two heat exchangers. One heat exchanger (labeled "HX2" in Figure 8.D-1) uses the incoming brine to boil R245fa at 1125 kPa. The brine exiting this heat exchanger (state 11) is 5°C warmer than the R245fa saturation temperature and it is used in the second heat exchanger (labeled HX1) to preheat the entering liquid R245fa at state 8 to the saturation temperature (state 9). The condenser uses water at 25°C to condense the working fluid to saturated liquid at 30°C (state 5). The interstage pressure, P[2], is to be determined. Pressure losses in the piping may be neglected. R245fa, as implemented in EES, is assumed to incompressible in the liquid state. Consequently, it is not possible to calculate the pump outlet states by assuming the ideal pump is isentropic. Use instead the fact that the minimum pump work per unit mass is the product of the inlet specific volume and the pressure rise. State any other assumptions you employ.

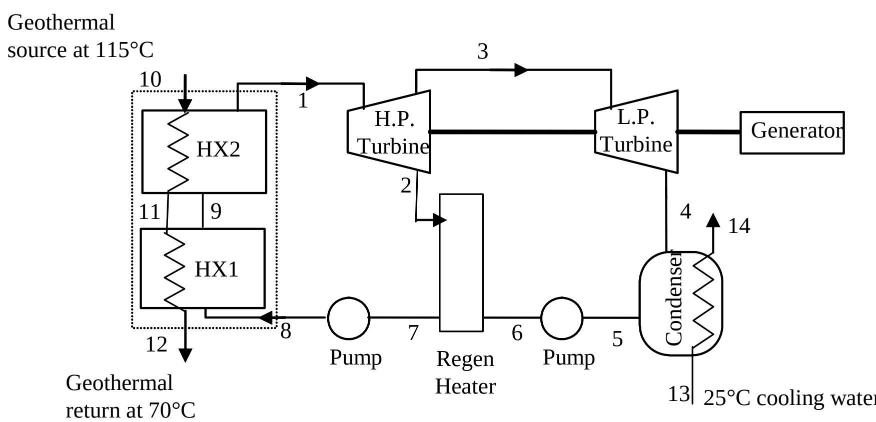

Figure 8.D-1: Geothermal power plant

a.) At what pressure should single-stage regeneration be done so as to maximize the thermal (first law) efficiency? What is the efficiency and the flow rates of the brine and R245fa for this interstage pressure?

b.) Recalculate the efficiency and flow rates of the brine and R245fa assuming that the there is no regenerative heater. (Set P[2] to P[5]). Do you recommend that the system be designed with the regenerative heater? Why or why not?

c.) Determine the exergy destruction rates for each component in the cycle for the optimum cycle with the interstage pressure as determined in part a. Which component is responsible for the highest exergy destruction rate?

8.D-2 A new power plant on a university campus is a cogeneration facility fueled by natural gas. The purpose of this problem is to conduct a First and Second law analysis of this system.

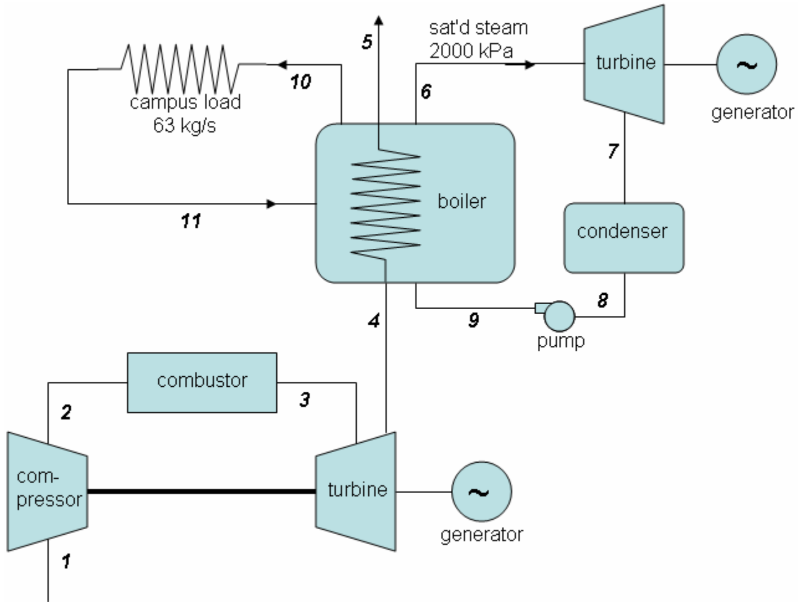

Figure 8.D-2: Cogeneration power plant

The system must supply 150 MW of electrical power and also provide 63 kg/s of saturated steam at 2000 kPa to the campus (state 10). The steam is returned as condensate at 25°C as 2000 kPa (state 11). The cogeneration plant that meets these energy needs is a combined gas turbine – steam Rankine cycle. The compressor isentropic efficiency is 0.78. The gas and steam turbine isentropic efficiencies are 0.84. The generator efficiencies are 0.94. The isentropic efficiency of the pump is 0.46. The maximum allowable temperature of the combustion products (which may be assumed to be air for property evaluation) is 1500 K (state 3). The temperature of the exhaust combustion gases (state 5) is 25°C higher than the temperature of the saturated steam. A cooling tower (not shown in the figure) provides cooling water sufficient to maintain the condenser outlet state as saturated liquid at 32°C (state 8). Conduct an energy and exergy analyses on this plant for a gas turbine pressure ratio of 10 under steady state operation. You may neglect pressure losses in the equipment – state any other assumptions you employ. Use you analyses to answer the following questions.

a. How you would define the First Law efficiency of this plant and what is its value?
b. What is the required mass flow rate of natural gas in kg/s if the natural gas is used with an efficiency of 83% of its lower heating value?
c. Determine the exergy destruction rate in each major component. Which component is responsible for the highest rate of exergy destruction?
d. What is the Second-Law efficiency assuming the exergy of natural gas is equal to its lower heating value?

8.D-3 A factory has an average electrical load of 1500 kW. In addition, 12 million Btu/hr are required to heat water from the supply temperature of 50°F to 180°F at 1 atm. A single-extraction steam turbine plant shown in Figure 8.D-3 is proposed to supply both the electrical and hot water loads. The following operating conditions have been fixed.

Boiler pressure	= 250 psia
Boiler outlet temperature	= 500 F
Extraction pressure	= 30 psia
Condenser pressure	= 1 psia
Pump efficiency	= 0.60
Turbine efficiencies	= 0.82 for high pressure and 0.75 for low pressure

Steam is to be extracted at a point between the high and low pressure turbines at a pressure that are to determine. The extracted steam will be isobarically heat-exchanged to supply the process hot water and exit the heat exchanger as saturated liquid. This liquid is then throttled to the condenser pressure and mixed with the fluid exiting the low-pressure turbine. The fuel will be natural gas which has a higher heating value (HHV) of 23,860 Btu/lb_m and an availability of 21930 Btu/lb_m. The boiler efficiency based on the HHV is 0.85. Heat losses from the turbines, pump and lines are assumed negligible and the generator efficiency may be assumed to be 0.90. Neglect pressure drops in the piping and heat exchangers.

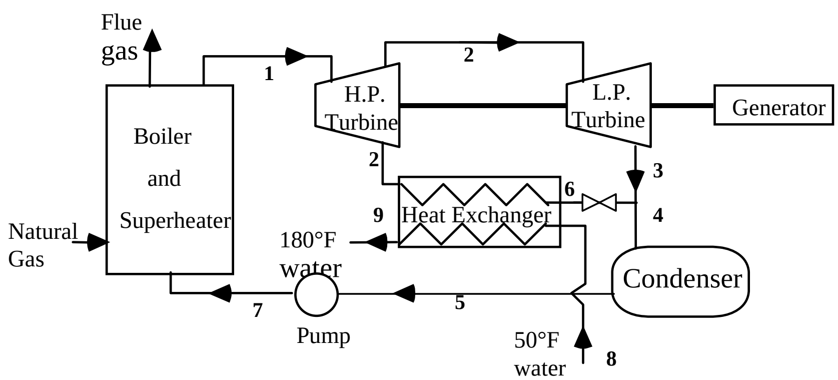

Figure 8.D-3: Cogeneration plant for heating water and generating power

a.) Determine the mass flow rate of water at state 9.

b.) Determine the First and Second-Law efficiency and the rate of consumption of natural gas for this process.

c.) Determine Second-Law efficiency for this process.

d.) Calculate the rate of exergy destruction for each system component. Based on the results, propose modifications that will improve the overall plant efficiency.

e.) For tax purposes, it is necessary to determine separate costs for the electricity and the hot water. If the natural gas cost is \$0.65 per therm (10^5 Btu) , what should be the costs of the electricity (per kW-hr) and the hot water (per million Btu)?

8.D-4 A meat processing company requires 2.5 kg/s of saturated steam at 8 bar and 4,500 kW of electricity to run their operations. The electricity is currently purchased from the local utility at an industrial cost of $0.05/kW-hr. The steam is produced on the premises by pumping well water at 15°C, 1 bar to 8 bar and then heating the water in the boiler at constant pressure to produce the steam. The boiler is fueled with natural gas which the company purchases at a cost of $0.50/therm. (A therm is 10^5 Btu). The boiler efficiency is 0.86. The boiler is failing and it needs to be replaced. The company is investigating alternatives. One alternative is the cogeneration system shown in the figure below. Here, a gas turbine system produces power that is used to drive a generator that has an efficiency of 0.93. The combustor burns natural gas with an efficiency that is the same as the boiler efficiency, 0.86. The compressor and turbine efficiencies are 0.84 and 0.88, respectively. Ambient air at 25°C, 1 bar enters the compressor. The turbine inlet temperature is 1400 K. The combustion gases exiting the turbine enter a steam generator where energy is transferred to the water. The steam generator has an effectiveness of 0.96. Effectiveness for this component is defined as the ratio $(T_4-T_5)/(T_4-T_8)$. The pump has an efficiency of 0.45. The system does not need to produce the entire 4,500 kW of electrical energy since additional electricity can be purchased from the utility as needed. However, since this system is replacing the boiler, it must produce all of the required steam. You may neglect pressure losses in the steam generator and in the combustor of the gas turbine cycle.

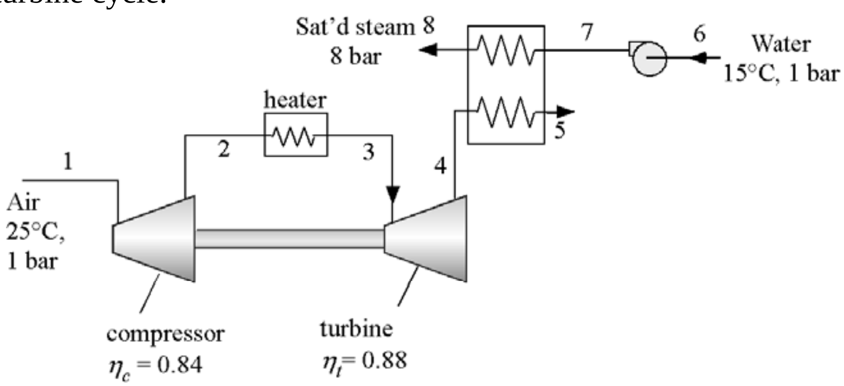

Figure 8.D-4: System to produce steam and electricity

a.) Calculate the current cost of providing electricity and steam in $/hr.
b.) Plot the efficiency of the gas turbine cycle (1-2-3-4) as a function of the pressure ratio (P_2/P_1) for pressure ratios between 3 and 15.
c.) Plot the required air flowrate in kg/s at state 1 as a function of pressure ratio. (Note: combustion gases can be treated as pure air and the mass of fuel can be neglected).
d.) Plot the electrical power in kW produced by the generator as a function of pressure ratio for ratios between 3 and 15.
e.) Calculate and plot the total operating costs to provide electricity and steam in $/hr for the proposed gas turbine cogeneration system as a function of pressure ratio.
f.) What pressure ratio do you recommend based on your results? What is the basis of your recommendation?

8.D-5 The purpose of this problem is to investigate the power – efficiency tradeoffs that govern the performance of the regenerative Rankine cycle shown in Figure 8.D-5. The cycle under consideration is a nominal 35 MW plant with the configuration shown in the figure. The heat source is molten salt from a solar thermal power system that is provided to the boiler/superheater at 540°C, 1 atm at a capacitance rate (mass flow rate specific heat product) of 375 kW/K. Steam exits that boiler at 480°C, 8 MPa. The rate of heat exchange between the molten salt and the steam ($\dot{Q}_b$) can be expressed as:

$$\dot{Q}_b = \varepsilon_b \dot{C}_b (T_{s,in} - T_{sat,b})$$

where

ε_b is the heat exchanger effectiveness for the boiler $= 1 - \exp(-NTU_b)$
$\dot{C}_b$ is the minimum capacitance rate in the boiler which in this case is 375 kW/K
$T_{s,in}$ is the temperature of the entering molten salt (540°C)
$T_{sat,b}$ is the saturation temperature of steam at the boiler pressure
NTU_b is the ratio of the boiler heat transfer conductance (UA_b) to $\dot{C}_b$

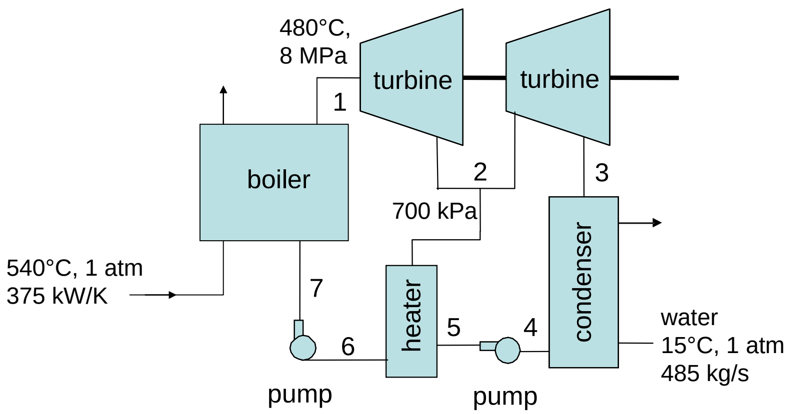

Figure 8.D-5: Regenerative cycle for a solar power system

Steam is expanded in the high pressure turbine to 700 kPa. A portion of the steam is extracted and used to heat the remaining fluid to produce saturated liquid at state 6 in an open feed water heater. The remaining fraction passes through the low pressure turbine and is condensed by heat exchange to cooling water that enters at 485 kg/s, 15°C, and atmospheric pressure. The rate of heat transfer in the condenser can be expressed as:

$$\dot{Q}_c = \varepsilon_c \dot{C}_c (T_{sat,c} - T_{c,in})$$

where

ε_c is the heat exchanger effectiveness for the condenser $= 1 - \exp(-NTU_c)$
$\dot{C}_c$ is the minimum capacitance rate in the condenser, which is due to the cooling water
$T_{c,in}$ is the temperature of the entering cooling water (15°C)
$T_{sat,c}$ is the saturation temperature of steam at the condenser pressure
NTU_c is the ratio of the condenser heat transfer conductance (UA_c) to $\dot{C}_c$

The sum of the boiler and condenser conductances (UA_b + UA_c) is 3500 kW/K. However, the distribution of conductance between the boiler and condenser can vary. An objective of this problem is to determine how the conductance should be allocated to maximize the net power of the cycle. Note that the steam always exits the boiler at 480°C, 8MPa, but the steam flow rate is controlled by the value of the boiler conductance. The pressure in the condenser is controlled by the value of the condenser conductance. States 4 and 6 are both saturated liquid states. The turbine and pumps can be assumed to operate ideally in this preliminary analysis. Also, neglect the pressure losses in the heat exchangers and piping. Prepare a plot of net power versus thermal efficiency for this Rankine cycle and answer the following questions.

a) what is the maximum power of the cycle and the corresponding boiler conductance?
b) compare the efficiency at maximum power to the relation developed for the Carnot cycle. Comment on the agreement.
c) What do expect the plot of power versus efficiency will look like if the turbine and pump efficiencies are 0.80 instead of 1?

8.D-6 A continuous power source is needed for an application outside of Earth's atmosphere. A solar-driven Carnot cycle has been proposed for this purpose. The Carnot cycle operates between the collector and radiator temperatures. The rate of solar energy input to the collector, $\dot{Q}_c$, is governed by the Stefan-Boltzmann law. Assuming black surfaces,

$$\dot{Q}_C = \sigma A_C F_{C,S} \left(T_S^4 - T_C^4\right)$$

where A_C = 2.5 m^2 is the collector area, $F_{C,S}$ = 0.75 is the collector-to-sun view factor, T_S = 5760 K is the equivalent blackbody temperature of the sun, T_C is the collector temperature and σ = 5.67x10^{-8} W/m^2-K^4 is the Stefan-Boltzmann constant. Similarly, the rate of heat rejection from the radiator, $\dot{Q}_R$, is given by:

$$\dot{Q}_R = \sigma A_R F_{R,E} \left(T_R^4 - T_E^4\right)$$

where A_R = 5 m^2 is the radiator area, $F_{R,E}$ = 1 is the radiator-to-space view factor, T_R is the radiator temperature, and T_E = 4 K is the equivalent tempreature of space.

a.) What is the maximum efficiency of this engine?
b.) What is the maximum power output?
c.) What is the efficiency at maximum power output?

8.D-7 The purpose of this problem is to investigate the relationship between efficiency and power production rate in a Carnot cycle in which heat transfer limitations are considered. Consider a Carnot heat engine having no internal irreversibilities that steadily operates on energy supplied from a gas stream having an entering temperature of 1,000°C and capacitance rate of 5,000 W/K. The engine rejects energy to a cooling water stream that enters the heat exchanger at 30°C with a capacitance rate of 10,000 W/K. The energy provided to the cycle is used to vaporize the working fluid, which is thereby maintained at unknown temperature T_h. Similarly, through a condensation process, heat rejection occurs at temperature T_l. The high temperature heat exchanger has an effectiveness of 0.70; the low temperature heat exchanger effectiveness of 0.80. Prepare a plot of the power output of this heat engine as a function of the cycle efficiency. How does this plot change if the capacitance rates of the hot and cold streams are increased or decreased by 10%? Indicate the efficiency you would shoot for in the design of this cycle. State reasons for your choice.

8.D-8 An internally-reversible heat engine steadily operates on energy supplied from a gas stream having that has an entering temperature of $T_{H,in}$ = 1,000°C and capacitance rate of $\dot{C}_H$ = 5,000 W/K as shown in Figure 8.D-8. The engine rejects energy to a cooling stream which enters the heat exchanger at $T_{C,in}$ = 30°C with a capacitance rate of $\dot{C}_C$ = 10,000 W/K. The high temperature heat exchanger has an effectiveness of ε_H = 0.70 and the low temperature heat exchanger effectiveness is ε_C = 0.80. The heat transfer processes to the ideal gas working fluid in the engine occur at constant pressure, just as in the ideal Brayton cycle, as indicated on the temperature-entropy diagram. The product of the mass flow rate and specific heat of the working fluid is $\dot{C}_{wf}$ = 20,000 W/K.

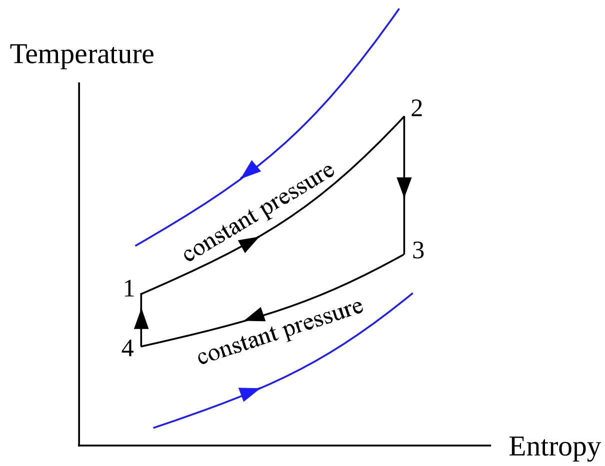

Figure 8.D-8: Temperature-entropy diagram for a heat engine operating as an ideal Brayton cycle.

a.) Prepare a plot of the power output of this heat engine as a function of the cycle efficiency.
b.) Calculate and plot the power output of a Carnot cycle (with the same external streams and heat exchangers) on the same axes.
c.) What can you conclude from these plots?

8.D-9 This problem considers two Carnot cycles operated in series as shown in Figure 8.D-9. The energy source is a hot stream that enters the high temperature heat exchanger at $T_{H,in}$ = 1,000°C at a capacitance rate (mass flow rate specific heat product) of $\dot{C}_H$ = 5,000 W/K. The heat exchanger effectiveness is ε_H = 0.70 for each of the two cycles. A second heat exchanger is used for heat rejection. The external cooling stream enters this heat exchanger at $T_{C,in}$ = 30°C with a capacitance rate of $\dot{C}_C$ = 10,000 W/K. The effectiveness of the low temperature heat exchanger is ε_C = 0.80 for each cycle.

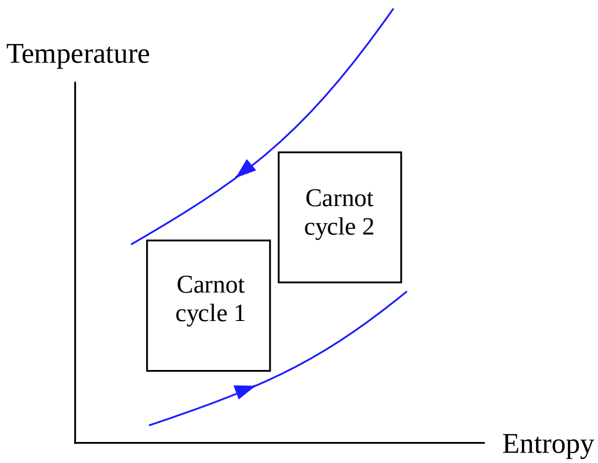

Figure 8.D-9: Power plant consisting of two Carnot cycles.

a.) Determine the power output and efficiency of each stage and the overall system efficiency at the conditions in which the combined work of both cycles is maximized.

b.) Compare the maximum power output and corresponding efficiency with the maximum power output and efficiency of a single stage Carnot cycle operated under the same conditions.

8.D-10 A power generation system is to be designed. The heat input into the cycle is 150 kW and the working fluid in the high temperature part of the cycle is maintained at 800°C as shown in Figure 8.D-19. The internal workings of the power cycle operate at the Carnot efficiency between its low temperature, T_L, and 800°C but T_L depends on the design of the external heat exchanger. Cooling is provided by condensing the working fluid in the engine at constant pressure with cooling water that enters at 30°C. The pressure losses in the heat exchanger are negligible for both the working fluid and the cooling water. The effectiveness of the heat rejection heat exchanger is a function of the cooling water flow rate. At a mass flow rate of 0.2 kg/s, the effectiveness is 0.50.

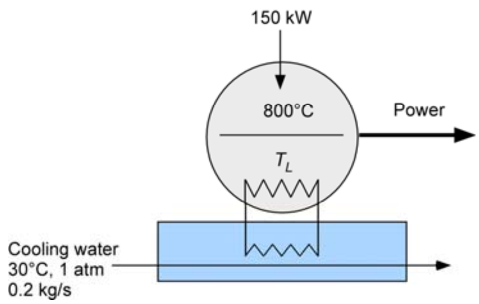

Figure 8.D-10: Thermal power generation system

a.) What is the efficiency of the power generation system?

b.) What is the overall heat transfer coefficient – area product for the cooling water heat exchanger?

c.) What is the maximum possible efficiency of the power generation system?

d.) How will the effectiveness of the condensing heat exchanger be affected if the cooling water flow rate is increased? (Provide a sentence of explanation.)

e.) How will the efficiency at maximum power be affected if the cooling water flow rate is increased? (Provide a sentence of explanation.)

8.D-11 A power module capable of supplying 100 W must be designed for a satellite that will orbit the earth. A heat-power engine is proposed (see Figure 8.D-11) that will operate between two fixed temperatures, as in the Carnot cycle. High temperature thermal energy is directly supplied to the engine at 225 K from a small nuclear reactor. However, the only way in which heat can be rejected from the engine is by radiation to outer space. The rate of heat rejection from the engine is $\dot{Q}_L = \sigma A\left(T_R^4 - T_{space}^4\right)$ where σ is the Stefan-Boltzmann constant = 5.67 x 10^{-8} W/m^2-K^4, A is the surface area of the radiator, T_R is the low temperature seen by the heat-power engine and T_{space} is the equivalent temperature of outer space, estimated to be 4 K. What is the minimum possible radiator area necessary for the power module?

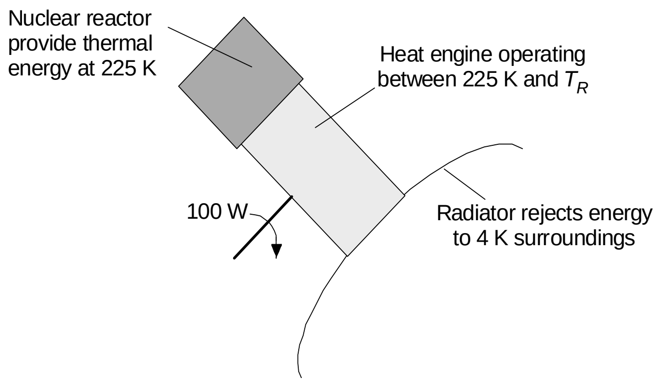

Figure 8.D-11: Nuclear powered heat engine in space

8.D-12. Most electric utilities in the U.S. provide base load electrical power with generators driven by Rankine steam cycles. Peak electrical loads are typically met with gas turbine-driven generators. Gas turbine systems offer the advantage of being able to start quickly, but they typically operate at lower efficiency than Rankine cycles. It has been proposed to use off-peak electrical power produced by the more efficient Rankine steam cycle to compress air into an underground cavern. When peak electricity is needed, the air is withdrawn and heated to 800°C by combusting natural gas, and run through turbines to produce power. The turbine exhausts to the environment. In a particular case, the cavern has a volume of 0.365 m^3. The cavern initially contains air at 25°C and 14 bar. During operation, the air pressure in the cavern is reduced from 14 bar to 11 bar in two hours while the air temperature of the air remaining in the cavern stays at 25°C as a result of heat transfer with the cavern walls. Since the air-fuel ratio is large for gas turbine systems, assume that the gas passing through the turbine is pure air with R=287 J/kg-K and c_p = 1030 J/kg-K. Neglect the mass of the fuel.

a.) Estimate the maximum total electrical energy that can be generated during the two hour period using the compressed air from the cavern to drive the gas turbine.

b.) Determine the thermal efficiency of the turbine for this 2 hour operation period.