INFLUENCE OF PINCH AND APPROACH POINT ON CONSTRUCTION OF A HEAT RECOVERY STEAM GENERATOR IN A COMBINED CYCLE

Constantin NEAGA[1], Laurenţiu FLOAREA[2]

Lucrarea de faţă este concepută ca o continuare detaliată a analizei problemei din lucrarea [1]. În articolul amintit, s-a analizat termodinamic ciclul mixt gaze-abur, scoţând în evidenţă în special debitele materiale care circulă prin sistemul centralei termice recuperatoare şi temperaturile acestora în diverse puncte ale circuitului. Analiza a insistat asupra dependenţei acestor temperaturi cu $\Delta t_s =$ =var şi $\Delta t_i = const$. Pe baza acestor interdependenţe s-au construit grafice care arată cuplul de valori capabil să determine puterea maximă a turbinei cu abur. În continuare autorii vor să arate şi dependenţa mărimilor schimbătoarelor de căldură ale generatorului recuperator de aceleaşi variabile.

The present paper is created as a detailed continuation of the analysis described in paper [1]. In the mentioned article, the combined cycle was thermodynamically analysed, emphasizing material debits which flow through recuperative thermoelectric power plant and their temperatures at various regions of the circuit. The analysis insisted on the dependance of these temperatures with respect to the pinch point variable and approach point constant. Based on these interdependencies, graphics which show the couple able to give the maximum power of the steam turbine were made. The authors would like to represent the dependencies of the heat exchangers surfaces areas as a function of the same variables.

Keywords: combined cycles; heat recovery steam generator; recuperative thermal power plant; pinch and approach point; optimization.

1. Introduction

Vital human needs can be achieved by rapid and efficient industrial growth, based on adequate extents of both electric and thermal energy. Steam generator represent a major component in any power plant; the steam produced by this equipment is used practically in any industrial area. With rising fuels and energy

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[1] Prof., Thermal Equipment of Conventional and Nuclear Power Plants Department, University POLITEHNICA of Bucharest, Romania

[2] PhD., Thermal Equipment of Conventional and Nuclear Power Plants Department, University POLITEHNICA of Bucharest, Romania, e-mail: laurentiu_floarea@yahoo.com

costs, the specialists are preoccupied in discovering new technologies of producing both electric and thermal energy, and efficient recovery of waste energy from various industrial processes.

Today, a number of alternative technologies for electric and thermal power generation are available: solar energy, wind energy, fuel cells, etc. Concerning the equipment based on fossil fuels combustion, combined cycle power plants are an important option in power generation, because of their high efficiency (50-58%), and also, because of their low pollution emissions, therefore a challenge in the actual context of global heating. As mentioned, high efficiency can be achieved due to the following reasons: improvements in gas turbine technologies, by achieving a high inlet temperature and also by heat recovery steam generators design improvements, and functional and constructive optimization of these equipments. In this context, the concept of „thermodynamic cascade" is used.

Steam and gas turbine equipments will work following a binary cycle, which represents a combination between gas turbine cycle, with high thermal level, and steam turbine cycle, with low thermal level.[13,14].

2. Paper Development

Nomenclature:

p_1, p_2 – low and high pressure, MPa;

t_{s1}, t_{s2} – corresponding saturation temperatures, $^{\circ}$C;

t_{e1}',t_{e1}'',t_{e2}',t_{e2}'' – Economizers inlet and outlet water temperatures, $^{\circ}$C;

i_{e1}',i_{e1}'',i_{e2}',i_{e2}'' - Economizers inlet and outlet water enthalpies, kJ/kg;

t_{n1},t_{n2} – Superheated steam temperatures(nominal temperatures) corresponding to both low and high pressure levels, $^{\circ}$C;

i_{n1}, i_{n2} – idem,superheated steam enthalpies, kJ/kg;

i_i', i_i'' – intermediary superheater inlet and outlet enthalpies, kJ/kg;

i_{tb1}, i_{tb2} – water enthalpies in the drum, kJ/kg (t_{tb1},t_{tb2} – idem,temperatures, $^{\circ}$C);

i_{cd} – wet steam enthalpy at condenser inlet, kJ/kg;

i'_{cd} – condensate enthalpy at saturation, kJ/kg;

i_d, i'_d – water enthalpy at deaerator inlet, saturation respectively, corresponding to water degasification pressure, kJ/kg;

p_{cd}, p_d – condenser, deaerator respectively pressure, MPa;

D_{n1}, D_{n2} – heat recovery steam generator nominal flow rates, corresponding to both pressure levels, kg/s.

D_d, D_{ej} – steam flow rates for both degasification and ejector operating;

P_{tg}, P_{ta} – gas and steam turbine power, kW;

B_g – flow rate of combusted fuel burned into the combustion chamber of the gas turbine cycle, m^3/s;

Q_2 – heat flux lost in the condenser with cooling water, kW;

Q_{cs} – thermal heat flow of the secondary circuit, kW;

Q_d – thermal heat flow available, recoverable, of the flue gas evacuated from the gas turbine, kW;

t_g, t_{gv2}, t_{ge2}, t_{gs1}, t_{gv1}, t_{ge1}, t_{ev} – flue gas temperatures, in different regions, before and after heat recovery steam generator heat exchangers, ^{o}C;

$I_g(\lambda_g, t_g)$ – flue gas enthalpy at gas turbine outlet for air excess λ_g and temperature t_g , kJ/m^3 [*];

Δt_s – superior temperature difference (pinch-point), ^{o}C;

Δt_i – inferior temperature difference (approach-point), ^{o}C;

E_1, E_2, V_1, V_2, S_1, S_2, S_I – heat recovery steam generators heat exchangers, for both pressure levels (economiser, evaporator, superheater);

PC_d – condensate preheater;

Ej – condensate and vacuum deaerator ejector;

s – fraction of thermal heat flow of the flue gases absorbed by SC_{cs} ;

SC_{cs} – secondary circuit heat exchanger;

P_{cd}, PA, PC, PT – condensate, feed, circulation and transfusion pumps;

TA, Cd, Rej, D_d , T – steam turbine, condenser, ejector cooler, deaerator, drum.

C – gains, RON/yr;

τ_{an} – power plant operating time, hrs/yr;

p_c – fuel costs, RON/m_N^3 ;

φ – heat conservation coefficient;

B_g – flow rate of gaseous fuel burnt in gas turbine combustion chamber, m_N^3 /s;

η - heat recovery steam generator efficiency;

r_n – investment recovery coefficient,1/yr;

a – amortization quota and supplementary costs;

I – investment with heat exchangers, RON;

i_{ta} – investment with steam turbine, RON.

p_s – cost of constructing 1 square meter of heat exchanger surface, RON/sq.m;

p_{cl} – drain cost, RON/sq.m drain;

τ_{vta} – steam turbine „lifetime", yrs.

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[*] Dimensions are expressed in normal physical state:273.15 K;0.1013 MPa.

Note: In I the costs associated with the components of recuperative power plant thermal circuit, such as condenser, ejector, regenerative preheaters, and also the costs of piping system were included.

In this work, the inlet parameters (gas turbine power, consumption of gaseous fuel, hot gas inlet temperature, hot gas exhaust temperature, feedwater temperature, and the nominal parameters of superheated steam) remain unchanged.
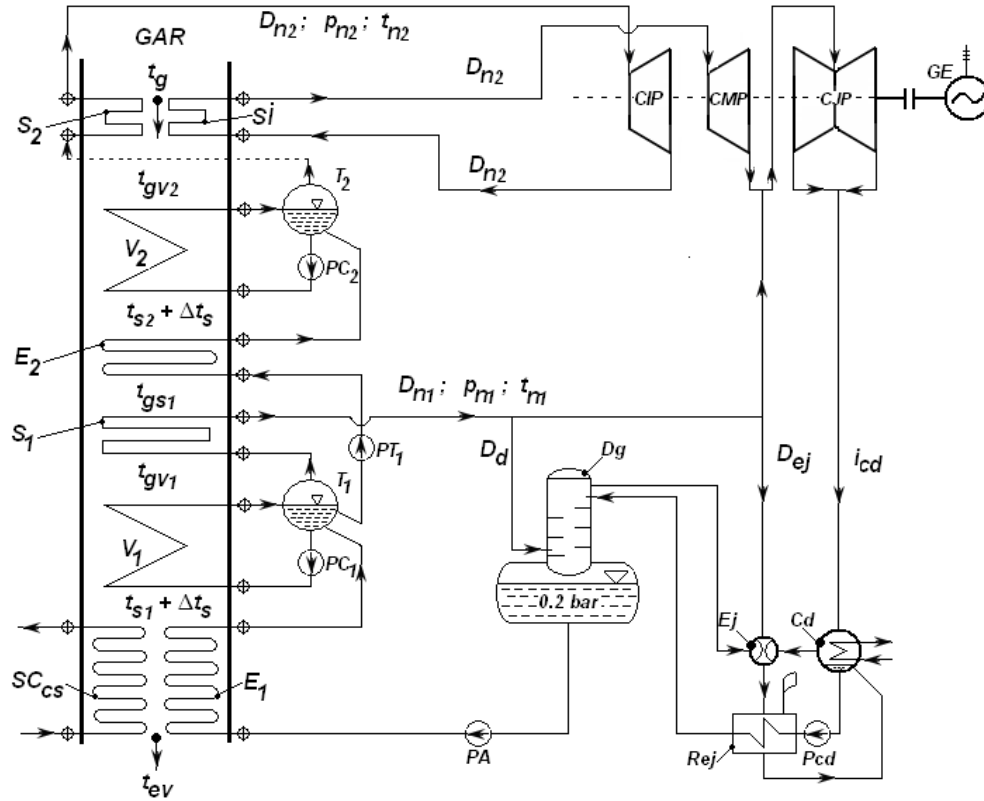

Fig.1. Thermodynamic representation of a recuperative thermoelectric power plant;GAR (HRSG-heat recovery steam generator);TA (ST) – steam turbine; Cd – condenser; Ej – ejector; Rej – ejector cooler; Dg – vacuum deaerator

In the entire work, we have considered that the entire thermal flow transferred to the heat recovery steam generator heat exchangers including the secondary circuit is constant and totally recovered:

$$Q_d = \varphi B_g [I_g(\lambda_g, t_g) - I_g(\lambda_g, t_{ev})] = const, kW. \tag{1}$$

In paper [1] a particular case study has been performed, corresponding to $\Delta t_s = 10^o C$, and $\Delta t_i = 8^o C$. This paper will show the results for various combinations of distinct values of pinch and approach point , as follows: one set

for Δt_s = 10°C, and Δt_i = 8°C,16 °C and 24 °C; Δt_s = 20°C, and Δt_i = 8°C,16 °C and 24 °C; Δt_s = 30°C, and Δt_i = 8°C,16 °C and 24 °C. The low pressure nominal parameters of heat recovery steam generator equipment are: p_{n1} = 0.6 MPa, t_{n1} = 270°C and the high pressure nominal parameters are p_{n2} = 14 MPa, t_{n1} = 470°C. The aim of these computations is to find a way how these temperature differences modify some important parameters of the recuperative cogenerative power plant. The above mentioned parameters are: steam turbine power, recovery factor, heat flow changed in the secondary circuit, and the heat exchangers surface areas [5;6;7;8;9]. These computations begin with a thermal balance of the recuperative cogenerative power plant equipped with a heat recovery steam generator without supplementary firing. The aim of this thermal balance is to find both low and high pressure nominal flow rates; these are unknown parameters:

$$Q_d = P_{ta} + Q_2 + Q_{cs} = const, kW. \tag{2}$$

Thermal balance of low pressure economizer is written. This is a mathematical link between the secondary circuit heat exchanger and the nominal flow rates:

$$\varphi B_g [I_g(\lambda_g, t_{ge1}) - I_g(\lambda_g, t_{ev})] = Q_{e1} + Q_{sccs} \tag{3}$$

In addition of these expressions,we need a thermal balance written for the reference surface composed by low pressure evaporator , low presure superheater and high pressure economizer. The purpose of this balance is to find a correlation between low and high pressure nominal flow rates:

$$\varphi B_g [I_g(\lambda_g, t_{s2} + \Delta t_s) - I_g(\lambda_g, t_{s1} + \Delta t_s)] = D_{n1} i_{n1} + D_{n2} i''_{e2} -$$
$$- (D_{n1} + D_{n2}) i''_{e1}, kW. \tag{4}$$

The correlation between thermal heat flow lost with cooling water in the condenser and the two nominal flow rates will result from the thermal balance written between the condenser, deaerator and ejector:

$$(D_{n1} + D_{n2})(i_{cd} - i'_{e1}) + D_d(i_{n1} - i_{cd}) + D_{ej}(i_{n1} - i_{cd}) = Q_2, kW. \tag{5}$$

Thermal balance of the steam turbine gives us the correlation between its power and the two nominal flow rates:

$$D_{n2} i_{n2} + D_{n2} i''_i + (D_{n1} - D_d - D_{ej}) i_{n1} = D_{n2} i'_i +$$
$$+ (D_{n1} + D_{n2} - D_d - D_{ej}) i_{cd} + P_{ta}, kW. \tag{6}$$

Introducing the expressions of steam turbine power, thermal heat flow lost with the cooling water and secondary circuit heat exchanger heat flow, as a function of nominal flow rates into the second equation, we will obtain a

correlation between the available thermal heat flow and the two nominal flow rates; coupling this equation with equation (4), we will obtain a system of two equations , with two unknown components. This is a very simple equation system. It has been solved in Mathcad , using the Given/Find function. Solving this system , we will obtain the values of the two nominal flow rates , for any kind of pinch and approach combinations. The variation of low and high pressure nominal flow rates as a function of pinch point is shown in Fig.2.
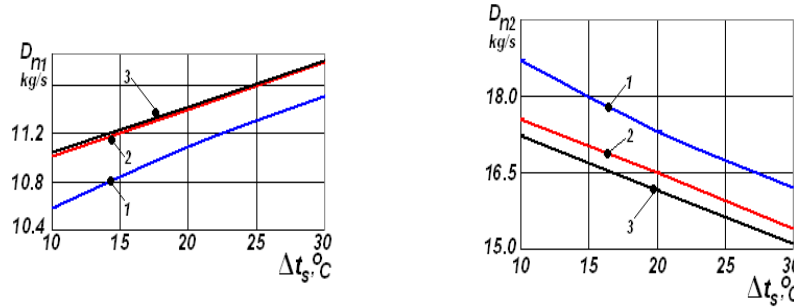

Fig.2. Low and high pressure nominal flow rates vs. pinch point ; 1 – $\Delta t_i = 8^oC$; 2 – $\Delta t_i = 16^oC$; $\Delta t_i = 24^oC$.

Observing this representation , it is obvious that , increasing the pinch point will lead to increasing low pressure nominal flow rate and decreasing high pressure nominal flow rate. The sum of the two nominal flow rates will also be decreased , which leads to decreasing steam turbine power , by varying the pinch point Δt_s .

Fig. 3 shows low pressure economizer constructive parameters versus pinch-point. By increasing pinch and approach point , from Fig. 3 it can be seen that both heat transfer surface area and economizer height will decrease.
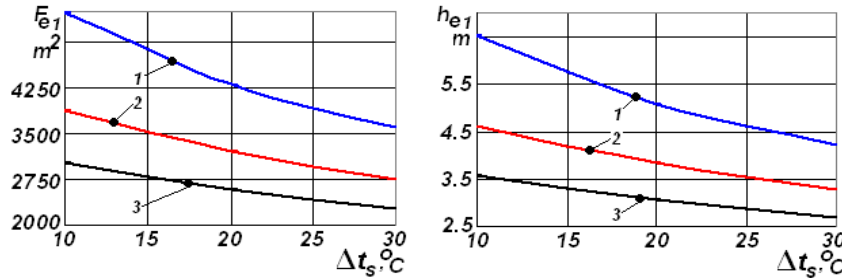

Fig.3. Pinch point vs low pressure economizer heat transfer surface area, and vs its height in heat recovery steam generator;1 – $\Delta t_i = 8^oC$; 2 – $\Delta t_i = 16^oC$; 3 - $\Delta t_i = 24^oC$.

In Fig. 4 the variation of low pressure evaporator constructive parameters with pinch-point is shown. It can be seen that the descending trend of both heat transfer surface area and height is maintained also regarding low pressure evaporator. It can be seen that the curves are close for $\Delta t_i = 8^oC$ and $\Delta t_i = 16^oC$ for both heat transfer surface area and height.
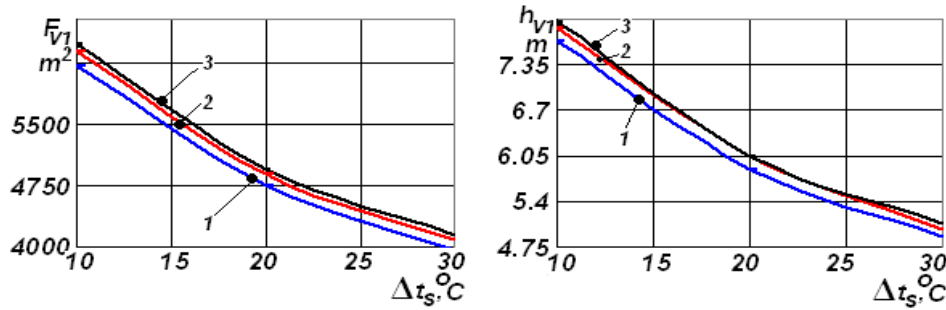
Fig.4. Pinch point vs low pressure evaporator heat transfer surface area and vs its height in heat recovery steam generator ; 1 – $\Delta t_i = 8^oC$; 2 – $\Delta t_i = 16^oC$; 3 - $\Delta t_i = 24^oC$

Low pressure superheater constructive parameters versus pinch-point is shown in Fig. 5. It is obvious that low pressure superheater heat transfer surface area and also its height will decrease with the increase of the pinch point. For $\Delta t_i = 16$ and $\Delta t_i = 24$, the values for both heat transfer surface area and low pressure superheater height are close.
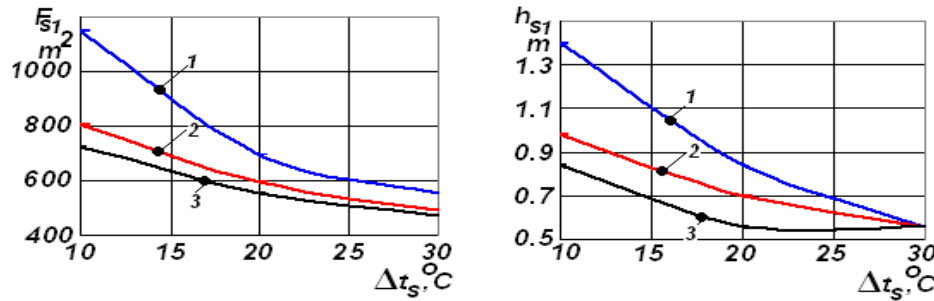

Fig.5. Pinch point vs low pressure superheater heat transfer surface area vs its height in heat recovery steam generator;1 – $\Delta t_i = 8^oC$;2 – $\Delta t_i = 16^oC$; 3 - $\Delta t_i = 24^oC$

Fig. 6 shows high pressure economizer constructive parameters versus pinch-point. Studying this figure , it can be seen that the descending trend of the low pressure economizer is maintained here. It can be seen that numerical values for $\Delta t_i = 16$ and $\Delta t_i = 24$ are relatively close.
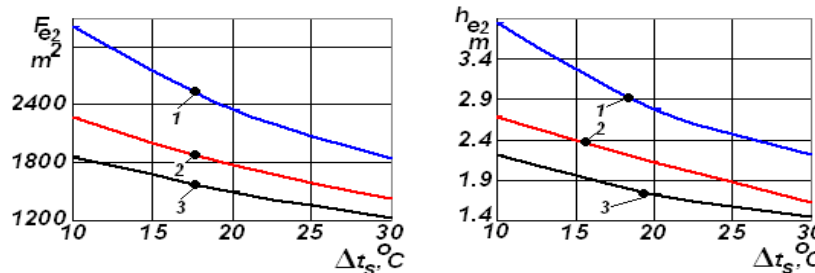

Fig.6. Pinch point vs high pressure economizer heat transfer surface area and vs its height in heat recovery steam generator; 1 – $\Delta t_i = 8^oC$; 2 – $\Delta t_i = 16^oC$; 3 - $\Delta t_i = 24^oC$.

High pressure evaporator constructive parameters versus pinch-point are shown in Fig. 7. Heat transfer surface area and its height will decrease by increasing both pinch and approach point. Unlike low pressure evaporator equipment , numerical values obtained are extremely close.
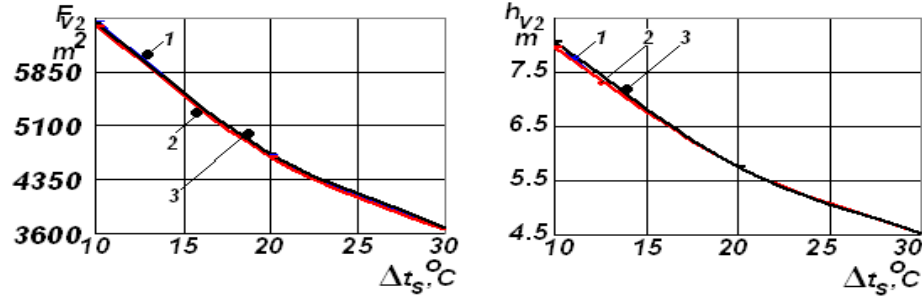

Fig.7. Pinch point vs high pressure evaporator heat transfer surface area and vs its height in heat recovery steam generator; 1 – Δt_i = 8°C; 2 – Δt_i = 16°C; 3 - Δt_i = 24°C

In Fig. 8 , the variation of high pressure superheater constructive parameters with pinch-point is shown. It can be seen that increasing both pinch and approach point , heat transfer surface area and its height will have a decreasing trend.
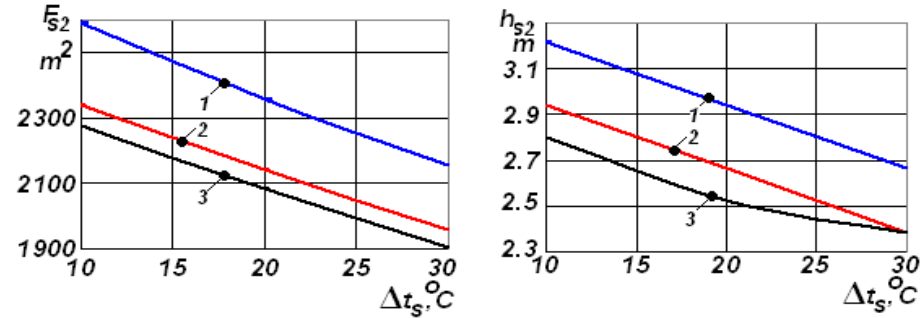

Fig.8.Pinch point vs high pressure superheater heat transfer surface area and vs its height in heat recovery steam generator; 1 – Δt_i = 8°C ; 2 – Δt_i = 16°C ; 3 - Δt_i = 24°C.

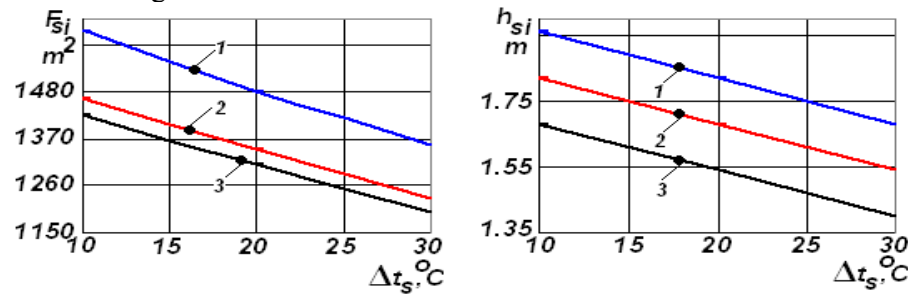The same situation can be observed in the case of intermediary superheater , shown in Fig. 9 .

Fig.9. Pinch point vs intermediary superheater heat transfer surface area and vs its height in heat recovery steam generator; 1 – Δt_i = =8°C;2 – Δt_i = 16°C; 3 - Δt_i = =24°C.

The secondary circuit heat exchanger , unlike the heat exchangers described above has an increasing trend shown in Fig. 10. It can be observed that both constructive parameters will increase by increasing pinch and approach point. Numerical values for $\Delta t_i = 8^oC$ $\Delta t_i = 16^oC$ are very close.
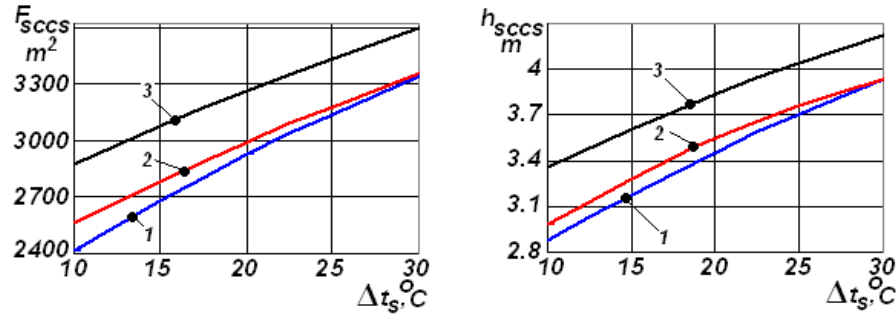

Fig.10. Pinch point vs secondary circuit heat exchanger heat transfer surface area and vs its height in heat recovery steam generator; $1 - \Delta t_i = 8^oC$; $2 - \Delta t_i = 16^oC$; $3 - \Delta t_i = 24^oC$

This heat exchanger will increase because of its high thermal heat flow transferred. As mentioned [1] , the lack of secondary circuit heat exchanger will lead to a higher value of exit gas temperature from heat recovery steam generator , higher that the one initially imposed $t_{ev} = 90^oC$. So , the initial assumption (thermal heat flow transmitted by the flue gases to heat recovery steam generator heat exchangers is constant and totally recovered) will not be respected. Consideration concerning the lay-out of secondary circuit heat exchanger in heat recovery steam generator , serial or parallel with low pressure economizer was made in [1].

The descending trend of heat transfer surface areas and its heights in heat recovery steam generator is obvious. However,as we can see in Fig.11 and Fig.12., steam turbine power and the recovery factor have a minimum value , for $\Delta t_s = 20^oC$ and $\Delta t_i = 16^oC$. As a direct consequence , we can say that extreme values of both pinch and approach points will lead to low steam turbine power. The loss of steam turbine power decrease , will be larger than gains resulted from smaller heat exchangers.

Another optimization component is finding the optimum gains resulted by residual flue gas evacuated from gas turbine recovery. This will be made by gain maximization method. Using of heat recovery steam generator will lead to fuel economy , but also to increasing costs.

$$C = 3600\tau_{an}p_c \frac{\varphi \cdot B_g\left[I_g\left(\lambda_g, t_g\right) - I_g\left(\lambda_g, t_{ev}\right)\right]}{\eta \cdot Q_i^{um}} - [(r_n + a_{e1})I_{e1} + (r_n + a_{sccs})I_{sccs} +$$

$$+(r_n + a_{v1})I_{v1} + (r_n + a_{s1})I_{s1} + (r_n + a_{e2})I_{e2} + (r_n + a_{v2})I_{v2} + (r_n + a_{s2})I_{s2} + \quad (7)$$

$$+(r_n + a_{si})I_{si} + \frac{i_{ta}P_{ta}}{\tau_{vta}} + (r_{ta} + a_{ta})i_{ta}P_{ta}], RON/yr.$$

$$I_s = (p_s + p_{cl})F_s. \quad (8)$$

Table 1 shows the computation results.

Table 1.

Mărimea	UM	Δt_s								
		10 °C			20 °C			30 °C		
		Δt_i =8°C	Δt_i =16°C	Δt_i =24°C	Δt_i =8°C	Δt_i =16°C	Δt_i =24°C	Δt_i =8°C	Δt_i =16°C	Δt_i =24°C
1	2	3	4	5	6	7	8	9	10	11
D_{n1}	kg/s	10.58	11	11.04	11.09	11.4	11.42	11.5	11.79	11.8
D_{n2}	kg/s	18.7	17.5	17.2	17.3	16.5	16.2	16.2	15.4	15.1
$D_{n1}+D_{n2}$	kg/s	29.3	28.6	28.3	28.4	27.9	27.6	27.7	27.2	26.9
Q_2	kW	61120	59685.6	59061	59707	58269	57652	58249	56850	56239
P_{ta}	kW	31540	32161	31719	30013	30965	30540	28829	29768	29359
Q_{cs}	kW	7810	8724	9792	10536	11339	12382	13181	13954	14973
t_{evcr}	°C	123.3	126.8	131.3	134.5	137.8	142.2	145.6	148.8	153.1
s	-	0.41	0.466	0.524	0.4917	0.538	0.587	0.5537	0.595	0.638
η_r	-	0.31361	0.3198	0.3154	0.2984	0.3079	0.3037	0.2867	0.296	0.2912
F_{e1}	m²	5474.8	3877	3028.3	4304.4	3218.7	2587.4	3600.2	2763.8	2262
h_{e1}	m	6.528	4.608	3.552	5.088	3.84	3.072	4.224	3.264	2.69
F_{sccs}	m²	2415.9	2561.4	2874.8	2926.6	2989.6	3264.6	3339.6	3355.7	3600.7
h_{sccs}	m	2.88	2.976	3.36	3.456	3.552	3.84	3.936	3.936	4.224
F_{v1}	m²	6229.8	6405.9	6492.2	4755.4	4886.3	4959	3975.5	4090	4152.1
h_{v1}	m	7.68	7.872	7.97	5.856	6.048	6.048	4.896	4.992	5.09
F_{s1}	m²	1148.3	806.6	725.6	692.7	592.7	557.6	549.7	488.8	468
h_{s1}	m	1.4	0.98	0.84	0.84	0.7	0.56	0.56	0.56	0.56
F_{e2}	m²	3209.9	2266.3	1858.3	2351	1770	1490	1846.2	1423.1	1218.7
h_{e2}	m	3.84	2.688	2.21	2.78	2.112	1.73	2.208	1.632	1.44
F_{v2}	m²	6552.2	6485.7	6526	4690.2	4667.8	4703.6	3670.8	3648.2	3680.2
h_{v2}	m	8.06	7.968	8.064	5.76	5.76	5.76	4.512	4.512	4.512
F_{s2}	m²	2512.1	2339.2	2274.9	2357.8	2142.7	2085.4	2154.9	1956	1905
h_{s2}	m	3.22	2.94	2.8	2.94	2.66	2.52	2.66	2.38	2.38
F_{si}	m²	1624	1463.4	1424	1479	1344.1	1309.2	1355.9	1230.1	1200
h_{si}	m	1.96	1.82	1.68	1.82	1.68	1.54	1.68	1.54	1.4

Fig.12 represents the variation of gains as a function of a pinch point and b approach point.

As we can see , increasing the pinch point , the gains will have an increasing trend , and increasing the approach point ,it can be seen that the gains have a maximum value at $\Delta t_i = 16^oC$.
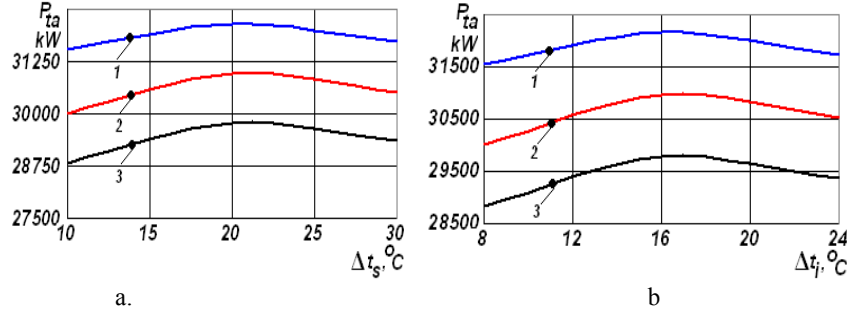

a.   b

Fig.11a. Pinch point vs steam turbine power; $1 - \Delta t_i = 8^oC$; $2 - \Delta t_i = 16^oC$; $3 - \Delta t_i = 24^oC$.
Fig.11b. Approach point vs steam turbine power; $1 - \Delta t_s = 10^oC$; $2 - \Delta t_s = 20^oC$; $3 - \Delta t_s = 30^oC$

a.   b.

Fig.12a. Pinch point vs recovery factor; $1 - \Delta t_i = 8^oC$; $2 - \Delta t_i = 16^oC$; $3 - \Delta t_i = 24^oC$.
Fig.12b. Approach point vs recovery factor; $1 - \Delta t_s = 10^oC$; $2 - \Delta t_s = 20^oC$; $3 - \Delta t_s = 30^oC$

3. Conclusions

- A well argumented mathematic model has been detailed , showing the tight links between several constructive and operating parameters of a recuperative thermoelectric power plant.
 - A secondary circuit heat exchanger was added in order to mantain the main assumption (the entire thermal heat flow of the flue gases at heat recovery steam generator inlet is mantained constant for all the computed situations).
 - Concerning heat recovery steam generator constructive elements computations , the variable is the pinch point ; in order to find steam turbine power and the recovery factor , we will take into account the approach point too.
 - Heat recovery steam generator constructive elements have a decreasing trend of heat transfer surfaces areas and their heights , among with increasing temperature differences ; the exception is the secondary circuit heat exchanger.

 - For low values of both pinch and approach points , steam turbine power and the recovery factor have a maximum value.

 - Recuperative thermoelectric power plant maximum gain will be for $\Delta t_s = 20^oC$ and $\Delta t_i = 16^oC$.

 - This paper present a great importance for the specialists in design , research and operating of these energetic equipment.

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