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COMPARATIVE ANALYSIS OF HEAT BALANCES FOR K-210-130-1 AND 13 CK 240 TURBINES

ION DOSA [1]

Abstract: Replacing and older equipment must be carried out as a result of an exhaustive technical and economical analysis. Energy efficiency is a major aspect, since energy represents a major part of operational cost for each company. This paper present the comparative analysis of heat balances of two turbines, the older K-210-130-1 and the most recent 13 CK 240 turbine, in order to emphasize the opportunity of replacing the remaining turbines with new ones.

Key words: steam turbine, heat balance, specific consumption.

1. INTRODUCTION

An important part of a power plant is the turbine which converts the thermal energy of steam into kinetic energy of steam flow and further transformed into mechanical work of turbine shaft rotation. Therefore improving the turbine efficiency will increase the amount of produced electric power.

The main advantages of steam turbines, compared with other prime movers, consists in their high speed, small size and the possibility to construct turbines of very high power ratings per single unit.

Both turbines presented in this paper are multistage condensing steam turbines for high steam conditions.

The K-210-130-1 turbine (fig. 1), is a one shaft, two-flow rotor, three-cylinder steam turbine (high-pressure cylinder, medium-pressure cylinder and low-pressure cylinder), with steam reheating after the high-pressure cylinder. The steam turbine is rated 210 MW, 3000 rot·min^{-1} [1]. The turbine is designed to operate with throttle pressure of 130 bar, and a temperature of 545 ºC, the high-pressure cylinder steam is reheated to 545 ºC at a pressure of 24.4 bar. The turbine exhaust pressure is 0.035 ata.

The 13 CK 240 and K-210-130-1 turbines are comparable; one difference resides in the construction of high-pressure and medium-pressure cylinder.

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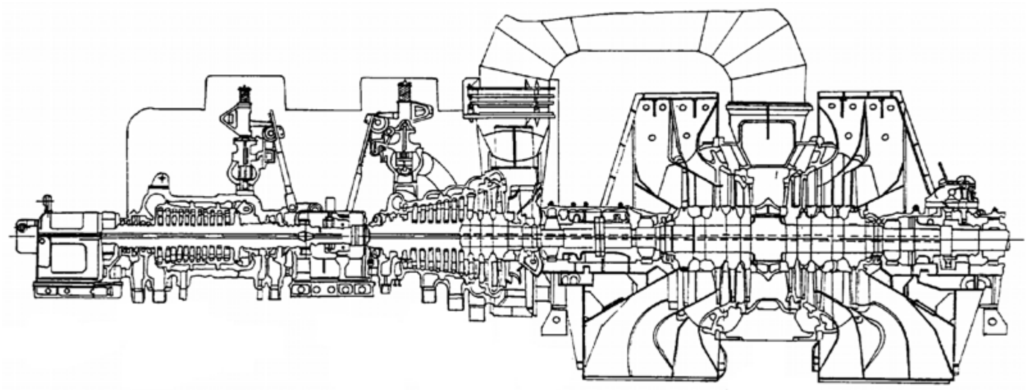

Fig. 1. Sectional view of K-210-130-1 turbine [1]

The inner rollers have a row of additional nozzles placed on double-cases, and the rotor has an extra row of rotor blades.

Another difference is in the shape of the nozzles of the high-pressure cylinder inlet (primary) steam, and for the reheated (secondary) steam at the medium-pressure cylinder inlet, in addition there are differences in the placement of control and stop valves, placed on the either side of turbine. As a result, the construction and operation principle of the latter, therefore for the entire turbine control system has changed.

The 13 CK 240 turbine is rated 235.022 MW, 3000 rot·min^{-1}. The turbine is designed to operate with throttle pressure of 127.5 bar, and a temperature of 545 ºC, the high-pressure cylinder steam is reheated to 545 ºC at a pressure of 23.83 bar. The turbine exhaust pressure is 0.035 ata.

The steam required to operate the turbines is provided in both cases by the Pp-55 boiler, with Dn=640 t·h^{-1} (2x320) rate of output steam flow at 130/127.5 bar pressure and a temperature of 545 ºC, and reheated steam with Dn=564 t·h^{-1} rate of steam flow, at 24.4/23.83 bar pressure and a temperature of 545 ºC, in matching thermal scheme.

The turbines are designed to provide steam to water heaters from which the hot water is directed into the district-heating system.

2. HEAT BALANCE CALCULATION OF STEAM TUBINES

Internal power developed by steam in a multi-stage turbine of different flow rates, can be written as a sum of internal powers in these stages [2]:

$$P_i = \sum_{j=1}^{n} D_j \cdot H_{i,j}, \quad kW \tag{1}$$

where: j is the index of summation; D_j flow rate of expanded steam in stage j, in kg·s^{-1}; $H_{i,j} = i_{1j} - i_{2rj}$, internal expansion in stage j, in kJ·kg^{-1}; i_{1j} and i_{2rj} effective enthalpy of steam at inlet and outlet of stage, in kJ·kg^{-1}.

The adiabatic expansion of steam is $H_{tj} = i_{lj} - i_{2tj}$ in kJ·kg^{-1}, where i_{2tj} is theoretical enthalpy of steam at outlet in kJ·kg^{-1}, as a result the internal efficiency is:

$$\eta_{ij} = \frac{H_{ij}}{H_{tj}} = \frac{i_{1j} - i_{2rj}}{i_{1j} - i_{2tj}} \tag{2}$$

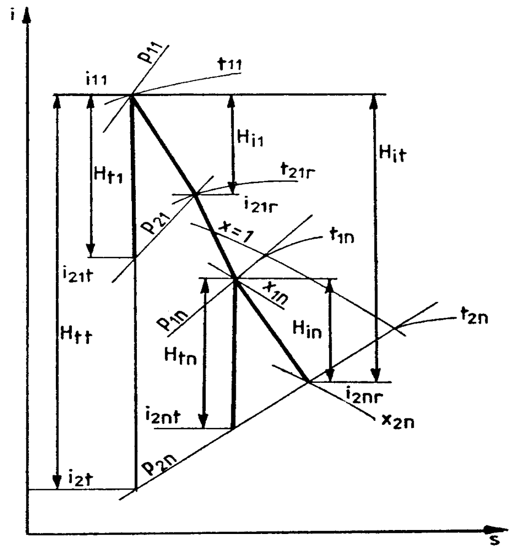

Fig. 2. The theoretical and actual process in a multi-stage steam turbine with no reheating [3]

In the same way can be defined the equivalent internal efficiency for specific bodies on the entire turbine (without reheating).

For turbines, turbine bodies or sections for which the entire expansion of steam is in the superheated range, enthalpies i_1 and i_{2r} can be calculated by measurements of pressure and temperature p_1, T_1 at inlet and p_2, T_2 at outlet.

Theoretical expansion can be found using diagrams or tables with thermodynamic properties of steam, and as a result the internal efficiency can be calculated directly using eq. (2).

The thermal power P_{ta} at turbine inlet can be found using [2][5]:

$$P_{ta} = D_0 \cdot i_{r0} + D_i \cdot \left(i_{rp6} - i_{r0}\right) \tag{3}$$

where: D_0 is the steam flow rate at inlet, in kg·s^{-1}; i_{r0} the actual enthalpy at inlet in kJ·kg^{-1}; D_i reheated steam flow rate, in kg·s^{-1}; i_{rp6} actual enthalpy at bleeder no.6 in kJ·kg^{-1}.

Generator loss and its efficiency can result from the energy balance of generator cooler:

$$\Delta P_{gen} = D_{rg} \cdot c_p \cdot \left(t_{r2g} - t_{r1}\right) \quad kW \tag{4}$$

where D_{rg} is the cooling water flow rate in kg·s^{-1}; t_{r2g} water temperature at outlet of generator cooler in °C; t_{r1} water temperature at inlet of generator cooler in °C.

Mechanical loss can be calculated using:

$$\Delta P_m = D_{ru} \cdot c_p \cdot \left(t_{r2u} - t_{r1}\right) \quad kW \tag{5}$$

where D_{ru} is the cooling water flow rate of oil cooler in $kg \cdot s^{-1}$, t_{r2u} water temperature at outlet of oil cooler in ºC; t_{r1} water temperature at inlet of oil cooler in ºC.

Lost thermal power with condenser cooling:

$$\Delta P_{cd} = D_{rc} \cdot c_p \cdot (t_{r2c} - t_{r1}) \quad kW \tag{6}$$

where D_{rc} is the cooling water flow rate in condenser $kg \cdot s^{-1}$, t_{r2c} cooling water temperature at outlet of condenser in ºC; t_{r1} cooling water temperature at inlet of condenser in ºC.

Other useful formulas can be found in [2][3][4][5][6][7][8][9].

3. RESULTS

As stated above, turbines can operate in condensing and extraction system, when the extracted steam is used in heat exchangers providing hot water for district heating.

In effective working conditions is difficult to balance the load of district heating to ensure the same load for both turbines. As a result, comparison of energy balances for turbines working in different conditions can't be done.

In order to overcome these difficulties, the energy balance will be calculated at rated parameters for operation in condensing system for both turbines.

The balance outline includes: the turbine, the generator, the condenser, the oil cooler, the generator cooler and the regenerative feedwater heating arrangement.

In Table 1 data for calculating the heat balance of K-210-130-1 and 13 CK 240 turbines can be found. The turbines are working at rated parameters, in condensation system.

Table 1 **Data for K-210-130-1 and 13 CK 240 turbines in condensation system**

Item	**Description**	**U.M.**	**Value**	
			K-210-130-1	**13 CK 240**
1.	Rated power P_g	MW	210	235.022
2.	Throttle pressure p_0	bar	130	127.5
3.	Throttle temperature t_0	ºC	545	545
4.	Steam flow rate D_0	$kg \cdot s^{-1}$	176,11	177.778
5.	Intermediate steam pressure p_i	bar	24.4	23.8
6.	Intermediate steam temperature t_i	ºC	545	545
7.	Intermediate steam flow rate D_i	$kg \cdot s^{-1}$	160.27	156.678
8.	Steam pressure at bleeder no. 7	bar	41	38.06
9.	Steam temperature at bleeder no. 7	ºC	390	365.5
10.	Steam flow rate at bleeder no. 7	$kg \cdot s^{-1}$	5.83	6.177
11.	Steam pressure at bleeder no. 6	bar	28.9	28.91
12.	Steam temperature at bleeder no. 6	ºC	350	326.4
13.	Steam flow rate at bleeder no. 6	$kg \cdot s^{-1}$	7.22	13.74

14.	Steam pressure at bleeder no. 5	bar	12.5	12.72
15.	Steam temperature at bleeder no. 5	ºC	460	445.5
16.	Steam flow rate at bleeder no. 5	$kg \cdot s^{-1}$	5.0	5.842
17.	Steam pressure at bleeder no. 4	bar	6.5	6.965
18.	Steam temperature at bleeder no. 4	ºC	380	359.1
19.	Steam flow rate at bleeder no. 4	$kg \cdot s^{-1}$	5.83	7.97
20.	Steam pressure at bleeder no. 3	bar	2.8	3.025
21.	Steam temperature at bleeder no. 3	ºC	280	253.7
22.	Steam flow rate at bleeder no. 3	$kg \cdot s^{-1}$	7.5	6.404
23.	Steam pressure at bleeder no. 2	bar	1.3	1.345
24.	Steam temperature at bleeder no. 2	ºC	200	167.8
25.	Steam flow rate at bleeder no. 2	$kg \cdot s^{-1}$	11.67	11.427
26.	Steam pressure at bleeder no. 1	bar	0.26	0.183
27.	Steam temperature at bleeder no. 1	ºC	80	60
28.	Steam flow rate at bleeder no. 1	$kg \cdot s^{-1}$	11.39	6.296
29.	Condenser temperature t_c	ºC	30	65
30.	Condenser pressure p_c	bar	0.035	0.034
31.	Temperature of main condensate t_{cd}	ºC	27	26.2
32.	Water temperature at deaerator outlet t_a	ºC	158	160
33.	Water pressure at deaerator outlet p_a	bar	6	6
34.	Water flow rate at regenerating boiler D_a	$kg \cdot s^{-1}$	183.33	191.6
35.	Condenser cooling water flow rate D_{rc}	$kg \cdot s^{-1}$	6944.44	6944.44
36.	Cold water temperature t_{r1}	ºC	17.6	18.1
37.	Water temperature at condenser outlet t_{r2c}	ºC	28.1	28
38.	Generator cooling water flow rate D_{rg}	$kg \cdot s^{-1}$	97.22	97.22
39.	Water temperature at generator cooler outlet t_{r2g}	ºC	38.3	39.9
40.	Oil cooler water flow rate D_{ru}	$kg \cdot s^{-1}$	108.33	108.33
41.	Water temperature at oil cooler outlet t_{r2u}	ºC	37	36.8

After performing calculus, results are presented in Table 2 along with efficiency indicators. Data regarding the heat balance is found in Table 3 for K-210-130-1 turbine and Table 4 for 13 CK 240 turbine.

Table 2 **Calculated values and efficiency indicators for K-210-130-1 and 13 CK 240 turbines**

Item	**Description**	**U.M.**	**Value**	
			K-210-130-1	**13 CK 240**
1.	Theoretical expansion in high-pressure body H_{tip}	$kJ \cdot kg^{-1}$	447.55	440.29
2.	Internal expansion in high-pressure body H_{iip}	$kJ \cdot kg^{-1}$	338.55	399.5
3.	Internal efficiency in high-pressure body η_{iip}	%	75.65	90.74
4.	Theoretical expansion in medium-pressure body H_{tmp}	$kJ \cdot kg^{-1}$	824.19	807.22
5.	Internal expansion in medium-pressure body H_{imp}	$kJ \cdot kg^{-1}$	688.99	750.05

6.	Internal efficiency in medium-pressure body η_{imp}	%	83.60	92.92
7.	Theoretical expansion in low-pressure body H_{tip}	$kJ \cdot kg^{-1}$	486.40	513.24
8.	Internal expansion in low-pressure body H_{iip}	$kJ \cdot kg^{-1}$	449.96	503.02
9.	Internal efficiency in low-pressure body η_{iip}	%	92.50	98.00
10.	Theoretical expansion in turbine H_h	$kJ \cdot kg^{-1}$	1736.55	1713.85
11.	Generator efficiency η_g	%	96.14	96.36
12.	Mechanical efficiency η_m	%	99.99	99.99
13.	Equivalent internal efficiency of expansion η_i	%	85.08	96.42
14.	Thermal efficiency of cycle η_t	%	42.88	46.25
15.	Total effective efficiency η_{ea}	%	39.63	43.07
16.	Gross heat consumption q_{bc}	$kJ \cdot kJ^{-1}$	3.125	2.875
17.	Consumption of conventional fuel b_{bc}	kg c.c.$\cdot$ kWh^{-1}	0.384	0.353
18.	Energy of steam at turbine inlet	$kJ \cdot kg^{-1}$	1192.44	1322

Table 3 **Summary of actual heat balance for K-210-130-1 turbine**

INPUT			**OUTPUT**		
Description	**MW**	**%**	**Description**	**MW**	**%**
Input thermal power of steam P_{ta}	652.166	100	**USEFUL POWER**		
			Generator power output P_g	210.000	32.20
			Recycled power by feedwater P_{cdr}	122.244	18.74
			TOTAL USEFUL	**332.244**	**50.94**
			Mechanical loss ΔP_m	8.800	1.35
			Generator loss ΔP_g	8.425	1.29
			Condenser loss ΔP_{cd}	305.287	46.81
			Unaccounted loss ΔP_{div}	-2.59	-0.39
			TOTAL LOST	**319.922**	**49.06**
TOTAL INPUT	**652.166**	**100**	**TOTAL OUTPUT**	**652.166**	**100**

Based on Table 3, the Sankey diagram of the actual heat balance for K-210-130-1 turbine was built (fig. 3).

Looking at Table 3 and Sankey diagram built on this basis we can draw the following conclusions:

- Steam pressure value is rated, therefore will not reduce the efficiency of the cycle and will not increase the consumption of steam and heat;

- Condenser is working well, fits the parameters, thus condensing pressure is good, and although is the main source of losses (46.81%), they cannot be reduced by other means than by reducing the flow of steam passing through the condenser, i.e. cogeneration.

- Mechanical efficiency, generator efficiency and equivalent internal efficiency of the entire turbine and for different turbine sections is within normal limits for turbines in this category of power.

In conclusion, if the turbine is functioning at rated parameters, improving its energy efficiency can be achieved only by constructive changes. This conclusion can be drawn also from the fact that the unit is used for district heating in the winter but at building was not equipped with adjustable extractions because it was designed to operate in condensing system.

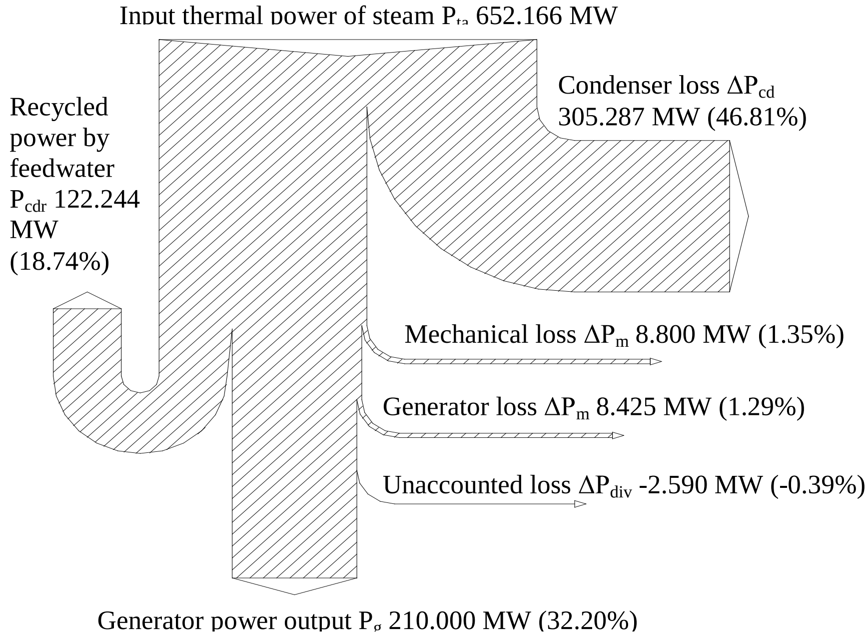

Fig.3. Sankey diagram of actual heat balance for K-210-130-1 turbine (condensation)

Table 4 **Summary of actual heat balance for 13 CK 240 turbine**

INPUT			**OUTPUT**		
Description	**MW**	**%**	**Description**	**MW**	**%**
Input thermal power of steam P_{ta}	661.448	100	**USEFUL POWER**		
			Generator power output P_g	235.022	35.53
			Recycled power by feedwater P_{cdr}	115.822	17.51
			TOTAL USEFUL	**350.844**	**53.04**
			Mechanical loss ΔP_m	8.481	1.28
			Generator loss ΔP_g	8.872	1.34
			Condenser loss ΔP_{cd}	287.841	43.52
			Unaccounted loss ΔP_{div}	5.410	0.82
			TOTAL LOST	**310.604**	**46.96**
TOTAL INPUT	**661.448**	**100**	**TOTAL OUTPUT**	**661.448**	**100**

Looking at Table 4 and Sankey diagram constructed on this basis we can draw the following conclusions:

- Throttle steam pressure is rated, therefore will not reduce the efficiency of the cycle and will not increase the consumption of steam and heat. Although throttle pressure is lower than in the case of K-210-130-1 turbine more power is obtained at the generator terminals, thus energy is used more judicious in 13 CK 240 steam turbine, of which construction is improved (extra stages and adjustable extraction);

- Condenser is working well, fits the parameters, thus condensing pressure is good, and although is the main source of losses (43.52%), they fell with 3.29% compared with the previous case, because in this case condenser temperature is lower;

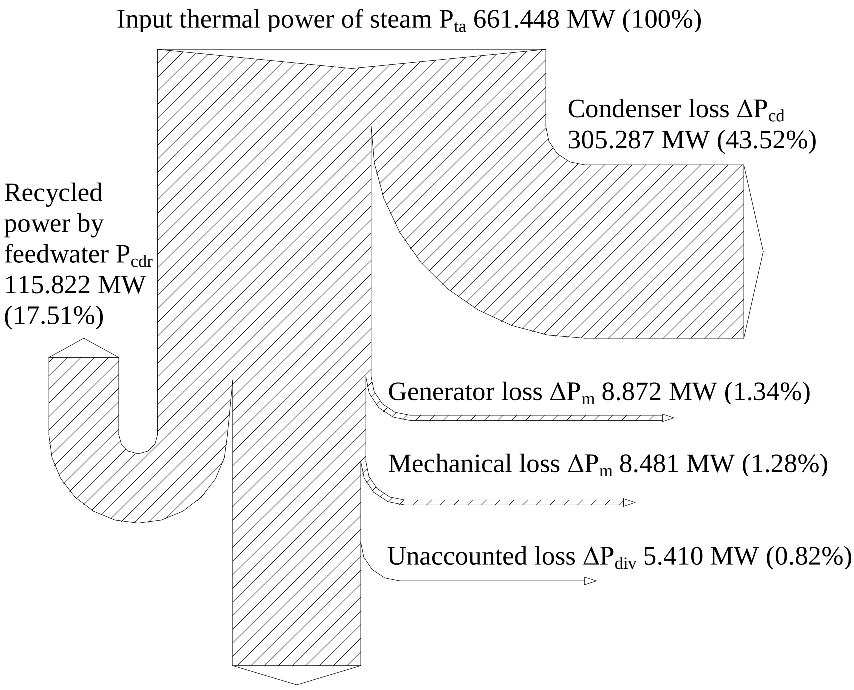

Fig. 4. Sankey diagram for actual heat balance of 13 CK 240 turbine (condensation)

- Mechanical efficiency, generator efficiency and equivalent internal efficiency of the entire turbine and for different turbine sections is within normal limits for turbines in this category of power.

- Energy efficiency of the turbine is greater, with available power output of 53.04% compared to 50.94% available useful power for K-210-13-1 turbine.

The 13 CK 240 turbine is working more efficiently then K-210-13-1 turbine in condensation system, and a comparison in district-heating system is not probative since K-210-130-1 turbine wasn't natively built for working in district heating system.

As known cogeneration improves overall efficiency of a thermal-electric plant, as a result, a heat balance for 13 CK 240 turbine working in district-heating system must be carried out.

Building the heat-balance at rated parameters in district-heating system was based on data measured at turbine testing, which are presented in Table 5.

The heat-balance outline includes: actual turbine, driven electric generator, condenser, oil cooler, generator cooler; deaerator for supplement water and condensate return from district-heating, deaerator for main condensate, regenerative preheating arrangement of condensate.

Table 5 **Data for 13 CK 240 turbine in district-heating system**

Item	**Description**	**U.M.**	**Value**
1.	Rated power P_g	MW	191.121
2.	Throttle pressure p_0	bar	127.5
3.	Throttle temperature t_0	ºC	545
4.	Steam flow rate D_0	$kg \cdot s^{-1}$	177.778
5.	Intermediate steam pressure p_i	bar	23.64
6.	Intermediate steam temperature t_i	ºC	545
7.	Intermediate steam flow rate D_i	$kg \cdot s^{-1}$	156.678
8.	Steam pressure at bleeder no. 7	bar	36.07
9.	Steam temperature at bleeder no. 7	ºC	360
10.	Steam flow rate at bleeder no. 7	$kg \cdot s^{-1}$	6.977
11.	Steam pressure at bleeder no. 6	bar	28.02
12.	Steam temperature at bleeder no. 6	ºC	320.4
13.	Steam flow rate at bleeder no. 6	$kg \cdot s^{-1}$	13.343
14.	Steam pressure at bleeder no. 5	bar	12.13
15.	Steam temperature at bleeder no. 5	ºC	443
16.	Steam flow rate at bleeder no. 5	$kg \cdot s^{-1}$	5.178
17.	Steam pressure at bleeder no. 4	bar	6.312
18.	Steam temperature at bleeder no. 4	ºC	350
19.	Steam flow rate at bleeder no. 4	$kg \cdot s^{-1}$	7.742
20.	Steam pressure at bleeder no. 3	bar	2.85
21.	Steam temperature at bleeder no. 3	ºC	251
22.	Steam flow rate at bleeder no. 3	$kg \cdot s^{-1}$	7.914
23.	Steam pressure at bleeder no. 2	bar	1.69
24.	Steam temperature at bleeder no. 2	ºC	195
25.	Steam flow rate at bleeder no. 2	$kg \cdot s^{-1}$	8.189
26.	Steam pressure at bleeder no. 1	bar	0.086
27.	Steam temperature at bleeder no. 1	ºC	50
28.	Steam flow rate at bleeder no. 1	$kg \cdot s^{-1}$	5.0246
29.	Condenser temperature t_c	ºC	50
30.	Condenser pressure p_c	bar	0.024
31.	Temperature of main condensate t_{cd}	ºC	20.6
32.	Water temperature at deaerator outlet t_a	ºC	164

33.	Water pressure at deaerator outlet p_a	bar	6
34.	Water flow rate at regenerating boiler D_a	$kg·s^{-1}$	191.6
35.	Condenser cooling water flow rate D_{rc}	$kg·s^{-1}$	6944.44
36.	Cold water temperature t_{r1}	ºC	18.1
37.	Water temperature at condenser outlet t_{r2c}	ºC	22.6
38.	Generator cooling water flow rate D_{rg}	$kg·s^{-1}$	97.22
39.	Water temperature at generator cooler outlet t_{r2g}	ºC	39.9
40.	Oil cooler water flow rate D_{ru}	$kg·s^{-1}$	108.33
41.	Water temperature at oil cooler outlet t_{r2u}	ºC	36.8

Calculated data in district-heating system, at rated parameters are presented in Table 6.

Table 6 **Calculated values and efficiency indicators for 13 CK 240 turbine**

Item	**Description**	**U.M.**	**Value**
1.	Theoretical expansion in high-pressure body H_{tip}	$kJ·kg^{-1}$	447.05
2.	Internal expansion in high-pressure body H_{iip}	$kJ·kg^{-1}$	409.9
3.	Internal efficiency in high-pressure body η_{iip}	%	91.69
4.	Theoretical expansion in medium-pressure body H_{tmp}	$kJ·kg^{-1}$	754.15
5.	Internal expansion in medium-pressure body H_{imp}	$kJ·kg^{-1}$	702.05
6.	Internal efficiency in medium-pressure body η_{imp}	%	93.09
7.	Theoretical expansion in low-pressure body H_{tjp}	$kJ·kg^{-1}$	521.19
8.	Internal expansion in low-pressure body H_{ijp}	$kJ·kg^{-1}$	507.21
9.	Internal efficiency in low-pressure body η_{ijp}	%	97.32
10.	Theoretical expansion in turbine H_h	$kJ·kg^{-1}$	1713.85
11.	Generator efficiency η_g	%	95.56
12.	Mechanical efficiency η_m	%	99.99
13.	Equivalent internal efficiency of expansion η_i	%	94.47
14.	Thermal efficiency of cycle η_t	%	60.99
15.	Total effective efficiency η_{ea}	%	55.92
16.	Gross heat consumption q_{bc}	$kJ·kJ^{-1}$	2.215
17.	Consumption of conventional fuel b_{bc}	kg c.c.· kWh^{-1}	0.272
18.	Energy of steam at turbine inlet	$kJ·kg^{-1}$	1075.05

Results required for building the actual heat balance of turbine 13 CK 240 were summarized in Table 7, and Sankey diagram shown in Fig. 5.

Table 7 **Summary of actual heat balance for 13 CK 240 turbine (district-heating)**

INPUT			**OUTPUT**		
Description	**MW**	**%**	**Description**	**MW**	**%**
Input thermal power of steam P_{ta}	663.226	97.75	**USEFUL POWER**		
			Generator output P_g	191.121	28.17
			Recycled power by feedwater P_{cdr}	115.822	17.07
			Extraction thermal power P_{ta2}	220.834	32.55

Input thermal power of district-heating condensate and supplement water P_{cdt}	15.231	2.25	**TOTAL USEFUL**	**527.777**	**77.79**
			Mechanical loss ΔP_m	8.481	1.25
			Generator loss ΔP_g	8.873	1.31
			Condenser loss ΔP_{cd}	130.837	19.28
			Unaccounted loss ΔP_{div}	2.49	0.37
			TOTAL LOST	**310.604**	**22.21**
TOTAL INPUT	**678.457**	**100**	**TOTAL OUTPUT**	**678.457**	**100**

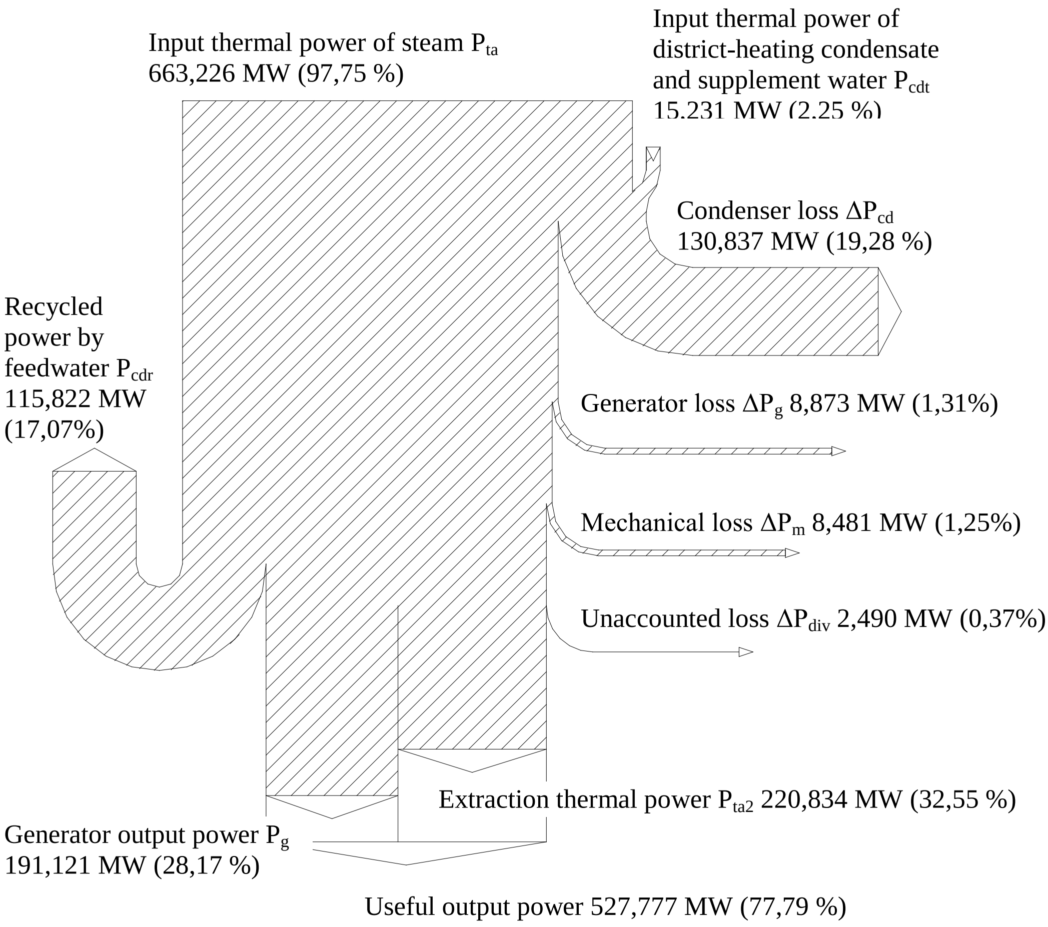

Fig. 5. Sankey diagram of heat balance for 13 CK 240 turbine (district-heating)

Heat balance analysis of turbine 13 CK 240 highlights the following:

- Using cogeneration, useful heat output reaches 77.79% of input thermal power, which is a good value in terms of energy use;

- The mechanical and generator loss remained relatively constant;
- Condenser loss was halved, because here comes a lower steam flow, and therefore waste heat from steam to cooling water is lower. Steam does not reach the condenser, is useful and is used for heating. As a result, generator output power is lower since part of the steam is no longer expanding in turbine.

Advantages of cogeneration are reflected in the data in Table 8, which noted:

- Internal efficiency in high, medium and low pressure cylinders are close to those in condensation system, but internal equivalent efficiency felt with 1.95%.
- Thermal efficiency of cycle η_t is 14.74 % higher, reaching 60.99 %, within usual limits as stated in literature for cogeneration.
- Actual efficiency of unit η_{ea} increased by 12.85 %, and the gross heat q_{bc} and fuel consumption b_{bc} decreased, with 0.66 $kJ \cdot kJ^{-1}$ for q_{bc} reaching 2.215 $kJ \cdot kJ^{-1}$ and b_{bc} reaching 0.272 kg c.c.$\cdot kWh^{-1}$, resulting substantial savings in heating period, that could reach to 0.081 x 191,121 x 180 days x 24 hours = 66877.06 t c.c., if we consider six months of heating.

4. COMPARISON OF TUBINES

Following the analysis of actual balances, the K-210-130-1 and 13 CK 240 turbines can be compared to choose the best value energy point of view turbine.

Comparisons will be made both on energy indicators (Table 9) obtained by calculations and by comparing the heat balance for the two turbines (Table 3 and Table 4) in condensing system.

Table 9 **Comparison of energy indicators**

Item	**Description**	**U.M.**	**Value**	
			K-210-130-1	**13 CK 240**
1	Theoretical expansion in high-pressure body H_{tip}	$kJ \cdot kg^{-1}$	440.29	447.55
2	Internal expansion in high-pressure body H_{iip}	$kJ \cdot kg^{-1}$	399.5	338.55
3	Internal efficiency in high-pressure body η_{iip}	%	90.74	75.65
4	Theoretical expansion in medium-pressure body H_{tmp}	$kJ \cdot kg^{-1}$	807.22	824.19
5	Internal expansion in medium-pressure body H_{imp}	$kJ \cdot kg^{-1}$	750.05	688.99
6	Internal efficiency in medium-pressure body η_{imp}	%	92.92	83.60
7	Theoretical expansion in low-pressure body H_{tjp}	$kJ \cdot kg^{-1}$	513.24	486.40
8	Internal expansion in low-pressure body H_{ijp}	$kJ \cdot kg^{-1}$	503.02	449.96
9	Internal efficiency in low-pressure body η_{ijp}	%	98.00	92.50
10	Theoretical expansion in turbine H_h	$kJ \cdot kg^{-1}$	1713.85	1736.55
11	Generator efficiency η_g	%	96.36	96.14
12	Mechanical efficiency η_m	%	99.99	99.99
13	Equivalent internal efficiency of expansion η_i	%	96.42	85.08
14	Thermal efficiency of cycle η_t	%	46.25	42.88
15	Total effective efficiency η_{ea}	%	43.07	39.63

Item	Description	U.M.	Value	
			K-210-130-1	**13 CK 240**
16	Gross specific heat consumption q_{bc}	$kJ \cdot kJ^{-1}$	2.875	3.125
17	Specific consumption of conventional fuel b_{bc}	kg c.c.· kWh^{-1}	0.353	0.384
18	Specific energy of steam at turbine inlet	$kJ \cdot kg^{-1}$	1322	1192.44

Analyzing the data in Table 9 can draw the following conclusions:

- Although the theoretical expansion in the high pressure body has similar values for the two turbines, internal expansion is higher for 13 CK 240 turbine, therefore. The internal efficiency of high pressure body is higher with 15.09%. This is due to larger number stages in the high pressure body of 13 CK 240 turbine;

- In the medium pressure body the situation is the same, the efficiency of 13 CK 240 turbine is higher with 9.32%, and in the low pressure body is 5.5% higher;

- Generator and mechanical efficiency of turbines is fairly equal as expected that turbines are very similar in terms of construction and relatively close in terms of powers;

- Equivalent internal efficiency of expansion η_i of 13 CK 240 turbine is 11.34% higher since the turbine above has a better efficiency in every body;

- The thermal efficiency η_t of the turbine 13 CK 240 cycle is 3.37% higher and also the actual efficiency η_{ea} of unit is 3.44% higher.

- In terms of consumption 13 CK 240 turbine is more efficient. Gross heat consumption q_{bc} is lower with 0.25 $kJ \cdot kJ^{-1}$ and specific fuel consumption of conventional fuel is 0.031 (kg c.c.)$\cdot kWh^{-1}$ less, the resulting fuel economy per year is 63,822.57 t c.c. assuming that the turbine will operate a total of 8760 $h \cdot year^{-1}$.

5. CONCLUSION

Comparison of heat balance for the two turbines emphasizes better efficiency of the turbine 13 CK 240 and the following categories will be highlighted:

- Generator power output P_g is 3.33% higher for 13 CK 240 turbine when compared to input steam thermal power.

- Recycled power by feedwater P_{cdr} is also lower by 1.23%.

- Condenser loss ΔP_{cd} is 3.29% lower for 13 CK 240 turbine.

In conclusion, the 13 CK 240 turbine is more energy efficient than K-210-130-1 turbine.

This result is because the 13 CK 240 turbine is the latest design with some improvements in terms of construction versus older turbine as described above.

If the K-210-130-1 turbine has bleeders, designed to operate in condensing system but adapted for district-heating by taking needed steam for heating from bleeders, 13 CK 240 has extractions designed from the start for combined heat and power production.

Therefore, it can be concluded that it is desirable to use the turbine 13 CK 240 instead of K-210-120-1 in the thermal power plant, given the high energy features in condensing and district-heating system.

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