The Regenerative Cycle

The thermal efficiency of a steam power cycle is increased when the pressure and hence the vaporization temperature in the boiler is raised. It is also increased by increased superheating in the boiler. Thus, high boiler pressures and temperatures favour high efficiencies.

Thus, in practice power plants seldom operate at pressures much above 10000 kPa (100 bar) or temperatures much above 873.15 K (600°C). The thermal efficiency of a power plant increases as the pressure and hence the temperature in the condenser is reduced.

Most modern power plants operate on a modification of the Rankine cycle that incorporates feedwater heaters. Water from the condenser, rather than being pumped directly back to the boiler, is first heated by steam extracted from the turbine. This is normally done in several stages, with steam taken from the turbine at several intermediate states of expansion. An arrangement with four feedwater heaters is shown in Fig. 8.5. The operating conditions indicated on this figure and described in the following paragraphs are typical, and are the basis for the illustrative calculations of Ex. 8.2.

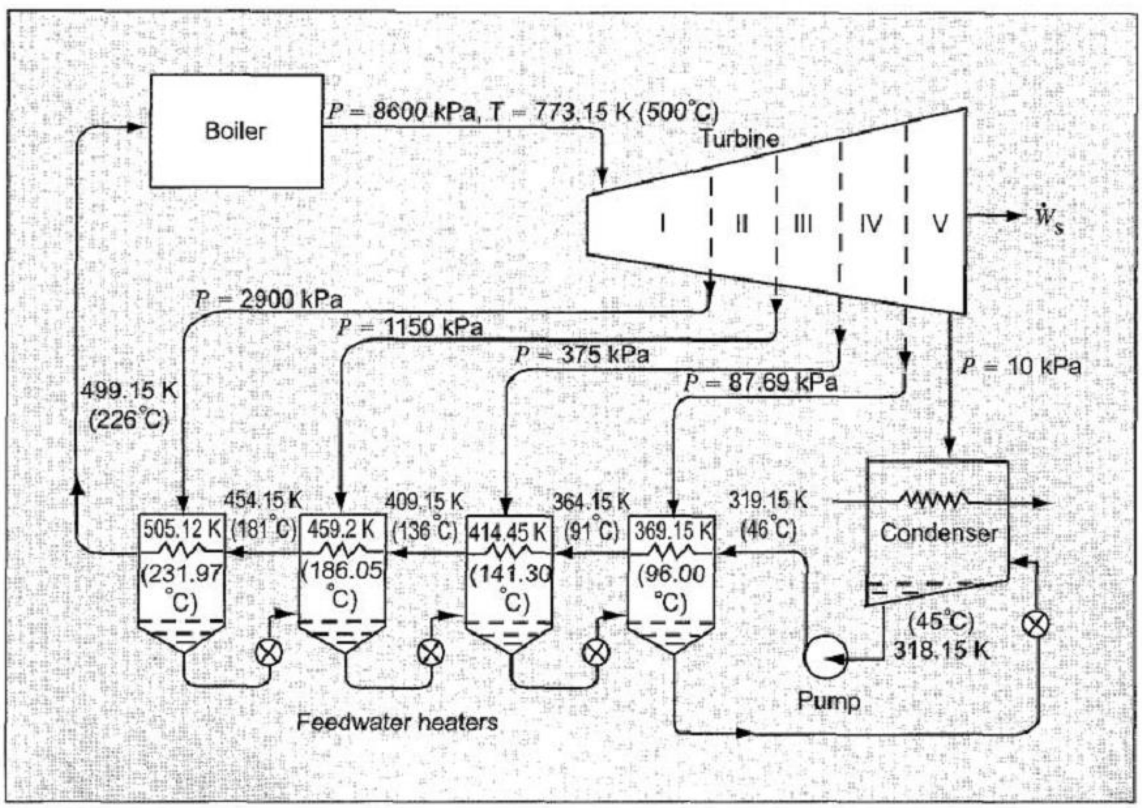

Figure 8.5 Steam power plant with feedwater heating

The feedwater pump, which operates under exactly the conditions of the pump in Ex. 7.10, causes a temperature rise of about 1 K (l°C), making the temperature of the feedwater entering the series of heaters equal to 3 19.15 K (46°C).

The saturation temperature of steam at the boiler pressure of 8600 kPa is 573.21 K (300.06°C), and the temperature to which the feedwater can be raised in the heaters is certainly less. This temperature is a design variable, which is ultimately fixed by economic considerations. However, a value must be chosen before any thermodynamic calculations can be made. We have therefore arbitrarily specified a temperature of 499.15 K (226°C) for the feedwater stream entering the boiler. We have also specified that all four feedwater heaters accomplish the same temperature rise. Thus, the total temperature rise of 499.15 - 319.15 = 180 K is divided into four 45 K (45°C) increments. This establishes all intermediate feedwater temperatures at the values shown on Fig. 8.5.

The steam supplied to a given feedwater heater must be at a pressure high enough that its saturation temperature is above that of the feedwater stream leaving the heater. We have here presumed a minimum temperature difference for heat transfer of no less than 5 K (5°C), and have chosen extraction steam pressures such that the T^{sat} values shown in the feedwater heaters are at least 5 K (5°C) greater than the exit temperatures of the feedwater streams. The condensate from each feedwater heater is flashed through a throttle valve to the heater at the next lower pressure, and the collected condensate in the final heater of the series is flashed into the condenser. Thus, all condensate returns from the condenser to the boiler by way of the feedwater heaters.

The purpose of heating the feedwater in this manner is to raise the average temperature at which heat is added in the boiler. This increases the thermal efficiency of the plant, which is said to operate on a regenerative cycle.

Example 8.2

Determine the thermal efficiency of the power plant shown in Fig. 8.5, assuming turbine and pump efficiencies of 0.75. If its power rating is 80,000 kW, what is the steam rate from the boiler and what are the heat-transfer rates in the boiler and condenser?

Solution 8.2

Initial calculations are made on the basis of 1 kg of steam entering the turbine from the boiler. The turbine is in effect divided into five sections, as indicated in Fig. 8.5. Because steam is extracted at the end of each section, the flow rate in the turbine decreases from one section to the next. The amounts of steam extracted from the first four sections are determined by energy balances.

This requires enthalpies of the compressed feedwater streams. The effect of pressure at constant temperature on a liquid is given by Eq. (7.25):

$$\Delta H = V(1 - \beta T)\Delta P \qquad \text{(const } T\text{)}$$

For saturated liquid water at 226°C (499.15 K), the steam tables provide:

$$P^{sat} = 2{,}598.2 \text{ kPa} \qquad H = 971.5 \text{ kJ kg}^{-1} \qquad V = 1{,}201 \text{ cm}^3 \text{ kg}^{-1}$$

In addition, at this temperature,

$$\beta = 1.582 \times 10^{-3} \text{ K}^{-1}$$

Thus, for a pressure change from the saturation pressure to 8,600 kPa:

$$\Delta H = 1{,}201[1 - (1.528 \times 10^{-3})(499.15)]\frac{(8{,}600 - 2{,}598.2)}{10^6} = 1.5 \text{ kJ kg}^{-1}$$

and $\qquad H = H(\text{sat. liq.}) + \Delta H = 971.5 + 1.5 = 973.0 \text{ kJ kg}^{-1}$

Similar calculations yield the enthalpies of the feedwater at other temperatures. All pertinent values are given in the following table.

t/°C	226	181	136	91	46
H/kJ kg^{-1} for water at t and P = 8,600 kPa	973.0	771.3	577.4	387.5	200.0

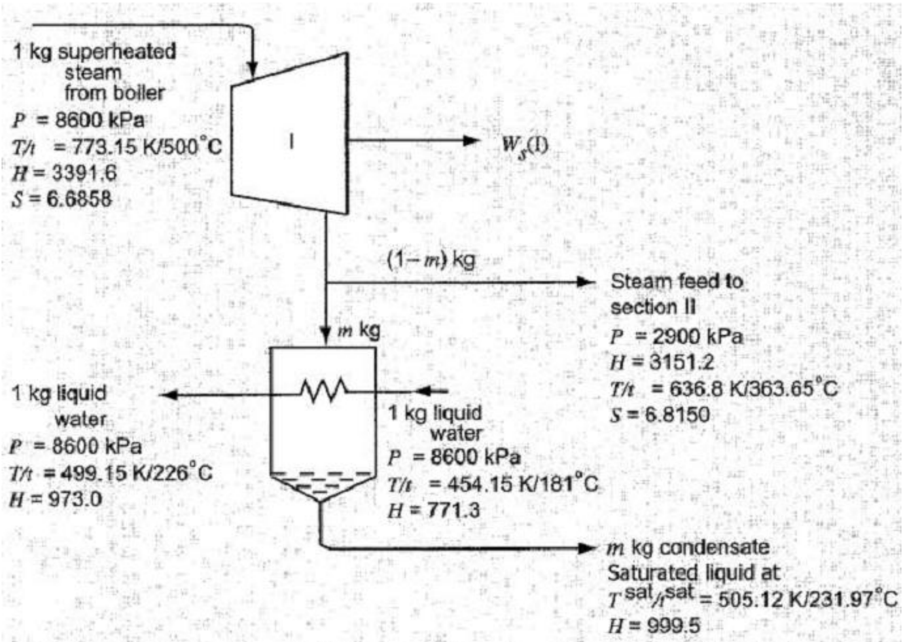

Figure 8.6: Section I of turbine and first feedwater heater.

Consider the first section of the turbine and the first feedwater heater, as shown by Fig. 8.6. The enthalpy and entropy of the steam entering the turbine are found from the tables for superheated steam. The assumption of isentropic expansion of steam in section I of the turbine to 2,900 kPa leads to the result:

$$(\Delta H)_S = -320.5 \text{ kJ kg}^{-1}$$

If we assume that the turbine efficiency is independent of the pressure to which the steam expands, then Eq. (7.16) gives:

$$\Delta H = \eta(\Delta H)_S = (0.75)(-320.5) = -240.4 \text{ kJ kg}^{-1}$$

By Eq. (7.14), $\quad W_s(\mathrm{I}) = \Delta H = -240.4 \text{ kJ}$

In addition, the enthalpy of steam discharged from this section of the turbine is:

$$H = 3{,}391.6 - 240.4 = 3{,}151.2 \text{ kJ kg}^{-1}$$

A simple energy balance on the feedwater heater results from the assumption that kinetic- and potential-energy changes are negligible and from the assignments, $\dot{Q} = -\dot{W}_s = 0$. Equation (2.30) then reduces to:

$$\Delta(\dot{m}H)_{fs} = 0$$

This equation gives mathematical expression to the requirement that the total enthalpy change for the process be zero. Thus on the basis of 1 kg of steam entering the turbine (Fig. 8.6):

$$m(999.5 - 3{,}151.2) + (1)(973.0 - 771.3) = 0$$

Whence, $m = 0.09374$ kg and $1 - m = 0.90626$ kg

On the basis of 1 kg of steam entering the turbine, $1 - m$ is the mass of steam flowing into section II of the turbine.

Section II of the turbine and the second feedwater heater are shown in Fig. 8.7. In doing the same calculations as for section I, we assume that each kilogram of steam leaving section II expands from its state *at the turbine entrance* to the exit of section II with an efficiency of 0.75 compared with isentropic expansion. The enthalpy of the steam leaving section II found in this way is:

$$H = 2{,}987.8 \text{ kJ kg}^{-1}$$

Then on the basis of 1 kg of steam entering the turbine,

$$W_s(\text{II}) = (2{,}987.8 - 3{,}151.2)(0.90626) = -148.08 \text{ kJ}$$

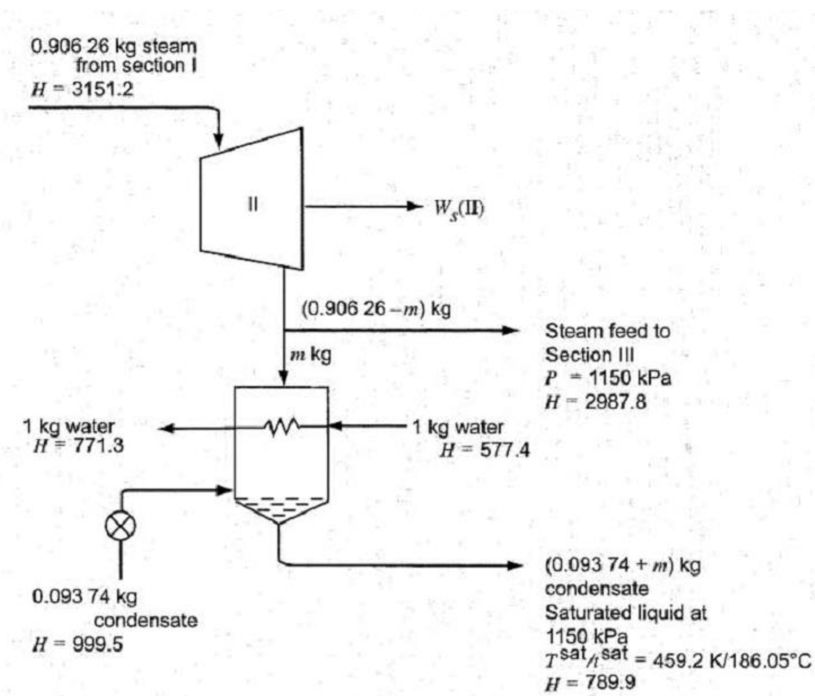

Figure 8.7: Section II of turbine and second feedwater heater.

An energy balance on the feedwater heater (Fig. 8.7) gives:

$$(0.09374 + m)(789.9) - (0.09374)(999.5) - m(2{,}987.8) + (1)(771.3 - 577.4) = 0$$

Whence, $$m = 0.07971 \text{ kg}$$

Note that throttling the condensate stream does not change its enthalpy.

These results and those of similar calculations for the remaining sections o the turbine are listed in the accompanying table. From the results shown,

$$\sum W_s = -804.0 \text{ kJ} \qquad \text{and} \qquad \sum m = 0.3055 \text{ kg}$$

	H/kJ kg^{-1} at section exit	W_s/kJ for section	t/°C at section exit	State	m/kg of steam extracted
Sec. I	3,151.2	−240.40	363.65	Superheated vapor	0.09374
Sec. II	2,987.8	−148.08	272.48	Superheated vapor	0.07928
Sec. III	2,827.4	−132.65	183.84	Superheated vapor	0.06993
Sec. IV	2,651.3	−133.32	96.00	Wet vapor $x = 0.9919$	0.06257
Sec. V	2,435.9	−149.59	45.83	Wet vapor $x = 0.9378$	

Thus for every kilogram of steam entering the turbine, the work produced i 804.0 kJ, and 0.3055 kg of steam is extracted from the turbine for the feedwate heaters. The work required by the pump is exactly the work calculated for th pump in Ex. 7.10, that is, 11.6 kJ. The net work of the cycle on the basis of 1 k of steam generated in the boiler is therefore:

$$W_s(\text{net}) = -804.0 + 11.6 = -792.4 \text{ kJ}$$

On the same basis, the heat added in the boiler is:

$$Q(\text{boiler}) = \Delta H = 3{,}391.6 - 973.0 = 2{,}418.6 \text{ kJ}$$

The thermal efficiency of the cycle is therefore:

$$\eta = \frac{|W_s(\text{net})|}{Q(\text{boiler})} = \frac{792.4}{2{,}418.6} = 0.3276$$

This is a significant improvement over the value 0.2961 of Ex. 8.1.

Because $\dot{W}_s(\text{net}) = -80{,}000 \text{ kJ s}^{-1}$,

$$\dot{m} = \frac{\dot{W}_s(\text{net})}{W_s(\text{net})} = \frac{-80{,}000}{-792.4} = 100.96 \text{ kg s}^{-1}$$

This is the steam rate to the turbine, used to calculate the heat-transfer rate in the boiler:

$$\dot{Q}(\text{boiler}) = \dot{m}\,\Delta H = (100.96)(2{,}418.6) = 244.2 \times 10^3 \text{ kJ s}^{-1}$$

The heat-transfer rate to the cooling water in the condenser is:

$$\begin{aligned}\dot{Q}(\text{condenser}) &= -\dot{Q}(\text{boiler}) - \dot{W}_s(\text{net}) \\ &= -244.2 \times 10^3 - (-80.0 \times 10^3) = -164.2 \times 10^3 \text{ kJ s}^{-1}\end{aligned}$$

Although the steam generation rate is higher than was found in Ex. 8.1, the heat-transfer rates in the boiler and condenser are appreciably less, because their functions are partly taken over by the feedwater heaters.