Design Features of Combined Cycle Systems

1.0 Introduction

As we have discussed in class, one of the largest irreversibilities associated with simple gas turbine cycles is the high temperature exhaust. Exhaust temperatures of around 800-1000 K imply a substantial waste of work potential. The largest irreversibility in Rankine cycles is associated with the boiler. Here, the steam temperature starts near ambient and exits at a temperature that seldom exceeds 900 K. The combustion gas stream that is providing this heat starts at the flame temperature (~2000 K) and exits around 450 K. Even in a counterflow configuration, the heat exchanger (which is all a boiler really is) has a large ΔT between the hot and cold streams, which results in substantial irreversibility. The most positive aspect of the Rankine cycle is that the heat rejection takes place via condensation (*i.e.,* isothermally), and by running the condenser at a vacuum the ΔT between the condenser and the cooling medium can be kept low. This substantially reduces the irreversibility associated with the heat rejection step.

The idea behind modern combined cycle systems is to use the high temperature gas turbine exhaust to power a Rankine cycle. In this way, each cycle attacks the weakest point in its partner's performance. In almost every way, the two cycles behave the same in the combined mode as they do as separate cycles, so there is very little new to learn in terms of analysis procedures or technology. As we will see, most of the performance issues are associated with the heat exchanger that sits between the two cycles. The industry calls this system the Heat Recovery Steam Generator, or HRSG. Most combined cycle design issues are associated with reducing the irreversibility of the HRSG.

2.0 Simple Combined Cycle Description

Figure 1 shows a simple combined cycle schematic.

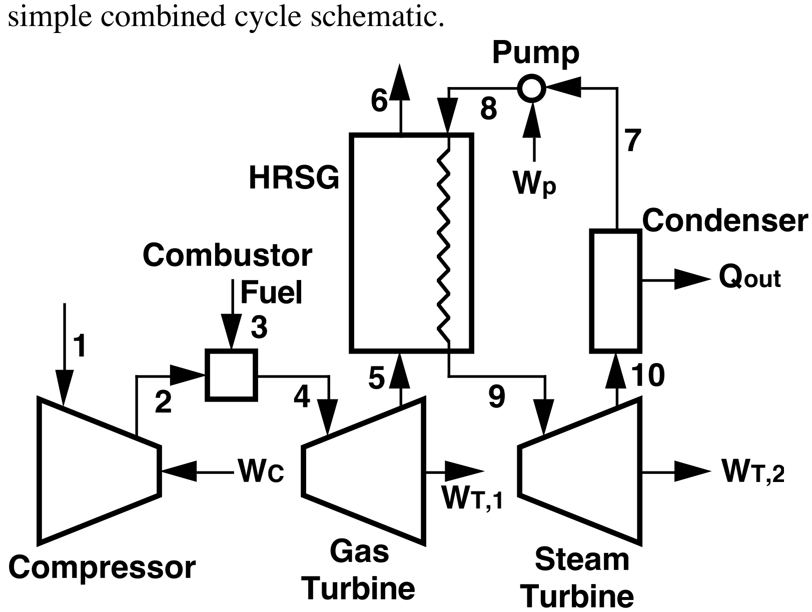

Figure 1. Simple combined cycle schematic.

Note that the HRSG is designed as a countercurrent heat exchanger. If we make a graph of both gas and steam temperature as we move through the exchanger, starting from the bottom and moving up, we will see a plot similar to Figure 2.

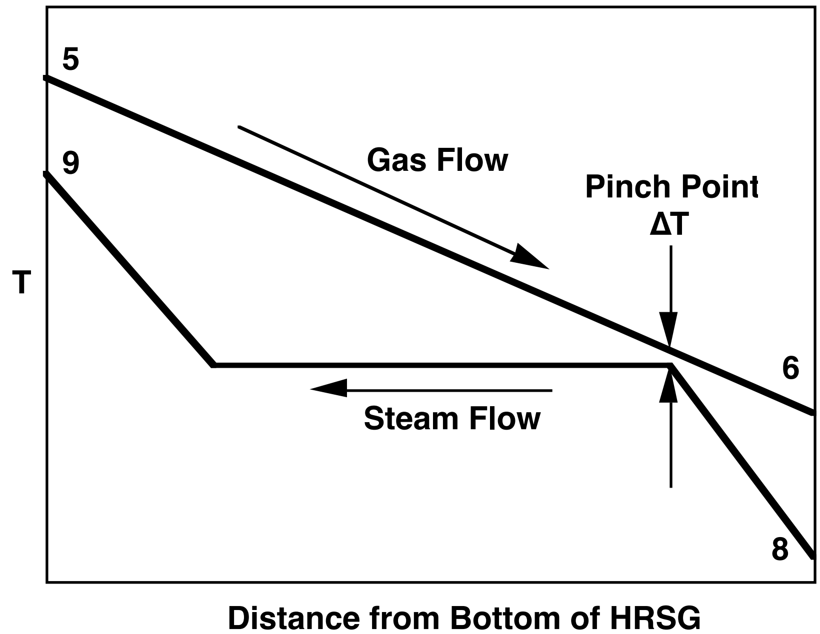

Figure 2. Temperature profiles through a simple HRSG.

The figure shows that the gas temperature tends to decline uniformly since it is giving up only sensible heat. As you might expect on the steam side, the feed water is first heated to the boiling point. Following this, the feedwater is boiled to saturated steam in a constant temperature process. Finally, the saturated steam is superheated before being discharged to the turbine.

Since irreversibility is associated with Q flowing across a ΔT, it is clear that a HRSG has a built-in problem. You will always have a region where the gas temperature is decreasing while the steam temperature is staying constant. Addressing the issue of how to reduce this ΔT, and thus reduce irreversibility, is the driving force behind most combined cycle design work.

In the next few sections we will present some of the most important design features and augmentations to combined cycle design.

3.0 Deaeration and Feedwater Heating

In normal Rankine cycles, a complex network of regenerative feedwater heaters is used to increase the temperature at which the feedwater enters the boiler. These are generally not used in combined cycles. Looking at Figure 2, if T_8 is increased due to regenerative feedwater heating, this will lead to an increase in T_6 since you will not need as much heat from the gas stream. This leads to an increased exhaust temperature, which obviously means an increased irreversibility. The key is to bring the feedwater in contact with the gas stream at as low a temperature as possible to get the most heat out of the gas stream.

There are, however, two reasons for using some feedwater heating. The first is to deaerate the steam (as is done in our power plant). The second is to prevent corrosion due to condensation on the exterior of cold feedwater tubes in the HRSG. We will explore both of these functions using Figure 3.

3.1 Air Ejection

Air inleakage into steam cycles is almost impossible to prevent. The major problem area is the condenser, which operates substantially below atmospheric pressure. Air also enters the steam cycle with makeup water. The main problem with air is that the O_2, when present in the superheated steam (*i.e.,* under high temperatures), becomes corrosive. To remove this air, a small amount of steam is sometimes extracted partway through the turbine (stream 12). This is added to the feedwater to just bring it to the boiling point (*i.e.,* a saturated liquid). Since the solubility of air in water decreases with temperature, the air bubbles off of the liquid and is mechanically separated and ejected. You see the same thing happen when you heat cold water in a saucepan on the stove. Initially, you see small bubbles collecting on the bottom of the pan. These are not steam bubbles, but air that has been driven out of solution.

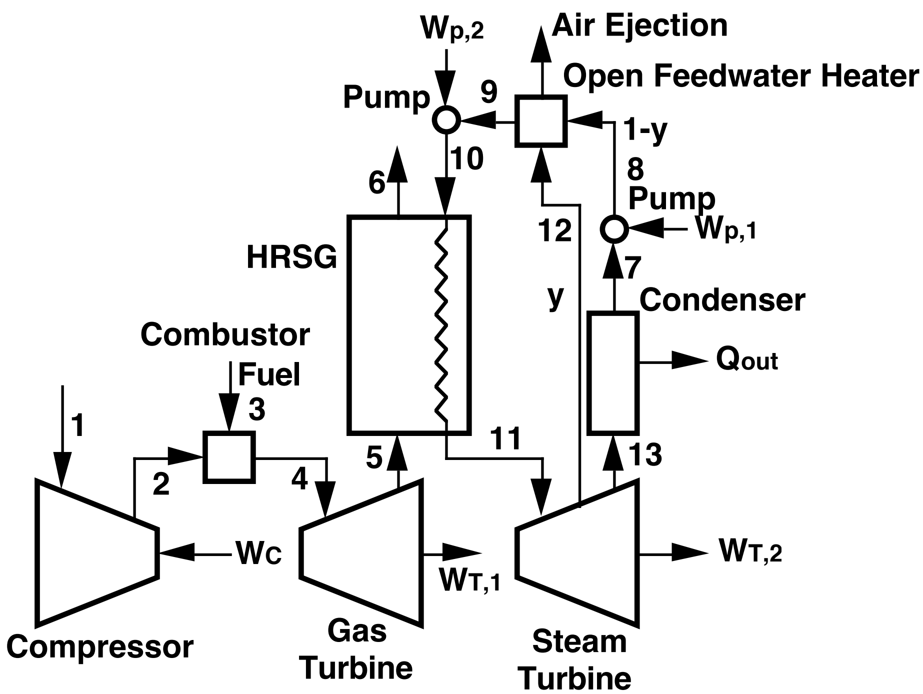

Figure 3. Simple combined cycle system with a single open feedwater heater.

For this reason, most simple Rankine cycle systems (and some combined cycle systems) will have a moderate amount of open feedwater heating. The higher temperature at 6 is not a problem in Rankine cycles because the stream at 6 is often used in a heat exchanger to preheat the combustion air, allowing for good heat recovery.

An alternative is to replace the open feedwater heater with a deaerating condenser. These systems make use of a complex condenser design involving vacuum pumps to directly remove the air in the condenser. This allows for colder feedwater to enter the HRSG than in the open feedwater heater case, resulting in higher heat extraction from the gas turbine exhaust and higher efficiency. There is no way to use the stream at 6 for air preheating in a combined cycle system since (1) preheating the compressor air increases the W_c, and (2) the temperature at point 2 is already higher than that of point 6, meaning you cannot use the exhaust to preheat the compressed air before the combustor. Hence, the deaerating condenser is generally the preferred solution for combined cycles.

3.2 Corrosion Prevention

This issue requires us think a little about the heat transfer processes in the HRSG.

From basic convective heat transfer, we know that the heat flow rate is given by:

$$\dot{Q} = hA_s(T_\infty - T_0) \tag{1}$$

Here, A_s is the exposed heat transfer surface, the two T's are the temperatures of the surface and the free stream fluid, and h is the heat transfer coefficient . Thus, h can be thought of as an effectiveness for heat transfer: in other words the amount of heat delivered over a 1 m^2 surface by 1 K temperature difference between the surface and the free stream fluid.

Values for h depend on the flow geometry, flow velocity, *etc*. However, approximate values can be given for the situations relevant in the HRSG. These are:

Situation	h (W/m^2-K)
Gas to surface heat transfer	20
Steam to surface heat transfer	20
Liquid water to surface heat transfer	700
Heat transfer during boiling over a surface	20,000

So we see that the heat transfer between a liquid and a surface (*e.g.,* feedwater heating) is about 35 times as fast as heat transfer between a gas and a surface (*e.g.,* the combustion product gas in contact with the feedwater tubes). This is the reason you have fins on the air side of a car radiator. You need extra surface area on the air side to overcome the poor heat transfer properties of air.

Since the liquid heat transfer is fast and the gas is slow, the temperature profile one would expect to see across a feedwater tube is shown in Figure 4.

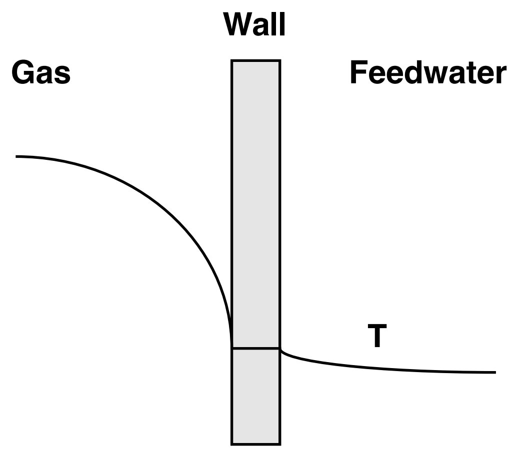

Figure 4. Temperature profile near the wall of a feedwater tube.

Here, most of the ΔT is across the gas boundary layer. This means that the tube wall in contact with the gas will be essentially at the feedwater temperature.

If the feedwater enters the HRSG at too low a temperature, one could get condensation on the air side of the feedwater tube, leading to a long-term corrosion problem. To prevent this, feedwater heating is sometimes used such that the feedwater enters the HRSG above the dewpoint temperature of the gas. (For 200% theoretical air, CH_4/air combustion products have a dewpoint of about 50°C.) For fuels that contain sulfur, the temperature needs to be above about 150°C to prevent H_2SO_4 condensation. For many natural gas-fired systems, the alternate solution is to use the deaerating condenser, take the thermodynamic benefits of the colder feedwater, and deal with the condensation by using advanced materials.

4.0 HRSG Heat Transfer Design

We are now ready to tackle the details of the HRSG design. Figure 5 shows a typical design where the numbered stations are referenced to Figure 3.

The feedwater enters near the gas exit and flow countercurrently against the gas flow. Upon reaching the saturation point, it exits (point 10a) and is placed in the steam drum. The steam drum consists of a horizontal cylinder that is filled about half way with saturated liquid under saturated vapor. The boiling takes place in the loop consisting of points 10b and 10c. The boiling within the HRSG reduces the density of the liquid vapor mix in the tube, and the vapor tends to rise into the steam drum. This sets up a natural circulation in which liquid is pulled out of the drum (point 10c) and moved into the tubes to replace the vapor that has moved upward. This natural circulation keeps the boiler tubes supplied with liquid and delivers the saturated vapor to the drum. The saturated vapor is tapped off from the top of the drum (point 10d), and it moves to the superheater. Here the steam is contacted with the hottest gas from the gas turbine cycle. The superheated steam is removed at point 11 for the turbine.

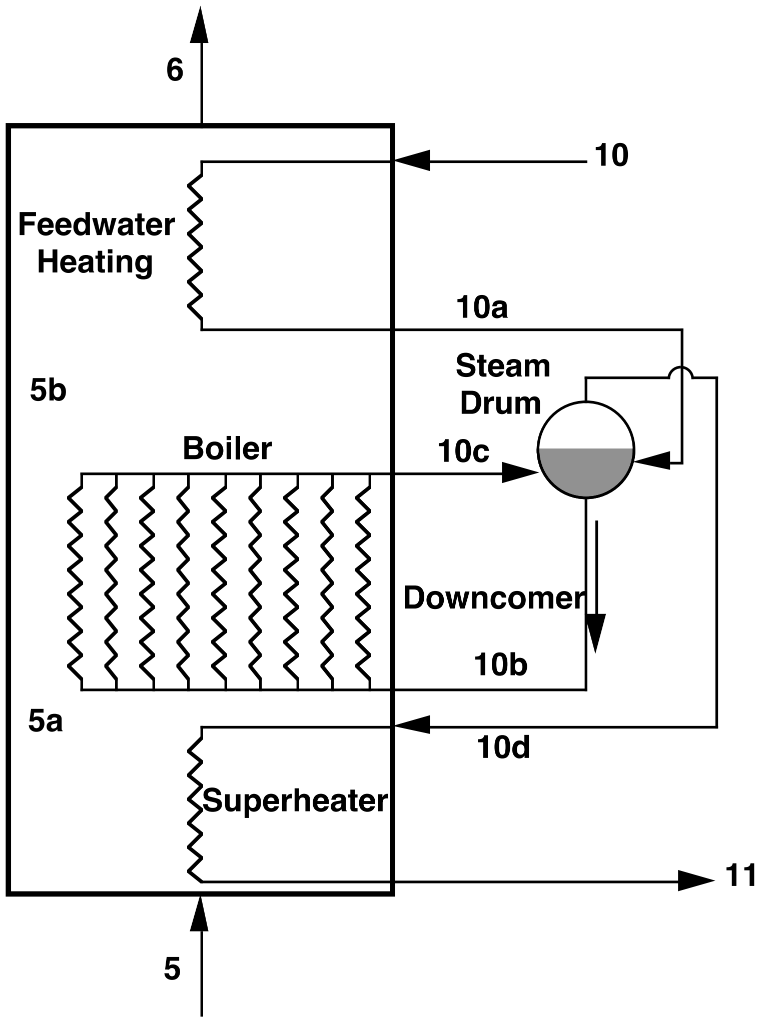

Figure 5. Details of the design of a HRSG.

In some systems a circulation pump is used in the boiler area to replace the natural circulation shown in Figure 5. The natural circulation systems need a significant height difference between the top and bottom of the downcomer to develop the buoyancy needed to drive the circulation. If space is at a premium, circulation pumps allow the boiler to occupy a more compact space, but with additional cost and complexity.

Since the boiling heat transfer is approximately 1000 times faster than the gas heat transfer in the boiler, extended surfaces are commonly used to increase the heat transfer area on the gas side. The are also used in the feedwater heater portion since the liquid heat transfer is about 35 times faster than the gas heat transfer. In the superheater, both the gas and steam are gases, so the heat transfer rates are similar, and there is no need for extended surfaces on the air side.

In later figures we will not show all the detail given in Figure 5, but it should be implicitly understood that this is how a HRSG is configured.

5.0 HRSG Thermodynamics

An availability analysis of the simple system shown in Figure 3 will reveal that the largest irreversiblilty lies in the combustor. Unfortunately, there is very little we can do about this without resorting to a direct conversion system (*i.e.,* a fuel cell). The only approach is to ensure that the combustor exit temperature is as hot as it can be. The next biggest irreversibility is in the HRSG, and this irreversibility is a direct consequence of the ΔT that occurs in the heat exchanger. Here, one side has a boiling fluid (*i.e.,* accepting heat at a constant T), while the other side does not have a phase change and thus changes temperature continuously as it gives up its heat. This is the situation illustrated in Figure 2.

If we make a heat exchanger, such as the HRSG, arbitrarily large, it will arrive at one of three limiting states:

1. The temperature of the steam at the outlet will rise to the temperature of the gas inlet.
2. The temperature of the gas outlet will fall to that of the feedwater inlet.
3. The pinch point ΔT will tend to zero.

In heat transfer, the ΔT at the limiting point is inversely proportional to the exchanger surface area. Thus, squeezing the last ΔT out of a HRSG requires a huge increase in surface area. The industry has found a rule of thumb from economics that says the most economical design (*i.e.,* one that balances capital vs. operating cost) is to set the limiting ΔT to 10 K. We must add the additional condition that the steam outlet temperature should not exceed 620°C since this is nominally the maximum steam temperature allowed by modern superheater tube materials and steam turbine components. This condition can be met by reducing the superheater surface area.

There are a number of ways to calculate the limiting condition. I like the following because it involves fewer steps than its competitors, and it leads you right to the most favorable condition. The goal of this HRSG analysis is to find the steam flow, and the implicit goal is to get the maximum steam flow possible since this will maximize the net work from the steam cycle.

The first step is to pick the steam pressure. This is a free variable (one that should be optimized as a "super iteration" over the procedure that follows). The next steps are as follows.

1. Find the steam flow via a first law across the entire HRSG. This is done via:

$$m_g(h_5-h_6)=m_s(h_{11}-h_{10})$$

   Here, the air flow is known from the Brayton cycle, h_{11} is set from the 620°C steam condition, h_{10} is known from the feedwater outlet temperature, h_5 is known from the Brayton cycle, and h_6 is set from the condition:

$$T_6=T_{10}+10K$$

   The only unknown is the steam flow, which is then calculated.

2. The next step is to establish whether the pinch point is a problem or not. This is done by performing a first law balance around the feedwater heater. With reference to Figure 5:

   $$m_g(h_{5b}-h_6)=m_s(h_{10a}-h_{10})$$

   Here, h_{10a} is the enthalpy of saturated liquid at the selected boiler pressure. Thus, everything is known except h_{5b}, which is calculated. Now the equilibrium code is used in the TP mode (iteratively) to find the temperature associated with point 5b. Finally, you must determine:

   $$\text{Is } T_{5b} \geq T_{10a}+10 \text{ K ?}$$

   If the answer is yes, then you are done, and you can use the calculated m_s to finish your evaluation of the steam part of the cycle. If the answer is no, then you need to recalculate m_s based on a pinch point differential of 10 K.

3. If the pinch point is controlling, then you set up a first law around the combined superheater and boiler:

   $$m_g(h_5-h_{5b})=m_s(h_{11}=h_{10a})$$

   Here, h_{5b} is selected from the condition:

   $$T_{5b}=T_{10a}+10 \text{ K}$$

   This establishes the correct m_s, which will be less than that calculated in part 1. The final step is to find the correct gas exhaust condition (point 6). This is done via a first law on the feedwater heater:

   $$m_g(h_{5b}-h_6)=m_s(h_{10a}-h_{10})$$

   This gives h_6, and the equilibrium code can then be used to find T_6 and s_6. With m_s set, the performance of the steam cycle can then be calculated.

Let's work a numerical example to illustrate. Let's choose the following parameters:

Gas Turbine:	$r_p=12$	$T_1=300$ K
	Compressor efficiency=0.85	$T_3=1700$ K
	Turbine efficiency=0.90	

To simplify the problem, let's take air standard assumptions as opposed to the more correct reacting gas approach. This will imply that stream 3 (the fuel) in Figure 3 is replaced by Q_{in}. On the steam side, assume:

Boiler pressure=5 MPa	Steam turbine efficiency=0.78
Feedwater heater pressure=200 kPa	Steam superheat=620°C

Condenser pressure=10 kPa

With these parameters selected, the following properties can be found for the various points on the cycle by the usual cycle evaluation procedures (referenced to Figures 3 and 5):

Point	T(K)	h(kJ/kg)	s(kJ/kg-k)
1	300	300.19	1.7058
2	655	665.43	1.7924
4	1700	1880.3	2.8885
5	1010	1057.4	2.9828
5a	?		
5b	?		
6	?		
7	319	191.83	0.6493
8	319	192.02	0.6493
9	393	504.70	1.5301
10	393	509.5	1.5301
10a	537	1154.23	2.9202
10d	537	2794.3	5.9734
11	893	3713.17	7.3096
12	531	2987.03	7.7378
13	339	2623.78	8.2635

The fraction of the steam that is extracted for the feedwater heater (y) is calculated via a first law balance on the feedwater heater to be: y=0.112.

Next, we work on the HRSG to get the total steam flow. First assume the ΔT at the exhaust is the limiting point. This means that:

$T_6=T_{10}+10K=403$ K This implies that $h_6=404.5$ kJ/kg. Now do a first law on the HRSG:

$$m_a(h_5-h_6)=m_s(h_{11}-h_{10})$$

or: $(1)[1057.3-404.5] = m_s[3713.17-509.3]$ which yields $m_s=0.204$ kg/s

Now we need to check the pinch point. This is done by performing a first law balance over the feedwater heater, since we know everything in such a balance except h_{5b}.

$$m_a(h_{5b}-h_6)=m_s(h_{10a}-h_{10})$$

or: $(1)[h_{5b}-404.5] = (0.204)[1154.23-509.5]$ which yields $h_{5b}=536.0$ kJ/kg, which in turn yields $T_{5b}=532$ K. This is less than T_{10a}, so we have a pinch point violation and we must scrap the calculation and start over assuming the pinch point is the controlling point.

Next we assume:

$T_{5b}=T_{10a}+10K=547$ K (a ΔT of 10 K for the pinch point). This implies that $h_{5b}=551.5$ kJ/kg. Now do a first law on the combined superheater and boiler:

$$m_a(h_5-h_{5b})=m_s(h_{11}-h_{10a})$$

or: (1)[1057.3-551.5] = m_s[3713.17-1154.23] which yields $m_s=0.1976$ kg/s

Now we need to find the exhaust point temperature (T_6). This is done by performing a first law balance over the whole HRSG

$$m_a(h_5-h_6)=m_s(h_{11}-h_{10})$$

or: (1)[1057.3-h_6] = m_s[3713.17-509.3] which yields $h_6=424.25$ kJ/kg, which in turn yields $T_6=423$ K and $s_6=2.0522$ kJ/kg-K.

Now we can pull the performance numbers together.

$W_c=ma(h_2-h_1)=365.24$ kW
$W_T=m_a(h_4-h_5)=823.0$ kW
$(W_{net})_{GT}=W_T-W_c=457.8$ kW
$Q_{in}=m_a(h_4-h_2)=1215$ kW
$(Q_{in})_{steam}=m_s(h_{11}-h_{10})=633.1$ kW
$(Q_{out})_{steam}=(1-y)m_s(h_{13}-h_7)=426.7$ kW
$(W_{net})_{steam}=(Q_{in}-Q_{out})_{steam}=206.4$ kW
$(W_{net})_{total}=664.2$ kW
Classical cycle efficiency=$(W_{net})_{total}/Q_{in}GT=55\%$

Finally, let's close an availability balance around this system. The availability of the Q_{in} coming into the cycle is equivalent to the Carnot work that could be generated by the Q. Asuming a 2000 K temperature for the high temperature reservoir:

$W_{max}=(Q_{in})_{GT}[1-(T_0/T_H)]=1033$ kW

$i_{compessor}=m_a \bullet T_0(s_2-s_1)=26.0$ kW
$i_{combustor}=m_a \bullet T_0[s_4-s_2-(h_4-h_2)/T_H]=146.6$ kW
$i_{turbine}=m_a \bullet T_0(s_5-s_4)=28.3$ kW
$i_{HRSG}=T_0[m_a(s_6-s_5)+m_s(s_{11}-s_{10})]=63.43$ kW
$i_{exhaust}=m_a[h_6-h_0-T_0(s_6-s_0)]=20.1$ kW
$i_{turbine}=T_0 \bullet m_s[y \bullet s_{12}+(1-y)s_{13}-s_{11}]=53.1$ kW
$i_{condenser}=T_0(1-y)m_s[s_7-s_{13}+(h_{13}-h_7)/T_0]=25.9$ kW
$i_{feedwater\ heater}=T_0 \bullet m_s[s_9-y \bullet s_{12}-[1-y)s_8]=5.2$ kW

The sum of the net work and the irreversibilities is 1032.8 kW, which matches W_{max} within rounding. Reducing these results to percentages:

W_{max}		100%	
Net Work		64.3	
Irreversibilities	Gas compressor		2.5
	Combustor	14.2	
	Gas turbine	2.7	
	HRSG	6.1	
	Gas exhaust	1.9	
	Steam turbine	5.1	
	Condenser	2.5	
	Open feedwater heater		0.5

So we see we have a pretty good system. The adoption of the air standard assumption instead of using a reacting system and an adiabatic combustor means the irreversibility for the combustor, and W_{max} will be somewhat unrealistic. However, the remaining irreversibilities and the net work will be in correct proportion to each other. (In any case, we can do little about the irreversibility for the combustor.)

This indicates that most of the remaining irreversibility is associated with the steam turbine and the HRSG. The savings for going with a higher efficiency turbine would have to be balanced against the capital cost. The HRSG irreversibility can be reduced by moving to a multi-pressure system, as illustrated in the next section.

6.0 Multi-Pressure Systems

Two-pressure systems are similar in most respects to the single pressure system illustrated in Figure 3. The only differences are around the HRSG. Figure 6 shows a basic two-pressure system. The feedwater enters at point 13 as usual, and his heated the saturation point. This is introduced into the first steam drum. A boiling loop converts some of this liquid to saturated vapor, and this leaves for the turbine as stream 15. The remaining liquid is pumped to the final

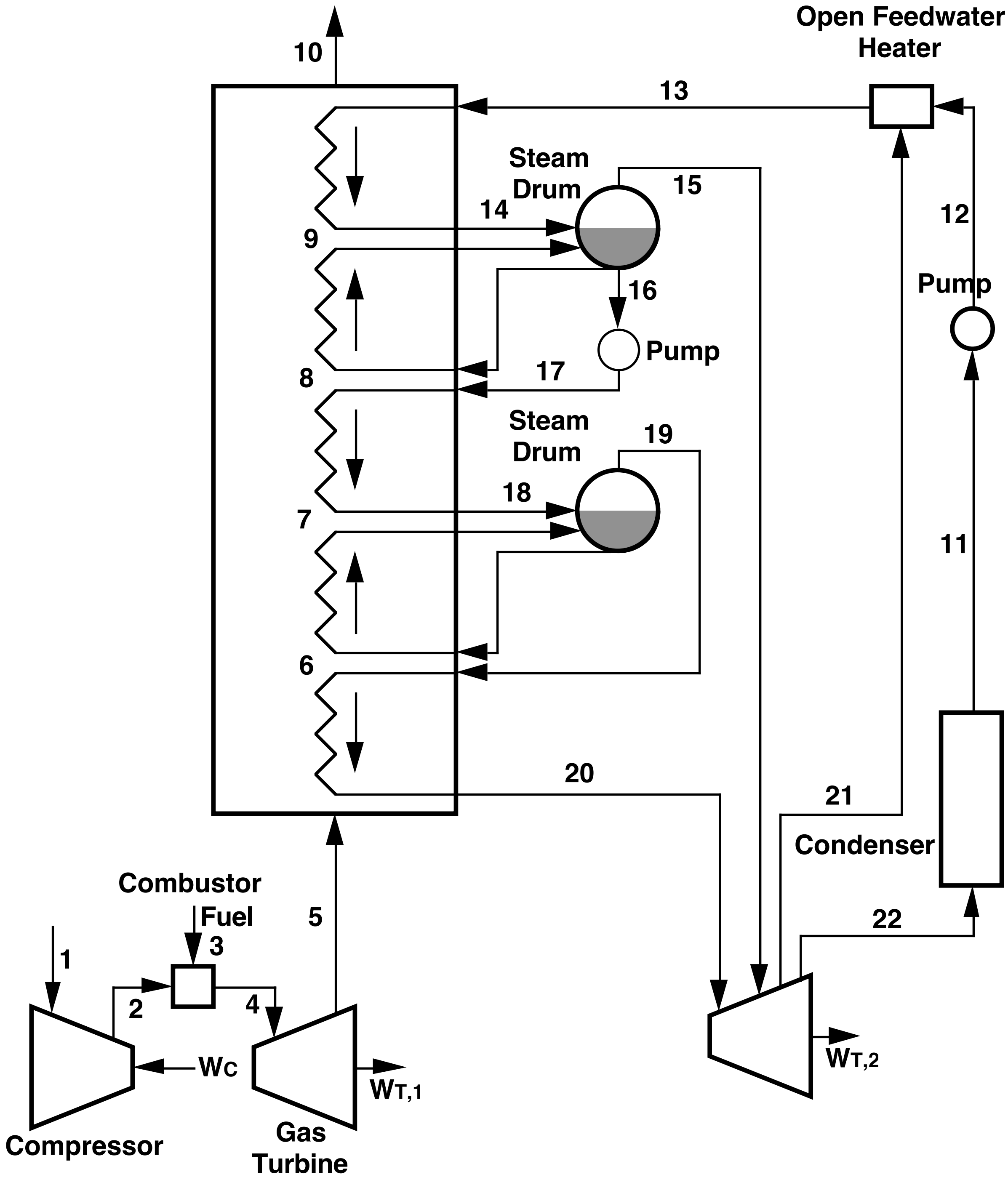

Figure 6. Schematic of a two-pressure combined cycle system.

pressure and run through a standard feedwater heater, boiling loop, and superheater, exiting for the turbine at point 20. The action of the two-pressure approach can probably be best appreciated by studying a temperature-distance diagram, as shown in Figure 7. After the feedwater is heated to point 14, part is boiled to point 15 and sent to the turbine. The rest is bypassed to a hotter point in the gas flow, where it is pumped to a higher pressure, and subjected to feedwater heating, boiling, and superheating. Note that this results in a narrowing of the ΔT in the HRSG, which means less irreversiblity and more net work. Real-world high-efficiency systems may repeat this pattern with up to five pressure levels to further reduce HRSG irreversibility.

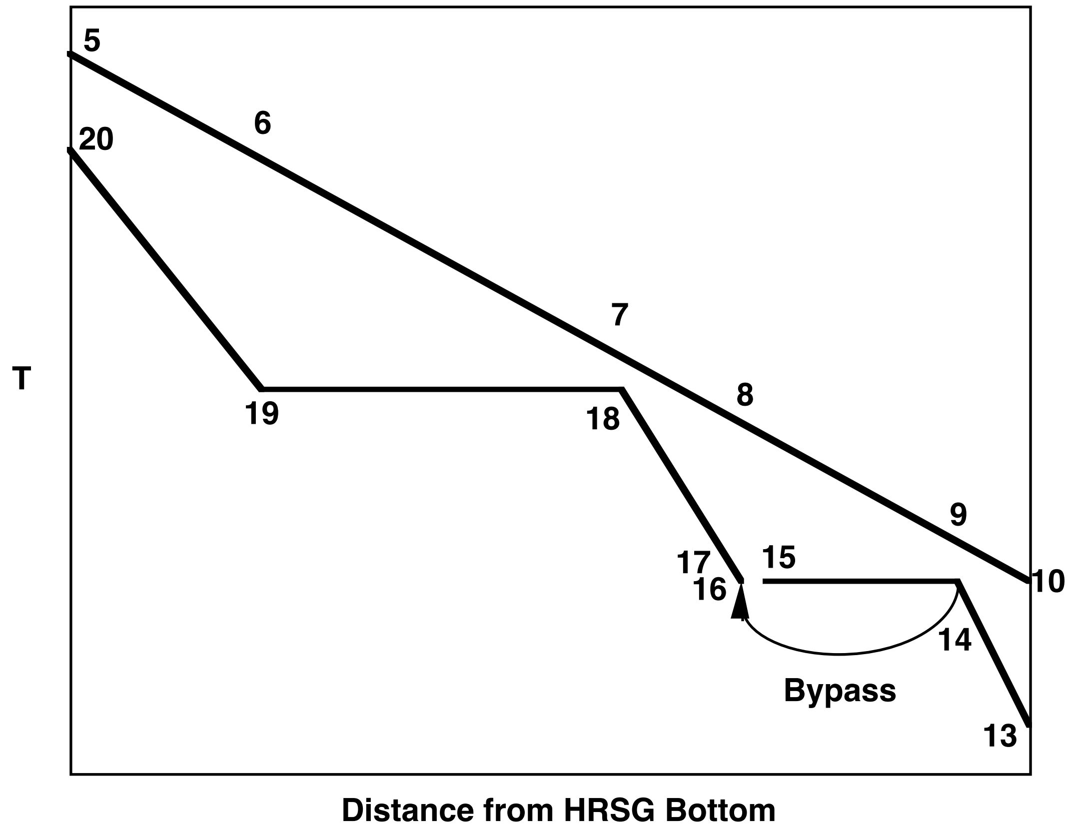

Figure 7. Temperature profile through a two-pressure HRSG.

Note that this system requires two inlets to the steam turbine at different locations. This complicates the usual procedure for finding the outlet conditions of the turbine (extraction and exhaust), using a known turbine efficiency.

The approach is to divide the turbine into two halves. The first half (points 20 to 15) is treated as usual to find the state of the steam just before stream 15 is added. Then this outlet stream is mixed with stream 15 adiabatically to determine the h and s of the steam entering the second stage. This steam is then expanded in the second stage by the usual procedures.