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COURSE INFORMATION SHEET

Course code:	ME 209
Course title:	Energy Conversion Systems
Pre-requisite(s):	NIL
Co- requisite(s):	NIL
Credits:	3 (L: 3, T: 0, P: 0)
Class schedule per week:	3
Class:	B. Tech
Semester / Level:	Fourth
Branch:	Mechanical Engineering

Course Objectives

This course envisions imparting the students to:

1	Provide basic knowledge of steam power cycle and different methods to improve the efficiency of the plant.
2	Develop comprehensive knowledge on boiler heat balance, steam turbine and condenser operation principles and to prepare the students to effectively use energy conversion theory in the practice of engineering.
3	Develop an intuitive understanding of energy conversion devices by emphasizing the scientific and engineering arguments.
4	Present a wealth of real-world engineering examples to give students a feel for how energy conversion principle is applied in engineering practice.
5	Classify, solve, and correlate Energy Conversion Systems problems.

Course Outcomes

After the completion of this course, students will be able to:

CO1	Understand the basic working principle of vapor power system.
CO2	Understand the combustion and energy equations to study the performance of boiler.
CO3	Apply the energy equation to evaluate the performance of nozzle.
CO4	Analyze impulse and reaction turbo machines for energy transfer.
CO5	Evaluate the performance of condenser.

SYLLABUS

Module	Lectures/hour
Module -I **Vapour Power Cycle:** Components of steam power system; Carnot vapour cycle and Rankine cycle; their comparisons; P-v, T-s & h-s diagrams; Deviation of actual vapour power cycle from ideal cycle; mean temperature of heat addition; Reheat cycle; Ideal regenerative cycle; feed water heaters	8
Module –II **Fuels and Combustions:** Classification of fuels; basic chemistry and combustion equations; conversion of volumetric to weight analysis and vise-versa; theoretical and excess air; Boiler performance: Equivalent evaporation; Boiler efficiency; Heat balance; Boiler Draught and its classification; Chimney height, maximum discharge and efficiency.	8
Module – III **Steam Nozzles:** Introduction; types of steam nozzles; nozzle efficiency; velocity of steam flow through the nozzle; discharge and condition of maximum discharge through a nozzle; physical significance of critical pressure ratio and choked flow; Supersaturated flow through nozzle; General relationship between area, velocity and pressure in nozzle flow.	8
Module - IV **Steam Turbines:** Classifications; compounding of turbines; working principle, velocity diagrams, diagram work and efficiency of impulse and reaction turbine; degree of reaction, Parsons turbine, condition for maximum efficiency impulse and reaction turbine; Losses in steam turbines, reheat factor and condition line; governing of steam turbine; Back-pressure and pass-out Turbine.	8
Module –V **Steam condensers:** Classification of condensers; sources of air leakage into the condenser; effects of air leakage in condenser; vacuum efficiency; condenser efficiency; cooling water calculations; Air ejector.	8

Textbooks:

1. Steam and Gas Turbines – R. Yadav, Central Publishing House
2. Elements of Heat Engine – Pandey & Saha
3. Thermal Engineering – R. K. Rajput
4. Power Plant Engineering – P.K. Nag; Tata McGraw-Hill publication

Reference Books:

1. Power Plant Technology- M.M.Ei. -Wakil, McGraw Hill
2. Theory and Practice of Heat Engine – D. A. Rangham; Camb. Univ. Press.

Gaps in the Syllabus (to meet Industry/Profession requirements)

1. Detailed analysis of combined power cycle.
2. Renewable energy conversion principles.
3. Major emissions and control.
4. Economics of energy conversion system.

POs met through Gaps in the Syllabus: 1, 2, 3, 4, 5

Topics beyond syllabus/Advanced topics/Design

1. Design of different energy conversion systems like nuclear reactors, turbines, and renewable energy devices.

POs met through Topics beyond syllabus/Advanced topics/Design: 1, 2, 3, 4, 5

Course Outcome (CO) Attainment Assessment Tools & Evaluation Procedure

Direct Assessment

Assessment Tool	% Contribution during CO Assessment
First Quiz	10
Mid Semester Examination	25
Second Quiz	10
Assignment	5
End Semester Examination	50

Indirect Assessment

1. Students' Feedback on Course Outcome.

Mapping of Course Outcomes onto Program Outcomes

Course Outcome	Program Outcomes (POs)											Program Specific Outcomes (PSOs)		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
CO1	3	2	3	2	1	1	1		1			2	3	2	1
CO2	3	2	3	2	1	1	1		1			3	3	2	1
CO3	3	2	3	2	1	1	1		1			2	3	2	1
CO4	3	2	3	2	1	1	1		1			2	3	2	1
CO5	3	2	3	2	1	1	1		1			2	3	2	1

Correlation Levels 1, 2 or 3 as defined below:

1: Slight (Low) 2: Moderate (Medium) 3: Substantial (High)

Mapping between COs and Course Delivery (CD) methods

CD Code	Course Delivery Methods	Course Outcome	Course Delivery Method Used
CD1	Lecture by use of Boards/LCD Projectors	CO1-5	CD1
CD2	Tutorials/Assignments	CO1-5	CD2
CD3	Seminars	-	-
CD4	Mini Projects/Projects	-	-
CD5	Laboratory Experiments/Teaching Aids	-	-
CD6	Industrial/Guest Lectures	-	-
CD7	Industrial Visits/In-plant Training	-	-
CD8	Self- learning such as use of NPTEL Materials and Internets	-	-
CD9	Simulation	-	-

Rankine cycle

It is ideal Vapor power cycle. No extraneous effect.

- Steam Boiler → Reversible Constant Pressure heating process of water to form steam.

- Steam Turbine → Reversible adiabatic expansion of Steam

- Condenser → Reversible Constant Pressure heat rejection of Steam. till it become Saturated liquid.

- Pump → Reversible adiabatic Compression of this liquid ending at initial pressure.

- When all process is ideal - It is called Rankine cycle.

- This is a reversible cycle.

Explanation

• For any given process, Steam approaching turbine must be dry (State 1); wet (1') or Superheated (state 1'')

But fluid approaching the pump must be Saturated Liquid (State 3)

• Fig. Ref. 489 - PK Nag [Fig 12.5]

For 1 kg Fluid.

SFEE for Boiler → $h_1 = Q_1 + h_4$

Turbine → $h_1 = W_T + h_2$

Condenser → $h_2 = Q_2 + h_3$

Pump → $h_4 = h_3 + W_P$

$$\eta = \frac{W_{net}}{Q_1} = \frac{W_T - W_P}{Q_1}$$

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Actual Vapour Cycle Process:

Ref Fig. 12·7 & 12·8·

· Actual cycle is different from ideal cycle. There are the main source of loss.

① Pumping loss:

Pressure drop due to friction and heat loss to the surrounding are the most important pumping loss.

② Turbine Losses:

Loss in turbine due to frictional effection and heat loss to the surrounding.

$$h_1 = h_2 + W_T + Q_{loss}$$

$$\eta_T = \frac{W_T \ (h_1 - h_2)}{h_1 - h_{2s}}$$ — Isentropic efficiency

For ordinary turbine, Q_{loss} is very small.

In actual turbine work is less than reversible ideal work output

$h_2 > h_{2s}$

Except for small turbine, ~~R~~ q_{loss} is negligible

Pumping Losses:

Mainly due to frictional loss and Heat transfer is ~~negkit~~ negligible.

$$\eta_p = \frac{h_{4s} - h_3}{W_p} = \frac{h_{4s} - h_3}{h_4 - h_3}$$

Condenser losses:

- It is very small.
- It include Loss of Pressure and cooling of condensate below the saturation temperature

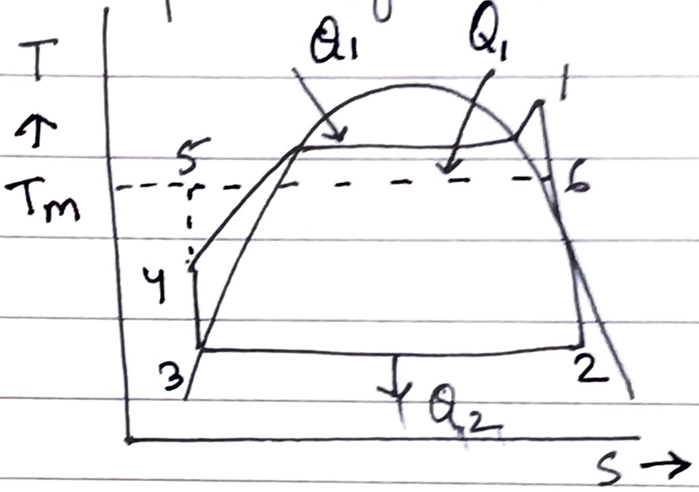
Comparison between Rankine and Carnot (Vapour) cycle

Carnot cycle	Rankine Cycle
① Ideal cycle, Not used Practically in Thermal Power plant	① Practical cycle, Used in thermal power plant.
② Efficiency is high	② Efficiency is less
③ Reversible Isothermal heat supply (Boiler)	3) Constant Pressure heat supply (Boiler)
④ Rev. Adiabatic Expansion [Turbine]	4) Rev. Adiabatic Expansion [Turbine]
⑤ Rev. Isothermal Heat rejection [Condenser]	5) Constant Pressure heat rejection [Condenser]
⑥ Revers. Adiabatic Compression	⑥ Rev. Adiabatic Compression
⑦ Boiler → Turbine → Condenser ↓ Compressor → Boiler	⑦ Boiler → Turbine ↓ Condenser → Pump ↑ Boiler

Carnot	Rankine
· Superheating of Steam is not possible	· Superheating of Steam is possible.
· Wet steam comes out from Condenser	· Saturated liquid comes out from Condenser.
· It is difficult to Compress wet Steam in Compressor	· It is easy to compress Saturated liquid in pump

Mean Temperature of Heat addition (T_m)

- Here in Rankine cycle, constant pressure heat addition is replaced by isothermal process of heat addition.



Heat supplied T_m

$Q_1 = h_6 - h_5$

$= h_1 - h_4$

$$ds = \frac{dQ}{T} \; ; \quad S_6 - S_5 = \frac{h_6 - h_5}{T_m}$$

$$\Rightarrow h_6 - h_5 = T_m(S_6 - S_5) = Q_1$$

Q_2 = Heat rejected $= h_2 - h_3$

$$ds = \frac{dQ}{T} \Rightarrow S_2 - S_3 = \frac{h_2 - h_3}{T_2}$$

$$\Rightarrow \boxed{h_2 - h_3 = Q_2 = T_2(S_2 - S_3)} = T_2(S_6 - S_5)$$

$$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2(S_6 - S_5)}{T_m(S_6 - S_5)}$$

$$\boxed{\eta = 1 - \frac{T_2}{T_m}}$$

Sometime $T_2 = T_0$ = Amb. Temp.

SEP	M	T	W	T	F	S	S	M	T	W	T	F	S	S	M	T	W	T	F	S	S
				1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	19	20	21	22	23	24	25	26	27	28	29	30									

· See Video of ~~Ra~~ Ideal regenerative cycle

· To improve the Rankine cycle —

Lot of Sensible heat is added in water

mean temperature at which Sensible heat is added is very very low.

· Now how to increase the efficiency of Rankine by reducing the energy supplied for Sensible heating.

Increasing the temperature of water to almost the exit temperature of the turbine. The reduces Heat requirement. Then it increases efficiency.

- water and Steam goes in counter direction.

$$\Delta T_{steam} = \Delta T_{water}$$

This process is exactly reversible.

Pump = Feed water Pump

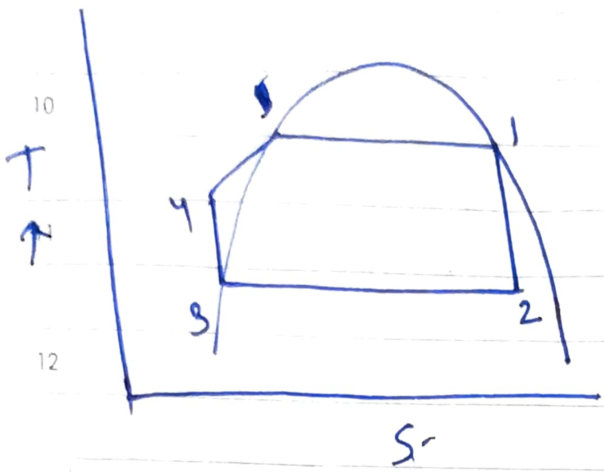
~~Slope of line 1-4' = Slope of line 4-4'.~~

Assume → Reversible heating

S	M	T	W	T	F	S	S	M	T	W	T	F	S	S	M	T	W	T	F	S	S
E				1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
P	19	20	21	22	23	24	25	26	27	28	29	30									

Ideal Regenerative cycle

9
10
12



$$\eta = 1 - \frac{T_L}{T_m}$$

Increase T_m, $\eta \uparrow$

1
2
3
4
5
6
7

→ way -

① Superheating

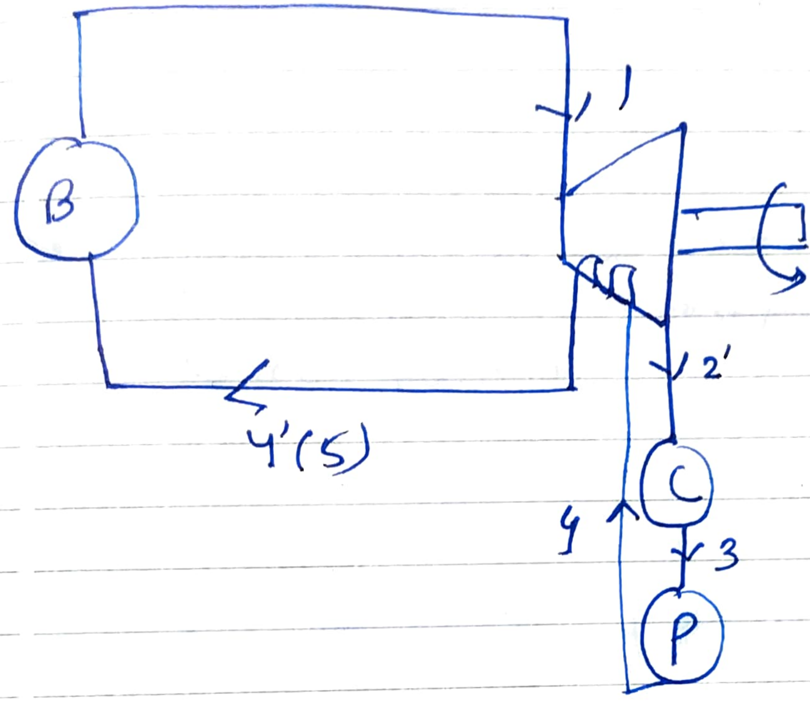
② Reheating

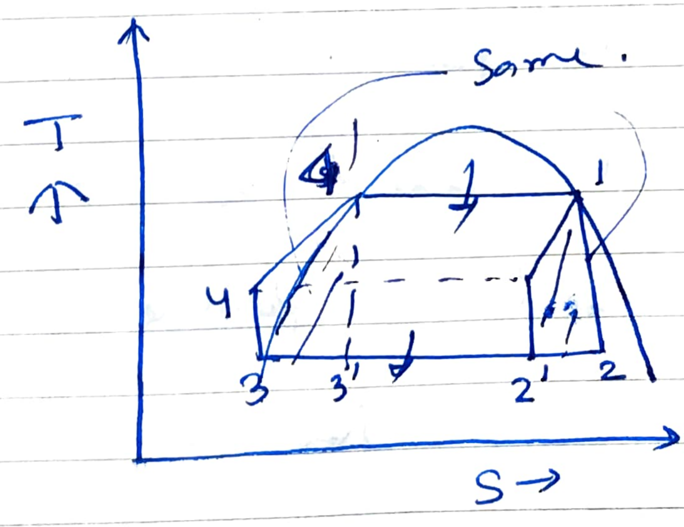
- Now Sensible heating 4-5 Process Can be elemented by passing feed water inside the casing of the turbine.

Then η of Rankine cycle become η of Carnot cycle.

Because heat addition happen isothermal.

18 SUNDAY





Renergention → 1 - 2' - 3' - 4'

Now ~~Car~~like Carnot cycle → Isothermal heat addition and isothermal heat rejection.

$W_T = h_1 - h_2'$

$Q_s = h_1 - h_4'$

Assumption → ① Reversible heat transfer or Feed Temperature difference between steam and feed water is infinitely small. ✓

② Efficiency of Renergative is Carnot Efficiency

Practical Not Possible

① Reversible Heat transfer → Very slow

② Mechanical not possible in Turbine

③ Blade erosion happen.

$$Q_1 = h_1 - h_4' = T_1 (s_1 - s_4')$$

$$Q_2 = h_2' - h_3 = T_2 (s_2' - s_3)$$

$$\Rightarrow s_4' - s_3 = s_1 - s_2'$$

$$\Rightarrow s_1 - s_4' = s_2' - s_3$$

$$\eta = 1 - \frac{Q_2}{Q_1}$$

$$\boxed{\eta = 1 - \frac{T_2}{T_1}}$$

Eff. of Ideal Regenerative cycle is equal to Carnot cycle.

Now Energy Balance equation for turbine; $h_1 - W_T - h_2' + h_4 - h_{4'} = 0$

$$W_T = (h_1 - h_2') - (h_4' - h_4)$$

$$\boxed{W_P = h_4 - h_3}$$

Example *A steam power station uses the following cycle:*
Steam at boiler outlet—150 bar, 550°C
Reheat at 40 bar to 550°C
Condenser at 0.1 bar.

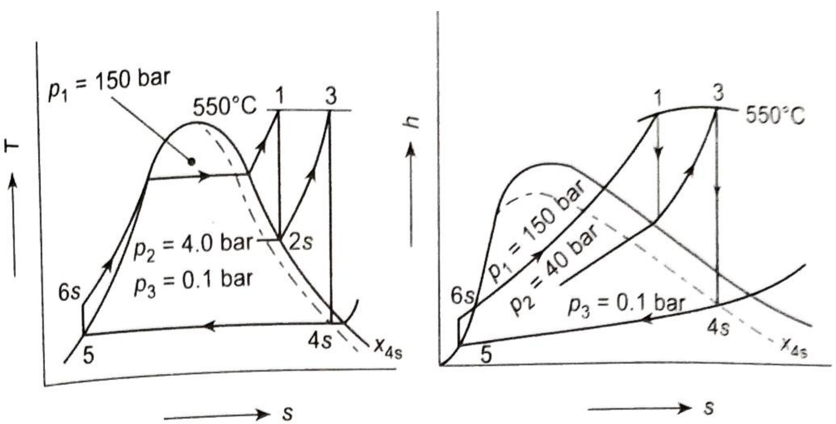
Using the Mollier chart and assuming ideal processes, find the (a) quality at turbine exhaust, (b) cycle efficiency, and (c) steam rate.

Solution The property values at different states are read from the Mollier chart.

$$h_1 = 3465, h_{2s} = 3065, h_3 = 3565,$$
$$h_{4s} = 2300 \text{ kJ/kg}, x_{4s} = 0.88, h_5 \text{ (steam table)} = 191.83 \text{ kJ/kg}$$

Quality at turbine exhaust = **0.88**

$$W_P = v\Delta p = 10^{-3} \times 150 \times 10^2 = 15 \text{ kJ/kg}$$
$$h_{6s} = 206.83 \text{ kJ/kg}$$
$$Q_1 = (h_1 - h_{6s}) + (h_3 - h_{2s})$$



(a) (b)

$$= (3465 - 206.83) + (3565 - 3065) = 3758.17 \text{ kJ/kg}$$
$$W_T = (h_1 - h_{2s}) + (h_3 - h_{4s})$$
$$= (3465 - 3065) + (3565 - 2300) = 1665 \text{ kJ/kg}$$
$$W_{net} = W_T - W_P = 1665 - 15 = 1650 \text{ kJ/kg}$$
$$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{1650}{3758.17} = \mathbf{0.4390, or\ 43.9\%}$$
$$\text{Steam rate} = \frac{3600}{1650} = \mathbf{2.18\ kg/kW\ h}$$

Example *In a single-heater regenerative cycle the steam enters the turbine at 30 bar, 400°C and the exhaust pressure is 0.10 bar. The feed water heater is a direct-contact type which operates at 5 bar. Find (a) the efficiency and the steam rate of the cycle and (b) the increase in mean temperature of heat addition, efficiency and steam rate, as compared to the Rankine cycle (without regeneration). Neglect pump work.*

Solution Figure gives the flow, *T-s*, and *h-s* diagrams. From the steam tables, the property values at various states have been obtained.

$$h_1 = 3230.9 \text{ kJ/kg}$$
$$s_1 = 6.9212 \text{ kJ/kg K} = s_2 = s_3$$
$$s_g \text{ at 5 bar} = 6.8213 \text{ kJ/kg K}$$

Since $s_2 > s_g$, the state 2 must lie in the superheated region. From the table for superheated steam $t_2 = 172°C, h_2 = 2796$ kJ/kg

$$s_3 = 6.9212 = s_{f0.1 \text{ bar}} + x_3 s_{fg0.1 \text{ bar}}$$
$$= 0.6493 + x_3 7.5009$$

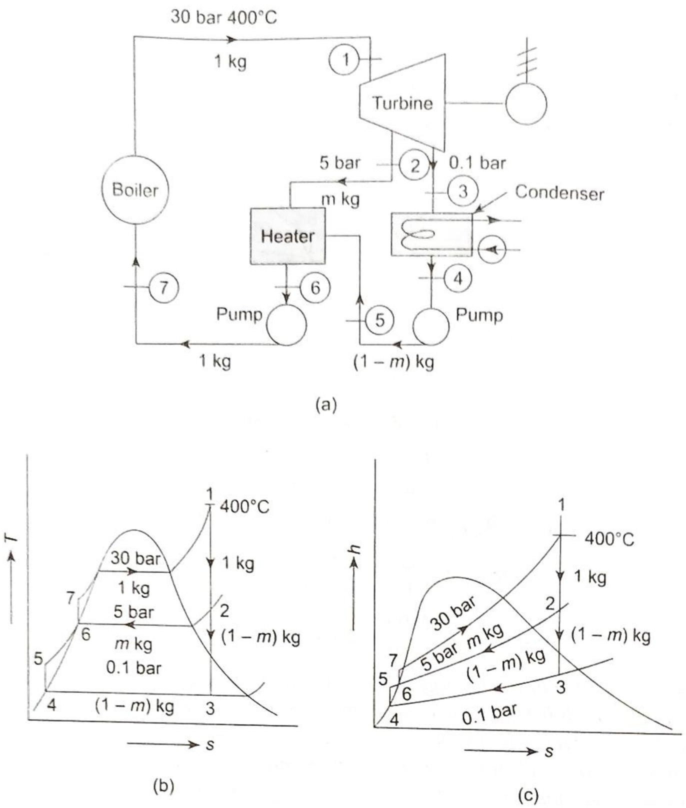
$$\therefore \quad x_3 = \frac{6.2719}{7.5009} = 0.836$$
$$\therefore \quad h_3 = 191.83 + 0.836 \times 2392.8 = 2192.2 \text{ kJ/kg}$$

Since pump work is neglected

$$h_4 = 191.83 \text{ kJ/kg} = h_5$$
$$h_6 = 640.23 \text{ kJ/kg} = h_7$$

Energy balance for the heater gives

$$m(h_2 - h_6) = (1 - m)(h_6 - h_5)$$
$$m(2796 - 640.23) = (1 - m)(640.23 - 191.83)$$
$$2155.77\, m = 548.4 - 548.4\, m$$
$$\therefore \quad m = \frac{548.4}{2704.17} = 0.203 \text{ kg}$$
$$\therefore \quad W_T = (h_1 - h_2) + (1 - m)(h_2 - h_3)$$
$$= (3230.9 - 2796) + 0.797\,(2796 - 2192.2)$$
$$= 916.13 \text{ kJ/kg}$$



$$Q_1 = h_1 - h_6 = 3230.9 - 640.23 = 2590.67 \text{ kJ/kg}$$

$$\eta_{\text{cycle}} = \frac{916.13}{2590.67} = \mathbf{0.3536, \text{ or } 35.36\%}$$

$$\text{Steam rate} = \frac{3600}{916.13} = \mathbf{3.93 \text{ kg/kW h}}$$

$$T_{m1} = \frac{h_1 - h_7}{s_1 - s_7} = \frac{2590.67}{6.9212 - 1.8607} = 511.95 \text{ K} = 238.8°\text{C}$$

$$T_{m1} \text{ (without regeneration)} = \frac{h_1 - h_4}{s_1 - s_4}$$

$$= \frac{3039.07}{6.9212 - 0.6493}$$

$$= 484.55 \text{ K}$$

$$= 211.4 °\text{C}$$

Increase in T_{m1} due to regeneration

$$= 238.8 - 211.4 = \mathbf{27.4°C}$$

$$W_T \text{(without regeneration)} = h_1 - h_3$$

$$= 3230.9 - 2192.2 = 1038.7 \text{ kJ/kg}$$

$$\text{Steam rate (without regeneration)} = \frac{3600}{1038.7} = 3.46 \text{ kg/kW h}$$

∴ Increase in steam rate due to regeneration

$$= 3.93 - 3.46 = \mathbf{0.476 \text{ kg/kW h}}$$

$$\eta_{\text{cycle}} \text{ (without regeneration)} = \frac{h_1 - h_3}{h_1 - h_4} = \frac{1038.7}{3039.07}$$

$$= 0.3418 \text{ or } 34.18\%$$

∴ Increase in cycle efficiency due to regeneration

$$= 35.36 - 34.18 = \mathbf{1.18\%}$$

Example *In a steam power plant the condition of steam at inlet to the steam generator is 20 bar and 300°C and the condenser pressure is 0.1 bar. Two feedwater heaters operate at optimum temperature. Determine: (a) the quality of steam at turbine exhaust, (b) net work per kg of steam, (c) cycle efficiency, and (d) the stream rate. Neglect pump work.*

Solution From Fig.

$$h_1 = 3023.5 \text{ kJ/kg}$$
$$s_1 = 6.7664 \text{ kJ/kg K} = s_2 = s_3 = s_4$$
$$t_{sat} \text{ at } 20 \text{ bar} \cong 212°\text{C}$$
$$t_{sat} \text{ at } 0.1 \text{ bar} \cong 46°\text{C}$$
$$\Delta t_{OA} = 212 - 46 = 166°\text{C}$$

$\therefore$ Temperature rise per heater $= \dfrac{166}{3} = 55°\text{C}$

$\therefore$ Temperature at which the first heater operates

$$= 212 - 55 = 157°\text{C} \cong 150°\text{C (assumed)}$$

Temperature at which the second heater operates $= 157 - 55 = 102°\text{C} \cong 100°\text{C}$ (assumed)

At 0.1 bar,

$$h_f = 191.83, h_{fg} = 2392.8, s_f = 0.6493$$
$$s_g = 8.1502$$

At 100°C,

$$h_f = 419.04, h_{fg} = 2257.0, s_f = 1.3069, s_g = 7.3549$$

At 150°C,

$$h_f = 632.20, h_{fg} = 2114.3, s_f = 1.8418, s_g = 6.8379$$

$$6.7664 = 1.8418 + x_2 \times 4.9961$$

$\therefore$ $x_2 = 0.986$

$\therefore$ $h_2 = 632.2 + 0.986 \times 2114.3$

$$= 2716.9 \text{ kJ/kg}$$

$$6.7664 = 1.3069 + x_3 \times 6.0480$$
$$x_3 = 0.903$$

$\therefore$ $h_3 = 419.04 + 0.903 \times 2257.0$

or $h_3 = 2457.1$ kJ/kg

$$6.7664 = 0.6493 + x_4 \times 7.5010$$
$$x_4 = 0.816$$

$\therefore$ $h_4 = 191.83 + 0.816 \times 2392.8$

$$= 2144.3 \text{ kJ/kg}$$

Since pump work is neglected, $h_{10} = h_9$, $h_8 = h_7$, $h_6 = h_5$. By making an energy balance for the hp heater

$$(1 - m_1)(h_9 - h_8) = m_1(h_2 - h_9)$$

Rearranging

$$m_1 = \frac{h_9 - h_7}{h_2 - h_7} = \frac{213.16}{2297.86} = 0.093 \text{ kg}$$

By making an energy balance for the *lp* heater,

$$(1 - m_1 - m_2)(h_7 - h_6) = m_2(h_3 - h_7)$$
$$(1 - 0.093 - m_2)(419.04 - 191.83) = m_2(2457.1 - 419.04)$$

$\therefore$ $m_2 = 0.091$ kg

$\therefore$ $W_T = 1(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$

$$= (3023.5 - 2716.9) + (1 - 0.093)(2716.9 - 2457.1) + (1 - 0.093 - 0.091)(2457.1 - 2144.3) = 797.48 \text{ kJ/kg}$$

$$Q_1 = h_1 - h_9 = 3023.5 - 632.2 = 2391.3 \text{ kJ/kg}$$

$$\eta_{cycle} = \frac{W_T - W_P}{Q_1} = \frac{797.48}{2391.3} = 0.3334 \text{ or } 33.34\%$$

$$\text{Steam rate} = \frac{3600}{W_{net}}$$

$$= \frac{3600}{797.48} = 4.51 \text{ kg/kW h}$$

EXAMPLE 10–2 An Actual Steam Power Cycle

A steam power plant operates on the cycle shown in Fig. 10–5. If the isentropic efficiency of the turbine is 87 percent and the isentropic efficiency of the pump is 85 percent, determine (*a*) the thermal efficiency of the cycle and (*b*) the net power output of the plant for a mass flow rate of 15 kg/s.

Solution A steam power cycle with specified turbine and pump efficiencies is considered. The thermal efficiency and the net power output are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–5. The temperatures and pressures of steam at various points are also indicated on the figure. We note that the power plant involves steady-flow components and operates on the Rankine cycle, but the imperfections at various components are accounted for.

(*a*) The thermal efficiency of a cycle is the ratio of the net work output to the heat input, and it is determined as follows:

Pump work input:

$$w_{\text{pump,in}} = \frac{w_{s,\text{pump,in}}}{\eta_P} = \frac{v_1(P_2 - P_1)}{\eta_P}$$

$$= \frac{(0.001009 \text{ m}^3/\text{kg})[(16{,}000 - 9) \text{ kPa}]}{0.85}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

$$= 19.0 \text{ kJ/kg}$$

Turbine work output:

$$w_{\text{turb,out}} = \eta_T w_{s,\text{turb,out}}$$

$$= \eta_T(h_5 - h_{6s}) = 0.87(3583.1 - 2115.3) \text{ kJ/kg}$$

$$= 1277.0 \text{ kJ/kg}$$

Boiler heat input: $q_{\text{in}} = h_4 - h_3 = (3647.6 - 160.1) \text{ kJ/kg} = 3487.5 \text{ kJ/kg}$

Thus,

$$w_{\text{net}} = w_{\text{turb,out}} - w_{\text{pump,in}} = (1277.0 - 19.0) \text{ kJ/kg} = 1258.0 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1258.0 \text{ kJ/kg}}{3487.5 \text{ kJ/kg}} = \mathbf{0.361 \text{ or } 36.1\%}$$

(*b*) The power produced by this power plant is

$$\dot{W}_{\text{net}} = \dot{m}(w_{\text{net}}) = (15 \text{ kg/s})(1258.0 \text{ kJ/kg}) = \mathbf{18.9 \text{ MW}}$$

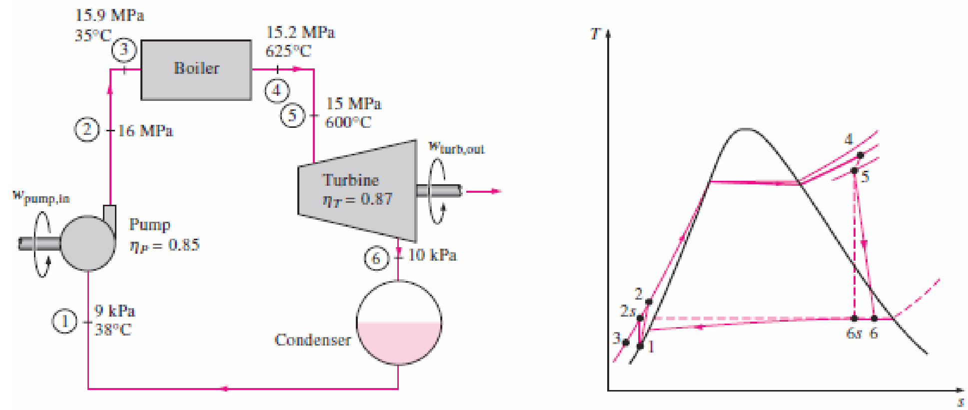


FIGURE 10–5
Schematic and *T-s* diagram for Example 10–2.

EXAMPLE : The Ideal Reheat Rankine Cycle

Consider a steam power plant operating on the ideal reheat Rankine cycle. Steam enters the high-pressure turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. If the moisture content of the steam at the exit of the low-pressure turbine is not to exceed 10.4 percent, determine (*a*) the pressure at which the steam should be reheated and (*b*) the thermal efficiency of the cycle. Assume the steam is reheated to the inlet temperature of the high-pressure turbine.

Solution A steam power plant operating on the ideal reheat Rankine cycle is considered. For a specified moisture content at the turbine exit, the reheat pressure and the thermal efficiency are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–13. We note that the power plant operates on the ideal reheat Rankine cycle. Therefore, the pump and the turbines are isentropic, there are no pressure drops in the boiler and condenser, and steam leaves the condenser and enters the pump as saturated liquid at the condenser pressure.

(*a*) The reheat pressure is determined from the requirement that the entropies at states 5 and 6 be the same:

State 6: $P_6 = 10 \text{ kPa}$

$$x_6 = 0.896 \quad \text{(sat. mixture)}$$

$$s_6 = s_f + x_6 s_{fg} = 0.6492 + 0.896(7.4996) = 7.3688 \text{ kJ/kg} \cdot \text{K}$$

Also,

$$h_6 = h_f + x_6 h_{fg} = 191.81 + 0.896(2392.1) = 2335.1 \text{ kJ/kg}$$

Thus,

State 5: $\left.\begin{array}{l} T_5 = 600^\circ\text{C} \\ s_5 = s_6 \end{array}\right\} \quad \begin{array}{l} P_5 = 4.0 \text{ MPa} \\ h_5 = 3674.9 \text{ kJ/kg} \end{array}$

Therefore, steam should be reheated at a pressure of 4 MPa or lower to prevent a moisture content above 10.4 percent.

(*b*) To determine the thermal efficiency, we need to know the enthalpies at all other states:

State 1: $\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} \quad \begin{array}{l} h_1 = h_{f @ 10 \text{ kPa}} = 191.81 \text{ kJ/kg} \\ v_1 = v_{f @ 10 \text{ kPa}} = 0.00101 \text{ m}^3\text{/kg} \end{array}$

State 2: $P_2 = 15 \text{ MPa}$

$$s_2 = s_1$$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3\text{/kg}) \times [(15{,}000 - 10)\text{kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= 15.14 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump,in}} = (191.81 + 15.14) \text{ kJ/kg} = 206.95 \text{ kJ/kg}$$

State 3: $\left.\begin{array}{l} P_3 = 15 \text{ MPa} \\ T_3 = 600^\circ\text{C} \end{array}\right\} \quad \begin{array}{l} h_3 = 3583.1 \text{ kJ/kg} \\ s_3 = 6.6796 \text{ kJ/kg} \cdot \text{K} \end{array}$

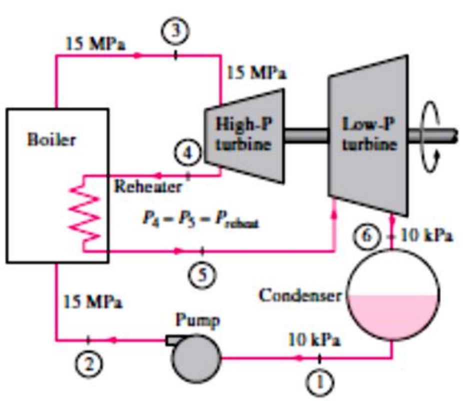
State 4: $\left.\begin{array}{l} P_4 = 4 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} \quad \begin{array}{l} h_4 = 3155.0 \text{ kJ/kg} \\ (T_4 = 375.5^\circ\text{C}) \end{array}$

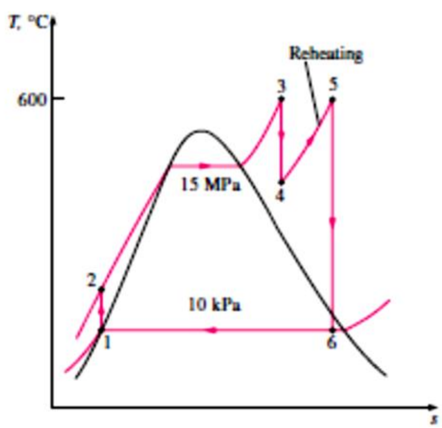
Thus

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4)$$
$$= (3583.1 - 206.95) \text{ kJ/kg} + (3674.9 - 3155.0) \text{ kJ/kg}$$
$$= 3896.1 \text{ kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = (2335.1 - 191.81) \text{ kJ/kg}$$
$$= 2143.3 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2143.3 \text{ kJ/kg}}{3896.1 \text{ kJ/kg}} = 0.450 \text{ or } 45.0\%$$





EXAMPLE : **The Ideal Regenerative Rankine Cycle**

Consider a steam power plant operating on the ideal regenerative Rankine cycle with one open feedwater heater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam leaves the turbine at a pressure of 1.2 MPa and enters the open feedwater heater. Determine the fraction of steam extracted from the turbine and the thermal efficiency of the cycle.

Solution A steam power plant operates on the ideal regenerative Rankine cycle with one open feedwater heater. The fraction of steam extracted from the turbine and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–18. We note that the power plant operates on the ideal regenerative Rankine cycle. Therefore, the pumps and the turbines are isentropic; there are no pressure drops in the boiler, condenser, and feedwater heater; and steam leaves the condenser and the feedwater heater as saturated liquid. First, we determine the enthalpies at various states:

State 1: $P_1 = 10\text{ kPa}$, Sat. liquid $\Big\}$ $h_1 = h_{f @ 10\text{ kPa}} = 191.81\text{ kJ/kg}$; $v_1 = v_{f @ 10\text{ kPa}} = 0.00101\text{ m}^3/\text{kg}$

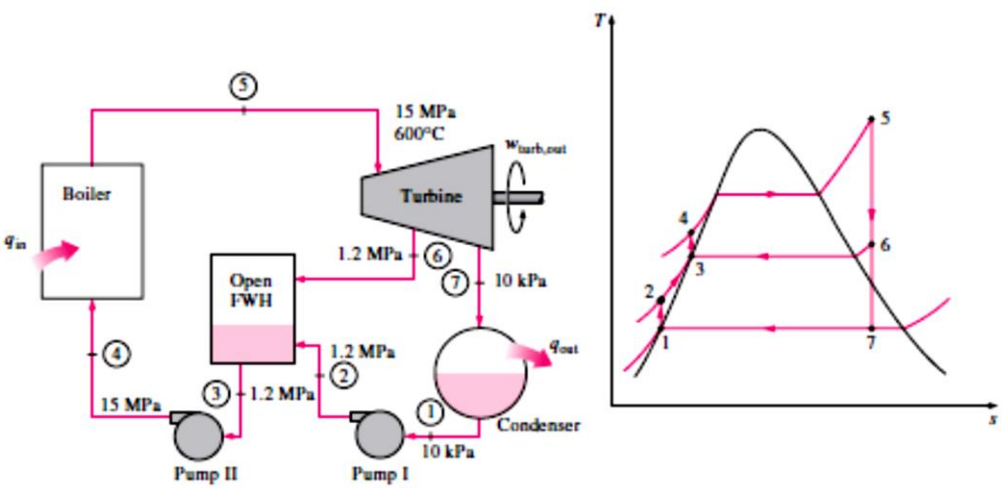
State 2: $P_2 = 1.2\text{ MPa}$

$s_2 = s_1$

$$w_{\text{pump I,in}} = v_1(P_2 - P_1) = (0.00101\text{ m}^3/\text{kg})[(1200 - 10)\text{ kPa}]\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 1.20\text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump I,in}} = (191.81 + 1.20)\text{ kJ/kg} = 193.01\text{ kJ/kg}$$



State 3: $P_3 = 1.2\text{ MPa}$, Sat. liquid $\Big\}$ $v_3 = v_{f @ 1.2\text{ MPa}} = 0.001138\text{ m}^3/\text{kg}$; $h_3 = h_{f @ 1.2\text{ MPa}} = 798.33\text{ kJ/kg}$

State 4: $P_4 = 15\text{ MPa}$

$s_4 = s_3$

$$w_{\text{pump II,in}} = v_3(P_4 - P_3)$$

$$= (0.001138\text{ m}^3/\text{kg})[(15{,}000 - 1200)\text{ kPa}]\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 15.70\text{ kJ/kg}$$

$$h_4 = h_3 + w_{\text{pump II,in}} = (798.33 + 15.70)\text{ kJ/kg} = 814.03\text{ kJ/kg}$$

State 5: $P_5 = 15\text{ MPa}$, $T_5 = 600°\text{C}$ $\Big\}$ $h_5 = 3583.1\text{ kJ/kg}$; $s_5 = 6.6796\text{ kJ/kg}\cdot\text{K}$

State 6: $P_6 = 1.2\text{ MPa}$, $s_6 = s_5$ $\Big\}$ $h_6 = 2860.2\text{ kJ/kg}$; $(T_6 = 218.4°\text{C})$

State 7: $P_7 = 10\text{ kPa}$

$$s_7 = s_5 \qquad x_7 = \frac{s_7 - s_f}{s_{fg}} = \frac{6.6796 - 0.6492}{7.4996} = 0.8041$$

$$h_7 = h_f + x_7 h_{fg} = 191.81 + 0.8041(2392.1) = 2115.3\text{ kJ/kg}$$

The energy analysis of open feedwater heaters is identical to the energy analysis of mixing chambers. The feedwater heaters are generally well insulated ($\dot{Q} = 0$), and they do not involve any work interactions ($\dot{W} = 0$). By neglecting the kinetic and potential energies of the streams, the energy balance reduces for a feedwater heater to

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \rightarrow \sum_{\text{in}} \dot{m}h = \sum_{\text{out}} \dot{m}h$$

or

$$yh_6 + (1 - y)h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6/\dot{m}_5$). Solving for y and substituting the enthalpy values, we find

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{798.33 - 193.01}{2860.2 - 193.01} = 0.2270$$

Thus,

$$q_{\text{in}} = h_5 - h_4 = (3583.1 - 814.03)\text{ kJ/kg} = 2769.1\text{ kJ/kg}$$

$$q_{\text{out}} = (1 - y)(h_7 - h_1) = (1 - 0.2270)(2115.3 - 191.81)\text{ kJ/kg}$$

$$= 1486.9\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1486.9\text{ kJ/kg}}{2769.1\text{ kJ/kg}} = 0.463\text{ or }46.3\%$$

EXAMPLE The Ideal Reheat–Regenerative Rankine Cycle

Consider a steam power plant that operates on an ideal reheat–regenerative Rankine cycle with one open feedwater heater, one closed feedwater heater, and one reheater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam is extracted from the turbine at 4 MPa for the closed feedwater heater, and the remaining steam is reheated at the same pressure to 600°C. The extracted steam is completely condensed in the heater and is pumped to 15 MPa before it mixes with the feedwater at the same pressure. Steam for the open feedwater heater is extracted from the low-pressure turbine at a pressure of 0.5 MPa. Determine the fractions of steam extracted from the turbine as well as the thermal efficiency of the cycle.

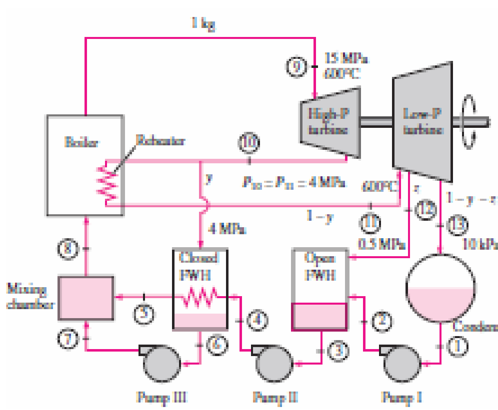
Solution A steam power plant operates on the ideal reheat–regenerative Rankine cycle with one open feedwater heater, one closed feedwater heater, and one reheater. The fractions of steam extracted from the turbine and the thermal efficiency are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible. 3 In both open and closed feedwater heaters, feedwater is heated to the saturation temperature at the feedwater heater pressure. (Note that this is a conservative assumption since extracted steam enters the closed feedwater heater at 376°C and the saturation temperature at the closed feedwater pressure of 4 MPa is 250°C).

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–19. The power plant operates on the ideal reheat–regenerative Rankine cycle and thus the pumps and the turbines are isentropic; there are no pressure drops in the boiler, reheater, condenser, and feedwater heaters; and steam leaves the condenser and the feedwater heaters as saturated liquid.

The enthalpies at the various states and the pump work per unit mass of fluid flowing through them are

$h_1 = 191.81$ kJ/kg	$h_9 = 3155.0$ kJ/kg
$h_2 = 192.30$ kJ/kg	$h_{10} = 3155.0$ kJ/kg
$h_3 = 640.09$ kJ/kg	$h_{11} = 3674.9$ kJ/kg
$h_4 = 643.92$ kJ/kg	$h_{12} = 3014.8$ kJ/kg
$h_5 = 1087.4$ kJ/kg	$h_{13} = 2335.7$ kJ/kg
$h_6 = 1087.4$ kJ/kg	$w_{\text{pump I,in}} = 0.49$ kJ/kg
$h_7 = 1101.2$ kJ/kg	$w_{\text{pump II,in}} = 3.83$ kJ/kg
$h_8 = 1089.8$ kJ/kg	$w_{\text{pump III,in}} = 13.77$ kJ/kg



The fractions of steam extracted are determined from the mass and energy balances of the feedwater heaters:

Closed feedwater heater:

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

$$yh_{10} + (1 - y)h_4 = (1 - y)h_5 + yh_6$$

$$y = \frac{h_5 - h_4}{(h_{10} - h_6) + (h_5 - h_4)} = \frac{1087.4 - 643.92}{(3155.0 - 1087.4) + (1087.4 - 643.92)} = 0.1766$$

Open feedwater heater:

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

$$zh_{12} + (1 - y - z)h_2 = (1 - y)h_3$$

$$z = \frac{(1 - y)(h_3 - h_2)}{h_{12} - h_2} = \frac{(1 - 0.1766)(640.09 - 192.30)}{3014.8 - 192.30} = 0.1306$$

The enthalpy at state 8 is determined by applying the mass and energy equations to the mixing chamber, which is assumed to be insulated:

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

$$(1)h_8 = (1 - y)h_5 + yh_7$$

$$h_8 = (1 - 0.1766)(1087.4)\text{ kJ/kg} + 0.1766(1101.2)\text{ kJ/kg}$$

$$= 1089.8\text{ kJ/kg}$$

Thus,

$$q_{in} = (h_9 - h_8) + (1 - y)(h_{11} - h_{10})$$

$$= (3583.1 - 1089.8)\text{ kJ/kg} + (1 - 0.1766)(3674.9 - 3155.0)\text{ kJ/kg}$$

$$= 2921.4\text{ kJ/kg}$$

$$q_{out} = (1 - y - z)(h_{13} - h_1)$$

$$= (1 - 0.1766 - 0.1306)(2335.7 - 191.81)\text{ kJ/kg}$$

$$= 1485.3\text{ kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1485.3\text{ kJ/kg}}{2921.4\text{ kJ/kg}} = 0.492\text{ or }49.2\%$$

Discussion This problem was worked out in Example 10–4 for the same pressure and temperature limits with reheat but without the regeneration process. A comparison of the two results reveals that the thermal efficiency of the cycle has increased from 45.0 to 49.2 percent as a result of regeneration.

The thermal efficiency of this cycle could also be determined from

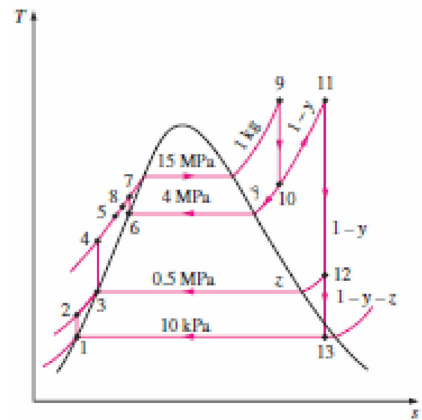
$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{w_{turb,out} - w_{pump,in}}{q_{in}}$$

where

$$w_{turb,out} = (h_9 - h_{10}) + (1 - y)(h_{11} - h_{12}) + (1 - y - z)(h_{12} - h_{13})$$

$$w_{pump,in} = (1 - y - z)w_{\text{pump I,in}} + (1 - y)w_{\text{pump II,in}} + (y)w_{\text{pump III,in}}$$

Also, if we assume that the feedwater leaves the closed FWH as a saturated liquid at 15 MPa (and thus at $T_5 = 342$°C and $h_5 = 1610.3$ kJ/kg), it can be shown that the thermal efficiency would be 50.6.



Example *Determine the work required to compress steam isentropically from 1 bar to 10 bar, assuming that at the initial state the steam exists as (a) saturated liquid and (b) saturated vapour. Neglect changes in kinetic and potential energies. What conclusion do you derive from this example?*

Solution The compression processes are shown in Fig. 12.40.

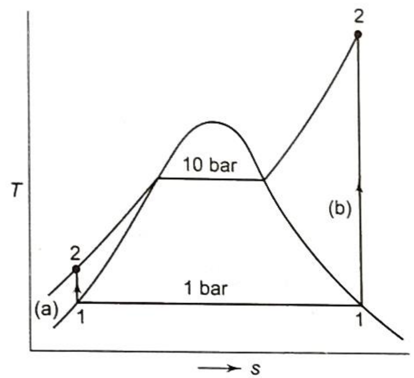
(a) Steam is a saturated liquid initially, and its specific volume is:

$$v_1 = (v_f)_{1\text{bar}} = 0.001043 \text{ m}^3/\text{kg}$$

Since liquid is incompressible, v_1 remains constant.

$$W_{rev} = \int_1^2 v\,dp = v_1(p_1 - p_2) = 0.001043\,(1 - 10) \times 10^2$$

$$= \mathbf{-\,0.9387\ kJ/kg.}$$



(b) Steam is a saturated vapour initially and remains a vapour during the entire compression process. Since the specific volume of a gas changes considerably during a compression process, we need to know how v varies with p to perform the integration $-\int v\,dp$. This relation is not readily available. But for an isentropic process, it is easily obtained from the property relation

$$Tds = dh - vdp = 0$$

or

$$v\,dp = dh$$

$$W_{rev} = \int_1^2 v\,dp = -\int_1^2 dh = h_1 - h_2$$

From steam tables,

$$h_1 = (h_g)_{1\text{bar}} = 2675.5 \text{ kJ/kg}$$

$$s_1 = (s_g)_{1\text{bar}} = 7.3594 \text{ kJ/kg K} = s_2$$

For $p = 10$ bar $= 1$ MPa and $s = 7.3594$ kJ/kg K, by interpolation

$$h_2 = 3195.5 \text{ kJ/kg}$$

$$W_{rev} = 2675.5 - 3195.5 = \mathbf{-\,520\ kJ/kg}$$

...se.

Example *Steam at 20 bar, 360°C is expanded in a steam turbine to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. (a) Assuming ideal processes, find per kg of steam the net work and the cycle efficiency. (b) If the turbine and the pump have each 80% efficiency, find the percentage reduction in the net work and cycle efficiency.*

Solution The property values at different state points (found from the steam tables are given below.

$$h_1 = 3159.3 \text{ kJ/kg} \qquad s_1 = 6.9917 \text{ kJ/kg K}$$

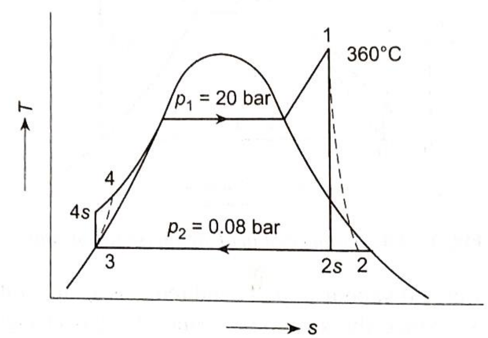
$$h_3 = h_{f_{p2}} = 173.88 \text{ kJ/kg} \qquad s_3 = s_{f_{p2}} = 0.5926 \text{ kJ/kg K}$$

$$h_{fg_{p2}} = 2403.1 \text{ kJ/kg} \qquad s_{gp2} = 8.2287 \text{ kJ/kg K}$$

$$v_{f_{p2}} = 0.001008 \text{ m}^3/\text{kg} \qquad \therefore \quad s_{fg_{p2}} = 7.6361 \text{ kJ/kg K}$$

Now $\quad s_1 = s_{2s} = 6.9917 = s_{f_{p2}} + x_{2s}\, s_{fgp2} = 0.5926 + x_2 \cdot 7.6361$

$$\therefore \quad x_{2s} = \frac{6.3991}{7.6361} = 0.838$$



$$\therefore \quad h_{2s} - h_{f_{p2}} + x_{2s} h_{fg_{p2}} = 173.88 + 0.838 \times 2403.1$$

$$= 2187.68 \text{ kJ/kg}$$

(a) $W_P = h_{4s} - h_3 = v_{f_{p2}}(p_1 - p_2) = 0.001008 \dfrac{\text{m}^3}{\text{kg}} \times 19.92 \times 100 \dfrac{\text{kN}}{\text{m}^2}$

$$= 2.008 \text{ k/kg}$$

$$h_{4s} = 175.89 \text{ kJ/kg}$$

$$W_T = h_1 - h_{2s}$$

$$= 3159.3 - 2187.68 = 971.62 \text{ kJ/kg}$$

$$W_{net} = W_T - W_P = \mathbf{969.61\ kJ/kg}$$

$$Q_1 = h_1 - h_{4s} = 3159.3 - 175.89$$
$$= 2983.41 \text{ kJ/kg}$$

$\therefore$
$$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{969.61}{2983.41} = \mathbf{0.325, \text{ or } 32.5\%}$$

(b) If
$$\eta_P = 80\%, \quad \text{and} \quad \eta_T = 80\%$$
$$W_P = \frac{2.008}{0.8} = 2.51 \text{ kJ/kg}$$
$$W_T = 0.8 \times 971.62 = 777.3 \text{ kJ/kg}$$

$\therefore$
$$W_{net} = W_T - W_P = 774.8 \text{ kJ/kg}$$

$\therefore$ % Reduction in work output
$$= \frac{969.61 - 774.8}{969.61} \times 100 = \mathbf{20.1\%}$$
$$h_{4s} = 173.88 + 2.51 = 176.39 \text{ kJ/kg}$$

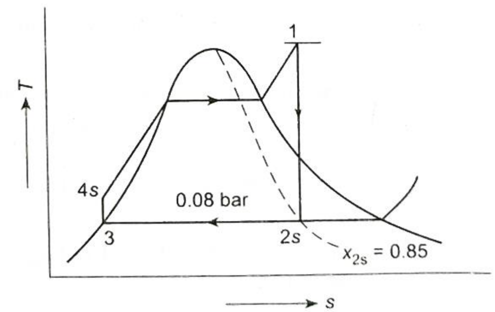
$\therefore$
$$Q_1 = 3159.3 - 176.39 = 2982.91 \text{ kJ/kg}$$

$\therefore$
$$\eta_{cycle} = \frac{774.8}{2982.91} = 0.2597, \text{ or } 25.97\%$$

$\therefore$ % Reduction in cycle efficiency
$$= \frac{0.325 - 0.2597}{0.325} \times 100 = \mathbf{20.1\%}$$

Example *A cyclic steam power plant is to be designed for a steam temperature at turbine inlet of 360°C and an exhaust pressure of 0.08 bar. After isentropic expansion of steam in the turbine, the moisture content at the turbine exhaust is not to exceed 15%. Determine the greatest allowable steam pressure at the turbine inlet, and calculate the Rankine cycle efficiency for these steam conditions. Estimate also the mean temperature of heat addition.*

Solution As state 2s (the quality and pressure are known.



$\therefore$
$$s_{2s} = s_f + x_{2s} s_{fg} = 0.5926 + 0.85\,(8.2287 - 0.5926)$$
$$= 7.0833 \text{ kJ/kg K}$$

Since
$$s_1 = s_{2s}$$

$\therefore$
$$s_1 = 7.0833 \text{ kJ/kg K}$$

At state 1, the temperature and entropy are thus known. At 360°C, $s_g = 5.0526$ kJ/kg K, which is less than s_1. So from the table of superheated steam, at $t_1 = 360°C$ and $s_1 =$ 7.0833 kJ/kg K, the pressure is found to be 16.832 bar (by interpolation).

$\therefore$ The greatest allowable steam pressure is
$$p_1 = \mathbf{16.832 \text{ bar}}$$
$$h_1 = 3165.54 \text{ kJ/kg}$$
$$h_{2s} = 173.88 + 0.85 \times 2403.1 = 2216.52 \text{ kJ/kg}$$
$$h_3 = 173.88 \text{ kJ/kg}$$
$$h_{4s} - h_3 = 0.001 \times (16.83 - 0.08) \times 100 = 1.675 \text{ kJ/kg}$$

$\therefore$
$$h_{4s} = 175.56 \text{ kJ/kg}$$
$$Q_1 = h_1 - h_{4s} = 3165.54 - 175.56$$
$$= 2990 \text{ kJ/kg}$$
$$W_T = h_1 - h_{2s} = 3165.54 - 2216.52 = 949 \text{ kJ/kg}$$
$$W_P = 1.675 \text{ kJ/kg}$$
$$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{947.32}{2990} = \mathbf{0.3168 \text{ or } 31.68\%}$$

Mean temperature of heat addition
$$T_{m1} = \frac{h_1 - h_{4s}}{s_1 - s_{4s}} = \frac{2990}{7.0833 - 0.5926}$$
$$= 460.66 \text{ K} = \mathbf{187.51°C}$$

The thermal efficiency of the cycle $\eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_1} = \frac{}{97891.5} = 0.3184$ or 31.84%

Ⓜ Example 12.8 *In a single-heater regenerative cycle, the steam enters the turbine at 30 bar, 400°C and the exhaust pressure is 0.10 bar. The feedwater heater is a direct-contact type which operates at 5 bar. Find (a) the efficiency and the steam rate of the cycle, and (b) the increase in mean temperature of heat addition, efficiency and steam rate, as compared to the Rankine cycle (without regeneration). Neglect pump work.* **[LO 12.8]**

Solution Figure 12.47 gives the flow, *T-s* and *h-s* diagrams. From the steam tables the property values at various states have been obtained.

$$h_1 = 3230.9 \text{ kJ/kg}$$
$$s_1 = 6.9212 \text{ kJ/kg K} = s_2 = s_3$$
$$s_g \text{ at 5 bar} = 6.8213 \text{ kJ/kg K}$$

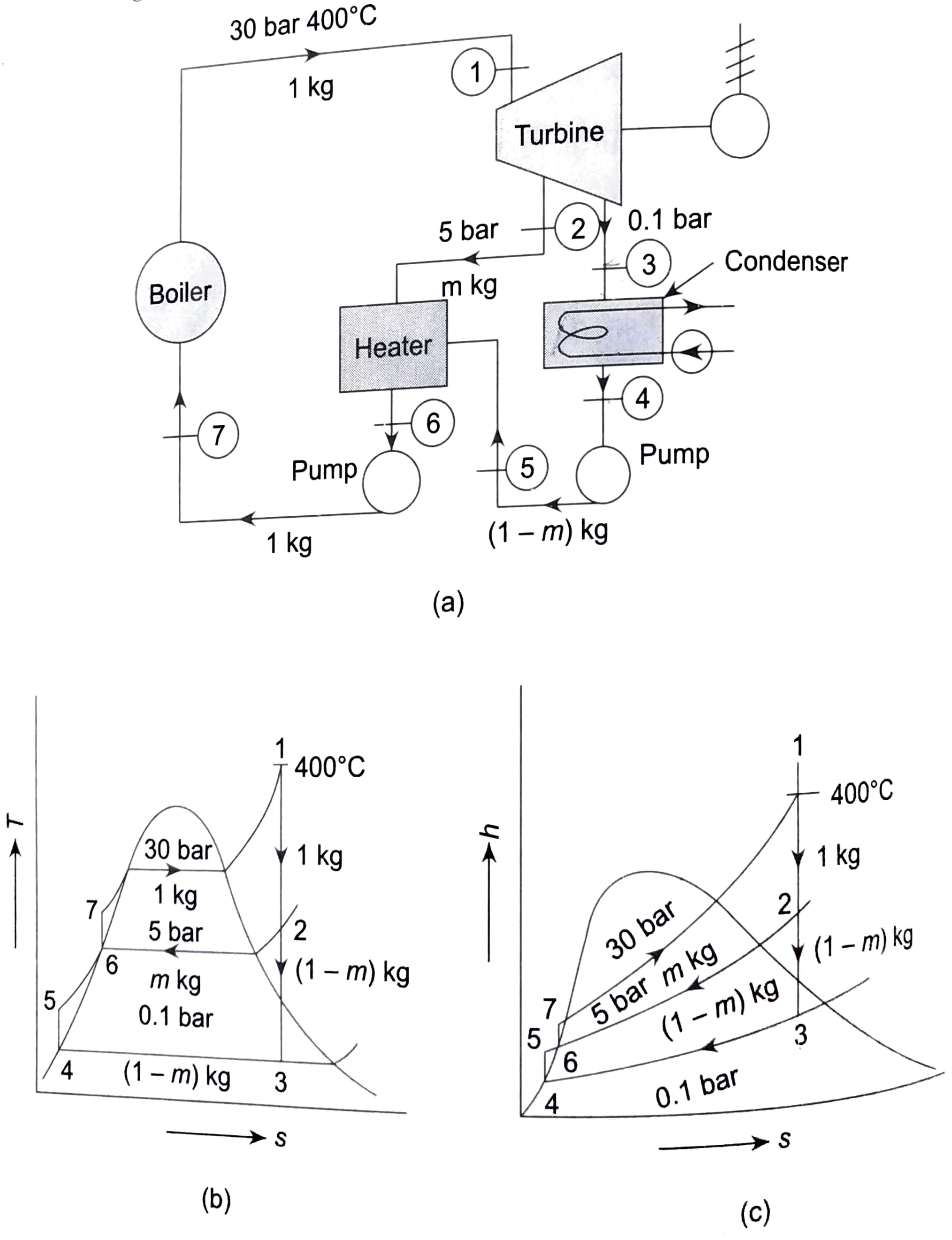


Fig. 12.47

Since $s_2 > s_g$, the state 2 must lie in the superheated region. From the table for superheated steam $t_2 = 172°C$, $h_2 = 2796$ kJ/kg

$$s_3 = 6.9212 = s_{f0.1\text{ bar}} + x_3 s_{fg0.1\text{ bar}}$$

$$= 0.6493 + x_3 7.5009$$

$$x_3 = \frac{6.2719}{7.5009} = 0.836$$

$\therefore$
$$h_3 = 191.83 + 0.836 \times 2392.8 = 2192.2 \text{ kJ/kg}$$

$\therefore$ Since pump work is neglected

$$h_4 = 191.83 \text{ kJ/kg} = h_5$$

$$h_6 = 640.23 \text{ kJ/kg} = h_7$$

Energy balance for the heater gives

$$m(h_2 - h_6) = (1 - m)(h_6 - h_5)$$

$$m(2796 - 640.23) = (1 - m)(640.23 - 191.83)$$

$$2155.77\, m = 548.4 - 548.4\, m$$

$\therefore$
$$m = \frac{548.4}{2704.17} = 0.203 \text{ kg}$$

$\therefore$
$$W_T = (h_1 - h_2) + (1 - m)(h_2 - h_3)$$

$$= (3230.9 - 2796) + 0.797\,(2796 - 2192.2)$$

$$= 916.13 \text{ kJ/kg}$$

$$Q_1 = h_1 - h_6 = 3230.9 - 640.23 = 2590.67 \text{ kJ/kg}$$

$$\eta_{\text{cycle}} = \frac{916.13}{2590.67} = \mathbf{0.3536, \text{ or } 35.36\%}$$

$$\text{Steam rate} = \frac{3600}{916.13} = \mathbf{3.93 \text{ kg/kW h}}$$

$$T_{m1} = \frac{h_1 - h_7}{s_1 - s_7} = \frac{2590.67}{6.9212 - 1.8607} = 511.95 \text{ K} = 238.8°C$$

$$T_{m1} \text{ (without regeneration)} = \frac{h_1 - h_4}{s_1 - s_4}$$

$$= \frac{3039.07}{6.9212 - 0.6493}$$

$$= 484.55 \text{ K}$$

$$= 211.4°C$$

Increase in T_{m1} due to regeneration

$$= 238.8 - 211.4 = \mathbf{27.4°C}$$

$$W_T \text{ (without regeneration)} = h_1 - h_3$$

$$= 3230.9 - 2192.2 = 1038.7 \text{ kJ/kg}$$

$$\text{Steam rate (without regeneration)} = \frac{3600}{1038.7} = 3.46 \text{ kg/kW h}$$

$\therefore$ Increase in steam rate due to regeneration

$$= 3.93 - 3.46 = \mathbf{0.476 \text{ kg/kW h}}$$

$$\eta_{cycle} \text{ (without regeneration)} = \frac{h_1 - h_3}{h_1 - h_4} = \frac{1038.7}{3039.07}$$

$$= 0.3418 \text{ or } 34.18\%$$

$\therefore$ Increase in cycle efficiency due to regeneration

$$= 35.36 - 34.18 = \mathbf{1.18\%}$$

In a steam power plant the condition of steam at inl

12.8 REGENERATIVE CYCLE

LO 12.8 Discuss the practical Regenerative cycle by employing the limitations of ideal cycle

In the practical regenerative cycle, the feedwater enters the boiler at a temperature between 4 and 4′ (Fig. 12.17), and it is heated by steam extracted from intermediate stages of the turbine. The flow diagram of the regenerative cycle with saturated steam at the inlet to the turbine, and the corresponding *T-s* diagram are shown in Figs. 12.18 and 12.19, respectively. For every kg of steam entering the turbine, let m_1 kg steam be extracted from an intermediate stage of the turbine where the pressure is p_2, and it is used to heat up feedwater [$(1 - m_1)$ kg at state 8] by mixing in heater 1. The remaining $(1 - m_1)$ kg of steam then expands in the turbine from pressure p_2 (state 2) to pressure p_3 (state 3) when m_2 kg of steam is extracted for heating feed water in heater 2. So $(1 - m_1 - m_2)$ kg of steam then expands in the remaining stages of the turbine to pressure p_4, gets condensed into water in the condenser and then pumped to heater 2, where it mixes with m_2 kg of steam extracted at pressure p_3. Then $(1 - m_1)$ kg of water is pumped to heater 1 where it mixes with m_1 kg of steam extracted at pressure p_2. The resulting 1 kg of steam is then pumped to the boiler where heat from an external source is supplied. Heaters 1 and 2 thus operate at pressures p_2 and p_3, respectively. The amounts of steam m_1 and m_2 extracted from the turbine are such that at the exit from each of the heaters, the state is saturated liquid at the respective pressures. The heat and work transfer quantities of the cycle are as follows:

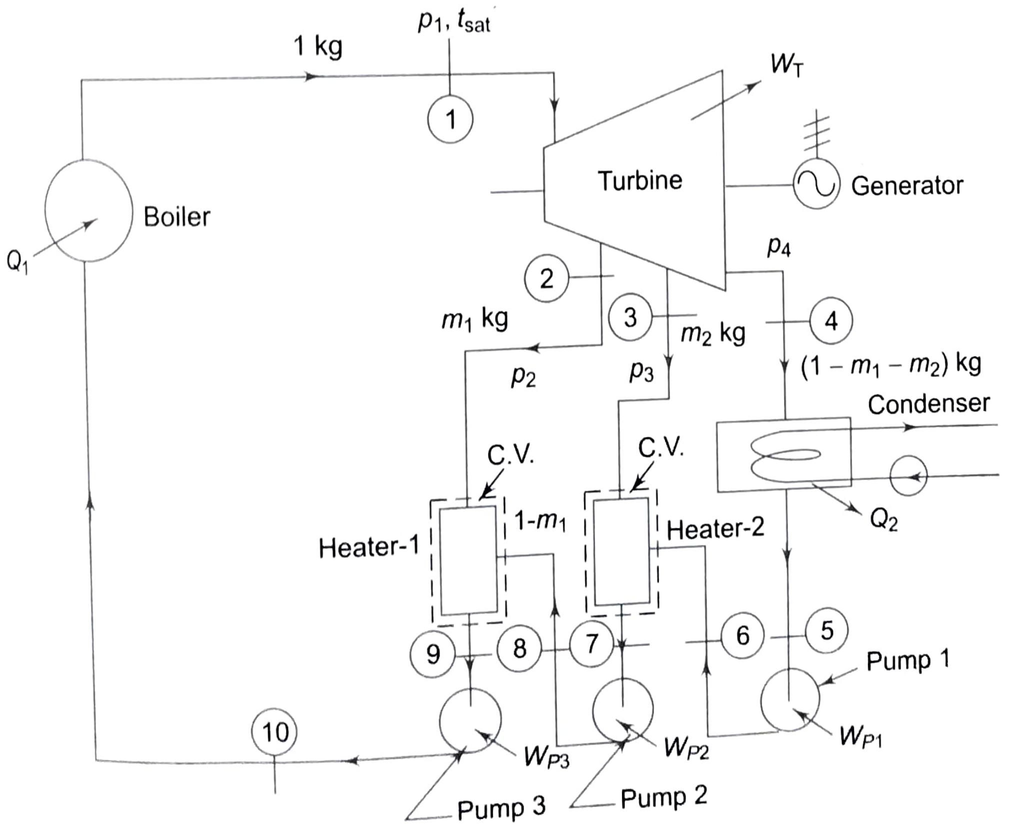


Fig. 12.18 *Regenerative cycle flow diagram with two feedwater heaters*

$$W_T = 1\,(h_1 - h_2) + (1 - m_1)\,(h_2 - h_3) + (1 - m_1 - m_2)\,(h_3 - h_4) \text{ kJ/kg} \quad (12.18)$$

$$W_P = W_{P1} + W_{P2} + W_{P3}$$
$$= (1 - m_1 - m_2)\,(h_6 - h_5) + (1 - m_1)\,(h_8 - h_7) + 1\,(h_{10} - h_9) \text{ kJ/kg} \quad (12.19)$$

$$Q_1 = 1\,(h_1 - h_{10}) \text{ kJ/kg} \quad (12.20)$$

$$Q_2 = (1 - m_1 - m_2)\,(h_4 - h_5) \text{ kJ/kg} \quad (12.21)$$

$$\text{Cycle efficiency, } \eta = \frac{Q_1 - Q_2}{Q_1} = \frac{W_T - W_P}{Q_1}$$

$$\text{Steam rate} = \frac{3600}{W_T - W_P} \text{ kg/kW h}$$

In the Rankine cycle operating at the given pressures, p_1 and p_4, the heat addition would have been from state 6 to state 1. By using two stages of regenerative feedwater heating, feedwater enters the boiler at state 10, instead of state 6, and heat addition is, therefore, from state 10 to state 1. Therefore,

$$(T_{m1})_{\text{with regeneration}} = \frac{h_1 - h_{10}}{s_1 - s_{10}} \quad (12.22)$$

and

$$(T_{m1})_{\text{without regeneration}} = \frac{h_1 - h_6}{s_1 - s_6} \quad (12.23)$$

Since

$$(T_{m1})_{\text{with regeneration}} > (T_{m1})_{\text{without regeneration}}$$

the efficiency of the regenerative cycle will be higher than that of the Rankine cycle.

The energy balance for heater 2 gives

$$m_1 h_2 + (1 - m_1)\,h_8 = 1 h_9$$

$$\therefore \quad m_1 = \frac{h_9 - h_8}{h_2 - h_8} \quad (12.24)$$

The energy balance for heater 1 gives

$$m_2 h_3 + (1 - m_1 - m_2)\,h_6 = (1 - m_1)\,h_7$$

or

$$m_2 = (1 - m_1)\,\frac{h_7 - h_6}{h_3 - h_6} \quad (12.25)$$

From Eqs. (12.24) and (12.25), m_1 and m_2 can be evaluated. Equations (12.24) and (12.25) can also be written alternatively as

$$(1 - m_1)\,(h_9 - h_8) = m_1\,(h_2 - h_9)$$

$$(1 - m_1 - m_2)\,(h_7 - h_6) = m_2\,(h_3 - h_7)$$

Energy gain of feedwater = Energy given off by vapour in condensation

Heaters have been assumed to be adequately insulated, and there is no heat gain from, or heat loss to, the surroundings.

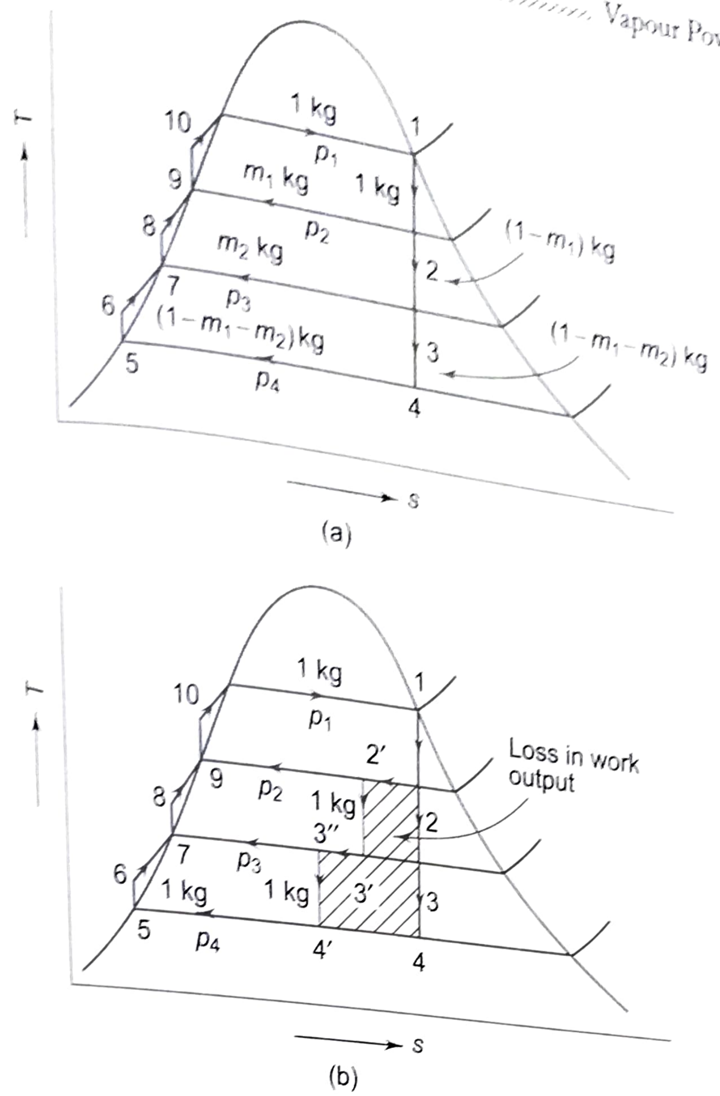


Fig. 12.19 *Regenerative cycle on T-s plot with decreasing mass of fluid (b) Regenerative cycle on T-s plot for unit mass of fluid*

Path 1–2–3–4 in Fig. 12.19 represents the states of a decreasing mass of fluid. For 1 kg of steam, the states would be represented by the path 1–2′–3′–4′. From Eq. (12.18),

$$W_T = (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$$
$$= (h_1 - h_2) + (h_{2'} - h_{3'}) + (h_{3'} - h_{4'}) \quad (12.26)$$

where

$$(1 - m_1)(h_2 - h_3) = 1\,(h_{2'} - h_{3'}) \quad (12.27)$$
$$(1 - m_1 - m_2)(h_3 - h_4) = 1\,(h_{3''} - h_{4'}) \quad (12.28)$$

The cycle 1–2–2′–3′–3″–4′–5–6–7–8–9–10–1 represents 1 kg of working fluid. The heat released by steam condensing from 2 to 2′ is utilised in heating up the water from 8 to 9.

$$1(h_2 - h_{2'}) = (h_9 - h_8) \quad (12.29)$$

Similarly,

$$1(h_{3'} - h_{3''}) = 1(h_7 - h_6) \quad (12.30)$$

From Eqs. (12.26), (12.29) and (12.30)

$$W_T = (h_1 - h_{4'}) - (h_2 - h_{2'}) - (h_{3'} - h_{3''})$$
$$= (h_1 - h_{4'}) - (h_9 - h_8) - (h_7 - h_6) \quad (12.31)$$

The similarity of Eqs. (12.17) and (12.31) can be noticed. It is seen that the stepped cycle 1–2′–3′–4′–5–6–7–8–9–10 approximates the ideal regenerative cycle in Fig. 12.17, and that a greater number of stages would give a closer approximation (Fig. 12.20). Thus the heating of feedwater by steam 'bled' from the turbine, known as regeneration, *carnotises* the Rankine cycle.

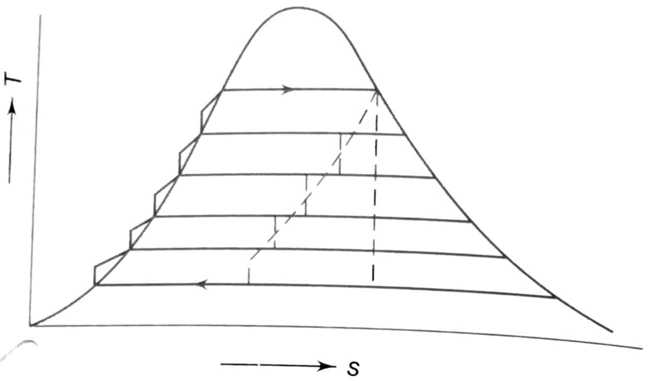


Fig. 12.20 *Regenerative cycle with many stages of feedwater heating*

The heat rejected Q_2 in the cycle decreases from $(h_4 - h_5)$ to $(h_{4'} - h_5)$. There is also loss in work output by the amount (Area under 2–2′ + Area under 3′–3″ – Area under 4–4′), as shown by the hatched area in Fig. 12.19 (b). So the steam rate increases by regeneration, i.e. more steam has to circulate per hour to produce unit shaft output.

The enthalpy-entropy diagram of a regenerative cycle is shown in Fig. 12.21.

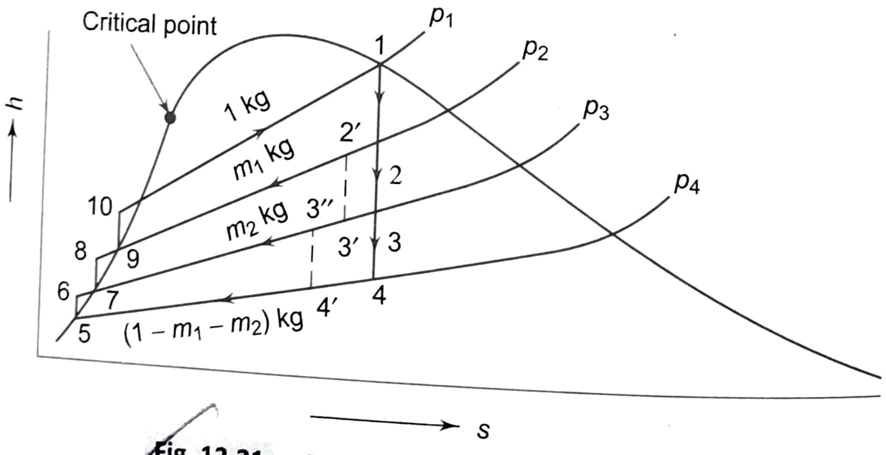


Fig. 12.21 *Regenerative cycle on h-s diagram*

12.9 REHEAT-REGENERATIVE CYCLE

The reheating of steam is adopted when the vaporisation pressure is high. The effect of reheat alone on the thermal efficiency of the cycle is very small. Regeneration or the heating up of feedwater by steam extracted from the turbine has a marked effect on cycle efficiency. A modern steam power plant is equipped with both. Figures 12.22 and 12.23 give the

LO 12.9

Combine the Reheat-Regenerative cycle for optimizing performance of modern steam power plant

...modern power plants is ...
... 12.14, for 1 kg of steam

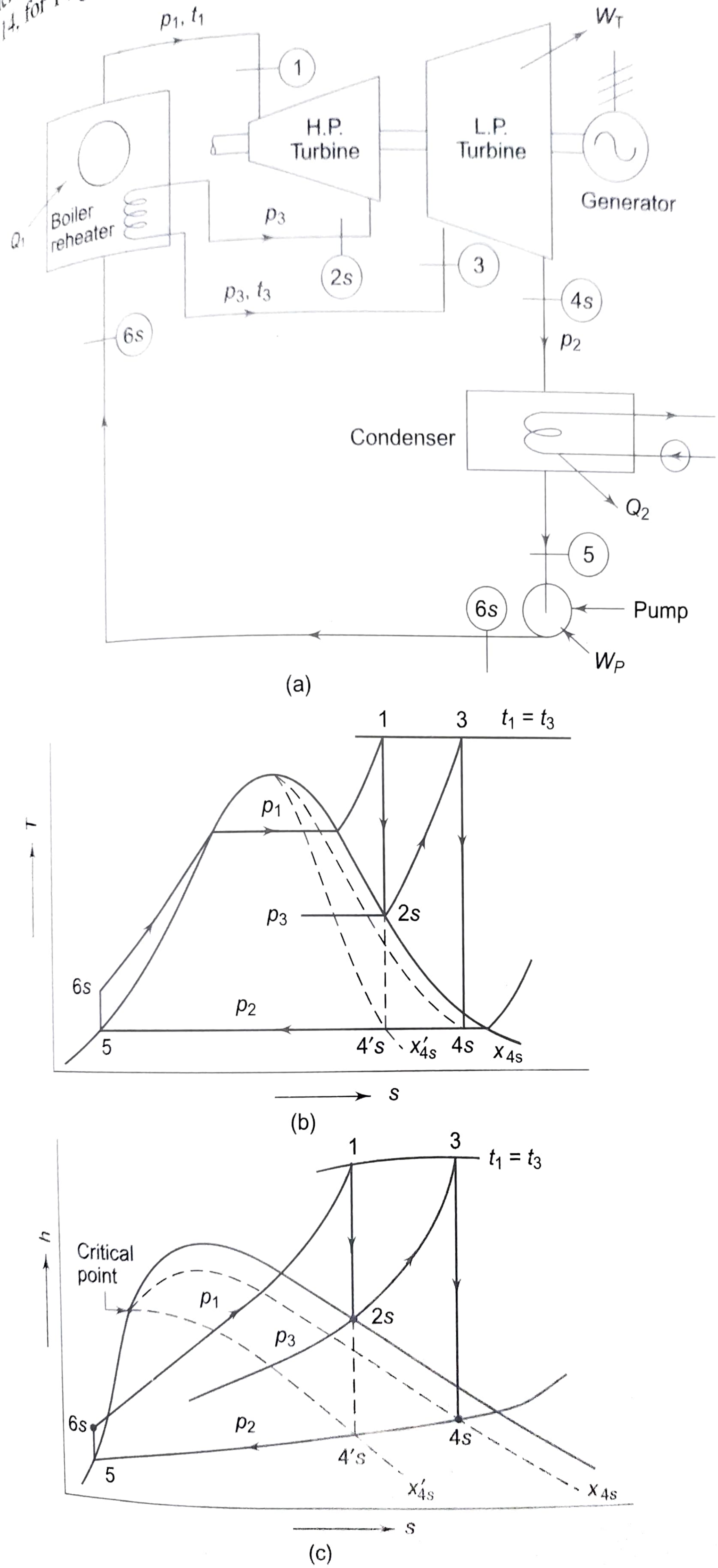


Fig. 12.14 *Reheat cycle*

$$Q_1 = h_1 - h_{6s} + h_3 - h_{2s}$$
$$Q_2 = h_{4s} - h_5$$
$$W_T = h_1 - h_{2s} + h_3 - h_{4s}$$
$$W_P = h_{6s} - h_5$$

$$\eta = \frac{W_T - W_P}{Q_1} = \frac{(h_1 - h_{2s} + h_3 - h_{4s}) - (h_{6s} - h_5)}{h_1 - h_{6s} + h_3 - h_{2s}} \tag{12.15}$$

$$\text{Steam rate} = \frac{3600}{(h_1 - h_{2s} + h_3 - h_{4s}) - (h_{6s} - h_5)} \text{ kg/kWh} \tag{12.16}$$

where enthalpy is in kJ/kg.

Since higher pressures are used in a reheat cycle, pump work may be appreciable. Had the high pressure p_1 been used without reheat, the ideal Rankine cycle wo

$h_2 = 2319.82$

$x_4 = 0.89$

Therefore, the maximum allowable pressure at the turbine inlet is 8 MPa.

Example 12.5 *A steam power station uses the following cycle:*

Steam at boiler outlet-150 bar, 550°C

Reheat at 40 bar to 550°C

Condenser at 0.1 bar.

Using the Mollier chart and assuming ideal processes, find the (a) quality at turbine exhaust, (b) cycle efficiency, and (c) steam rate. [LO 12.6]

Solution The property values at different states (Fig. 12.44) are read from the Mollier chart.

$$h_1 = 3465,\ h_{2s} = 3065,\ h_3 = 3565,$$

$$h_{4s} = 2300 \text{ kJ/kg},\ x_{4s} = 0.88,\ h_5 \text{ (steam table)} = 191.83 \text{ kJ/kg}$$

Quality at turbine exhaust = **0.88**

$$W_P = v\,\Delta p = 10^{-3} \times 150 \times 10^2 = 15 \text{ kJ/kg}$$

$$h_{6s} = 206.83 \text{ kJ/kg}$$

$$Q_1 = (h_1 - h_{6s}) + (h_3 - h_{2s})$$

$$= (3465 - 206.83) + (3565 - 3065) = 3758.17 \text{ kJ/kg}$$

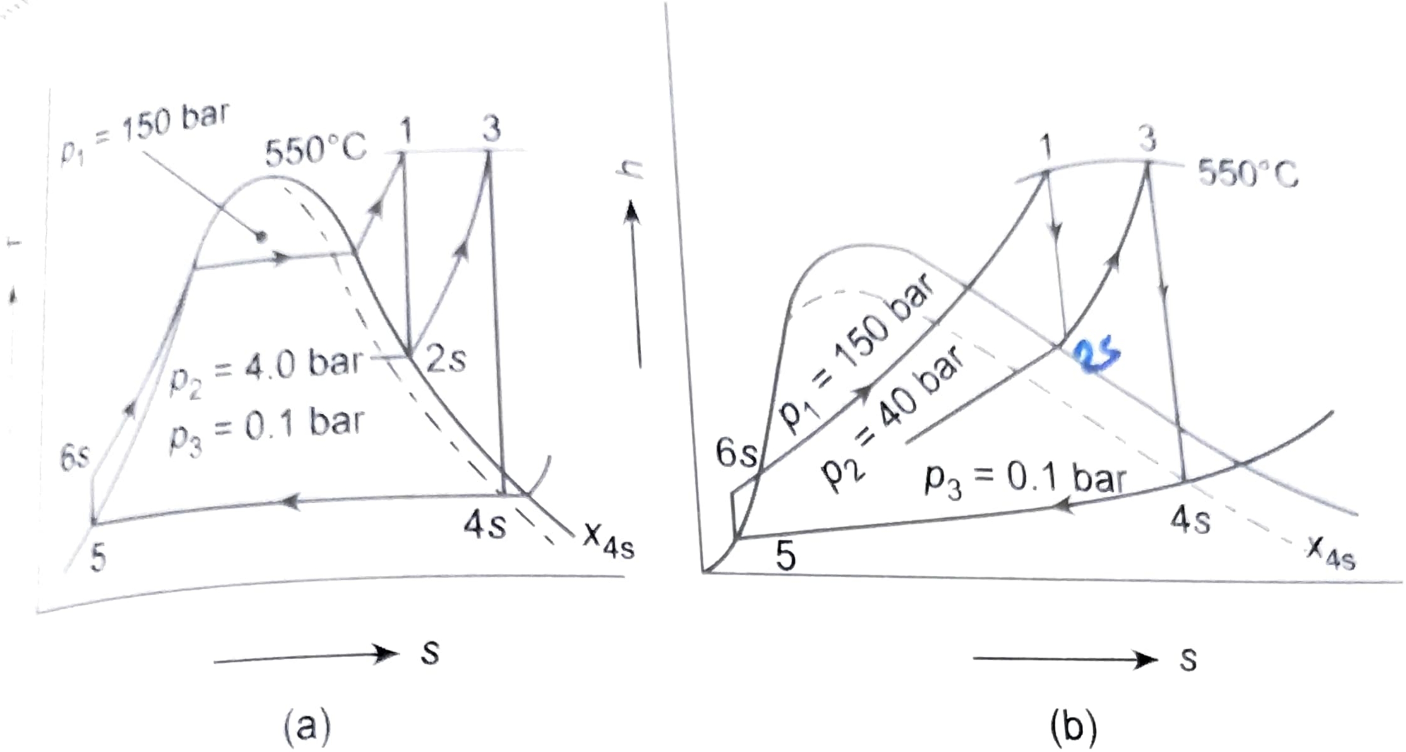


Fig. 12.44

$$W_T = (h_1 - h_{2s}) + (h_3 - h_{4s})$$

$$= (3465 - 3065) + (3565 - 2300) = 1665 \text{ kJ/kg}$$

$$W_{\text{net}} = W_T - W_P = 1665 - 15 = 1650 \text{ kJ/kg}$$

$$\eta_{\text{cycle}} = \frac{W_{\text{net}}}{Q_1} = \frac{1650}{3758.17} = \mathbf{0.4390, \text{ or } 43.9\%}$$

$$\text{Steam rate} = \frac{3600}{1650} = \mathbf{2.18 \text{ kg/kW h}}$$

$$\text{Steam Rate} = \frac{1}{W_T - W_p} \frac{kg}{kJ} \cdot \frac{1 \, kJ/s}{1 \, kW}$$

$$= \frac{1}{W_T - W_p} \frac{kg}{kWs}$$

$$= \frac{3600}{W_T - W_p} \frac{kg}{kWh}$$

Rate of steam flow (kg/h) required to produce unit shaft output (1 kW).

cycle efficiency is sometimes expressed ~~as~~ alternatively as heat rate which is the rate of heat output (Q_1) required to produce unit work output (1 kW)

$$\text{Heat Rate} = \frac{3600\, Q_1}{W_T - W_P} = \frac{3600}{\eta_{cycle}} \frac{kJ}{kWh}$$

Fig. 12.21 *Regenerative cycle on h-s diagram*

12.9 REHEAT-REGENERATIVE CYCLE

The reheating of steam is adopted when the vaporisation pressure is high. The effect of reheat alone on the thermal efficiency of the cycle is very small. Regeneration or the heating up of feedwater by steam extracted from the turbine has a marked effect on cycle efficiency. A modern steam power plant is equipped with both. Figures 12.22 and 12.23 give the

...flow and T-s diagrams of a steam plant with reheat and three stages of feedwater heating. Here

$$W_T = (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1)(h_4 - h_5) + (1 - m_1 - m_2)(h_5 - h_6) + (1 - m_1 - m_2 - m_3)(h_6 - h_7) \text{ kJ/kg}$$

$$W_P = (1 - m_1 - m_2 - m_3)(h_9 - h_8) + (1 - m_1 - m_2)(h_{11} - h_{10}) + (1 - m_1)(h_{13} - h_{12}) + 1(h_{15} - h_{14}) \text{ kJ/kg}$$

$$Q_1 = (h_1 - h_{15}) + (1 - m_1)(h_4 - h_3) \text{ kJ/kg}$$

and

$$Q_2 = (1 - m_1 - m_2 - m_3)(h_7 - h_8) \text{ kJ/kg}$$

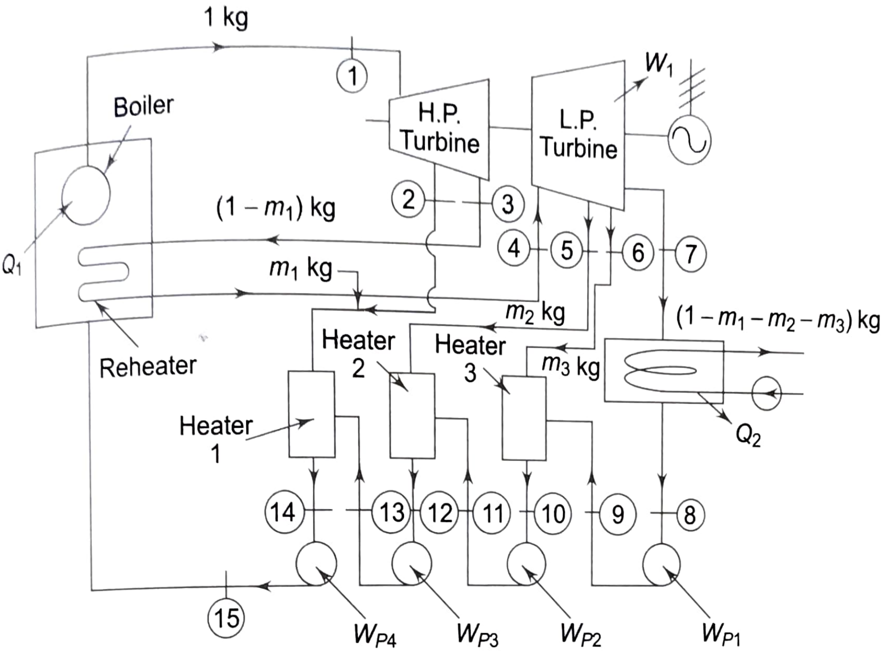


Fig. 12.22 *Reheat-regenerative cycle flow diagram*

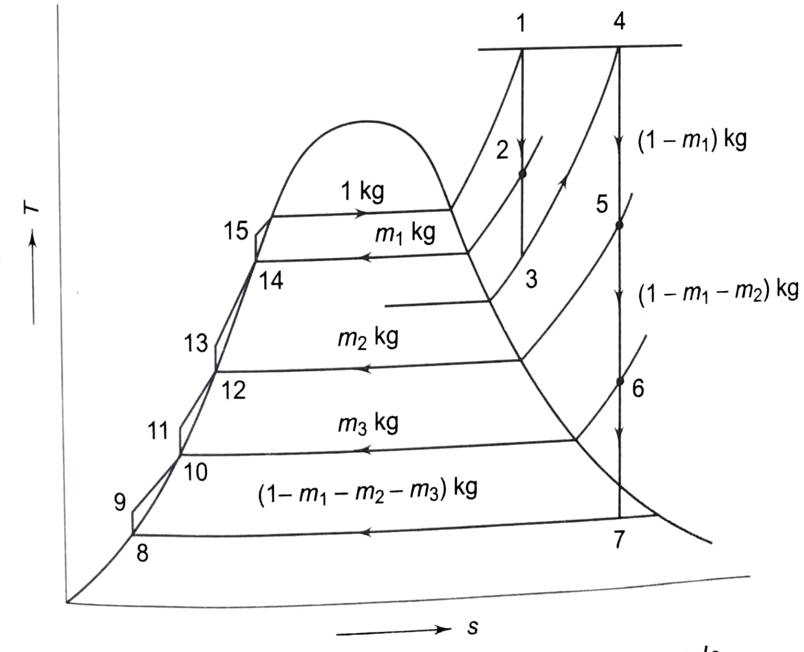


Fig. 12.23 *T-s diagram of reheat-regenerative cycle*

The energy balance of heaters 1, 2 and 3 give

$$m_1 h_2 + (1 - m_1) h_{13} = 1 \times h_{14}$$

$$m_2 h_5 + (1 - m_1 - m_2) h_{11} = (1 - m_1) h_{12}$$

$$m_3 h_6 + (1 - m_1 - m_2 - m_3) h_9 = (1 - m_1 - m_2) h_{10}$$

from which m_1, m_2 and m_3 can be evaluated.

[12.1] Determined the work required to compress steam isentropically from 1 bar to 10 bar, when steam exit as (a) Saturated liquid
(b) Saturated vapor.

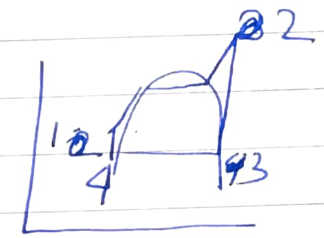
(A) Saturated Liquid State

$v_1 = v_{f_{1 bar}} = 0.001043 \ m^3/kg$

Since liquid is incompressible; v_1 remain constant.

$W_{rev} = \int_4^2 v \cdot dP = v_1 (P_4 - P_1)$

$= -0.9387 \ kJ/kg$



[B] Saturated Vapour

$W = h_4 - h_1$

$=$

where $h_4 = h_g)_{1 bar} = 2675.5 \ kJ/kg$

$h_1 = h_g)_{10 bar} = 3195.5 \ kJ/kg$

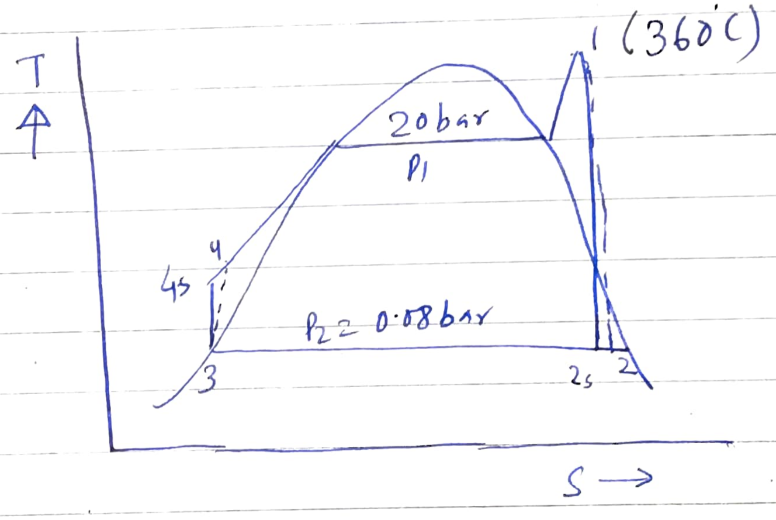
$W_{rev} = 2675.5 - 3195.5$

$= -520 \ kJ/kg$

(12.2) Steam at 20 bar, 360°C is expanded in steam turbine to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler.

(a) Assume ideal processes find per kg of steam of the net work and the cycle efficiency.

(b) If the turbine and pump efficiency have each 80%, find the percentage reduction in the net work and cycle efficiency.



Properties of different state

$h_1 = 3159.3$ kJ/kg $s_1 = 6.9917$ kJ/kgK

$h_3 = h_{fP2} = 173.88$ kJ/kg ; $s_3 = s_{fg2} = 0.5926$ kJ/kgK

$h_{fgP2} = 2403.1$ kJ/kg ; $s_{gP2} = 8.2287$ kJ/kgK

$v_{fP_2} = 0.001008\ m^3/kg$; $s_{fgP_2} = 7.6361\ kJ/kgK$.

$s_1 = s_{2s} = 6.9917$

$= s_{fP_2} + x_{2s}\, s_{fgP_2}$

$x = 0.838$

$h_{2s} = 2187.68\ kJ/kg$

(a) $W_p = h_{4s} - h_3 = v_{fP_2}(P_1 - P_2)$

$= 0.001008\ \frac{m^3}{kg} \times 19.92\ \frac{kN}{m^2}$

$W_p = 2.008\ k/kg$

$h_{4s} = 175.89\ kJ/kg$

$W_T = h_1 - h_{2s} = 3159.3 - 2187.68$

$= 971.62\ kJ/kg$

$W_{net} = W_T - W_p = 971.62 - 2.008$

$= 969.61\ kJ/kg$.

$Q_1 = h_1 - h_{4s} = 3159.3 - 175.89$

$= 2983.41\ kJ/kg$.

$\eta = \frac{W_{net}}{Q_1} = \frac{969.61}{2983.41} = 0.325 \approx 32.5\%$

(b) $\eta_P = 80\%$; $\eta_T = 80\%$

$$\eta_P = \frac{h_{4s} - h_3}{h_4 - h_3} = \frac{h_{4s} - h_3}{W_P}$$

$$W_P = \frac{2.008}{0.8} = 2.51 \text{ KJ/Kg}$$

$$\eta_T = 80\% = \frac{W_T}{h_1 - h_{2s}}$$

$$W_T = 0.8 \times (971.62)$$

$$W_T = 777.3 \text{ KJ/Kg}$$

$$W_{net} = W_T - W_P = 774.8 \text{ KJ/Kg}$$

% reduction in work output $= \frac{969.61 - 774.8}{969.61}$

$= 20.1\%$

$$h_{4s} = 173.88 + 2.51 = h_3 + W_P$$

$$= 173.88 + 2.51$$

$$h_{4s} = 176.39 \text{ KJ/Kg}$$

$Q_1 = 3159.3 - 176.39 = 2982.91$ KJ/Kg

$\eta = \frac{W_{net}}{Q_1} = \frac{774.8}{2982.91} = 25.97\%$

10

11

12 Reduction in efficiency $= \frac{0.325 - 0.2597}{0.325}$

1 $= 20.1\%$

2

...liquid form for the same pressure rise.

Ⓔ Example 12.2 *Steam at 20 bar, 360°C is expanded in a steam turbine to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. (a) Assuming ideal processes, find per kg of steams of the net work and the cycle efficiency. (b) If the turbine and the pump have each 80% efficiency, find the percentage reduction in the net work and cycle efficiency.*

[LO 12.2]

Solution The property values at different state points (Fig. 12.41) found from the steam tables are given below.

$$h_1 = 3159.3 \text{ kJ/kg} \qquad s_1 = 6.9917 \text{ kJ/kg K}$$

$$h_3 = h_{fp2} = 173.88 \text{ kJ/kg} \qquad s_3 = s_{fp2} = 0.5926 \text{ kJ/kg K}$$

$$h_{fgp2} = 2403.1 \text{ kJ/kg} \qquad s_{gp2} = 8.2287 \text{ kJ/kg K}$$

$$v_{fp2} = 0.001008 \text{ m}^3\text{/kg} \qquad \therefore \quad s_{fgp2} = 7.6361 \text{ kJ/kg K}$$

Now $\quad s_1 = s_{2s} = 6.9917 = s_{fp2} + x_{2s}\, s_{fgp2} = 0.5926 + x_2 \cdot 7.6361$

$$\therefore \quad x_{2s} = \frac{6.3991}{7.6361} = 0.838$$

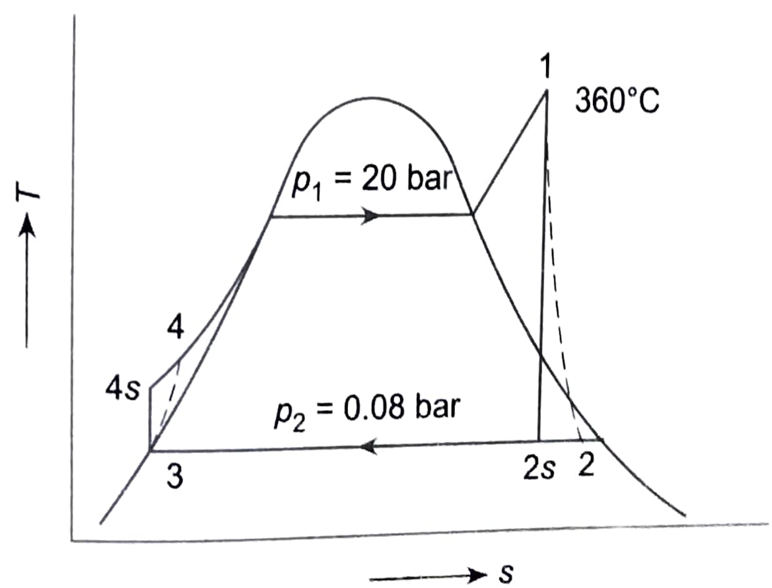


Fig. 12.41

$$\therefore \quad h_{2s} = h_{fp2} + x_{2s} h_{fgp2} = 173.88 + 0.838 \times 2403.1 = 2187.68 \text{ kJ/kg}$$

(a) $W_p = h_{4s} - h_3 = v_{fp2}(p_1 - p_2) = 0.001008 \dfrac{\text{m}^3}{\text{kg}} \times 19.92 \times 100 \dfrac{\text{kN}}{\text{m}^2}$

$= 2.008$ k/kg

$$h_{4s} = 175.89 \text{ kJ/kg}$$

$$W_T = h_1 - h_{2s}$$

$$= 3159.3 - 2187.68 = 971.62 \text{ kJ/kg}$$

$$\therefore \quad W_{net} = W_T - W_P = \mathbf{969.61\ kJ/kg}$$

$$Q_1 = h_1 - h_{4s} = 3159.3 - 175.89$$

$$= 2983.41 \text{ kJ/kg}$$

$$\therefore \quad \eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{969.61}{2983.41} = \mathbf{0.325,\ or\ 32.5\%}$$

(b) If $\eta_P = 80\%$, and $\eta_T = 80\%$

$$W_P = \frac{2.008}{0.8} = 2.51 \text{ kJ/kg}$$

$$W_T = 0.8 \times 971.62 = 777.3 \text{ kJ/kg}$$

$$\therefore \quad W_{net} = W_T - W_P = 774.8 \text{ kJ/kg}$$

∴ % reduction in work output

$$= \frac{969.61 - 774.8}{969.61} \times 100 = \mathbf{20.1\%}$$

$$h_{4s} = 173.88 + 2.51 = 176.39 \text{ kJ/kg}$$

$$\therefore \quad Q_1 = 3159.3 - 176.39 = 2982.91 \text{ kJ/kg}$$

$$\therefore \quad \eta_{cycle} = \frac{774.8}{2982.91} = 0.2597, \text{ or } 25.97\%$$

∴ % reduction in cycle efficiency

$$= \frac{0.325 - 0.2597}{0.325} \times 100 = \mathbf{20.1\%}$$

Ⓜ **Example 12.3** *A cyclic steam power plant is to be designed for a steam temperature at turbine inlet of 360°C and an exhaust pressure of 0.08 bar. After isentropic expansion of steam in the turbine, the moisture content at the turbine exhaust is not to exceed 15%. Determine the greatest allowable steam pressure at the turbine inlet, and calculate the Rankine cycle efficiency for these steam conditions. Estimate also the mean temperature of heat addition.* [LO 12.2]

Solution As for state 2s (Fig. 12.42), the quality and pressure are known.

$$\therefore \quad s_{2s} = s_f + x_{2s} s_{fg} = 0.5926 + 0.85\,(8.2287 - 0.5926)$$

$$= 7.0833 \text{ kJ/kg K}$$

Since $s_1 = s_{2s}$

$$\therefore \quad s_1 = 7.0833 \text{ kJ/kg K}$$

At state 1, the temperature and entropy are thus known. At 360°C, $s_g = 5.0526$ kJ/kg K, which is less than s_1. So from the table of superheated steam, at $t_1 = 360$°C and $s_1 = 7.0833$ kJ/kg K, the pressure is found to be 16.832 bar (by interpolation).

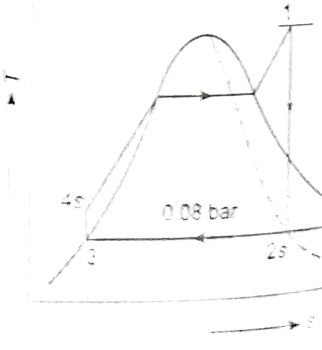


Fig. 12.42

The greatest allowable steam pressure is

$$\therefore \quad p_1 = \mathbf{16.832\ bar}$$

$$h_1 = 3165.54 \text{ kJ/kg}$$

$$h_{2s} = 173.88 + 0.85 \times 2403.1 = 2216.52 \text{ kJ/kg}$$

$$h_3 = 173.88 \text{ kJ/kg}$$

$$h_{4s} - h_3 = 0.001 \times (16.83 - 0.08) \times 100 = 1.675 \text{ kJ/kg}$$

$$\therefore \quad h_{4s} = 175.56 \text{ kJ/kg}$$

$$Q_1 = h_1 - h_{4s} = 3165.54 - 175.56$$

$$= 2990 \text{ kJ/kg}$$

$$W_T = h_1 - h_{2s} = 3165.54 - 2216.52 = 949 \text{ kJ/kg}$$

$$W_p = 1.675 \text{ kJ/kg}$$

$$\eta_{\text{cycle}} = \frac{W_{\text{net}}}{Q_1} = \frac{947.32}{2990} = \mathbf{0.3168 \text{ or } 31.68\%}$$

Mean temperature of heat addition

$$T_{m1} = \frac{h_1 - h_{4s}}{s_1 - s_{4s}} = \frac{2990}{7.0833 - 0.5926}$$

$$= 460.66 \text{ K} = \mathbf{187.51°C}$$

Ⓜ **Example 12.4** *During an job trai...*