Chapter Two

The Rankine cycle

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The Ideal Rankine Cycle

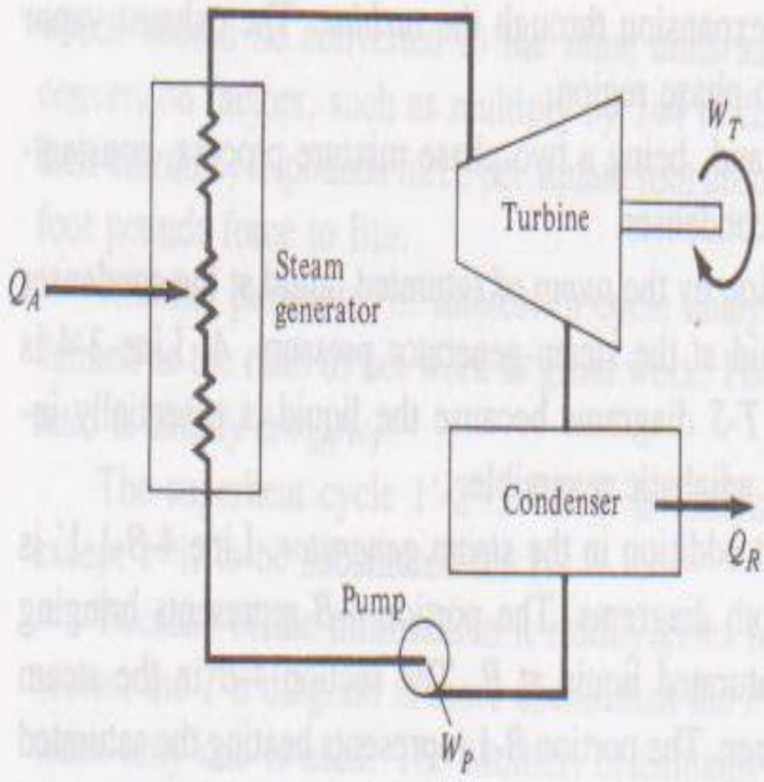


Figure 2-1 Schematic flow diagram of a Rankine cycle.

Schematic Diagram of ideal simple Rankine

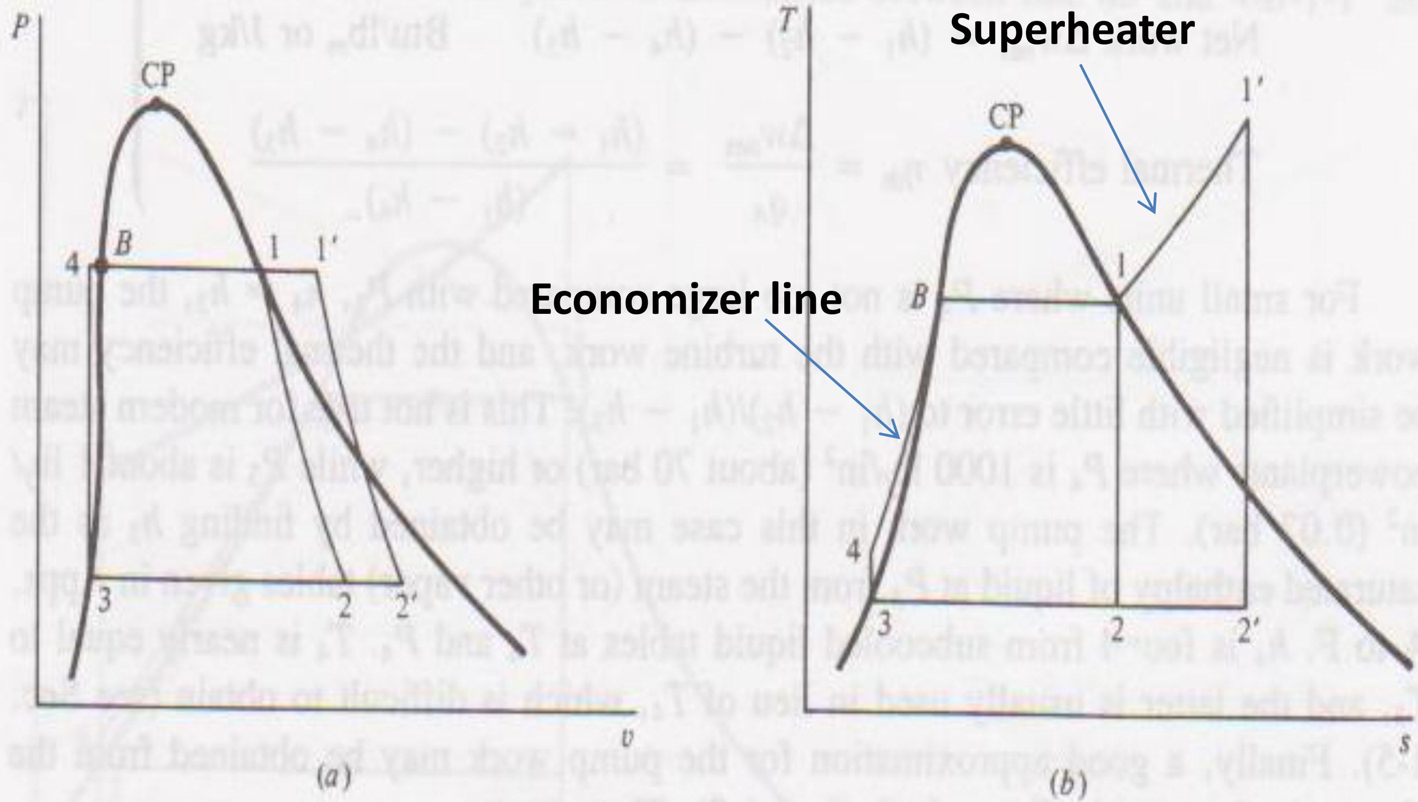


Figure 2-2 Ideal Rankine cycles of the (*a*) *P-v* and (*b*) *T-s* diagrams. 1-2-3-4-*B*-1 = saturated cycle. 1^1-2^1-3-4-*B*-1^1 = superheated cycle. CP = critical point.

Heat Addition Types In The Steam Generator

- Sensible heat addition in the economizer and the superheater Line (4-B, B-$1^{/}$).

- Latent heat transfer in the boiler (B-1).

$$\left.\begin{array}{ll}
\text{Heat added } q_A = h_1 - h_4 & \text{Btu/lb}_m \text{ or J/kg} \\
\text{Turbine work } w_T = h_1 - h_2 & \text{Btu/lb}_m \text{ or J/kg} \\
\text{Heat rejected } |q_R| = h_2 - h_3 & \text{Btu/lb}_m \text{ or J/kg} \\
\text{Pump work } |w_p| = h_4 - h_3 & \\
\text{Net work } \Delta w_{\text{net}} = (h_1 - h_2) - (h_4 - h_3) & \text{Btu/lb}_m \text{ or J/kg} \\
\text{Thermal efficiency } \eta_{\text{th}} = \dfrac{\Delta w_{\text{net}}}{q_A} = \dfrac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)} &
\end{array}\right\} \quad (2\text{-}1)$$

$$|w_p| = v_3(P_4 - P_3) \qquad (2\text{-}2)$$

The Externally Irreversible Rankine Cycle

- External irreversibility is a result of the temperature difference between the primary heat source and the working fluid.
- Temperature difference between condensing working fluid and the heat sink fluid, which is usually the condenser cooling water.

1-Effect of the heat source type (heat exchanger)

- There are two types of heat exchangers: 1- parallel flow 2- Counter flow

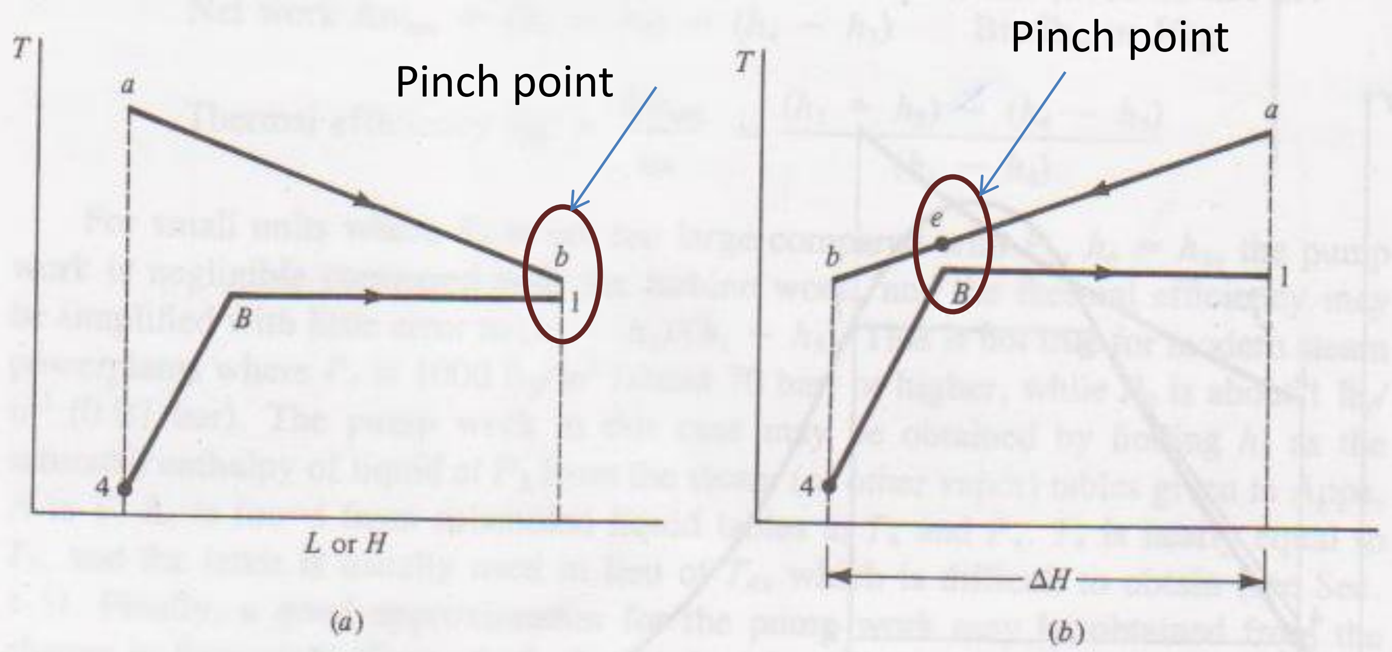


Figure 2-4 Effect of flow direction on external irreversibility: (a) parallel flow, (b) counterflow.

- Pinch-point: is the minimum approach between the working fluid line and the primary heat source line and it must be finite
- Too small a pinch-point temperature difference results in lower irreversibility and higher efficiency, but costly steam generator.
- Too large a pinch-point temperature difference results in more irreversibility and small but cheap steam generator.
- The most economical point is obtained by optimization that takes into account the (a) fixed charges (based on the capital costs) (b) operating costs (based on efficiency and thence fuel costs).
- Counter flow heat exchangers are preferred over parallel flow ones from both thermodynamics and heat transfer point of view.

2-Effect of the type of heat source fluid

- There are different types of heat source fluids such as: 1-combustion gases 2- water from a pressurized-water reactor, or molten sodium from a liquid metal fast breeder reactor.
- These fluids has different mass flow rate and specific heat c_p . Water has higher c_p than gases.

Assuming that a differential amount of heat dQ exchanged between the two fluids is proportional to a path length dL and that $dQ = \dot{m}c_p\, dT$, where dT is the change in primary-fluid temperature in dL, the slope of line ab is then proportional to the reciprocal of $\dot{m}c_p$ or

$$\frac{dT}{dL} \propto \frac{1}{\dot{m}c_p} \quad \text{(primary fluid)} \qquad (2\text{-}3)$$

- For a given pinch point temperature difference over all temperature difference between the primary and the working fluid is greater in the case of gases than water especially in the boiler section, which determines whether or not to use superheat or reheat.

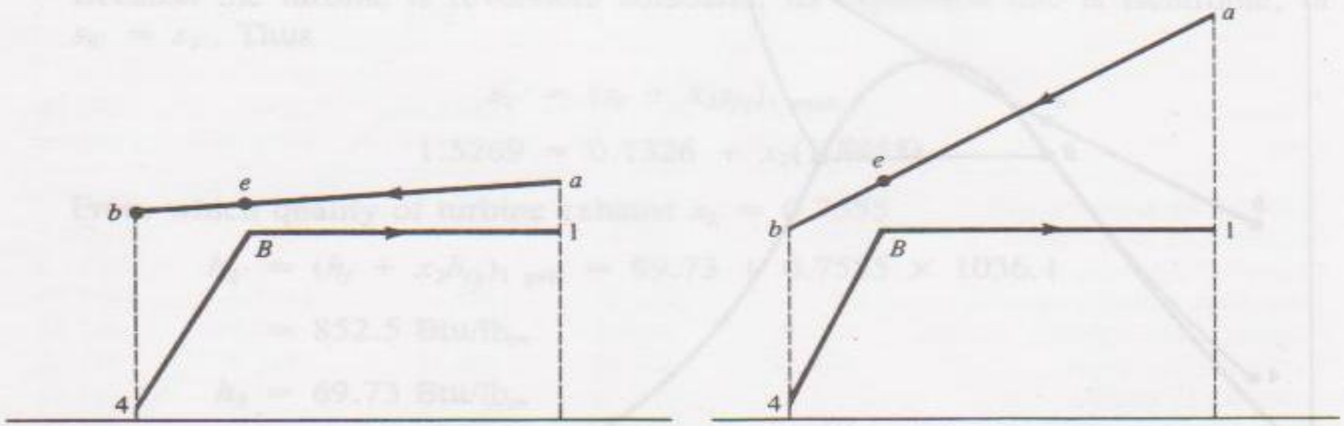


Figure 2-5 Effect of primary fluid type on external irreversibility: (*a*) water, (*b*) gases or liquid metal.

We note that there are two distinct regions where the external irreversibility exists at the higher-temperature end of the cycle. These are: (1) between the primary fluid and the working fluid in the boiler section, i.e., between *ae* and *B*-1, and (2) between the primary fluid and the working fluid in the economizer section, i.e., between *be* and 4-*B*. We shall deal with these in turn in the next two sections.

Superheat

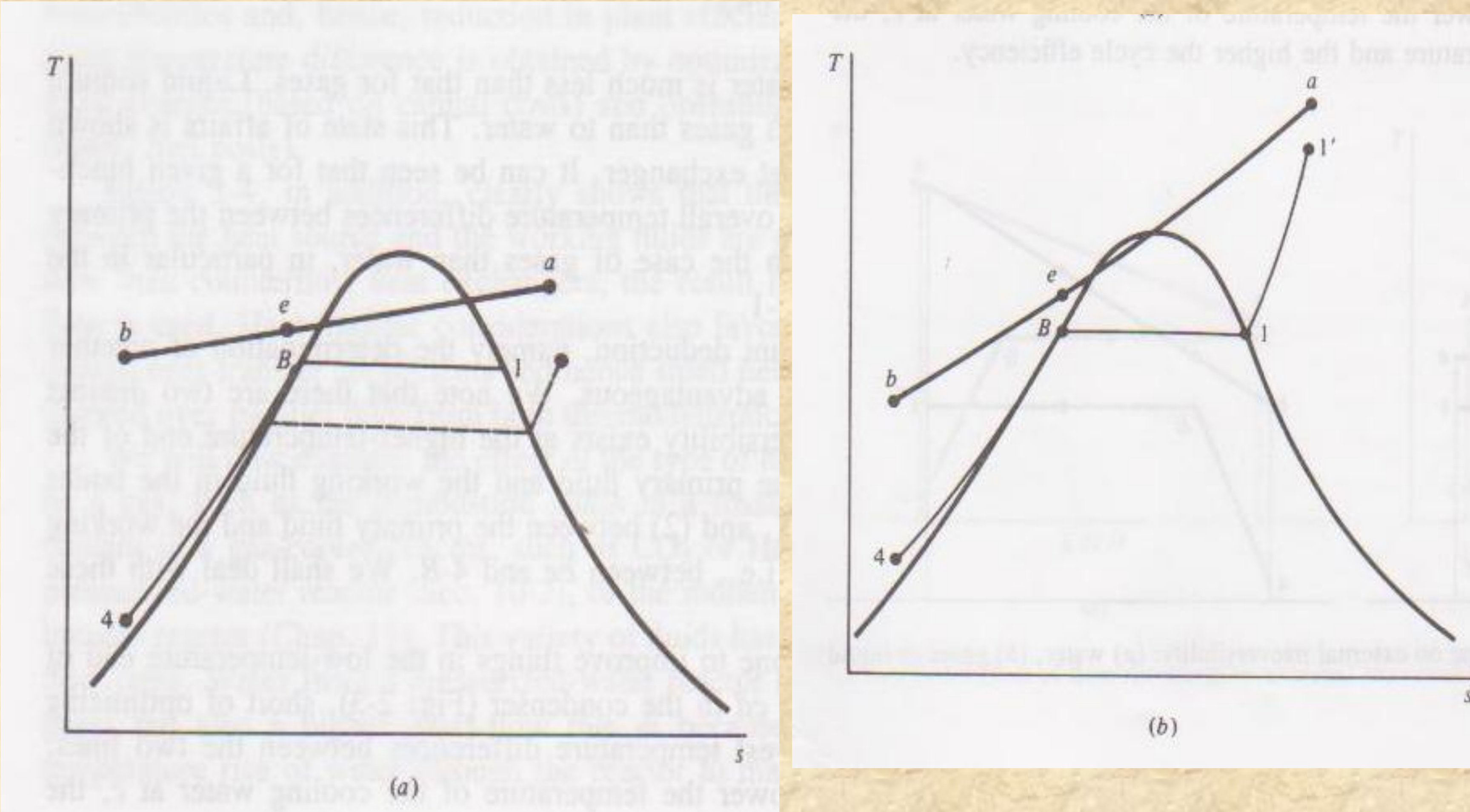


Figure 2-6 Superheat with (*a*) water as primary fluid, (*b*) gases or liquid metal as primary fluid.

- For a given pinch point temperature difference gases and liquid metals have larger and increasing temperature difference (between line ae and B-1) as the working fluid boils from B-1 than is the case of water where the slope of line ae is much lower.
- Due to the large temperature difference between line ae and B-1 in the gases case, more irreversibility is produced due to larger heat loss. To overcome this problem, superheat is needed when gas and liquid metals are used as primary heat sources.
- In the case of water, superheat is not practical as the differences between ae and B-1 vary little.
- If superheat is to be used with water, then the boiling temperature of water is lowered and thence the saturation pressure, which results in reducing the cycle efficiency rather than increasing it. This is why pressurized water reactors mainly do not use superheat.
- Superheat is also results in drier steam at turbine exhaust, which helps protecting the turbine blades from corrosion.

Reheat

- It is an additional improvement in cycle efficiency with gaseous primary fluids as in fossil-fueled and gas-cooled power plants is achieved by the use of reheat.

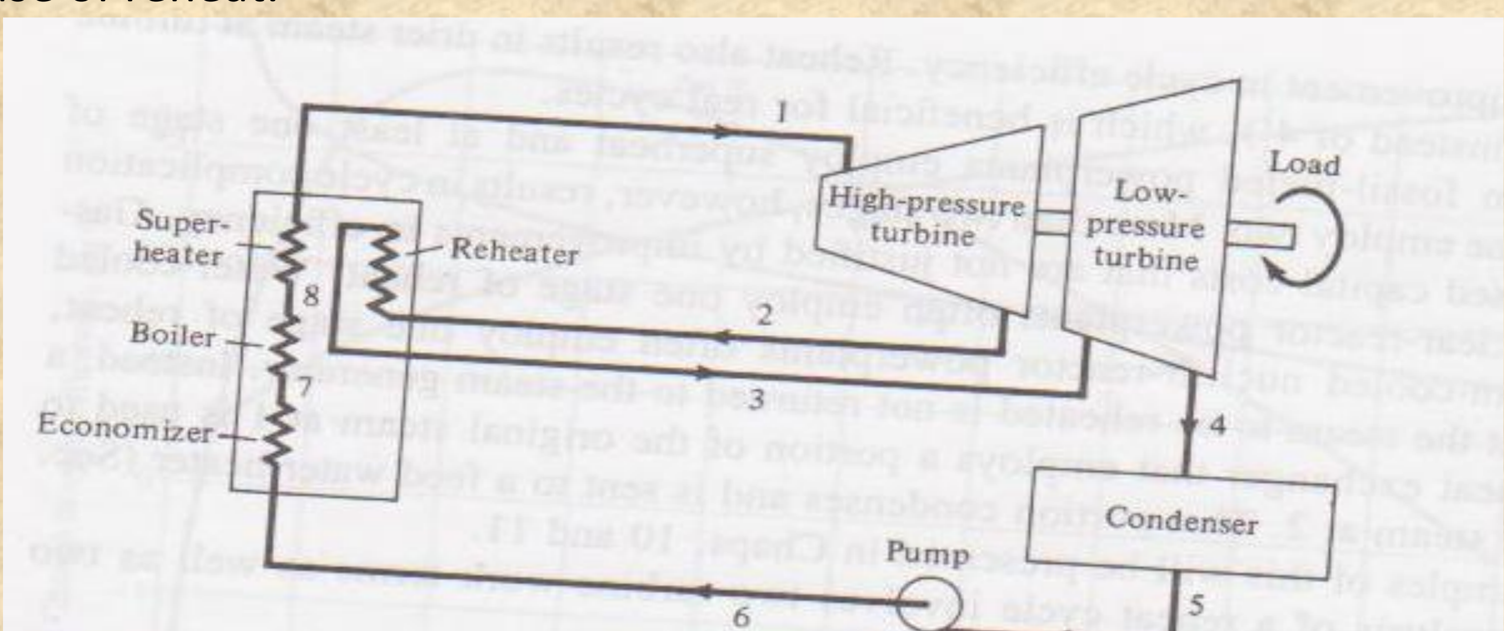


Figure 2-7 Schematic of a Rankine cycle with superheat and reheat.

- Reheat allows heat addition twice that results in increasing the average temperature at which heat is added and keeps the boiler-superheat-reheat portion close to the primary fluid line ae, which results in improvement in the cycle efficiency. One of the benefits as well is more drier at the turbine exit.

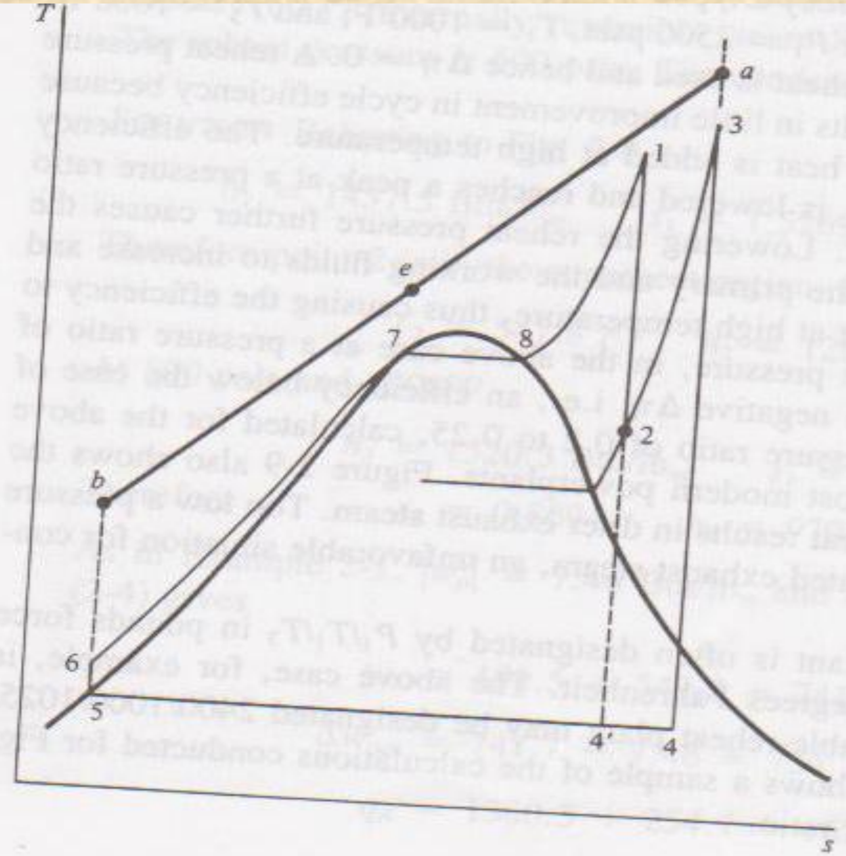
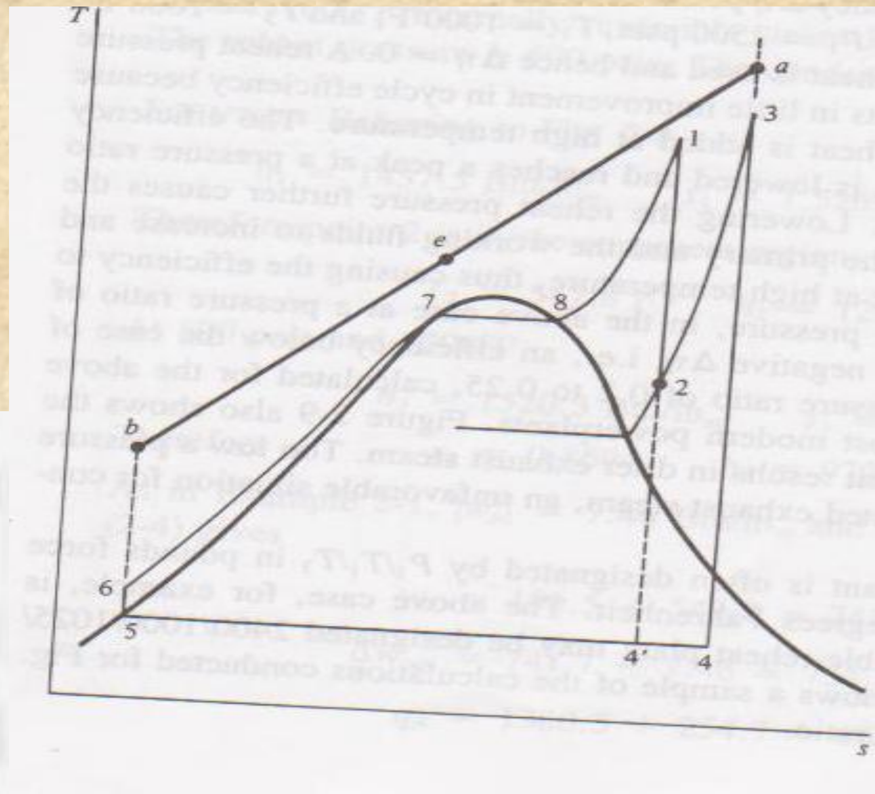


Figure 2-8 *T-s* **diagram of Rankine cycle of Fig. 2-7.**



$$\left.\begin{aligned}
w_T &= (h_1 - h_2) + (h_3 - h_4) \\
|w_p| &= h_6 - h_5 \\
\Delta w_{\text{net}} &= (h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5) \\
q_A &= (h_1 - h_6) + (h_3 - h_2) \\
\eta_{\text{th}} &= \frac{\Delta w_{\text{net}}}{q_A}
\end{aligned}\right\} \qquad (2\text{-}4)$$

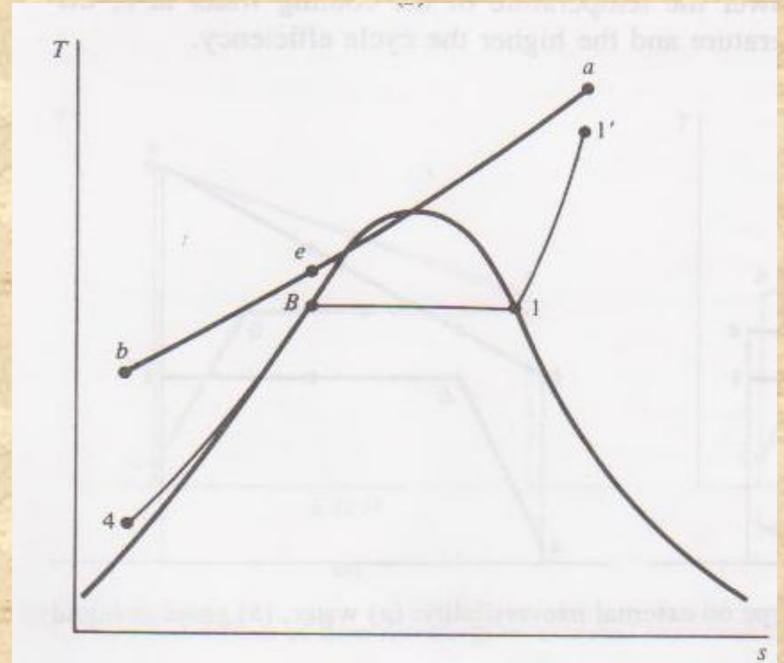
- Modern power plants have superheat and at least one stage of reheat. Some employ two stages. More than that results in cycle complications and increased capital costs that are not justified by improvements in efficiency.
- Gas cooled nuclear-reactor power plants often employ one stage of reheat. Water-cooled and sodium-cooled nuclear-reactor power plants also have on reheat stage, except that the steam is not reheated in the steam generator unite. It is reheated in outer heat exchanger unit.
- A superheat-reheat power plant is often designated by P_1 / T_1 / T_3

Effect of Reheat Pressure

- The reheat pressure affects the cycle a lot and it should be with a range. Assume that the initial pressure is P_1 and the reheat pressure is P_2 then P_2 / P_1 should be between 0.2-0.25 (20%-25%).
- Too close to the initial pressure results in little improvement in cycle efficiency.
- Too low reheat pressure results in negative efficiency difference.

Regeneration

- A great deal of external irreversibility occurs in the economizer section.
- To overcome this problem it is recommended to make use of the heat in cycle and admit water at point B rather than 4, this is called regeneration, where internal heat is exchanged between the expanding steam in the turbine and the water before heat addition. This theoretically indicates the elimination of the economizer



(b)

Theoretical Suggestion

This suggestion is not practical because of the following:

1. The vapour in the turbine does not have enough heat transfer surface to warm the water.
2. The water mass flow rate is very large so the effectiveness of such heat exchanger will be low.
3. The vapour in the turbine will have low quality, which is bad for its performance.

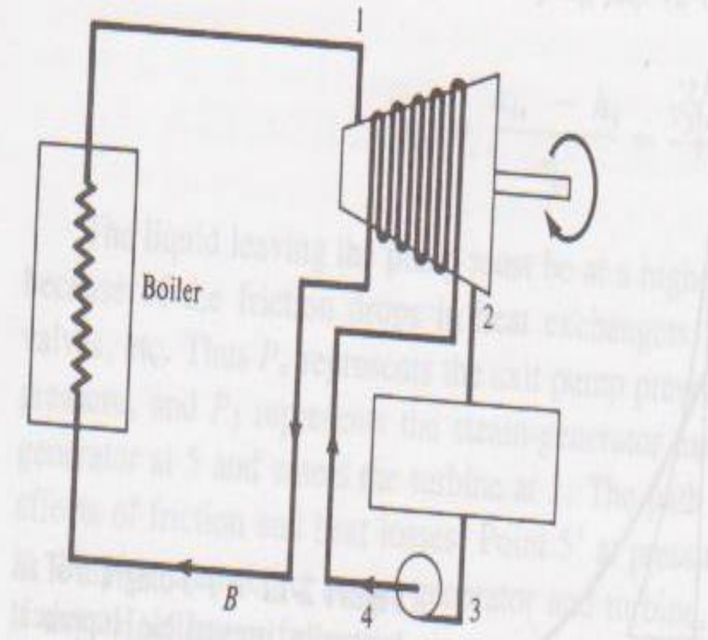


Figure 2-11 Ideal regeneration of a Rankine cycle.

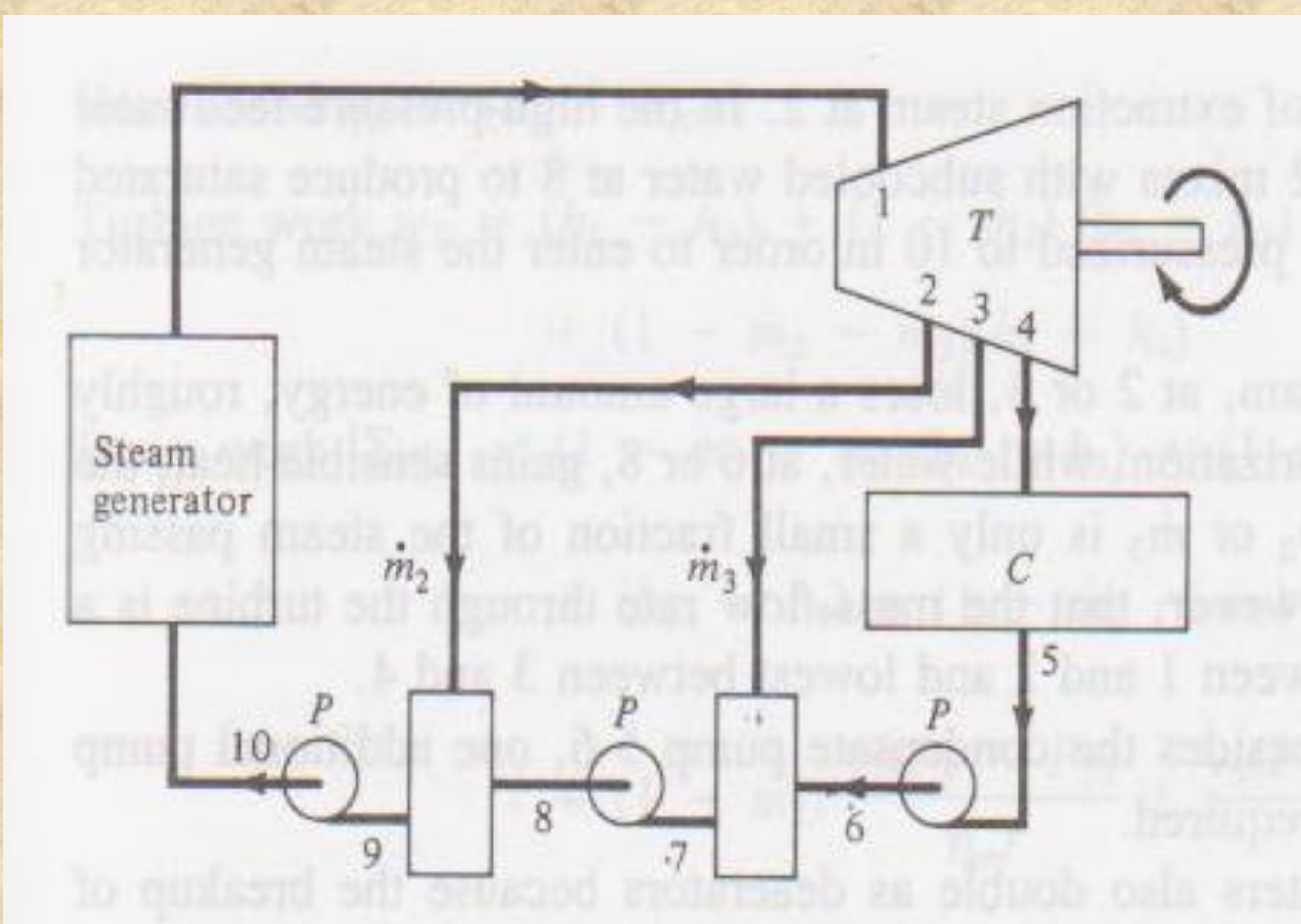
Feedwater Heating

- Feedwater heating means that the compressed water is heated by bled steam from the turbine at finite steps before entering the steam generator. This results in reducing the economizer area but not eliminating it at all.
- Most modern steam power plants use between 5-8 feedwater heaters.
- There are three types of feedwater heating:

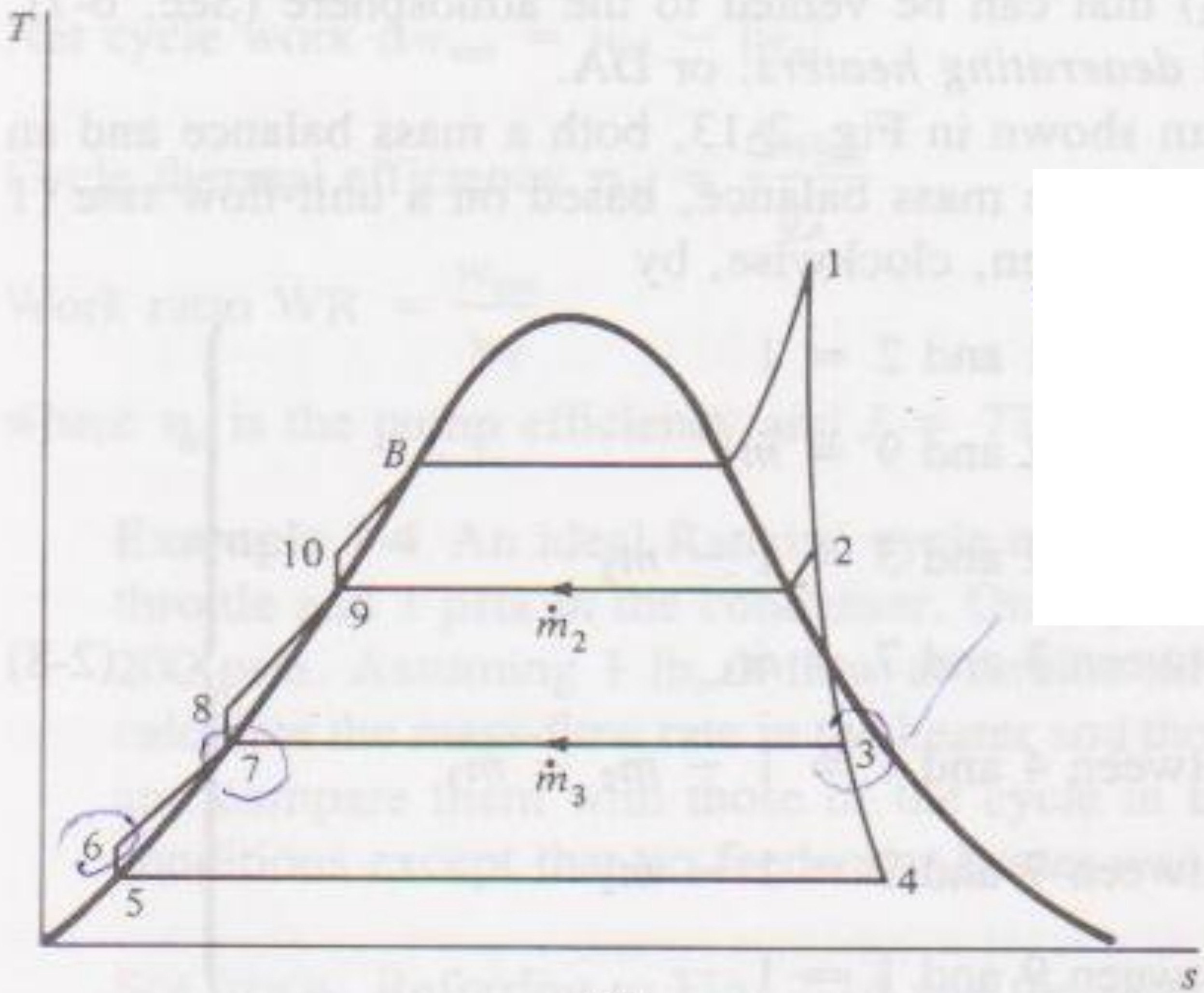
1. Open or direct contact type.
2. Closed type with drains cascaded backward.
3. Closed type with drains pumped forward.

Open Or Direct Contact Feedwater Heaters

- In the open feedwater heater the extracted steam from the turbine is mixed directly with the incoming subcooled water to produce saturated steam at the extraction steam pressure.
- The amount of bled steam should equal to that would saturate the subcooled water it is going to mix with. If it is much less it may negate the advantage of the feedwater heater. On the other hand if it is more it will affect the turbine work by causing losses, also it would result in two-phase mixture in the pump.
- Open type feedwater heaters is treated as mixing chambers.
- The mass flow rate in the turbine is a variable quantity in the case of feedwater heating.
- Besides the condensate pump there is one additional pump per open feedwater heater.
- Open feedwater heaters are also called deaerating heaters or DA, as the breakup of water in the mixing process results in non-condensable gases such as air, O_2 , CO_2, H_2.



(a)



(b)

$$\left.\begin{array}{l}\text{Mass flow between 1 and 2} = 1 \\ \text{Mass flow between 2 and 9} = \dot{m}_2 \\ \text{Mass flow between 2 and 3} = 1 - \dot{m}_2 \\ \text{Mass flow between 3 and 7} = \dot{m}_3 \\ \text{Mass flow between 4 and 7} = 1 - \dot{m}_2 - \dot{m}_3 \\ \text{Mass flow between 7 and 9} = 1 - \dot{m}_2 \\ \text{Mass flow between 9 and 1} = 1 \end{array}\right\} \quad (2\text{-}8)$$

where $\dot{m}_2$ and $\dot{m}_3$ are small fractions of 1. Energy balances are now done on the high- and low-pressure feedwater heaters, respectively

$$\dot{m}_2(h_2 - h_9) = (1 - \dot{m}_2)(h_9 - h_8) \quad (2\text{-}9)$$

and

$$\dot{m}_3(h_3 - h_7) = (1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_7) \quad (2\text{-}10)$$

Closed-Type Feedwater heaters With drains Cascaded Backward

- This type of feedwater heaters is the simplest and most commonly used.
- It is shell and tube heat exchanger with no moving parts.
- The feedwater passes in the tubes and the bled steam in the shell.
- Only one pump is required as the steam does not mix with the feedwater. This pump is doubles to be also a boiler feed pump.
- If a deaerating heater is used then another pump should be used after it to be the boiler feed pump.
- The bled steam is feed back to the next lower pressure feedwater heater by throttling and then led back to the condenser, which is called cascade from high pressure to low pressure.

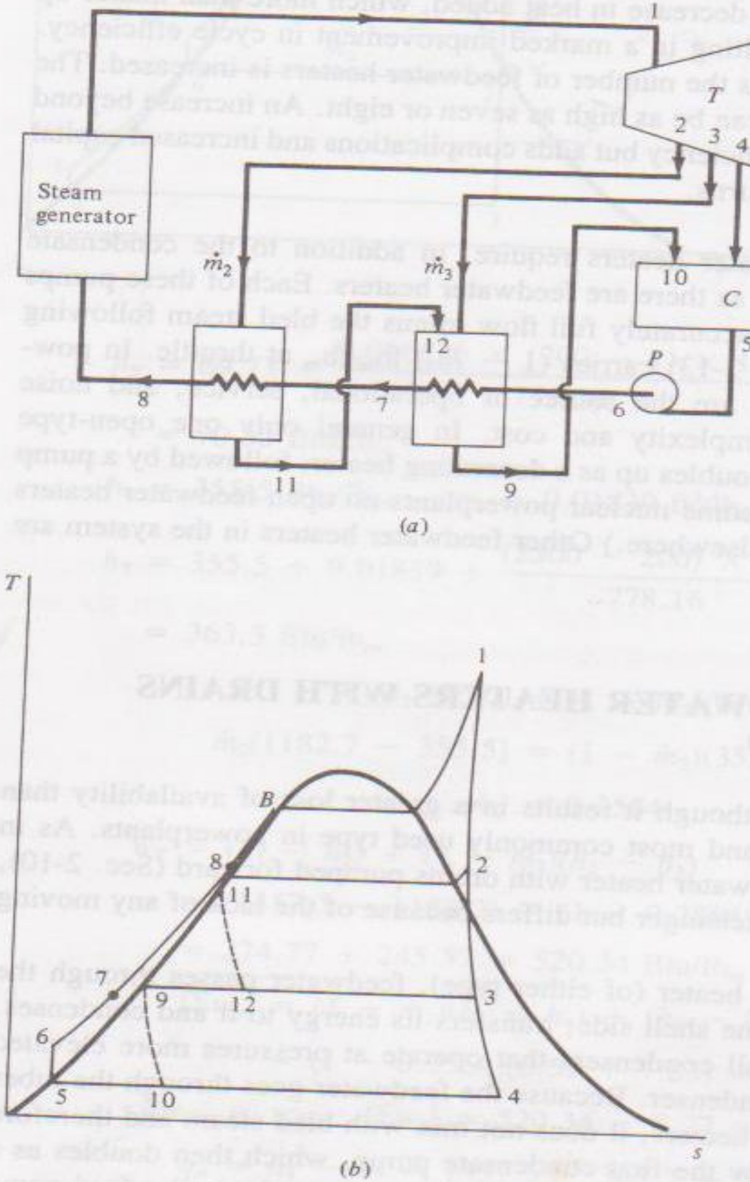
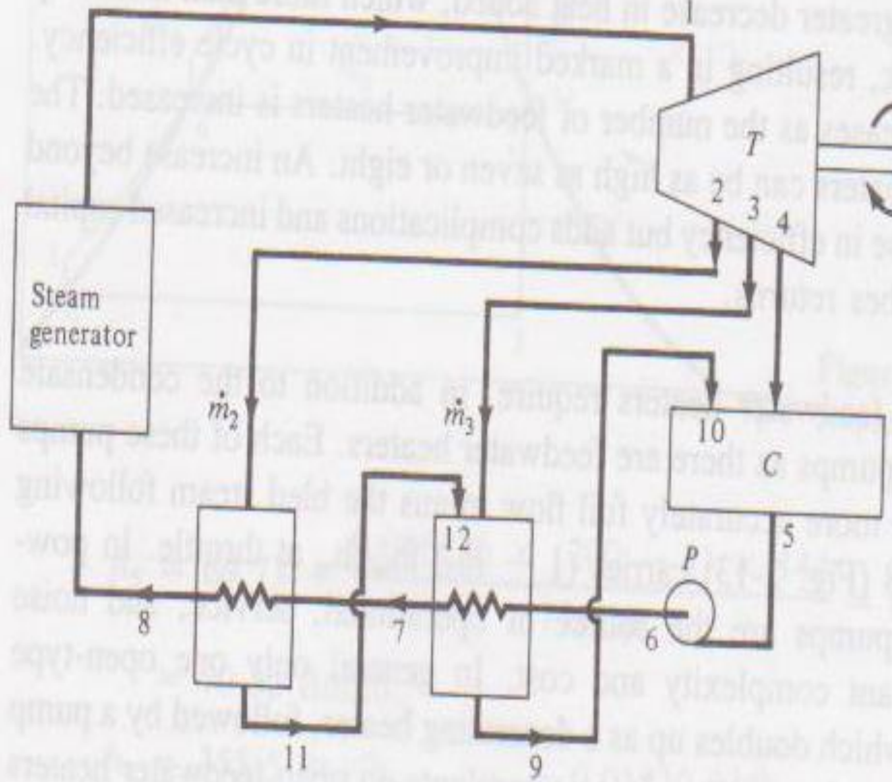


Figure 2-15 Schematic flow and *T-s* diagrams of a nonideal superheat Rankine cycle with two closed-type feedwater heaters with drains cascaded backward.



$$
\left.
\begin{aligned}
&\text{Mass flow between 1 and 2} &&= 1\\
&\text{Mass flow between 2 and 3} &&= 1 - \dot{m}_2\\
&\text{Mass flow between 3 and 10} &&= 1 - \dot{m}_2 - \dot{m}_3\\
&\text{Mass flow between 10 and 1} &&= 1\\
&\text{Mass flow between 2 and 12} &&= \dot{m}_2\\
&\text{Mass flow between 3 and 12} &&= \dot{m}_3\\
&\text{Mass flow between 12 and 10} &&= \dot{m}_2 + \dot{m}_3
\end{aligned}
\right\} \qquad (2\text{-}13)
$$

The energy balances on the high- and low-pressure heaters are now given, respectively, by

$$\dot{m}_2(h_2 - h_{11}) = h_8 - h_7 \qquad (2\text{-}14)$$

and

$$\dot{m}_3(h_3 - h_9) + \dot{m}_2(h_{12} - h_9) = h_7 - h_6 \qquad (2\text{-}15)$$

Recalling that a throttling process is a constant enthalpy process so that

$$h_{12} = h_{11} \quad \text{and} \quad h_{10} = h_9$$

- There is always a temperature difference between the bled steam entering the feedwater heater and the exit temperature of the subcooled water in the pipes, this is called Terminal Temperature Difference (TTD) and it is represented as: TTD = Saturation temperature of bled steam – exit water temperature. The value of TTD varies with the heater pressure.
- TTD is positive and the optimum design TTD is 5^0 F. Too small TTD is good for the cycle efficiency, but would require a larger heater, which may not be justified economically. On the other hand too large a value will hurt the cycle efficiency.
- TTD can be negative at the case of high pressure feedwater heater, as the exit water may have a temperature higher than the saturation temperature of that steam, therefore TTD ranges from 0 to -5^0 F. The drain here is slightly subcooled.
- In the low pressure heater the steam can have a drain cooler, thus physically it is composed from condensing side and drain cooler side.
- Based on the previous discussion there are four possible section zones of the closed heaters: 1- condenser 2- condenser and drain cooler (DC) 3- desuperheater, condenser, drain cooler 4- desuperheater and condenser.

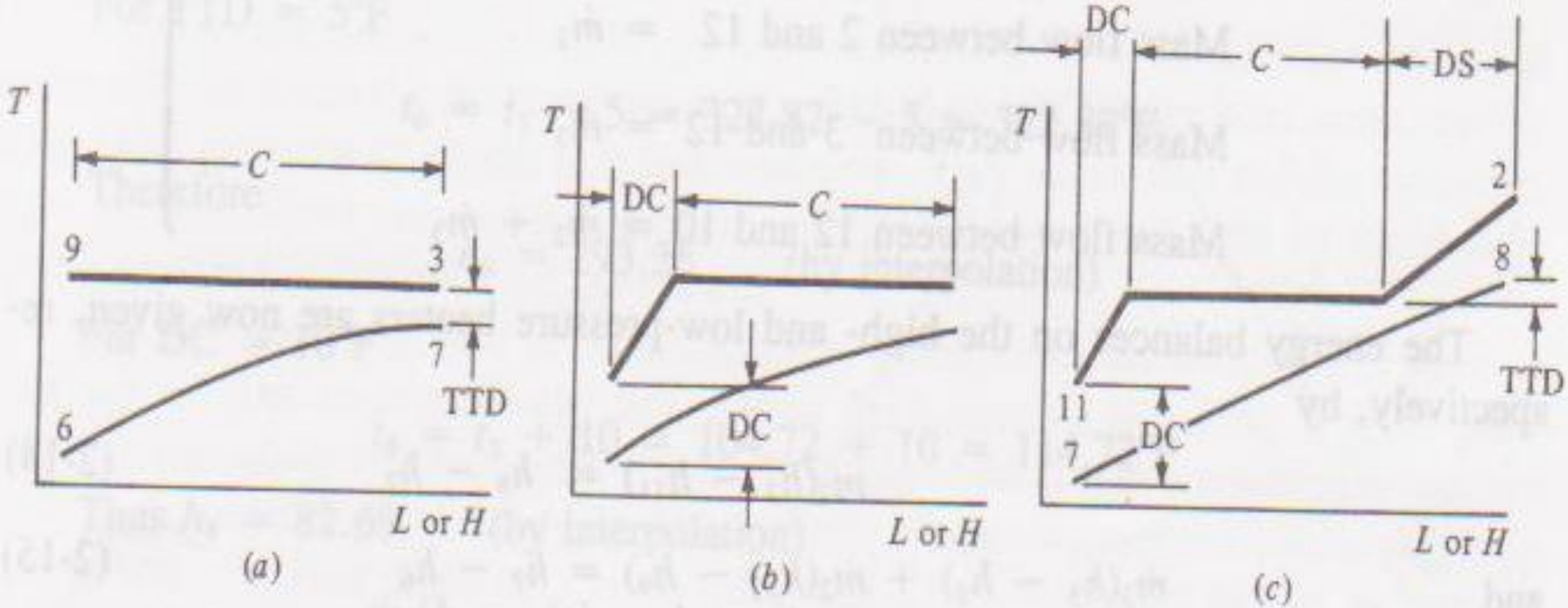


Figure 2-16 Temperature-enthalpy diagrams of (*a*) and (*b*) low-pressure and (*c*) high-pressure feedwater heaters of Fig. 2-15. TTD = terminal temperature difference, DS = desuperheater, C = condenser, DC = drain cooler.

Efficiency And Heat Rate

- The thermal efficiency a loan (net work/ heat added) is not enough to accurately analyze the real cycle with all its auxiliaries and irreversibility.
- The Gross efficiency is the one calculated based on gross work or power of the turbine-generator, which is before any of the power is tapped to the plant, such as the pumps work, computers, lighting, heating and AC, etc.
- The net efficiency is calculated based on the net work of the plant.
- The efficiency is also a measure of the economy of the power plant as it affects capital, fuel and operating costs.
- The heat rate (HR) is a measure of fuel economy and it is defined as the amount of heat added, usually in Btu, to produce a unit a mount of work, usually in Kilowatt hours (kWh), so it is HR/ kWh.
- The lower the heat rate the higher the efficiency.

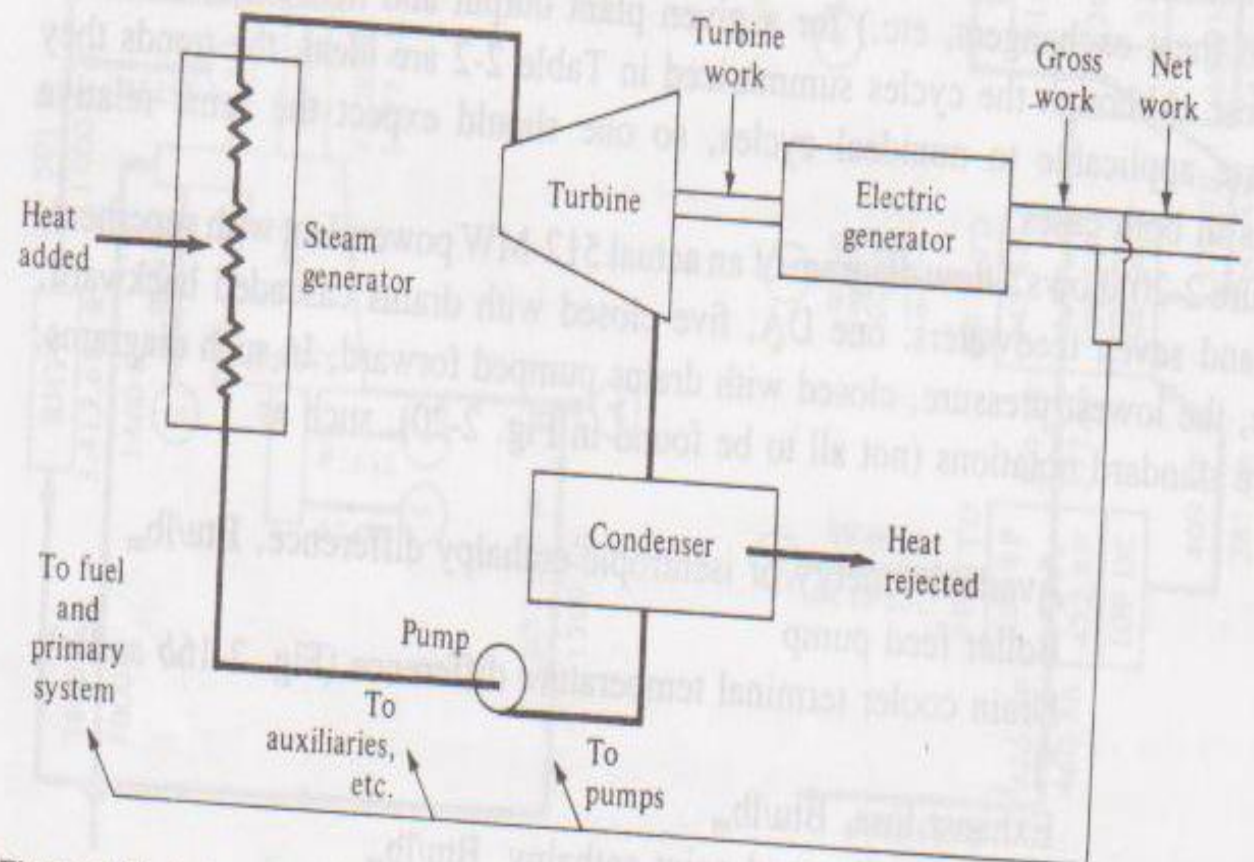


Figure 2-21 Schematic of a powerplant showing turbine, gross and net work.

$$\text{Net cycle HR} = \frac{\text{heat added to cycle, Btu}}{\text{net cycle work kWh}}$$

$$= \frac{\text{rate of heat added to cycle, Btu/h}}{\text{net cycle power, kW}}$$

$$\text{Gross cycle HR} = \frac{\text{rate of heat added to cycle, Btu/h}}{\text{turbine power output, kW}}$$

$$\text{Net station HR} = \frac{\text{rate of heat added to steam generator, Btu/h}}{\text{net station power, kW}}$$

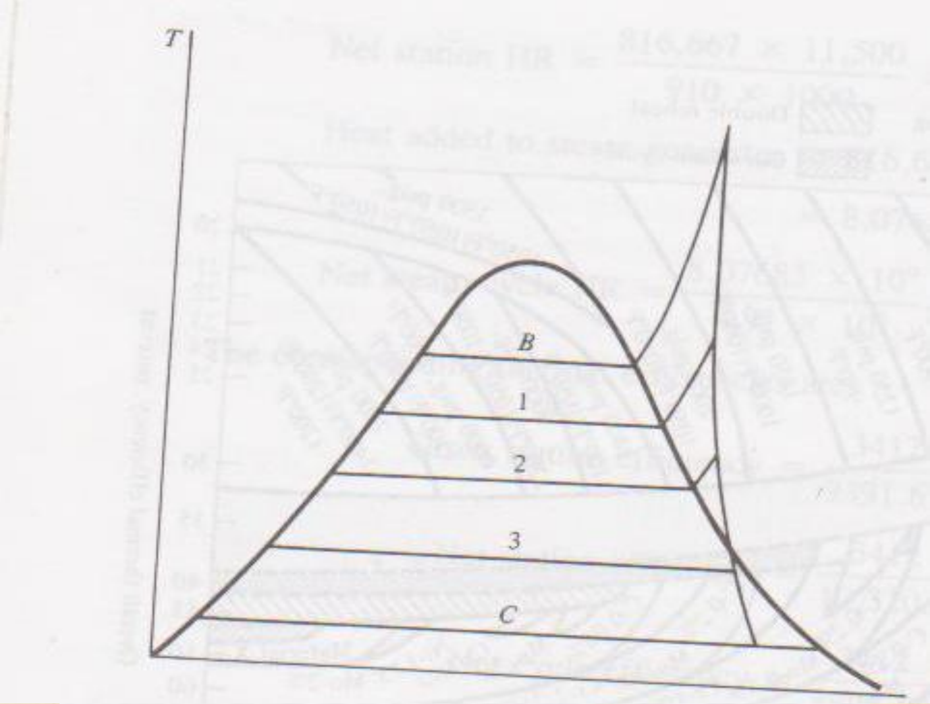
$$\text{Gross station HR} = \frac{\text{rate of heat added to steam generator, Btu/h}}{\text{gross turbine-generator power, kW}}$$

and there are as many thermal efficiencies as there are heat rates. Because 1 kWh = 3412 Btu, the heat rate of any kind is related to the corresponding thermal efficiency by

$$\text{HR} = \frac{3412}{\eta_{th}} \qquad (2\text{-}27)$$

The Placement of Feed Water Heaters

- What are the pressures at which steam is to be bled from the turbine that will result in the maximum increase in efficiency?.
- The heat exchange in feedwater heaters is due to temperature difference between the boiler and the heater as well as between the condenser and the heater.



$$\Delta T_{\text{opt}} = \frac{T_B - T_C}{n + 1}$$

- For heaters at the high pressure side they would be constructed with a desuperheate zone, a condensing zone and probably a drain cooler zone.
- At low pressure side the feedwater heater may receive wet steam and thus it is mainly a condensing section and maybe drain cooler.
- Other considerations that may dictate the exact positions are: 1- the placement of deareating heater 2- the existence of a convenient point to bled the steam from the turbine 3- turbine design, casing and others.

The Supercritical Pressure Cycle

- It is the cycle that operates over the critical pressure of the steam, which is mainly 3208 psia for the steam.
- Supercritical cycle steam generator is mainly a once-through type.
- A disadvantage of the supercritical-pressure cycle is that in the expansion process a more wet steam would pass through the turbine exit, so it is mainly joined with one stage or two stage reheat process.

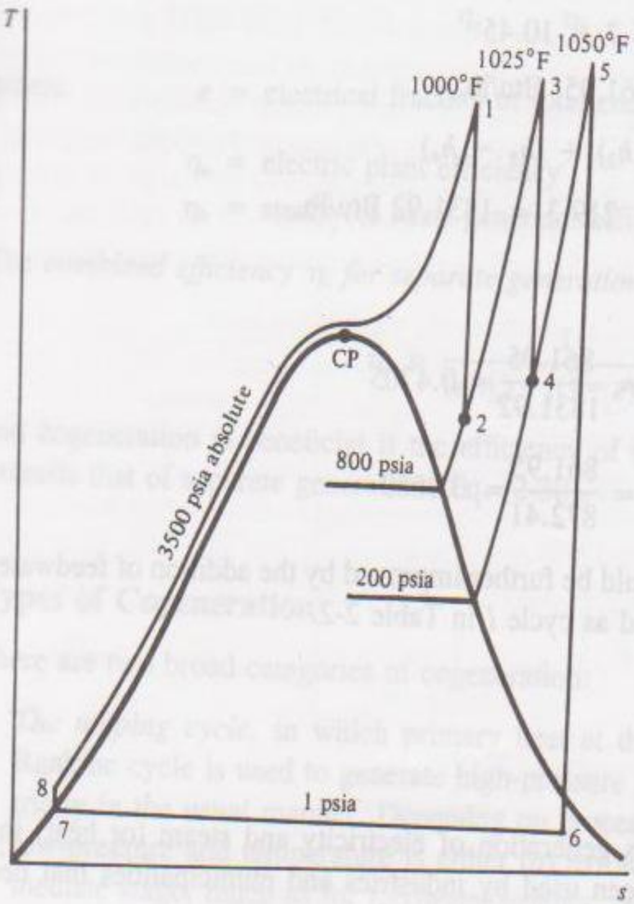


Figure 2-26 *T-s* diagram of an ideal supercritical, double-reheat 3500/1000/1025/1050 steam cycle.

Cogeneration

- Cogeneration is the simultaneous generation of electricity and steam (heat) in a single power plant. Cogeneration is used in industries, municipalities that need process steam or (heat) as well as electricity e.g. Chemical industries, paper mills.
- Cogeneration is advised for industries if they can produce electricity cheaper, or more conveniently than buying it.
- From energy resource view, cogeneration is beneficial only if it saves primary energy when compared with separate generation of electricity and steam (heat).
- The cogeneration efficiency is given by the electricity energy generated and the heat energy in process steam according to the total heat added to the plant.

$$\eta_{co} = \frac{E + \Delta H_s}{Q_A} \tag{2-29}$$

where E = electric energy generated

ΔH_s = heat energy, or heat energy in process steam

= (enthalpy of steam entering the process)

− (enthalpy of process condensate returning to plant)

Q_A = heat added to plant (in coal, nuclear fuel, etc.)

For separate generation of electricity and steam, the heat added per unit *total* energy output is

$$\frac{e}{\eta_e} + \frac{(1-e)}{\eta_h}$$

where e = electrical fraction of total energy output = $\dfrac{E}{(E+\Delta H_s)}$

η_e = electric plant efficiency

η_h = steam (or heat) generator efficiency

The *combined efficiency* η_c *for separate generation* is therefore given by

$$\eta_c = \frac{1}{(e/\eta_e) + [(1-e)/\eta_h]} \tag{2-30}$$

and cogeneration is beneficial if the efficiency of the cogeneration plant Eq. (2-29) exceeds that of separate generation, Eq. (2-30).

Types of Cogeneration

- <u>The topping cycle</u>; where the primary heat at the higher temperature end of the Rankine cycle is used to generate high pressure and temperature steam and electricity in the usual manner. Depending on the process requirements the process steam is either: 1- extracted from the turbine at an intermediate stage, like feed water heating 2- taken at the turbine exhaust where it is called a back pressure turbine.
- <u>The bottoming cycle ;</u> in which primary heat is used at high temperature directly for process requirements. An example is the high-temperature cement kiln. The waste heat is then used to generate electricity. The bottoming cycle is not that much in interest as the topping cycle.
- There are different arrangements for cogeneration in a topping cycle as: 1- Steam-electric power plant with back pressure turbine and it is most suitable only when the electric demand is low compared with heat demand 2- steam-electric power plant with steam extraction from a condensing turbine 3- Gas-turbine power plant with a heat recovery boiler (using the gas turbine exhaust to generate steam 4- combined steam gas turbine cycle power plant, where the steam turbine is either of back pressure type or of the extraction-condensing type and it is most suitable only when the electric demand is high.

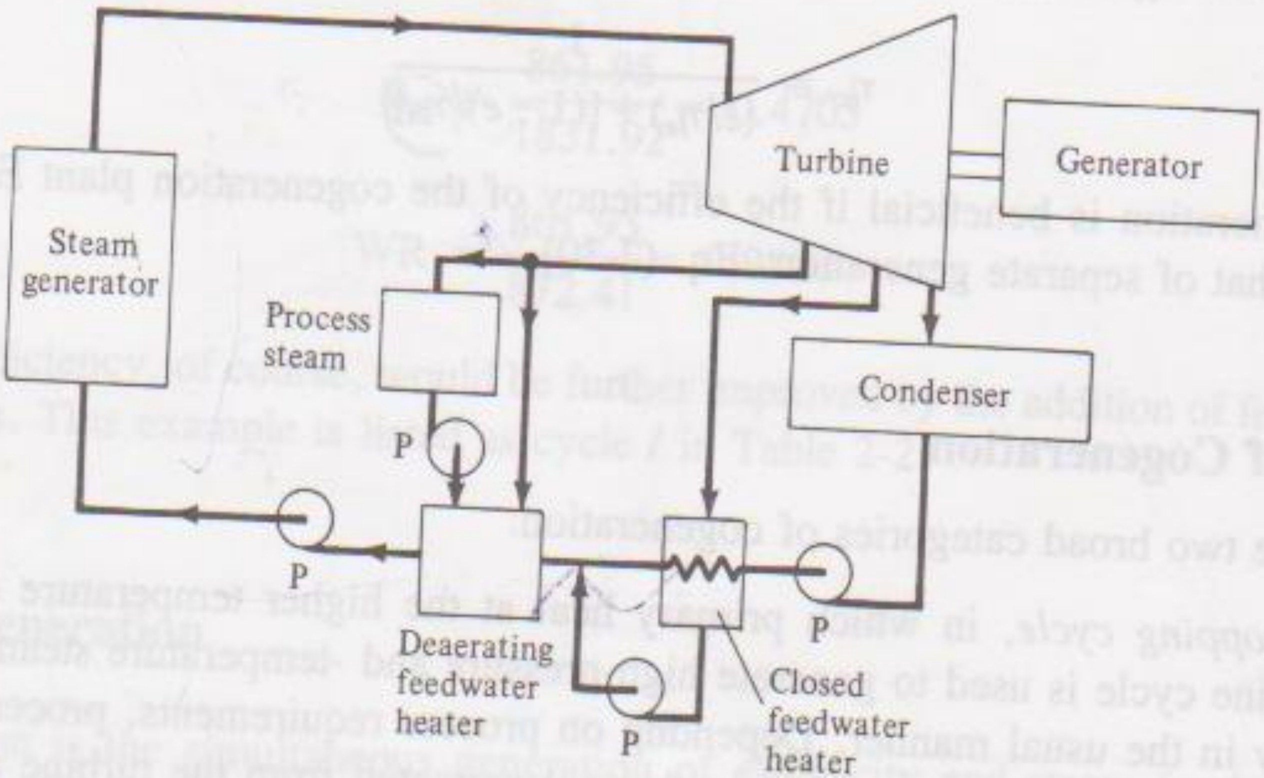


Figure 2-27 Schematic of basic cogeneration plant with extraction-condensing turbine.

Economics of Cogeneration

- A privately owned cogeneration plant is advisable from an economic point of view if the cost of electricity generated by it is less than if purchased from a utility unless the utility is not available.
- Power plant costs are of two kinds: 1- capital costs and it is given in total (currency name e.g. $ or JD) or as unit capital costs in JD/ KW net. Capital cost determine if the utility is sound enough to obtain financing 2- production costs, which are mainly calculated annually or as desired. They are given in filse/KWh (mills/KWh in USA). Production costs are the measure of the cost of power generated and they are composed of:

1. The fixed charges against the capital costs
2. The fuel costs
3. Operation and maintenance costs.

all in mills per kilowatt hour. They are therefore given by:

$$\text{Production costs} = \frac{\text{total } (a+b+c) \text{ \$ spent per period } \times 10^3}{\text{KWh (net) generated during same period}} \tag{2-31}$$

where the period is usually taken as one year.

For a congeneration plant, it is important to calculate the production costs of electricity as an excess over the generating cost of steam alone, and to compare it with the cost of electricity when purchased from a utility. It is now necessary to introduce the *plant operating factor* POF, defined for all plants as

$$\text{POF} = \frac{\text{total net energy generated by plant during a period of time}}{\text{rated net energy capacity of plant during same period}} \tag{2-32}$$

where the period is again usually taken as one year. For estimation purposes, it is common to take POF = 0.80. A plant operating with POF = 0.8 is the same as if it operated only at rated capacity for 80 percent of the time or for 0.8(365 × 24) = 7008 h/yr, which is usually rounded out to 7000 h/yr.

The excess cost of electricity for a cogeneration plant may now be obtained from

$$\text{Electric cost} = [(C_{co} - C_h)r + (\text{OM}_{co} - \text{OM}_h) + (F_{co} - F_h)] \frac{10^3}{7000\,P} \text{ mills/kWh} \tag{2-33}$$

where C = capital costs, \$

r = annual fixed charges against the capital cost, fraction of C