Exercise on Geothermal Binary Cycle
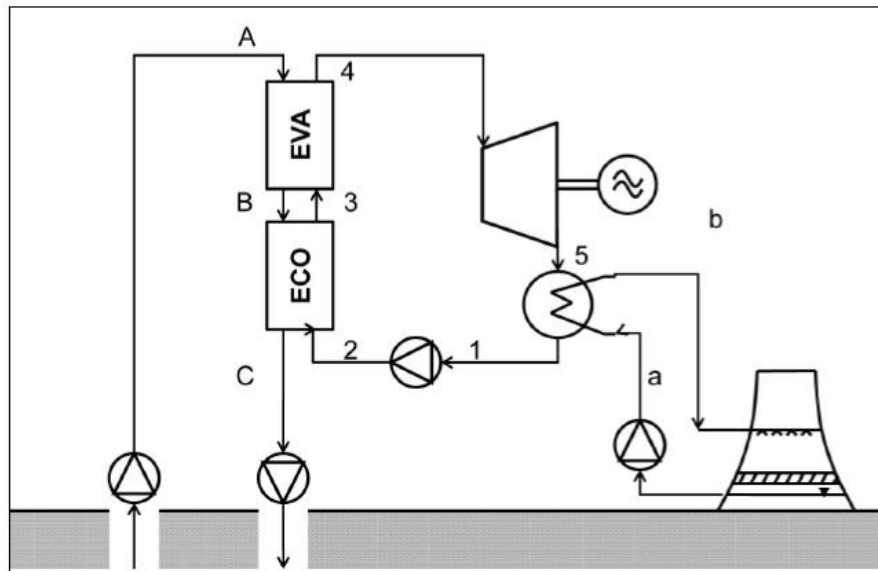
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System Description

The figure shows the schematic view of a binary cycle plant. The binary cycle plant employs an Organic fluid to extract the energy from a low to medium temperature source. In this case, the cycle is used to harness the energy of a geothermal fluid and to convert it into electrical power. For this reason, three loops are used in the system which are as follows:

- **Geothermal Fluid Loop (Water):** Geothermal fluid at 160°C is enetring from point A in evaporator. In the evaporator, this fluid is used to vaporize the organic fluid of the binary cycle and leaves evaporator with some temperature difference with the organic fluid, knows as pinch temeprature. Afterwards, it enters inside the economizer in order to preheat the organic fluid and leaves the economizer at minimum temperature to avoid salt precipoitation.
- **Binary Cycle Loop (Isopentane):** The heat released by geothermal fuid in the economizer and evaporator is gained by isopentane to change its state from subcooled liquid to saturated vapors. These vapors are then expanded by a turbine to generate electrical power. The expanded isopentane then enters into a condenser and reject its heat to condenser fluid. The isopentane leaves condenser as a saturated liquid which is then recirculated in the system via pump.
- **Cooling water Loop (Water):** This loop is used to absorb the rejected heat from the binary cycle loop. Afterwards, the cooling water circulates through the cooling tower to regenerate itself.

Mathematical Modelling of the System

- ## Geothermal Loop (Water)

Point	Description	State	Temperature
A	Evaporator Inlet	Saturated liquid	$T_A = 160°C = 433.15$ K
B	Evaporator outlet/Economizer Inlet	Subcooled liquid	$T_B = T_{eva} + \Delta T_{pinch}$
C	Economizer Outlet	Subcooled liquid	$T_C = 70°C = 343.15$ K

Where ΔT_{pinch} is the evaporator pinch point = 5°C = 5 K & T_{eva} is the evaporator temperature.
The mass flow rate of geothermal fluid is 900 ton/hr or 250 kg/s.

- ## Organic Cycle Loop (Isopentane)

Point	Description	State	Temperature (K)	Pressure (bar)	Enthalpy (kJ/kg)	Entropy (kJ/kg K)
1	Condenser Outlet/ Pump Inlet	Saturated Liquid	$T_1 = T_{cond}$	P_1 @ T_{cond}	h_1 @ T_{cond}	s_1 @ T_{cond}
2	Pump Outlet/ Economizer Inlet	Subcooled liquid	$T_2 = T_1 + (h_2 - h_1)/c_{p,iso}$	$P_2 = P_{eva} * (\dfrac{1}{1 - \dfrac{dP}{P_{in}}})$	$h_2 = h_1 + V_1(P_2 - P_1) / \eta_{isen,pump}$	$s_2 = s_1 + c_{p,iso} * \ln(\dfrac{T_2}{T_1})$
3	Economizer Outlet/ Evaporator Inlet	Subcooled liquid	$T_3 = T_{eva} - \Delta T_{sub\,cooling}$	$P_3 = P_{eva}$	$h_3 = h_{3L} + c_p *(T_3 - T_{3L})$	$s_3 = s_{3L} + c_{p,iso} * ln(T_3/T_{3L})$
3L	Evaporator	Saturated Liquid	$T_{3L} = T_{eva}$	$P_{3L} = P_{eva}$	h_{3L} @ T_{eva}	s_{3L} @ T_{eva}
4	Evaporator Outlet/ Turbine Inlet	Saturated Vapor	$T_4 = T_{eva}$	$P_4 = P_{eva}$	h_4 @ T_{eva}	s_4 @ T_{eva}
5s	Turbine Outlet/ Condenser Inlet	Superheated vapor	$T_{5s} = T_4 * (\dfrac{P_5}{P_4})^{\gamma-1/\gamma}$	$P_{5s} = P_1$	$h_{5s} = h_4 + c_{p,iso} * (T_{5s} - T_4)$	$s_{5s} = s_4$
5	Turbine Outlet/ Condenser Inlet	Superheated vapor	$T_5 = T_4 + (h_5 - h_4)/c_{p,iso}$	$P_5 = P_1$	$h_5 = h_4 + (h_{5s} - h_4) * \eta_{isen,turb}$	$s_5 = s_4 + c_{p,iso} * ln(T_5/T_4) - R^* * ln(P_5/P_4)$

Where $\dfrac{dP}{P_{in}}$ is the relative pressure drop in the economizer, R^* is the specific gas constant.

The mass flow rate of isopentane can be calculated using energy balance at the evaporator:

$$\dot{m}_{iso} * (h_4 - h_3) = \dot{m}_{geo} * c_{p_w} * (T_A - T_B)$$

$$\dot{m}_{iso} = \frac{\dot{m}_{geo} * c_{p_w} * (T_A - T_B)}{(h_4 - h_3)}$$

- Cooling Water Loop

Point	Description	State	Temperature (K)
a	Condenser inlet	Subcooled liquid	$T_a = 291.15$ K
b	Condenser outlet	Subcooled liquid	$T_b = T_a + 7$K

The mass flow rate of cooling water can be calculated using the energy balance at the condenser.

$$\dot{m}_{cond} * c_{p_w} * (T_b - T_a) = \dot{m}_{iso} * (h_5 - h_1)$$

$$\dot{m}_{cond} = \frac{\dot{m}_{iso} * (h_5 - h_1)}{c_{p_w} * (T_b - T_a)}$$

- Performance Parameters

Parameter Description	Formulation
Thermal power received by the cycle	$$\dot{Q}_{in,cycle} (kW) = \dot{m}_{iso} * (h_4 - h_2)$$
Available Geothermal Power	$$\dot{Q}_{available} (kW) = \dot{m}_{geo} * c_{pw}(T_A - T_C)$$
Auxiliary Power Input	$$\dot{W}_{aux} = \dot{W}_{pump,geo} + \dot{W}_{pump,cond}$$ $$= (\dot{m}_{geo} * v_{geo} * \frac{P_{drop,geo}}{\eta_{isen,pump} * \eta_{elec,pump}}) + (\dot{m}_{cond} * v_{cond} * \frac{P_{drop,cond}}{\eta_{isen,pump} * \eta_{elec,pump}})$$
Net Electrical Power Output	$$\dot{W}_{net,elec} = \dot{W}_{elec,turb} - \dot{W}_{elec,pump}$$ $$= \left(\dot{m}_{iso} * (h_4 - h_{5s}) * \eta_{isen,turb} * \eta_{elec,gen}\right) - (\dot{m}_{iso} * \frac{(h_{2s} - h_1)}{\eta_{isen,pump} * \eta_{elec,pump}})$$
Net Plant Power Output	$$\dot{W}_{net,plant} = \dot{W}_{net,elec} - \dot{W}_{aux}$$
Electrical Efficiency of the Plant	$$\eta_{elec} = (\dot{W}_{net,plant}/\dot{Q}_{in,cycle}) * 100$$
Thermal Efficiency of the Plant	$$\eta_{thermal} = (\dot{Q}_{in,cycle}/\dot{Q}_{available}) * 100$$

Solution

A: Determine the evaporation temperature (pressure) that maximizes the power output, considering the following evaporation temperatures: 70 °C, 80 °C, 90 °C, 100 °C, 110 °C, 120 °C.
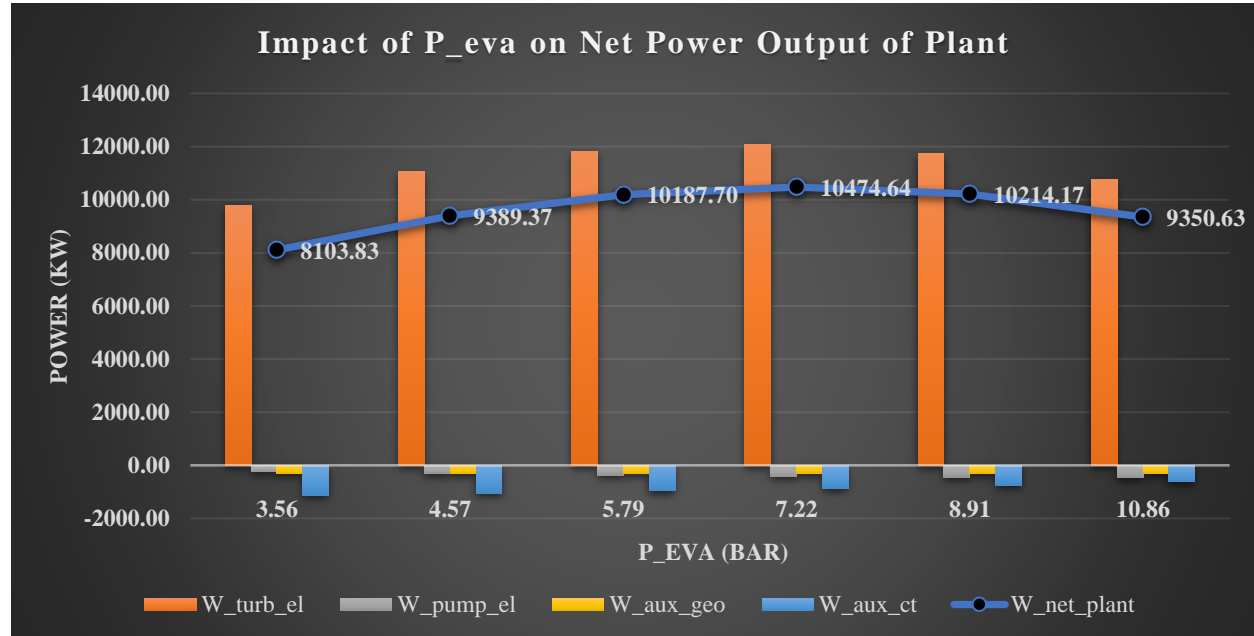

It can be observed from the figure that when the evaporator pressure (P_{eva}) of binary cycle is increased, net power output of plant ($W_{net,plant}$) also increases. However, after a certain rise in P_{eva}, $W_{net,plant}$ starts decreasing. The maximum value of the $W_{net,plant}$ came out as 10474.64 kW at $P_{eva} = 7.22$ bars.

The overall profile of $W_{net,plant}$ can be understood by analysisng the impact of P_{eva} on mass flow rate and enthalpy difference across turbine and pumps. By increasing P_{eva}, mass flow rate of both fluids (isopentane & cooling water) decreases where as enthalpy difference across all the pumps and turbine increases. As the power constitute the product of mass flow rate and enthalpy difference,the power trend of pump and turbine can be explained as follows:

- With the rise in P_{eva}, initially the enthalpy difference across turbine dominates mass flow rate of isopentane. Consequently we see a rise in turbine power output. However, after $P_{eva} = 7.22$ bars, the trend is reversed and the decrease in mass flow starts shadowing the increase in enthalpy difference. As a result, decrease in turbine power output is observed in the profile.
- For the binary cycle pump, power is dominated by increase in enthalpy difference over mass flow rate. As a consequence, the pump input power demand increases with increase in P_{eva}.
- For the cooling water pump, the mass flow decreases significantly with pressure increase and prevail the input pump power.

It can be summarized that the $W_{net,plant}$ depends on the behavior of turbine output and pump input. As we increase the evaporator pressure, the power consumed by the pumps follow either only increasing or decreasing behavior depending on the prevailance of mass flowrate or enthalpy difference. Whereas, turbine output first increases and then decreases, which is the main reason why $W_{net,plant}$ shows similar behavior.

B: On T-s diagram, draw the optimal thermodynamic cycle (the one characterized by the evaporation temperature (pressure) that enables the maximization of net power.
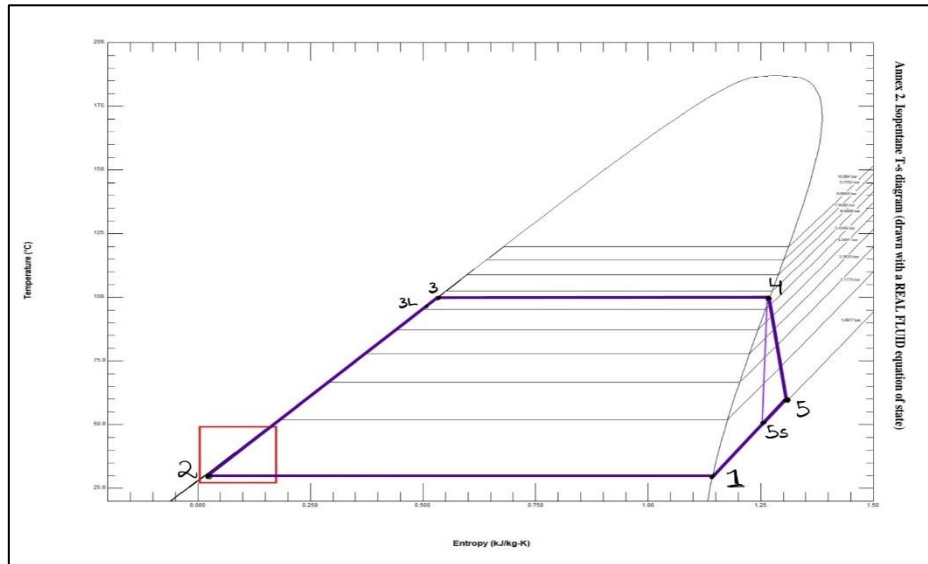
C: Draw the scheme of the plant, showing in each point the values for the mass flow, the temperature, pressure, enthalpy, entropy, and vapor quality.
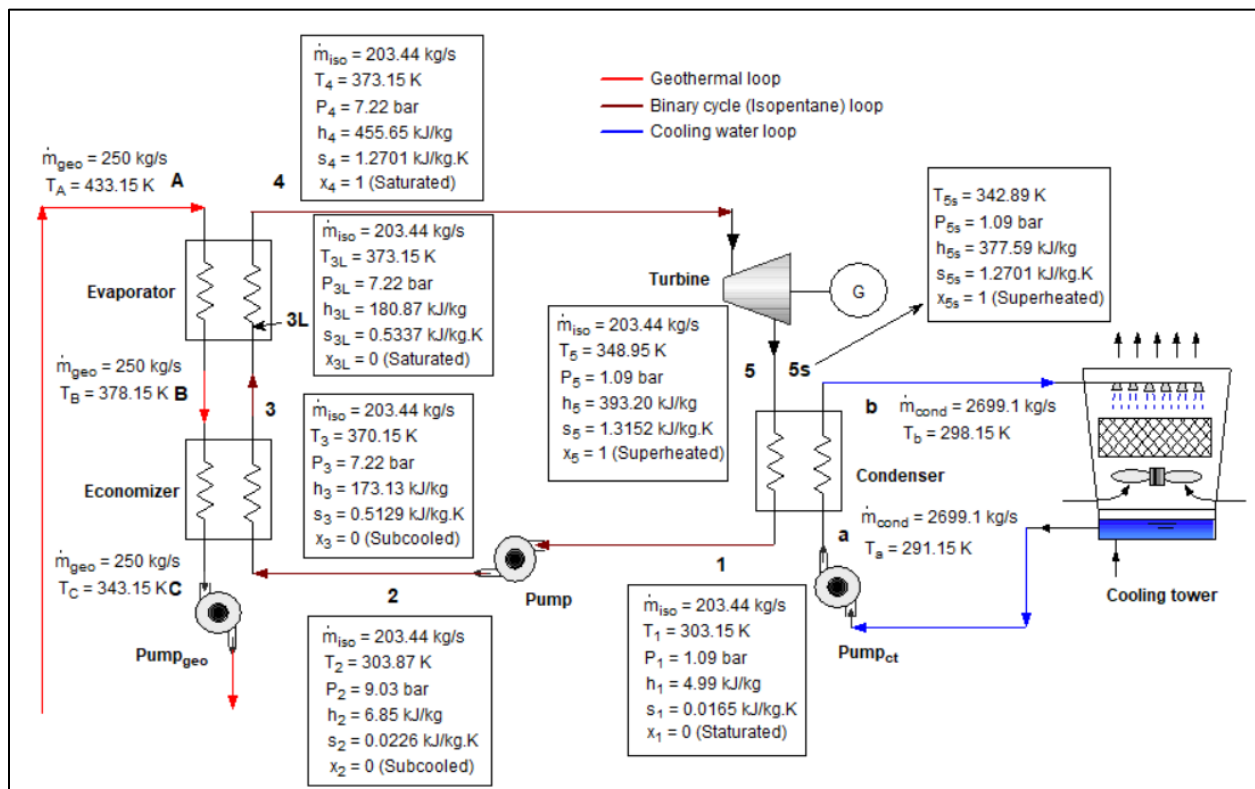
D: Determine electrical efficiency and thermal recovery efficiency.

From system performance modelling, we have:

1) Electrical Efficiency

$$\eta_{elec} = (\dot{W}_{net,plant}/\dot{Q}_{in,cycle}) * 100 \text{ ----------- eq (1).}$$

Whereas,

$$\dot{W}_{net,plant} = \dot{W}_{net,elec} - \dot{W}_{aux} = [(\dot{m}_{iso} * (h_4 - h_5) * \eta_{elec,gen}) - (\dot{m}_{iso} * \frac{(h_2 - h_1)}{\eta_{elec,pump}})] - [(\dot{m}_{geo} *$$

$$v_{geo} * \frac{P_{drop.geo}}{\eta_{isen,pump} * \eta_{elec,pump}}) + (\dot{m}_{cond} * v_{cond} * \frac{P_{drop,cond}}{\eta_{isen,pump} * \eta_{elec,pump}})]$$

By putting the values of parameters from part c, we have:

$$\dot{W}_{net,plant} = [(203.44 * (455.65 - 393.20) * 0.95) - (203.44 * \frac{(6.85 - 4.99)}{0.9})] - [(250 * 0.001 *$$

$$\frac{8*100}{0.7*0.9}) + (2699.1 * 0.001 * \frac{2*100}{0.7*0.9})]$$

$\dot{W}_{net,plant} = 10474.64$ kW

& $\dot{Q}_{in,cycle} = \dot{m}_{iso} * (h_4 - h_2) = 203.44*(455.65-6.85)$

$\dot{Q}_{in,cycle} = 91301.94$ kW

Using eq (1), we have:

$$\eta_{elec} = \frac{10474.64}{91301.94} * 100 = 11\%$$

2) Thermal Efficiency

$$\eta_{thermal} = (\dot{Q}_{in,cycle}/\dot{Q}_{available}) * 100 \text{ ----------- eq (1).}$$

Whereas,

$$\dot{Q}_{available} = \dot{m}_{geo} * c_{pw}(T_A - T_C) = 250*4.18*(433.15-343.15)$$

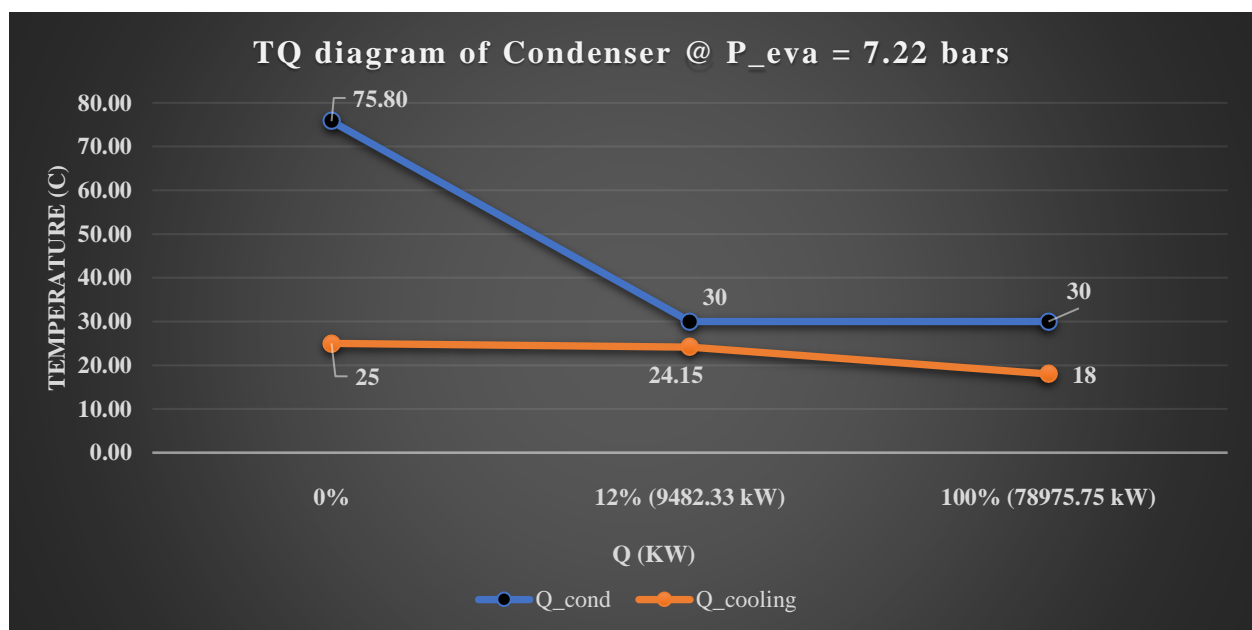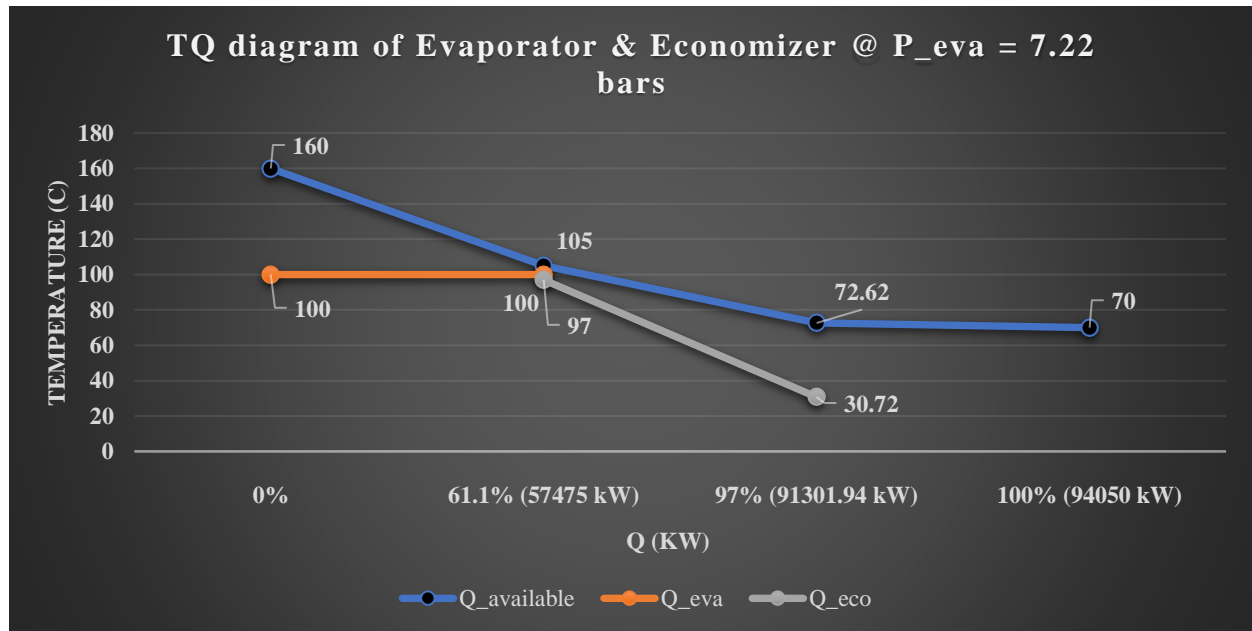$\dot{Q}_{available} = 94050$ kW

& $\dot{Q}_{in,cycle} = 91301.94$ kW

By putting the values in eq (1), we have:

$$\eta_{thermal} = \left(\frac{91301.94}{94050}\right) * 100 = 97\%$$

E: Draw the T-Q diagram for each heat exchanger.

TQ diagram of Evaporator & Economizer @ P_eva = 7.22 bars

TQ diagram of Condenser @ P_eva = 7.22 bars

F: Calculate the evaporator, economizer, and condenser surface.

Equipment	Thermal Power (kW)	Temperature Difference (K)		Log Mean Temperature Difference (K)	Area (m²)
		ΔT_1	ΔT_2		
Economizer	$\dot{Q}_{eco} = \dot{m}_{iso} * (h_3 - h_2)$	$\Delta T_{1,eco}$ $= T_B - T_3$	$\Delta T_{2,eco}$ $= T_C - T_2$	$\Delta T_{ml,eco}$ $= (\Delta T_{1,eco} - \Delta T_{2,eco})$ $/\ln (\frac{\Delta T_{1,eco}}{\Delta T_{2,eco}})$	A_{eco} $= \frac{\dot{Q}_{eco}}{\Delta T_{ml,eco} * U_{eco}}$
	$\dot{Q}_{eco}$ $= 203.44$ $* (173.13 - 6.85)$ $= 33825.94\ kW$	$\Delta T_{1,eco}$ $= 378.15$ $- 370.15$ $= 8\ K$	$\Delta T_{2,eco}$ $= 343.15$ $- 303.87$ $= 39.28\ K$	$\Delta T_{ml,eco} = 19.66\ K$	$A_{eco} = 2151.29\ m^2$
Evaporator	$\dot{Q}_{eva} = \dot{m}_{iso} * (h_4 - h_3)$	$\Delta T_{1,eva}$ $= T_A - T_4$	$\Delta T_{2,eva}$ $= T_B - T_3$	$\Delta T_{ml,eva}$ $= (\Delta T_{1,eva} - \Delta T_{2,eva})$ $/\ln (\frac{\Delta T_{1,eva}}{\Delta T_{2,eva}})$	A_{eva} $= \frac{\dot{Q}_{eva}}{\Delta T_{ml,eva} * U_{eva}}$
	$\dot{Q}_{eva}$ $= 203.44$ $* (455.65 - 173.13)$ $= 57475\ kW$	$\Delta T_{1,eva}$ $= 433.15$ $- 373.15$ $= 60\ K$	$\Delta T_{2,eva}$ $= 378.15$ $- 370.15$ $= 8\ K$	$\Delta T_{ml,eva} = 25.81\ K$	$A_{eva} = 2474.76\ m^2$
Condenser	$\dot{Q}_{cond} = \dot{m}_{iso} * (h_5 - h_1)$	$\Delta T_{1,cond}$ $= T_5 - T_b$	$\Delta T_{1,cond}$ $= T_1 - T_a$	$\Delta T_{ml,cond}$ $= (\Delta T_{1,cond} - \Delta T_{2,cond})$ $/\ln (\frac{\Delta T_{1,cond}}{\Delta T_{2,cond}})$	A_{cond} $= \frac{\dot{Q}_{cond}}{\Delta T_{ml,cond} * U_{cond}}$
	$\dot{Q}_{cond}$ $= 203.44$ $* (393.20 - 4.99)$ $= 78975.75\ kW$	$\Delta T_{1,cond}$ $= 348.95$ $- 298.15$ $= 50.80\ K$	$\Delta T_{1,cond} =$ $303.15 -$ $291.15 =$ $12\ K$	$\Delta T_{ml,cond} = 26.88\ K$	A_{cond} $= 5875.24\ m^2$