TECHNICAL OPTIMIZATION OF A TWO-PRESSURE LEVEL HEAT RECOVERY STEAM GENERATOR

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Pe lângă utilizarea surselor alternative de energie, una dintre soluțiile pentru reducerea poluării este utilizarea de echipamente de înaltă eficiență precum ciclurile mixte gaze-abur ce folosesc gazele fierbinți evacuate de către turbina cu gaze ca agent cald într-un generator de abur recuperator ce alimentează o turbină cu abur, eficiența totală fiind semnificativ mai mare decât dacă cele două cicluri ar fi exploatate separat. Pentru această lucrare s-a realizat o optimizare tehnologică în privința utilizării mai multor nivele de presiune a aburului, ducând la creşterea eficienței şi de asemenea reducând semnificativ costurile de investiție datorită reducerii în dimensiune a schimbătoarelor de căldură.

Beside using alternative energy sources one of the pollution reduction solutions is to use high efficiency equipment like the combined gas-steam cycles, that use the hot exhaust gas evacuated by the gas turbine as a heat agent in a heat recovery steam generator that powers a steam turbine, significantly increasing the total efficiency than using the two cycles separately. For this work a technological optimization regarding the use of multiple steam pressure stages was done, leading to an even bigger efficiency and also significantly reducing the investment costs due to the reduction in size of the heat exchangers.

Keywords: Combined cycle, Heat recovery steam generator (HRSG), Optimization.

1. Introduction

A heat recovery steam generator is the link between the gas turbine cycle and the steam turbine cycle. Its role is to recover the heat of the hot exhaust gas produced by the gas turbine and to transfer it to the Rankine cycle, by converting water into steam which powers the steam turbine. By dividing the HRSG in pressure stages, thus recuperating more heat, the cost of the HRSG can be lowered and the power output increased. Usually the three main types of heat exchanger found in a HRSG are: economizers that heat the water almost to the boiling point, evaporators with the role of boiling the water and superheaters with the role of further increasing the steam temperature. The price of the heat exchangers goes up in the order just mentioned due to the type of material used, each chosen for the corresponding temperature regime.

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In paper [1] the author made a thermodynamic optimization of multi pressure heat recovery steam generator in combined cycle. The introduction of more than one steam pressure level will increase the total recovered heat and directly the HRSG steam production and consequently will increase the power generated by the steam turbine and the overall efficiency of the combined cycle. In these conditions the exhaust gas temperature at the stack may drop below 100˚C - the boiling temperature of water at normal conditions. The gas turbine cycle in a relation to a power unit that is based on a conventional steam cycle has a compact structure. The temperature of the exhausted gas from the gas turbine can be as high as 500˚C [2,3]. In this paper, this is the temperature used for the exhaust gases.

The scope of this paper is to optimize and highlight the influence of the pressure values chosen in the sizing of the HRSG on the generated power and in the final cost of the equipment.

2. Parameters

In this paper an analysis for a two pressure stage HRSG has been chosen. More pressure stages would have greatly increased the complexity of the problem without adding anything new. Fig. 1 shows the combined cycle configuration chosen for this paper.

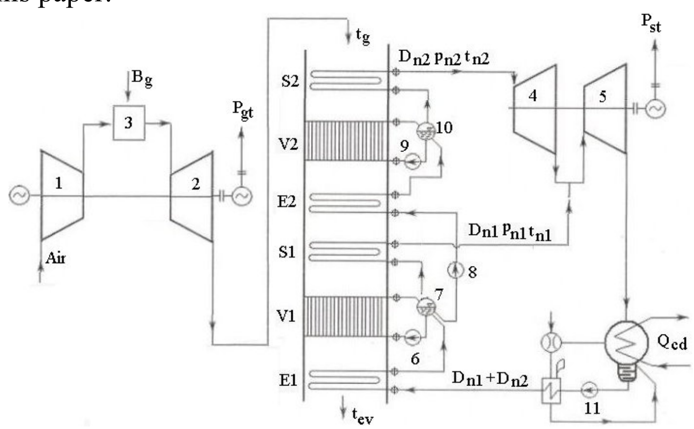


Fig. 1. Combined cycle gas steam: 1. Compressor, 2. Gas turbine (GT); 3. Combustion chamber, 4. High pressure steam turbine; 5. Low pressure steam turbine; 6. First pressure stage feed pump; 7. First stage drum; 8. Pump; 9. Second stage feed pump; 10. Second stage drum; 11. Condenser pump; B_g - fuel flow; P_{gt} - gas turbine power; E1, E2 - economizer corresponding to the first pressure level, respectively with the second pressure level; V1, V2 - evaporator corresponding to the first pressure level, respectively with the second pressure level; S1, S2 - super heater, corresponding to the first pressure level, respectively with the second pressure level; D_{n1}, D_{n2} - nominal flow for the first pressure level, respectively for the second pressure level; p_{n1}, p_{n2} - nominal pressure for the first level, respectively for the second level; t_{n1}, t_{n2} - nominal temperature for the first level, respectively for the second level; Q_{cd} - condenser heat loss.

The HRSG in the figure is composed of two sets of economizers, evaporators and superheaters, one for each pressure stage. Variations on the way in which the two steam outputs from the HRSG are used in the Rankine cycle can be devised but this isn't the scope of the paper.

Table 1 presents the usual nominal temperatures and pressure ranges for one to three pressure stages [4,5,6]. The values presented are just for reference and may differ from one manufacturer to another.

Table 1

Parameters for a different number of steam pressure stages.

Types of combined cycles		One stage	Two stages	Three stages
High steam pressure	p [bar]	40 - 70	55 -85	110 - 140
	t [˚C]	480 - 540	500 - 565	520 - 565
Medium steam pressure	p [bar]	-	-	20 - 35
	t [˚C]	-	-	520 - 565
Low steam pressure	p [bar]	-	3 - 8	4 - 6
	t [˚C]	-	200 - 260	200 - 230

3. The design of the HRSG

3.1 Energy balances of the HRSG

In order to calculate all the sizing and to obtain a technical optimization for this cycle, some heat balances will be made over the heat exchangers so that all the unknown parameters can be calculated. At the end the power of the steam turbine can be calculated. Fig. 2 shows a temperature heat diagram for the cycle to better understand the process and the notations used [7,8,9].

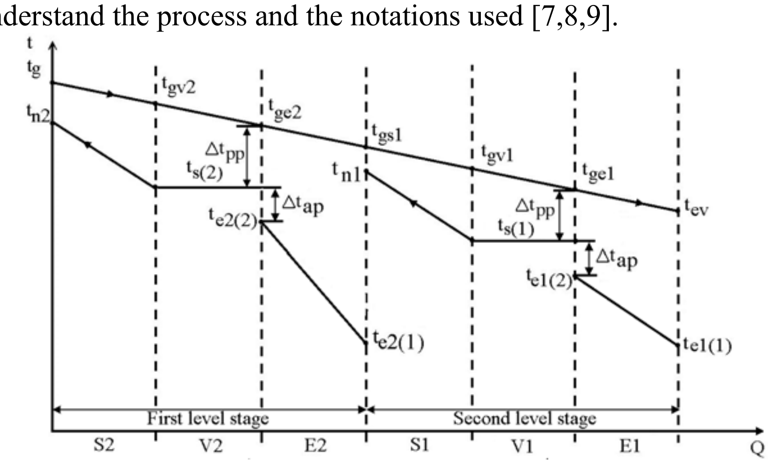


Fig. 2. Temperature-heat diagram for a HRSG with two stages of pressure: $t_{ei(1)}$ - economizer inlet temperature; $t_{ei(2)}$ - economizer outlet temperature; t_{si}- saturation temperature; t_{ni} - nominal temperature; t_{ev} - exhaust temperature; t_{gei} - economizer gas temperature; t_{gvi} - evaporator gas temperature; t_{gs1} - first stage superheater gas temperature; t_g - gas temperature; Δt_{ap} - approach point temperature; Δt_{pp} - pinch point temperature; *the "i" index in the previous notations corresponds to the pressure stage.

From the heat balance between superheater and evaporator belonging to the second pressure level the nominal flow in the second stage D_{n2} [m^3/s] can be found:

$$\varphi \cdot B_g \left[I_g\left(\lambda_g, t_g\right) - I_g\left(\lambda_g, t_{ge2}\right)\right] = D_{n2} \cdot \left(i_{n2} - i_{e2(2)}\right), \tag{1}$$

where: : φ [-] - heat conservation coefficient; Bg[m^3/s] - fuel flow; I_g[kJ/m^3] - combustion products enthalpy; λ_g[-] - air excess coefficient; i[kJ/kg] - working agent enthalpy. The enthalpies used in the balances are those corresponding to the temperature presented in Fig. 2.

From the heat balance for the second superheater, t_{gv2} (second stage evaporator gas temperature) is calculated by knowing its corresponding enthalpy $I_g\left(\lambda_g, t_{gv2}\right)$:

$$\varphi \cdot_{B_g \cdot} \left[I_g\left(\lambda_g, t_g\right) - I_g\left(\lambda_g, t_{gv2}\right)\right] = D_{n2} \cdot \left[i_{n2} - i''_{(2)} \right]. \tag{2}$$

From the heat balance for E2, S1 and V1, the nominal flow in the first stage D_{n1} [m^3 /s] can be found:

$$\varphi \cdot_{B_g \cdot} \left[I_g\left(\lambda_g, t_{ge2}\right) - I_g\left(\lambda_g, t_{ge1}\right)\right] = D_{n2} \cdot \left(i_{e2(2)} - i_{e2(1)}\right) + D_{n1} \cdot \left(i_{n1} - i_{e2(1)}\right). \tag{3}$$

Knowing this the condenser heat that needs to be removed is:

$$Q_{cd} = \left(D_{n1} + D_{n2} - D_{ej} - D_d\right) \cdot \left(i_{cd} - i'_{cd}\right), \tag{4}$$

where: D_{ej}[m^3/s] - ejector flow; D_d[m^3/s] - degasser flow; i_{cd}[kJ/kg] - condenser enthalpy.

The total flow going through the degasser and the ejector is determined from:

$$D_{n1} i_{n1} + \left(D_{n1} + D_{n2} - D_{ej} - D_d\right) i_{cd} = Q_{cd} + \left(D_{n1} + D_{n2}\right) i'_d + \left(D_{n1} - D_{ej} - D_d\right) i_{n1}, \tag{5}$$

where i'_d [kJ/kg] is the enthalpy of the saturated water in the degasser.

The steam turbine power, P_{st} [MW] is determined with:

$$D_{n2} \cdot i_{n2} + \left(D_{n1} - D_{ej} - D_d\right) \cdot i_{n1} = P_{st} + \left(D_{n1} + D_{n2} - D_{ej} - D_d\right) \cdot i_{cd}. \tag{6}$$

By continuing to apply heat balances to all the heat exchangers all the parameters can be determined in each point of the HRSG.

3.2 Calculation of the heat exchange surfaces

The general equations needed in the calculation of the heat exchanger surfaces F_{sc}[m^2] are:

$$F_{sc} = Q / (k \cdot \Delta t_{m\ln}), \tag{7}$$

where: Q[KW] is the heat transferred by the specific heat exchanger, k [W/(m^2·K)] is the global heat exchange coefficient and $\Delta t_{m\ln}$ [degrees] is the logarithmic average temperature difference. The last two are calculated with (8) [5]:

$$k = \psi \cdot \alpha_c ,\ \alpha_c = 0.2 \cdot \lambda \cdot \frac{w}{\upsilon},\ w = \frac{D_g}{F_g}, \Delta_{tm\ln} = \frac{\left(\Delta_{TMAX} - \Delta_{TMIN}\right)}{\ln\left(\Delta_{TMAX} / \Delta_{TMIN}\right)}, \tag{8}$$

where: ψ[-] is the coefficient of fouling of the heat exchanger; α_c[W/m^{2}K] - convection coefficient from gas to metal wall; λ[W/(m·K)] - gas thermal conductivity; w [m/s] - flue gas velocity over the heat exchangers; υ[m2/s] - gas kinematic viscosity; D_g[m^3/s] - gas flow; F_g[m^2] - free flowing section over the heat exchanger coils; Δ_{TMAX} [degrees] and Δ_{TMIN} [degrees] are the temperature differences at the input and the output of the heat exchanger.

A few other concepts that are unique to HRSG's are pinch point and approach point. All pinch and approach points should be sized for unfired conditions and will change during fired conditions. The approach point Δt_{ap} is used in the sizing of the economizer. As can be seen in Fig. 2, the approach point is the difference between the economizer water outlet temperature and the saturation temperature of the steam. A good approach point is in value between 5-8˚C [10,11,12], values obtained after a technical optimization.

The pinch point Δt_{pp} is used in sizing the heat transfer surface area of the HRSG. As seen in Fig. 2, the pinch point is the difference between the saturation temperature and the HRSG exit temperature. It is desirable to make the pinch point as small as possible without increasing the cost of the HRSG to much. It is usually in value between 8-10˚C [2,3], values obtained after a technical optimization.

4. Technical optimization

The only part of the cycle that will be optimized is the HRSG. The gas turbine cycle and steam turbine itself will not be considered in this paper. The initial data is given in Table 2. Three different nominal pressures were taken for the first pressure stage and other three for the second stage. The gas turbine exhaust temperature was assumed to be the same in all the cases. Also the feed water temperature and the desired nominal temperatures were assumed the same so that a comparison between the final results should be possible. Hence the only parameters influencing the HRSG are the two nominal pressures.

The initial data presented in this paper was chosen to show the behavior of the results based on the variation of the pressure. In practice some of the pressure options may be impossible to realize due to other factors. This type of analysis can be made for any other steam stages pressure combination. Also the nominal temperature can be optimized to further increase the power of the steam turbine at the lowest possible cost.

Table 2

Initial data for the optimization

	CASE NO. 1			CASE NO. 2			CASE NO. 3		
	1.1	1.2	1.3	2.1	2.2	2.3	3.1	3.2	3.3
t_g[°C]	500	500	500	500	500	500	500	500	500
$t_{e1(1)}$,$t_{e1(2)}$ [°C]	60	60	60	60	60	60	60	60	60
p_{n1}[bar]	3	3	3	9	9	9	15	15	15
t_{n1}[°C]	200	200	200	200	200	200	200	200	200
p_{n2}[bar]	100	130	160	100	130	160	100	130	160
t_{n2}[°C]	470	470	470	470	470	470	470	470	470

The formulas presented before were used in the calculations of the parameters for each of the nine cases presented in Table 2. For each of the cases will result the total surface of all the heat exchangers and the power of the steam turbine.

In Table 3 the power of the steam turbine and the total surface of the HRSG are given. Also the Rankine cycle and the total combined cycle efficiencies are shown. In the calculation of the total efficiency the gas turbine cycle is supposed to have 40% efficiency.

Table 3

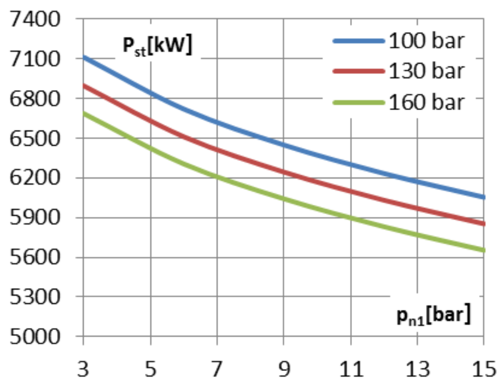
Final results

	1.1	1.2	1.3	2.1	2.2	2.3	3.1	3.2	3.3
P_{st} [MW]	7.11	6.89	6.68	6.44	6.24	6.04	6.05	5.85	5.65
F_{HRSG}[m^2]	9979	10283	10568	8700	8935	9202	9263	8794	8838
Q_u[MW]	16.86	17.17	17.45	14.68	15.05	15.37	13.31	13.72	14.07
η_{stc}	0.259	0.251	0.244	0.235	0.228	0.220	0.221	0.213	0.206
η_{mixt}	0.577	0.573	0.568	0.563	0.559	0.555	0.555	0.551	0.547
Q_u/ F_{HRSG} [kW/m^2]	1.690	1.670	1.651	1.687	1.684	1.670	1.437	1.560	1.592

Figures 3 to 8 present the variation of the results based on the nominal pressures. Regarding the steam turbine it can be seen that the generated power will be lower if any of the two nominal pressures increases.

From the plots of the total surface of the heat exchangers in the HRSG an optimum zone can be delimited where the equipment will be cheaper and more compact. A cost estimation can be made by multiplying the total surface with the price per unit of metal chosen.

Still this optimum doesn't consider the fact that the HRSG will recover more heat at the lowest chosen steam pressure value, thus producing more power. To consider this, the value of the HRSG's recovered heat should be divided to the calculated surface to obtain a ratio that shows the energy obtained per unit of surface.



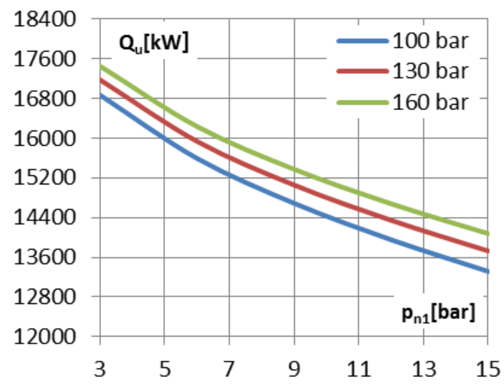
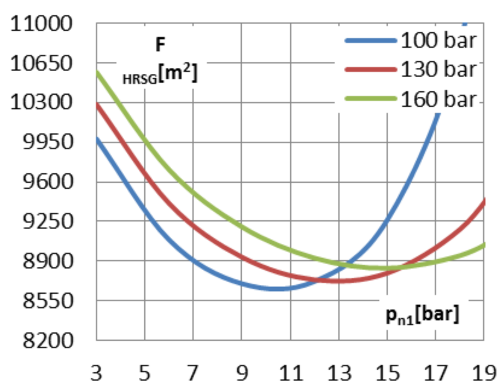


Fig. 3. Steam turbine power depending on p_{n1} and p_{n2}.

Fig. 4. Recovered heat depending on p_{n1} and p_{n2}.



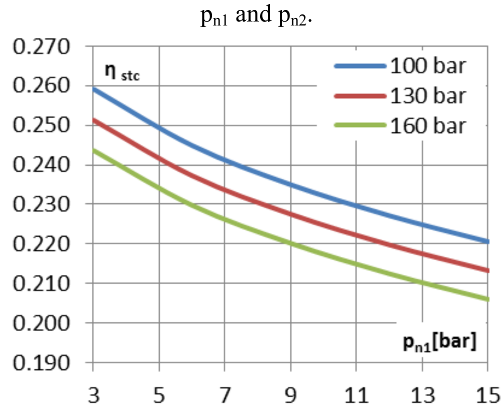
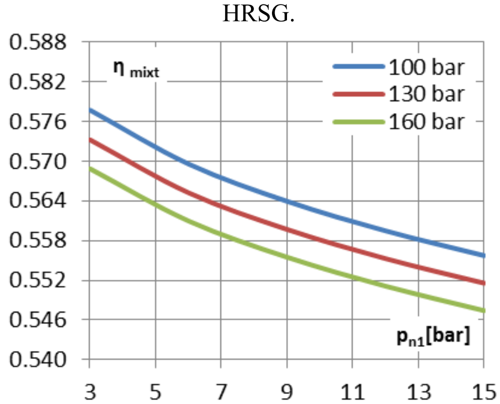


Fig. 5. Total surface of the heat exchangers in the HRSG.

Fig.6. Steam turbine cycle efficiency depending on p_{n1} and p_{n2}.



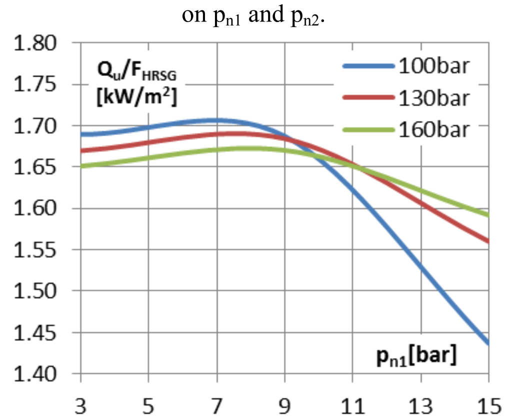


Fig. 7. Combined cycle efficiency depending on p_{n1} and p_{n2}.

Fig. 8. Recovered heat (HRSG) per total surface of the heat exchanger ratio.

5. Conclusions

This technical analysis should be made before the construction of any combined cycle with one or more pressure stages. From the example presented in the paper, it can be seen that if the manufacturer is careful at choosing the right nominal pressures the surface of the steam generator heat exchangers could be smaller and directly the investment costs lower.

The steam pressure stages combination should be chosen with a thorough analysis, the plots presented showing that the power output of the steam turbine should be balanced to the corresponding optimum zone where the surface of the HRSG is lower. From Fig. 8 we conclude that for the initial data chosen the HRSG with two steam pressure stages will have an optimum related to the total surface and recovered heat situated around p_{n1}=7 bar for p_{n2}=100 bar, p_{n1}=8 bar for p_{n2}=130 bar and p_{n1}=8.5 bar for p_{n2}=160 bar. These options will recover the most heat related to the heat exchanger total surface. This analysis can be done for any number of steam pressure stages.

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