Efficiency Assessment of Condensing Steam Turbine

DOSA ION,
Department of Mechanical Engineering,
Industrial Engineering and Transportation
University of Petrosani
Address Str. Universitatii nr. 20, 332006, Petrosani, jud. Hunedoara
ROMANIA
i_dosa@hotmail.com http://www.upet.ro

PETRILEAN DAN CODRUT
Department of Mechanical Engineering,
Industrial Engineering and Transportation
University of Petrosani
Address Str. Universitatii nr. 20, 332006, Petrosani, jud. Hunedoara
ROMANIA
dcpetrilean@yahoo.com http://www.upet.ro

Abstract: The paper presents energetic characteristics of K 200-130-1 steam turbine after operating at different loads. Turbine energy efficiency has an significant impact on overall steam power plant efficiency. The K 200-130-1 steam turbine's rated thermal efficiency is 44.7 %. The turbine is operational since 1968, therefore an efficiency assessment can highlight its present technical condition.

Key-Words: Steam turbine, energy performance, efficiency assessment, heat balance

1 Introduction

The steam turbine converts the thermal energy of pressurized steam into useful mechanical work, driving the electrical generator; therefore its efficiency has a major impact on the amount of electricity produced and in the end on overall power plant efficiency.

As energy demand rises constantly over the last decades, energy efficiency is an important aspect of modern economy. High efficiency power generation can reduce the primary energy consumption, meeting Directive 2012/27/EU on energy efficiency which emphasized the need to increase energy efficiency in order to achieve the objective of saving 20 % of the Union's primary energy consumption by 2020 compared to projections [1].

Typical combustion turbine heat rates are 10,181–10,972 $kJ \cdot kWh^{-1}$ (33-35% efficient higher heating value) [2].

Bhatt and Rajkumar [3] present the results of study on performance enhancement of 22 coal fired thermal power stations with capacities from 30 to 500 MW. The oldest, 30 MW units have served for over 30yr and the newer 500 MW units, have been in operation for a shorter period of time. Turbine efficiencies are in the range 31.00 to 41.90% as compared to the design range of 34.80–43.97%, while isentropic efficiencies ranged 74.13–86.40% as compared to values of 83.20–89.10%.

Since K 200-130-1 steam turbine is operational since 1968 results presented in [3] are relevant for this paper, providing a term of comparison for turbine efficiency.

Paper [4] shows that for turbines running for over 30 yr, expected efficiency degradation is approximately 5%, therefore the turbine analyzed in present paper is expected to have an efficiency exceeding 29%.

2 Problem Presentation

Steam turbines are complex equipment, with a long life-cycle; as a result, many of them are still in use [3] after more than 30 yr of service.

Technology improvements driven by need of higher efficiency led to a new generation of steam turbines.

Since a steam turbine is costly equipment a decision must be taken regarding replacing or retrofitting after careful consideration of actual technical condition and efficiency.

First step in decision making is assessment of energy performance of steam turbine, which requires heat balance calculations.

ISBN: 978-1-61804-195-1

Energy auditing in Romania is regulated by the state and supervised by the regulatory authority ANRE, and must be carried out according to the published guide [5].

Heat balance calculations examples for various installations and equipments can be found in literature [6][7].

2.1 A brief presentation of steam turbine

K 200-130-1 [8] steam turbine is a condensing type turbine and was designed to operate at 3,000 rpm, 13 MPa, and 545 °C with one steam reheat to a temperature of 545 °C at a pressure of 2.44 MPa. The exhaust pressure is 0.0034 MPa. The turbine has seven bleeder connections for regenerative feed water heating to a maximum of 242 ºC. From the High Pressure Turbine (HPT) steam is directed to reheater at a pressure of 2.89 MPa and a temperature of 350 ºC from which is returned to the Reheat Turbine (RT). The Low Pressure Turbine (LPT) is of a double-flow design.

Steam for turbine is provided by Pp-330/140-P55 type steam generator, a once-through coal-fired boiler [8].

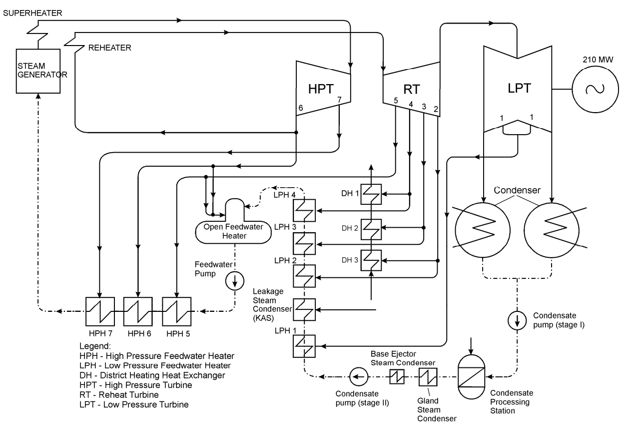

Fig. 1. Schematic diagram of 200 MW unit [8]

Construction of the steam generator is carried out in two distinct bodies, symmetrical with the axis of the group, operating in parallel to the K-200-130-1 steam turbine. The steam output of generator (one body) is 330 $t{\cdot}h^{-1}$, at a pressure of 140 bar and 550 °C for live steam and 24.4 bar at 550 ºC temperature for reheat steam.

Feed water parameters at steam generator rated load are: pressure 188 bar, temperature 242 °C.

The unit is equipped with an electric generator having 210 MW output power at 15.75 kV and 0.85 power factor.

Fig. 1 presents the schematic diagram of 200 MW unit. At the time of construction the steam turbine was used only for electricity production. Later, as nearby city grew bigger, 3 heat exchangers where added in order to provide district heating. The steam required for water heating is drawn from bleeder 4, 3 and 2.

2.2 Balance outline

Balance outline consist in: the generator terminals for electricity output; main flow control valve of turbine and ramification of parallel pipeline which draws steam from turbine for technological purpose to the de-aerator pressure reduction and cooling station abbreviated SRRD, on steam side; outlet of HPH 7 and inlet of demineralized makeup water on water side; inlet sections of cooling water used in condenser, electrical generator and turbine lubricating oil system.

Balance outline contains the steam turbine, the condenser, the regenerative cycle, the regenerative feed water heaters; feed water and condensate pumps.

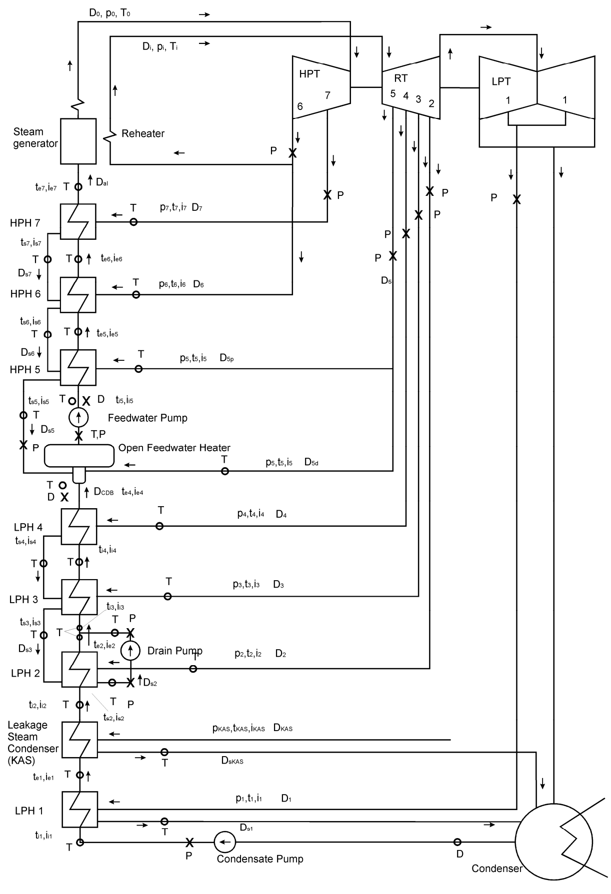

Fig. 2. Schematic diagram illustrating condensate, drain and feed water flow

2.3 Measured data

As regulations require, for heat balance calculations measurements must be carried out for at least 3 different loads. The loads for performance tests were fixed to 460 $t \cdot h^{-1}$ - 70%, 560 $t \cdot h^{-1}$ – 85% and 320 $t \cdot h^{-1}$ – 94%. Rate of steam flow to the turbine D_0 is 660 $t \cdot h^{-1}$, at rated load.

Old systems for the management and control in power plant were replaced with the new computing Distributed Control System (DCS) similar with one described in paper [9]. In addition, measuring equipment was placed in different locations, symbolized with **x** in Fig. 2. Also data from indicator panels located in the control room of the unit can be obtained.

After analyzing data available from DCS and measuring equipment in place, points for measuring additional data were chosen, symbolized with **O** in Fig. 2. As an adequate amount of data was available from DCS regarding temperature pressure and rate flow of feed water through HPH 5, 6 and 7, condensate at outlet of LPH 4, measuring additional data in order to calculate the steam flow rate of bleeders from their heat balance has been decided.

Steam required by de-aerator (open feed water heater) was drawn off from bleeder no. 5.

3 Results obtained

In order to increase the accuracy of flow rate computing for bleeders, flow rate of condensate at the outlet of steam condenser was measured with Flexim ultrasonic clamp-on flow meter.

As ambient and cooling water temperature was high, flow rate of cooling water was increased on average with 10% for generator cooling and lubricating oil system cooling and with 5.2 to 7.2% for condenser.

Table 1. Turbine performance characteristics

Nom.	Load		
	70%	85%	94%
High pressure turbine			
Theoretical enthalpy drop H_{tip}, $kJ \cdot kg^{-1}$	542.65	496.85	483.34
Actual enthalpy drop H_{iip}, $kJ \cdot kg^{-1}$	361.80	362.24	359.15
Isentropic efficiency η_{iip}, %	66.67	72.91	74.31
Reheat or intermediate pressure turbine			
Theoretical enthalpy drop H_{tmp}, $kJ \cdot kg^{-1}$	745.05	761.74	797.24
Actual enthalpy drop H_{imp}, $kJ \cdot kg^{-1}$	593.21	615.47	688.94
Isentropic efficiency η_{imp}, %	79.62	80.80	86.42
Low pressure turbine			
Theoretical enthalpy drop for H_{tjp}, $kJ \cdot kg^{-1}$	492.69	473.11	438.99
Actual enthalpy drop H_{ijp}, $kJ \cdot kg^{-1}$	400.84	350.81	309.65
Isentropic efficiency η_{ijp}, %	81.36	74.15	70.53
Theoretical enthalpy drop in turbine H_h, $kJ \cdot kg^{-1}$	1,388	1,371	1,376
Electrical generator efficiency η_g, %	98.74	98.71	98.75
Mechanical efficiency η_m, %	98.63	98.82	99.06
Turbine isentropic efficiency η_i, %	63.39	62.48	64.56
Thermal efficiency η_t, %	40.78	40.12	41.08
Actual efficiency of turbine-generator aggregate η_{ea}, %	39.71	39.13	40.16
Specific heat consumption q_{bc}, $kJ_{th} \cdot kJ_e^{-1}$	2.906	2.894	2.839
Specific fuel consumption b_{bc}, (kg e.f.) $\cdot$ kWh^{-1}	0.357	0.355	0.349
Specific energy of main steam e_{sp}, $kJ \cdot kg^{-1}$	1,132	1,099	1,123
Heat rate, $kJ \cdot kWh^{-1}$	9,065	9,199	8,959
Steam rate ***d***, $kg \cdot kWh^{-1}$	3.392	3.496	3.416

Table 2. Actual hourly energy balance 70% load

INPUT		
Nom.	**MWh**	**%**
Energy of steam (main and reheat) P_{ta}	488.046	99.97
Energy of makeup water P_{aad}	0.153	0.03
TOTAL INPUT	**488.199**	**100.0**
OUTPUT		
USEFUL OUTPUT		
Output power P_g	144.839	29.67
Energy recovered in regenerative cycle P_{cdr}	123.324	25.26
Energy of steam extracted for technological use P_{SRRD}	4.432	0.91
TOTAL USEFUL	**272.595**	**55.84**
LOSSES		
Mechanical loss ΔP_m	2.044	0.42
Generator loss ΔP_m	1.842	0.38
Heat rejected by condenser P_{cd}	209.291	42.87
Unaccounted losses ΔP_{div}	2.426	0.50
TOTAL ENERGY LOSS	**215.604**	**44.16**
TOTAL OUTPUT	**488.199**	**100.0**

Table 3. Actual hourly energy balance 85% load

INPUT		
Nom.	**MWh**	**%**
Energy of steam (main and reheat) P_{ta}	608.935	99.87
Energy of makeup water P_{aad}	0.777	0.13
TOTAL INPUT	**609.712**	**100.0**
OUTPUT		
USEFUL OUTPUT		
Output power P_g	175.168	28.73
Energy recovered in regenerative cycle P_{cdr}	161.330	26.46
Energy of steam extracted for technological use P_{SRRD}	4.962	0.81
TOTAL USEFUL	**341.460**	**56.00**
LOSSES		
Mechanical loss ΔP_m	2.106	0.35
Generator loss ΔP_m	2.289	0.38
Heat rejected by condenser P_{cd}	259.989	42.64
Unaccounted losses ΔP_{div}	3.868	0.63
TOTAL ENERGY LOSS	**268.252**	**44.00**
TOTAL OUTPUT	**609.712**	**100.0**

Table 4. Actual hourly energy balance 94% load

INPUT		
Nom.	**MWh**	**%**
Energy of steam (main and reheat) P_{ta}	659.249	99.92
Energy of makeup water P_{aad}	0.545	0.08
TOTAL INPUT	**659.794**	**100.0**
OUTPUT		
USEFUL OUTPUT		
Electric output power P_g	193.599	29.34
Energy recovered in regenerative cycle P_{cdr}	177.483	26.90
Energy of steam extracted for technological use P_{SRRD}	5.413	0.82
TOTAL USEFUL	**376.495**	**57.06**
LOSSES		
Mechanical loss ΔP_m	1.861	0.28
Generator loss ΔP_m	2.446	0.37
Heat rejected by condenser P_{cd}	274.059	41.54
Unaccounted losses ΔP_{div}	4.934	0.75
TOTAL ENERGY LOSS	**283.299**	**39.18**
TOTAL OUTPUT	**659.794**	**100.0**

Optimal energy balance was computed for the optimal steam flow rate of 634 $t \cdot h^{-1}$, value provided by the manufacturer of the turbine. Therefore, some results obtained for 94% load representing 620 $t \cdot h^{-1}$, are expected to be close to the optimal values. Other values used in calculus are those provided by the manufacturer as rated values.

Table 5. Optimal hourly energy balance

INPUT		
Nom.	**MWh**	**%**
Energy of steam (main and reheat) P_{ta}	669.149	100.0
Energy of makeup water P_{aad}	0.0	0.0
TOTAL INPUT	**669.149**	**100.0**
OUTPUT		
USEFUL OUTPUT		
Output power P_g	210.0	31.38
Energy recovered in regenerative cycle P_{cdr}	183.144	27.37
Energy of steam extracted for technological use P_{SRRD}	6.410	0.96
TOTAL USEFUL	**399.554**	**59.71**
LOSSES		
Mechanical loss ΔP_m	10.467	1.56
Generator loss ΔP_m	3.111	0.46
Heat rejected by condenser P_{cd}	261.675	39.11
Unaccounted losses ΔP_{div}	-5.658	-0.85
TOTAL ENERGY LOSS	**269.595**	**39.18**
TOTAL OUTPUT	**669.149**	**100.0**

Table 6. Turbine performance characteristics for optimal energy balance

Nom.	Optimal
Theoretical enthalpy drop for HPT H_{tip}, $kJ \cdot kg^{-1}$	443.39
Actual enthalpy drop for HPT H_{iip}, $kJ \cdot kg^{-1}$	339.76
HPT isentropic efficiency η_{iip}, %	76.63
Theoretical enthalpy drop for RT H_{tmp}, $kJ \cdot kg^{-1}$	800.31
Actual enthalpy drop for RT H_{imp}, $kJ \cdot kg^{-1}$	689.53
RT isentropic efficiency η_{imp}, %	86.16
Theoretical enthalpy drop for LPT H_{tjp}, $kJ \cdot kg^{-1}$	567.95
Actual enthalpy drop for LPT H_{ijp}, $kJ \cdot kg^{-1}$	506.18
LPT isentropic efficiency η_{ijp}, %	89.12
Theoretical enthalpy drop in turbine H_h, $kJ \cdot kg^{-1}$	1.486
Electrical generator efficiency η_g, %	98.54
Mechanical efficiency η_m, %	95.32
Turbine isentropic efficiency η_i, %	73.40
Thermal efficiency η_t, %	46.00
Actual efficiency of turbine-generator aggregate η_{ea}, %	43.21
Specific heat consumption q_{bc}, $kJ_{th} \cdot kJ_e^{-1}$	2.521
Specific fuel consumption b_{bc}, (kg e.f.) $\cdot$ kWh^{-1}	0.310
Specific energy of main steam e_{sp}, $kJ \cdot kg^{-1}$	1,192
Heat rate, $kJ \cdot kWh^{-1}$	8,332
Steam rate ***d***, $kg \cdot kWh^{-1}$	3.214

Table 7. Turbine efficiency comparison

Nom.	Load			Optimal	Expected
	70%	85%	94%		
HPT	66.7	72.9	74.3	76.63	0.72÷0.80
RT	79.6	80.8	86.4	86.16	0.80÷0.90
LPT	81.4	74.2	70.5	89.12	0.70÷0.78

4 Conclusions

Analyzing data in Table 1, values for mechanical efficiency η_m are found in the range of 98.63 to 99.06 higher than expected 95.32%. The reason why these unusual values occur results from the way they were calculated, from the heat balance of the turbine lubricating oil cooler. Since friction losses are unlikely to be smaller than rated, therefore must be a problem in the lubricating oil system and the oil is inadequately cooled. But heat resulting from friction in turbine bearings must be discharged, otherwise they overheat and the turbine will shut down.

At the time of measurements, turbine outer casing was removed and fans were used to blow air in order to cool the bearings and prevent turbine to shut down. This observation suggested at the time, that there was a problem with the cooling of bearings and is consistent with data obtained from measurements analyzed above.

Since mechanical efficiency is used to compute other efficiency characteristics, an inappropriate value can affect results. Therefore for other calculations 95% value for mechanical efficiency was set.

Values for generator efficiency are in range as for TVV 200-2-A and TVV 200-2-A Y3 electrical generator the rated efficiency is 98.6% [8].

Notes on comparison data in Table 1 to 7 can be summarized as follows:

1. Isentropic efficiency η_{iip} for HPT is 66.67%, 72.91% and 74.31% for loads of 70%, 85% and 94% compared to optimal 76.63%. Isentropic efficiency η_{imp} for RT is 79.62%, is 80.80% and 86.42% compared to optimal 86.16 %. Isentropic efficiency η_{ijp} for LPT is 81.36%, 74.15% and 70.53% compared to optimal 89.12 %.

Values for isentropic efficiency are in the typical range, excepting values of HPT for 70% load. For HPT isentropic efficiency is directly proportional to electric output power, while for LPT efficiency decreases as heat rejected by condenser increases.

2. The amount of heat rejected by condenser represents the greatest loss as expected, having values of 209.291 MWh (42.87%), 259.989 MWh (42.64 %) and 274.059 MWh (41.54%), for loads of 70%, 85% and 94%, compared to optimal 261.675 MWh (39.11%).

3. Electrical efficiency is 29, 67 %, 28, 73% and 29, 34 %, for loads of 70%, 85% and 94%, compared to optimal 31, 38 %.

An important aspect of turbine functioning must be highlighted here, namely exhaust pressure at the outlet of LPT. It's a known fact that high exhaust pressure has a negative influence on work done by the turbine and its output power. At the time of measurements condenser cooling water temperature at condenser inlet was 26 °C, 26.19 ºC and 27.9 °C. Condenser pressure was 0.073 bar for 70% load, 0.09 bar for 85% load and 0.091 bar for 94% load with corresponding temperatures of exhaust steam: 42.35 °C, 46.26 °C, 45.84 °C. For comparison rated steam exhaust parameters at LPT are supposed to be 0.03547 bar and 27 °C.

Loss of power due to actual steam exhaust conditions were evaluated, computing energy balance for the same isentropic efficiency of LPT, while increasing power output and assuming rated exhaust conditions. Results highlighted that lost power was 5% for 70% load, 5.9% for 85% load and 5.5% for 94% load.

Influence of exhaust conditions on isentropic, thermal and electrical efficiency are presented in Fig. 3.

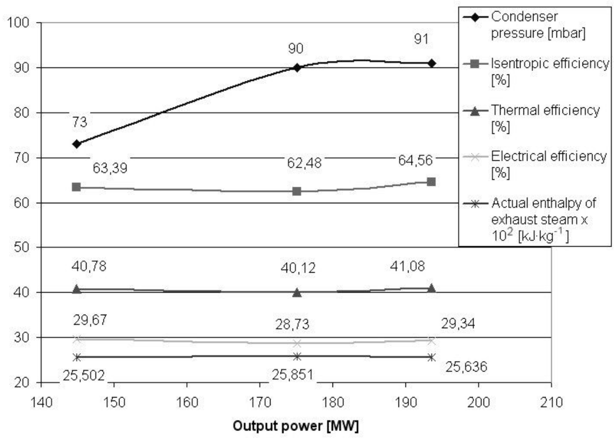

Fig. 3. Influence of exhaust conditions

There is an obvious relation between poorer exhaust conditions having the highest enthalpy (2,585.1 $kJ \cdot kg^{-1}$) and the minimum values of isentropic, thermal and electrical efficiencies corresponding to that point.

Analyzing the energy efficiency parameters in Table 1 compared with data in Table 6, remarks can be summarized:

Specific heat consumption was 2.906, 2.894 and 2.839 $kJ_{th} \cdot kJ_e^{-1}$, for 70, 85 and 94 % load compared to 2.521 $kJ_{th} \cdot kJ_e^{-1}$ optimal;

Specific fuel consumption was 0,357, 0,355 and 349 (kg e.f.)·kWh^{-1} for 70, 85 and 94 % load compared to 0.310 (kg e.f.)·kWh^{-1} optimal;

Heat rate was 9,065, 9,199 and 8,959 kJ·kWh^{-1} for 70, 85 and 94 % load compared to 8,332 kJ·kWh^{-1} optimal.

Leĭzerovich in [10] reveals that K 200 series turbines are the typical turbines of 1960s. Rated thermal efficiency for the 210 MW output turbine was 44.7% and heat rate of 8,045 kJ·kWh^{-1}, but current efficiencies for turbines in use are between 41.7 to 42.57% corresponding to hate rates of 8,632 to 8,457 kJ·kWh^{-1}. Values obtained as a result of energy balance calculations are slightly worse than expected due to actual operating conditions.

Before enhancing efficiency of turbine, problems with the lubricating oil system must be corrected.

More important from efficiency point of view is to restore the optimal pressure in condenser improving its heat transfer capability by maintaining heat exchange surfaces clean. At the same time actions must be considered in order to improve the sealing of condenser to prevent air infiltration, and a proper maintenance of ejectors to guarantee the vacuum.

Applying measures to improve efficiency will lead in the end to thermal and electrical efficiency values close to optimal, but modern supercritical steam turbines have higher efficiency.

Conclusively, to improve energy efficiency of studied turbine taking into account its extended operating time must be done by retrofitting. Retrofitting is required also as the unit supplies district heating and hot water.

Retrofitting older steam turbines is a widespread practice [11] [12], performed usually by the manufacturer of equipment [13]. Benefits of retrofitting are higher power output, over 10 MW, increased efficiency by 5-7% [13] and reduced consumption of fuel to 26.1 g · kWh^{-1} [11].

A good experience in this direction is the retrofitting of K 200-130-1 turbine at Deva power plant, Romania [14]. Scope of turbine retrofit includes among others: new HP (high pressure) and IP (intermediate pressure) turbines with steam admission systems, LP (low pressure) turbine retrofit, adaptation of the turbine to district heating mode of operation as well as control and safety systems modernization. After retrofitting heat rate of the new modernized 13K215 turbine was 7,765 kJ·kWh^{-1}, electric efficiency 33.88% in condensing operation and cogeneration efficiency was 63.65%.

Unfortunately recent economical crisis halted the efforts of modernization in power industry. This paper intended to highlight the benefits of retrofitting in power industry hoping that these efforts will be resumed as soon as possible.

References:

[1] http://ec.europa.eu/energy/efficiency/eed/eed_en.htm.

[2] http://www.npc.org/study_topic_papers/4-dtg-electricefficiency.pdf.

[3] Bhatt, M. S. and Rajkumar, N. (1999), *Performance enhancement in coal fired thermal power plants. Part II: steam turbines.* Int. J. Energy Res., 23: 489–515. doi: 10.1002/(SICI)1099-114X(199905)23:6<489::AID-ER494>3.0.CO;2-T

[4] http://www.geindustrial.com/publibrary/checkout/GEA-13450A?TNR=Service%20and%20Maintenance|GEA-13450A|generic

[5] ***, *Guide to development and analysis of energy balance.* M.O. of Romania, part.I, nr.792/11.11.2003.

[6] I.Gh. Carabogdan, et al., *Energy Balances - problems and applications*. Tehnica Publishing House, Bucharest, (1986).

[7] T. Berinde, et al., *Elaboration and analysis of energy balance in the industry*. Tehnica Publishing House, Bucharest, (1976).

[8] ***, *Technical Instructions and Operation Manual for K 200-130-1 turbine.*

[9] e2010.drustvo-termicara.com/papers/download/107

[10] Leizerovich, A. Sh., Steam turbines for modern fossil-fuel power plants, ISBN 0-88173-548-5, Published by The Fairmont Press, Inc., 2008.

[11] http://energyland.info/news-print-80620

[12] http://www.turboatom.com.ua/en/press/news/1996.html

[13] www.myshared.ru/slide/46040/

[14] Bugała, G. Karpiuk, R. Kietliński, K. Stasik, *B. Retrofitting turbine K 200-130-1 turbine at Deva power plant*, Scientific Papers of the University of Warsaw. Conferences, 5 Research Conference on Thermal Energy Problems / symposium (V, 04-07.12.2001, Warsaw, Poland), vol. Z 21, pg. 39-48.