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Materials:

* water

Sources:

Çengel & Boles Thermodynamics: An engineering approach 8th Ed. McGraw Hill. 2015

Purpose:

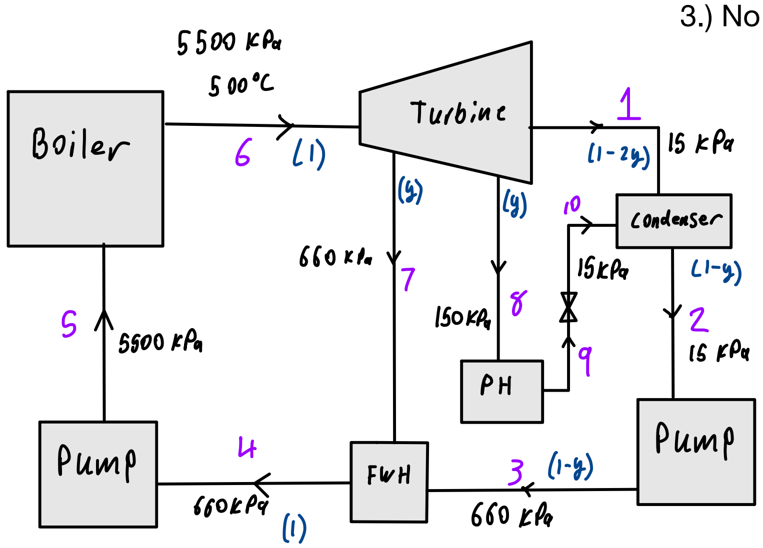
Evaluate the regenerative steam power cycle and its various components, such as the turbine, condenser, and feedwater heating. Perform a comprehensive analysis of the given steam power plant system, including determining the utilization factor, constructing P-v and T-s diagrams, calculating the turbine work, mass flow rates, heat transfer rates, and identifying the thermodynamic states at various points in the cycle. Additionally, determine the mass fractions of steam extracted for feedwater heating and space heating.

Design Considerations:

1.) The water is pure
2.) The pumps and turbine are not isentropic
3.) No losses due to heat or friction

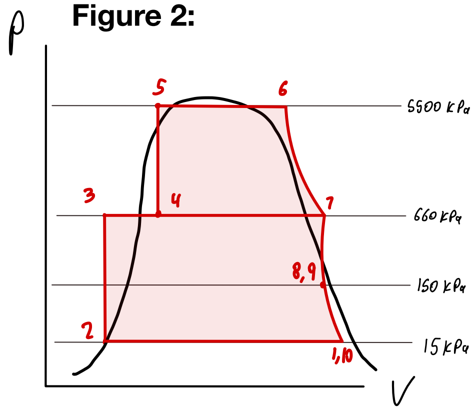
Drawings:

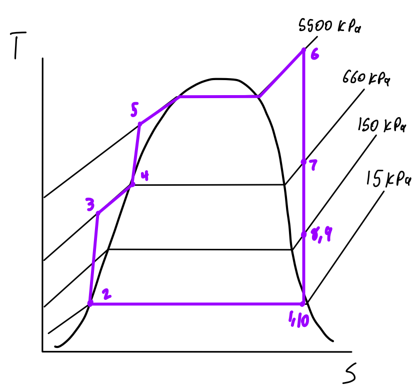
Figure 1:



B)

Figure 2:

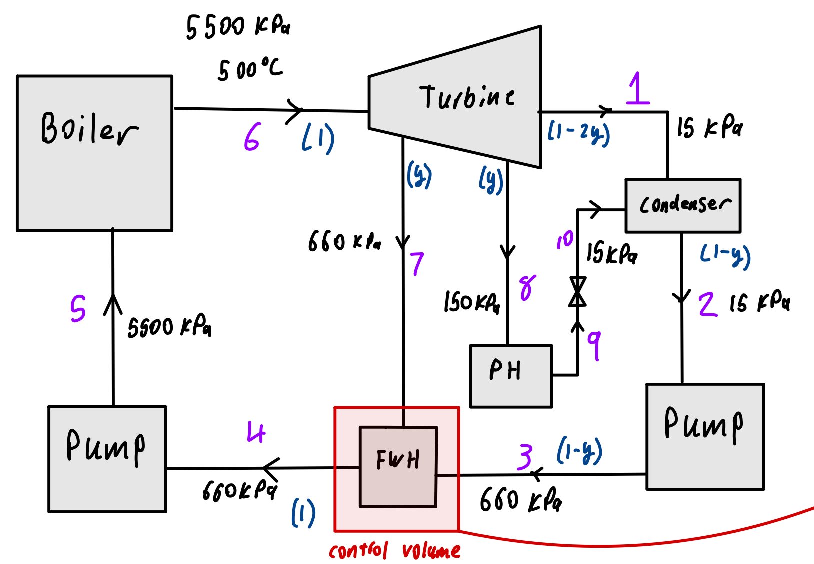




A manufacturing facility uses a steam power plant to generate electricity and provide heating steam to the work spaces. The steam plant operates on the regenerative cycle, with one stage of feed heating. Steam enters the turbine at 5.5 MPa and 500 °C, expands to 660 kPa, where extraction of y 1 steam fraction for feed-water heating occurs. The remaining steam expands through the turbine to 150 kPa,where extraction of y2 steam fraction for space heating occurs. The system is designed in such a way that y2 =y1 . The remaining steam expands to 15 kPa, where it is condensed. The fluid returns from the space heating as a saturated liquid, gets throttled to condenser pressure to then enter the condenser. Determine:

(a) the utilization factor;
(b) P-v and T-s diagrams;
(c) the turbine work per kg;
(d) for a 50,000 kW load, the mass flow steam entering the turbine;
(e) the heat supplied to the working areas for the conditions in (d);
(f) all the states in the cycle;
(g) mass fractions y2 and y 1 ;
(h) the rate of heat removed in the condenser.

Procedure and calculations:



G)

$$\dot{m}_{(1)}(h_4) = \dot{m}_{(1-y)}(h_3) + \dot{m}_{(y)}(h_7)$$

$$h_4 = (1-y)h_3 + (y)h_7$$

$$656.66 = (1-y)226.6 + (y)2847.36$$

$$y = 0.175548$$

$$W_{p\,in} = (1-2y)W_{p_1} + W_{p_2}$$

$$W_{p_1} = h_3 - h_2$$

$$W_{p_1} = V_2(P_3 - P_2)$$

$$h_3 = v_2(P_3 - P_2) + h_2$$

$$W_{p_2} = h_5 - h_4$$

$$W_{p_2} = V_4(P_5 - P_4)$$

$$h_5 = v_4(P_5 - P_4) + h_4$$

F)

Isenthalpic   Isenthalpic   Isentropic

1	2 Pump →	3	4 Pump →	5	6	7	8	9	10
$P_1 = 16kPa$	$P_2 = 15kPa$	$P_3 = 660 kPa$	$P_4 = 660 kPa$	$P_5 = 5500 kPa$	$P_6 = 5500 kPa$	$P_7 = 660 kPa$	$P_8 = 150 kPa$	$P_9 = 150 kPa$	$P_{10} = 15 kPa$
$h_1 = 2246$	$h_2 = 225.94$	$h_3 = 226.6$	$h_4 = 686.7$	$h_5 = 692.05$	$T_6 = 500°C$	$h_7 = 2847.3$	$h_8 = 2580.57$	$h_9 = 467.13$	$h_{10} = 225.94$
	$v_2 = 0.001014$		$v_4 = 0.0011048$		$h_6 = 3428.9$	$s_7 = 6.93035$	$s_8 = 6.93035$		
					$s_6 = 6.93035$				
	Saturated		Saturated		Superheated	Superheated	Evaporated	Saturated	Saturated

Mix - $s_1 > s_{fg_1}$

$x_1 = \frac{s_1 - s_{f_1}}{s_{fg_1}}$

$x_1 = \frac{6.93 - 0.755}{7.25} = 0.85$

$h_1 = h_{f_1} + x_1 \cdot h_{fg_1}$

$h_1 = 226 + 0.85 \cdot 2372$

$h_1 = 2246$ KJ/kg

Table A-5

$h_2 = 225.94$ KJ/kg

$h_3 = v_2 (P_3 - P_2) + h_2$

$h_3 = 0.001014(660 - 15) + 225.94$

$h_3 = 226.6$ KJ/kg

Table A-5

Interp:

$h_4 = 686.7$ KJ/kg

$V_4 = 0.0011048$ m³/kg

$h_5 = v_4 (P_5 - P_4) + h_4$

$h_5 = 0.0011048(5500 - 660) + 686.7$

$h_5 = 692.05$ KJ/kg

Table A-6

Interp.

$h_6 = 3428.9$ KJ/kg

$S_6 = 6.93035$ KJ/kg · K

Table A-6

Interp.

$h_7 = 2847.36$ KJ/kg

Mix - $s_8 > s_{fg}$

$x_8 = \frac{s_8 - s_{f_8}}{s_{fg_8}}$

$x_8 = \frac{6.93 - 1.4337}{5.7694} = 0.9494$

$h_8 = h_{f_8} + x_8 \cdot h_{fg_8}$

$h_8 = 467.13 + 0.9494 \cdot 2226$

$h_8 = 2580.57$ KJ/kg

Table A-5

$h_9 = 467.13$ KJ/kg

$h_{10} = 225.94$ KJ/kg

C)

$$W_T = (y)(h_6 - h_7) + (y)(h_7 - h_8) + (1-y)(h_8 - h_1)$$

$$W_T = (0.175548)(3428.9 - 2847.36) + (0.175548)(2847.36 - 2580.57) + (0.82445)(2580.57 - 2246)$$

$$\boxed{W_T = 424.743 \text{ KJ}}$$

D)

$$\dot{W}_{net} = \underbrace{\boxed{\dot{m}(W_T)}}_{\dot{W}_{out}\ \text{Turbine}} - \underbrace{\boxed{\dot{m}_{(1-y)}(h_3 - h_2) - \dot{m}(h_5 - h_4)}}_{\dot{W}_{in}\ \text{Pump}}$$

$$50{,}000 \text{ KW} = \dot{m}_T \left[(W_T) - (1-y)(h_3 - h_2) - (h_5 - h_4) \right]$$

$$\dot{m}_T = 50{,}000 / \left((424.74) - (0.82445)(226.6 - 225.94) - (692.011 - 686.664) \right)$$

$$\boxed{\dot{m}_T = 119.373 \text{ Kg/s}}$$

E)

$$q_{in} = h_6 - h_5$$

$$q_{in} = 3428.9 - 692.05 = 2736.9 \text{ KJ/Kg}$$

$$\dot{Q}_{in} = \dot{m}(q_{in}) = 119.373(2736.9) = 326710 \text{ KJ/s}$$

$$\dot{Q}_{process} = \dot{m}(y)(h_8 - h_9) = (119.373 \cdot 0.175548)(2580.57 - 467.13) = \boxed{44288.6 \text{ KJ/s}}$$

A)

$$\varepsilon_u = \frac{\dot{W}_{net} + \dot{Q}_{process}}{\dot{Q}_{in}}$$

$$\varepsilon_u = \frac{50{,}000 + 44288.6}{326710} = 0.2886 = \boxed{28.86\%}$$

H)

$$Q_{condenser} = (1-2y)h_1 + (y)h_{10} - (1-y)h_2$$

$$Q_c = (0.6489)(2246) + (0.175548)(225.94) - (0.82445)(225.94)$$

$$\boxed{Q_c = 1310.84 \text{ KJ/Kg}}$$

State	P (kPa)	h (kJ/kg)	s (kJ/kg °K)	v (m^3/kg)
1	15	2246.019		
2	15	225.94		0.001014
3	660	226.594		
4	660	686.664		0.001105
5	5500	692.0112		
6	5500	3428.9	6.93035	
7	660	2847.36	6.93035	
8	150	2580.569	6.93035	
9	150	467.13		
10	15	225.94		

q_in	2736.889
Q*_in	326709.7
Q_c	1310.838
W_out	424.7428
W_in	5.886448
W_net	418.8563
W*_net	50000
m_t	119.3727

Calculations:

P	hf	hfg	s	sf	sfg	x	h
15	225.94	2372.3	6.93035	0.7549	7.2522	0.851528	2246.01

P	hf	hfg	s	sf	sfg	x	h
150	467.13	2226	6.93035	1.4337	5.7894	0.949433	2580.56

Interpolations:

P	h	s		P	s	h
5000	3434.7	6.9781		0.6	6.9683	2850.6
5500	3428.9	6.93035		0.66	6.92312	2847.36
6000	3423.1	6.8826		0.8	6.8177	2839.8

Seek

y	LHS	RHS
0.1	686.664	488.67
0.15	686.664	619.71
0.17	686.664	672.12
0.175	686.664	685.23
0.175548	686.664	686.6643

Summary:
The utilization factor, which indicates the cycle efficiency compared to an ideal case, was determined to be 28.86%. The turbine work output per kg of steam was calculated as 424.74 kJ. To meet the specified 50,000 kW load, the required mass flow rate of steam entering the turbine is 119.37 kg/s. The heat supplied to the work spaces for this load condition is 44,288.6 kJ/s. All the thermodynamic states throughout the cycle were found, including pressures, temperatures, and vapor qualities. The mass fraction of steam extracted from the turbine for feedwater heating and space heating purposes is 0.175548. Finally, the rate of heat rejection in the condenser was calculated as 1,310.838 kJ/kg of steam.

Analysis:
This regenerative steam cycle has some inefficiencies that limit its overall utilization factor to 28.86%. A major irreversibility occurs due to the large entropy increase from expanding the steam across the turbine. The steam inlet conditions of 5.5 MPa and 500°C are quite superheated, which is advantageous for maximizing work output. However, the quality degradation from extracting low pressure streams reduces the cycle efficiency.
The turbine work output of 424.74 kJ/kg is reasonable given the inlet temperature and pressure ratio across the turbine. To meet the 50,000 kW load, a substantial mass flow of 119.37 kg/s of steam is required. Providing 44,288.6 kJ/s of heat to the work spaces is a major output, met by extracting the 0.175548 mass fraction at 660 kPa and the same fraction at 150 kPa.
The large 1,310.838 kJ/kg heat rejection in the condenser indicates that better heat recovery from the low pressure exhaust could improve cycle performance. Additional feedwater heating stages could be implemented to extract more work from the steam before condensing. Overall, while functional, there are areas for improving the efficiency of this regenerative cycle design through better heat recovery and optimizing extraction pressures.