Combined Cycle Assignment

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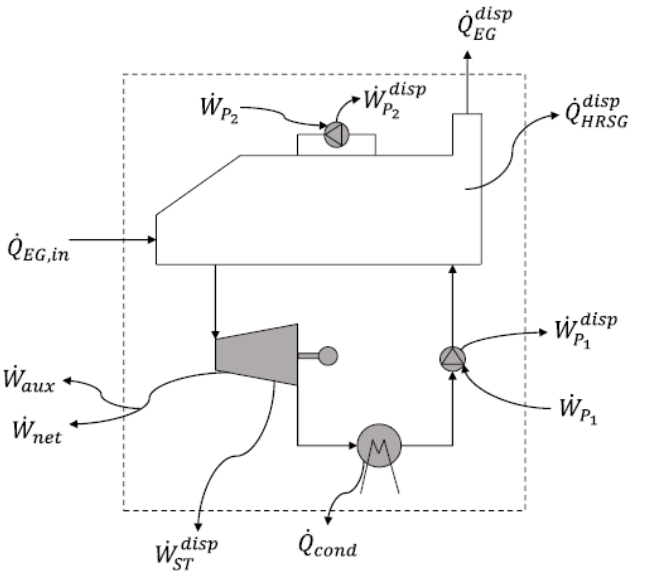
- Energy balance proof at optimum $P_{eva} = 47\ bars$ (calculation shown in excel file attached)



$$\dot{Q}_{in,EG} + \dot{W}^{el}_{p1} + \dot{W}^{el}_{p2} = \dot{W}^{disp}_{EG} + \dot{Q}^{disp}_{HRSG} + \dot{W}^{el\,disp}_{p1} + \dot{W}^{el\,disp}_{p2} + \dot{W}^{el}_{ST} + \dot{W}^{el\,disp}_{ST} + \dot{Q}_{Cond} + \dot{W}^{el}_{aux}$$

Putting values of parameters in the above equation, we have:

$$428027.80 + 24.93 + 2652.30 = 154331.90 + 2736.96 + 1.745 + 185.66 + 107107.18 + 2265.96 + 159724.38 + 4351.25$$

$$430705.04 = 430705.04$$

L.H.S = R.H.S, hence proved.

I. Calculate the variation of the gas turbine outlet temperature due to the presence of a heat recovery steam generator.

Isentropic GT outlet when coupled with HRSG = $T_{CC,s} = T_{SC}\left(\frac{P_{CC,s}}{P_{SC}}\right)^{\gamma-1/\gamma}$

Data:

T_{SC} = 550°C = 823.15 K
$P_{CC,s} = P_{amb} + \Delta P_{HRSG}$ = 101325 + 3000 = 104325 Pa
P_{SC} = 101325 Pa
C_p = 1.2 kJ/(kg.K)
M = 28.4 kg/mol

$\bar{R} = R/M$ = 8.314/28.4 = 0.2927 kJ/(kg.K)
$\gamma = C_p/(C_p - \bar{R}) = 1.2/(1.2 - 0.2927)$ = 1.323 kJ/(kg.K)
$\eta_{is,GT}$ = 0.925

Putting the values in the equation we have,

$T_{CC,s} = 823.15\left(\frac{104325}{101325}\right)^{(1.323-1)/1.323}$ = 829.03 K = 555.88°C

The Real GT outlet when coupled with HRSG is given as

$T_{CC} = T_{SC} + \eta_{is,GT}(T_{CC,s} - T_{SC})$ = 550 + 0.925(555.88-550) = 555.44°C

Therefore, the variation in gas turbine outlet temperature in presence of HRSG is given as,

$$\boxed{\Delta T = T_{CC} - T_{SC} = 555.44 - 550 = 5.44°C}$$

II. Optimize the evaporation temperature to maximize the net electrical power and

➢ Plot the heat transfer T-Q diagram between exhaust gases and water in the optimized case.

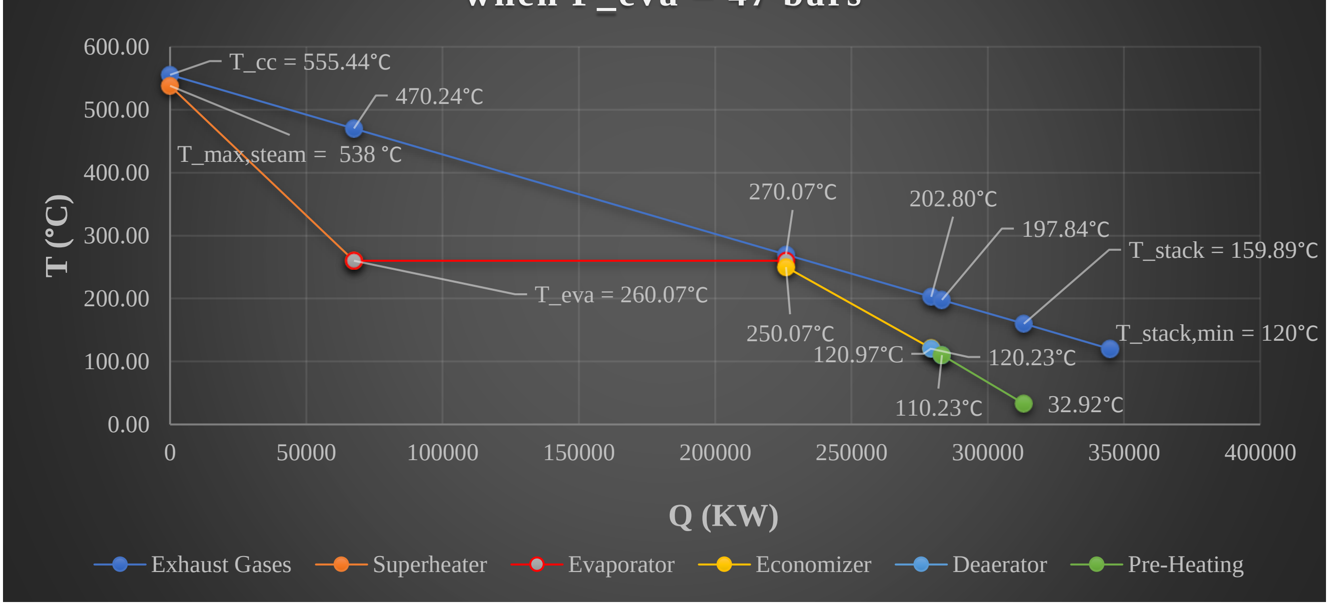


Figure 1: TQ diagram between exhaust gases and water when P_eva = 47 bars

➢ Plot and comment on the trends of the variables of interest for the designer and which would be useful in the comparative analysis of the cycle at various evaporative pressures (efficiencies, powers, exchanged thermal powers, flow rates). Compare and justify the trends of plant efficiency, cycle efficiency, heat recovery efficiency, recovery efficiency and the steam flow rate.

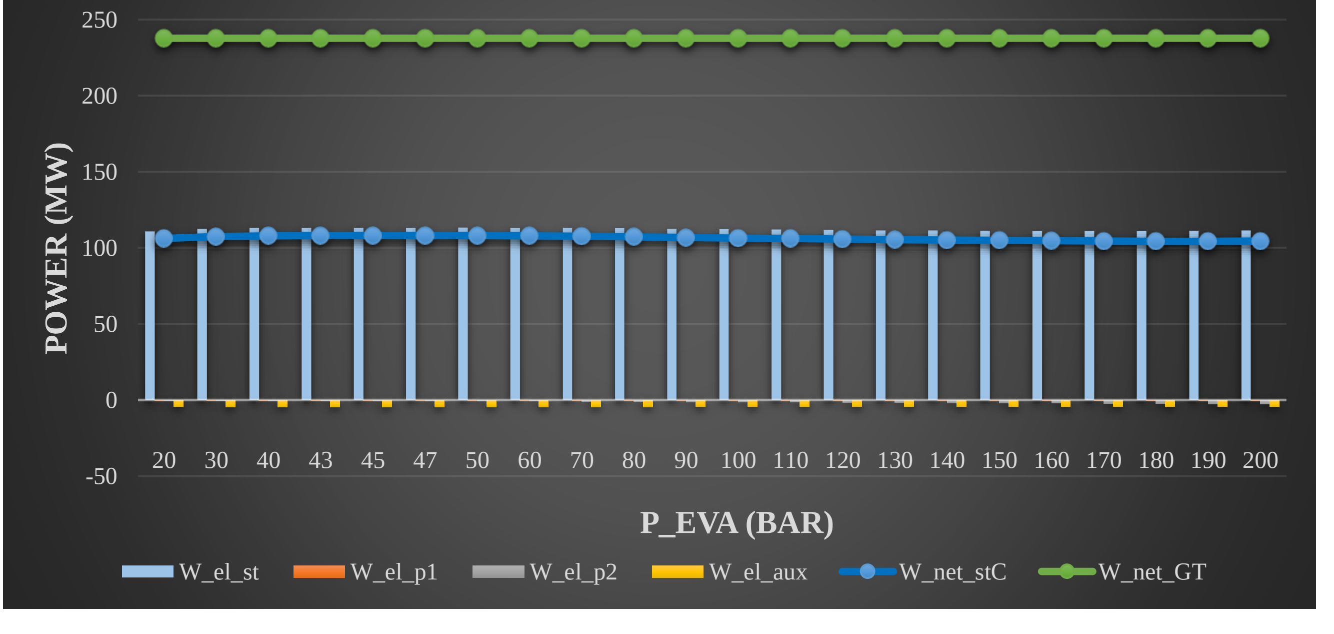


Figure 2: Impact of P_eva on Input & Output Powers

Figure 2 shows the impact of evaporator pressure on input and output powers of the plant. It can be observed that at P_eva = 47 bars, W_net_stC is maximum (108030 kW). Moreover, the turbine power output of the steam cycle is very huge compared to the input power required by the pumps and auxiliaries. Therefore, the behavior of the net power output of the steam plant also depends on the turbine power output. This is mainly the reason why the net power output peaks at 47 bars where turbine power output is also maximum. Furthermore, it can also be observed that evaporator pressure has no impact on the gas turbine power output. The only impact steam cycle creates on the power output of gas turbine cycle is due to the pressure drop at HRSG which results in some power loss due to greater turbine outlet temperature compared to simple cycle configuration.

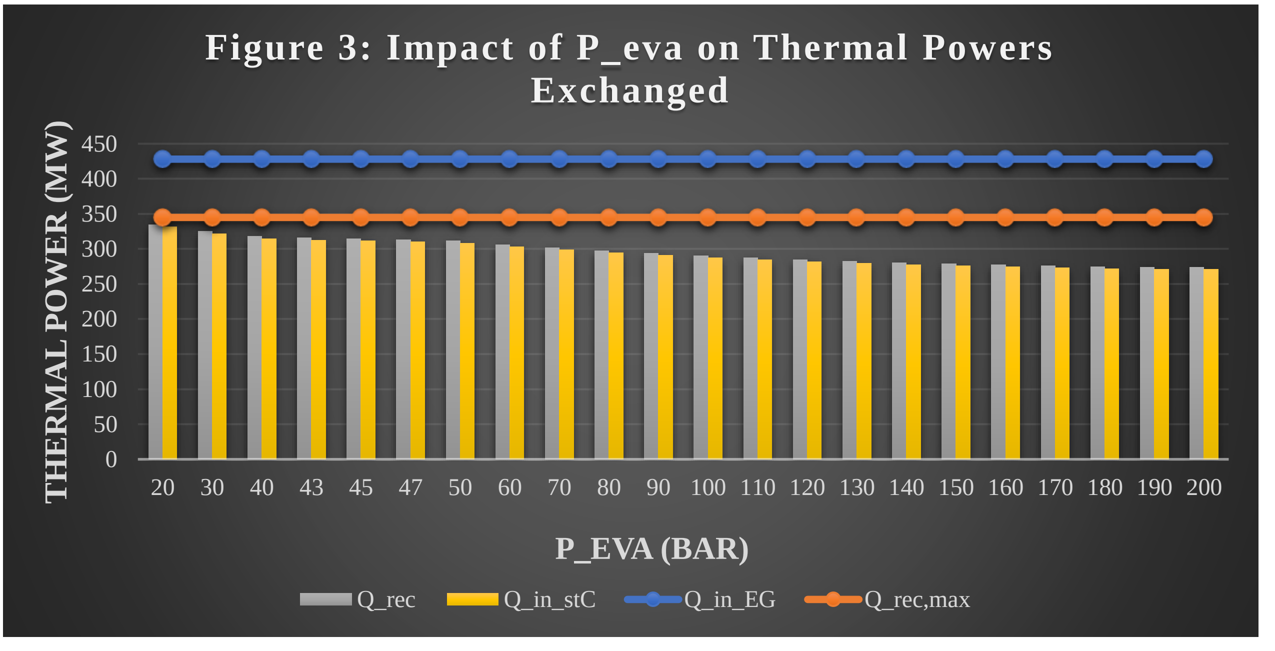
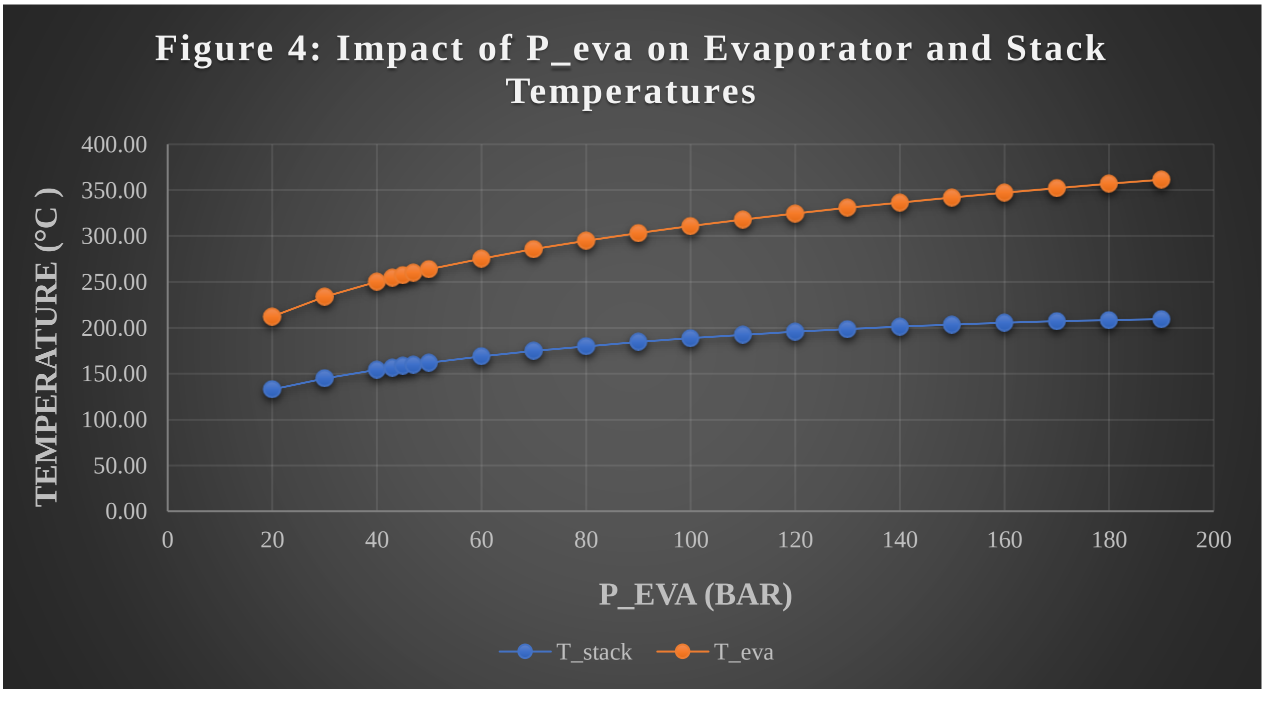


Figure 3 shows the impact of P_eva on various thermal power of the systems. It can be observed that the thermal power content of the exhaust gases and the max thermal power that can be absorbed by water remain constant and are independent of P_eva because of the predefined limits, that are, T_amb and T_stack_min, respectively. However, the actual thermal provided by the exhaust gases and the thermal power received by water decrease with the increase in evaporator pressure. The explanation of this decrease is provided with the help of Figure 4.



It can be observed in figure 4 that with the increase in P_eva, T_eva also increases. This implies that exhaust gases can transfer less heat to the evaporator, which ultimately results in higher stack temperature. As the result, we see a decrease in thermal provided by the exhaust gases and the thermal power received by water with evaporator pressure in figure 3.

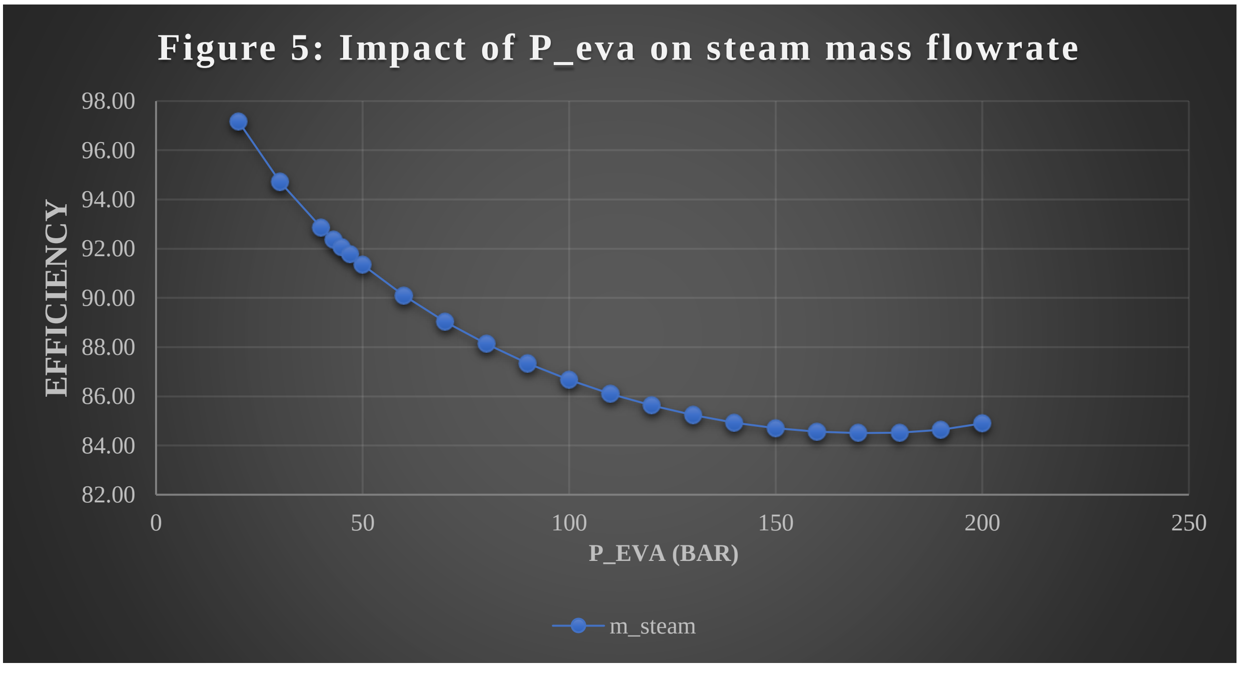


Figure 5 shows the impact of evaporator pressure on the mass flowrate of steam. It can be observed that the steam mass flow rate decreases with the increase in evaporator pressure. The mass flowrate of steam depends on the thermal power absorbed by water at evaporator after HRSG losses (Q_in_stC) and the enthalpy difference across the turbine (h_8-h_6). As we know, Q_in_stC decreases while the enthalpy difference across turbine increases. This is mainly the reason why the mass flowrate of the steam cycle continuously decreases. Decreasing mass flowrate is also the reason why the turbine power output of the steam cycle shows this behavior and peaks at 47 bars. Because steam mass flowrate and enthalpy difference across the turbine works as opposite forces. Therefore, the direction of trend of turbine power output depends on the force that is dominant at that instant with the increase in evaporator pressure.

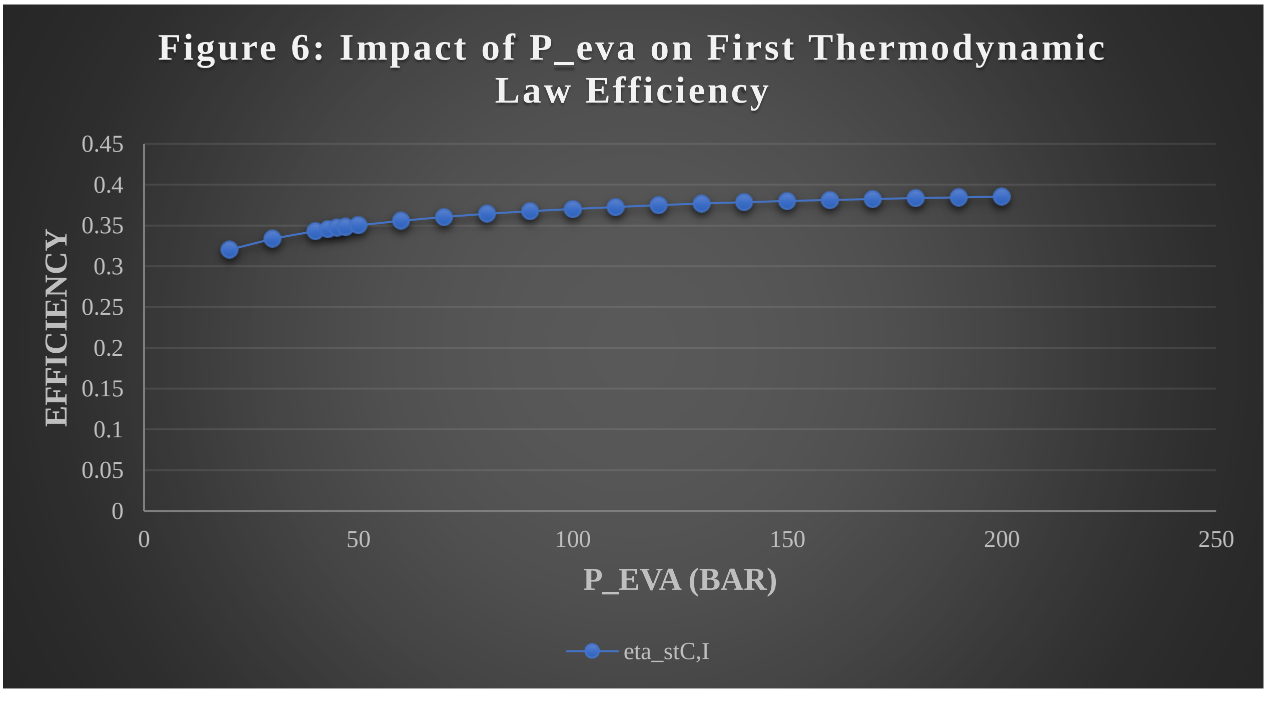


Figure 6 shows the impact of evaporator pressure on the first thermodynamic law efficiency (eta_stC,I). It can be observed that as the evaporator pressure increases, the first law efficiency also increases. The first law efficiency depends on net power output of the turbine (W_net_stC) and thermal power received by water after HRSG losses (Q_in_stC). As shown in figure 2 and figure 3, respectively, W_net_stC decreases very slowly after peaking 47 bars, however, Q_in_stC decreases rapidly. That is the main reason we observe an increase in eta_stC,I with the increase in P_eva.

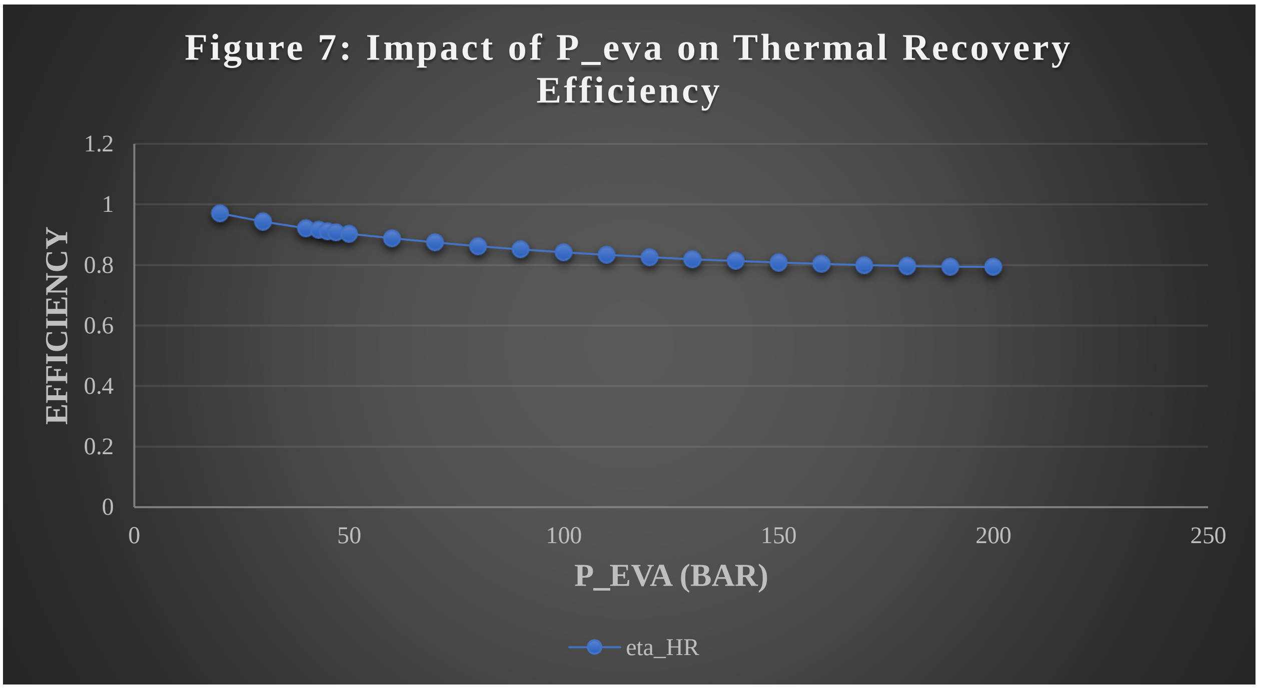


Figure 7 shows the impact of evaporator pressure on thermal recovery efficiency (eta_HR). As observed, this efficiency decreases with the increase in evaporator pressure. This efficiency depends on the energy recovered from the exhaust gases (Q_rec) and the energy that can technically be extracted from the exhaust gases (Q_rec,max). And as shown in figure 3, the Q_rec decreases while Q_rec,max is independent of evaporator pressure. Therefore, eta_HR decreases with increase in evaporator pressure.

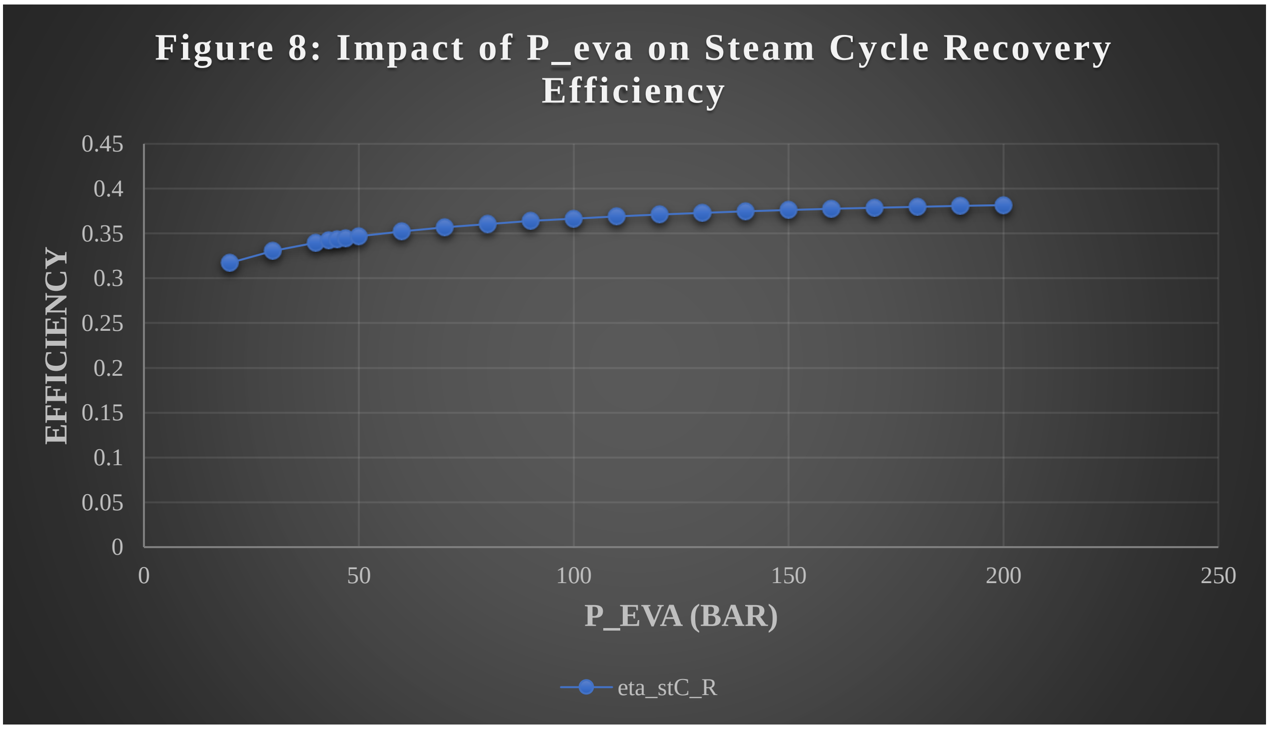


Figure 8 shows the impact of evaporator pressure on steam cycle recovery efficiency (eta_stC_R). As shown in the figure, as evaporator pressure increases, the efficiency also increases. It depends on the net turbine power output (W_net_StC) and the energy recovered from the exhaust gases (Q_rec). As shown previously in figure 2 and figure 3, W_net_stC decreases very slowly after peaking at 47 bars, however, Q_rec decreases sharply. Therefore, we observe an overall increase in steam cycle recovery efficiency.

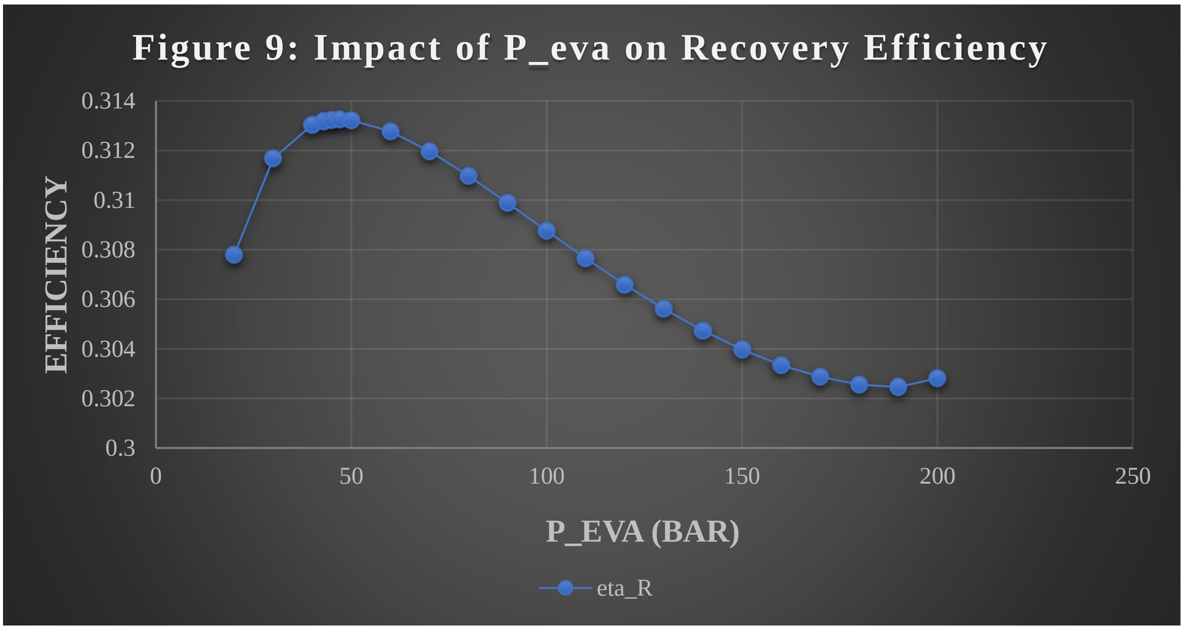


Figure 9 shows the impact of evaporator pressure on recovery efficiency (eta_R). It can be observed that as the evaporator pressure increases, recovery efficiency increases at first, peaks at 47 bars and then it starts decreasing. This efficiency depends on the net turbine power output (W_net_stC) and the maximum energy that can be technically recovered from exhaust gases (Q_rec,max). We know that the Q_rec,max is independent of P_eva while W_net_stC also increases at first, peaks at 47 bars and then starts decreasing. This is the main reason why recovery efficiency shows this behavior.

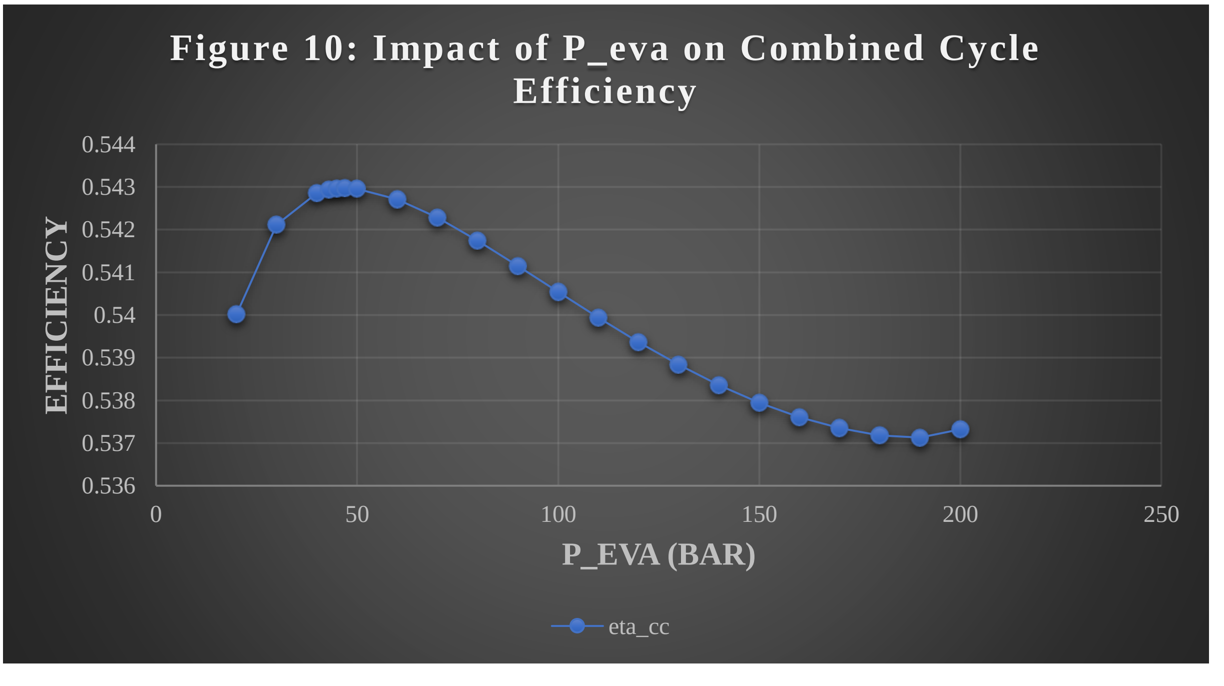


Figure 10 shows the impact of evaporator pressure on combined cycle efficiency (eta_cc). As evaporator pressure increases, the combined cycle efficiency increases at first, peaks at 47 bars and then decreases. This efficiency depends on the net turbine power output of steam cycle (W_net_stC) and the net power output of the gas turbine cycle (W_net_GT). We know that the W_net_GT is independent of P_eva while W_net_stC also increases at first, peaks at 47 bars and then starts decreasing. This is the main reason why combined cycle efficiency shows this behavior.