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APPLICATION OF SELECTED BALANCE METHODS TO THE EVALUATION OF PERFORMANCE PARAMETERS OF THE EXTRACTION BACK-PRESSURE POWER UNIT IN A COMBINED HEAT AND POWER PLANT

SUMMARY

Thermal measurements of the extraction back-pressure power unit in the combined heat and power plant for a wide range of working loads were utilised to develop an algorithm for computing performance parameters of the power unit. The calculation procedures use the enthalpy balance method and the entropy balance method. Adequacy of the two methods was compared on the basis of performance characteristics of the investigated power unit.

Keywords: *combined heat and power plant, extraction back-pressure power unit, thermal efficiency, entropy balance*

ZASTOSOWANIE WYBRANYCH METOD BILANSOWYCH DO OCENY EFEKTYWNOŚCI PRACY BLOKU UPUSTOWO-PRZECIWPRĘŻNEGO ELEKTROCIEPŁOWNI

Na podstawie pomiarów cieplnych bloku upustowo-przeciwprężnego elektrociepłowni, przeprowadzonych w szerokim zakresie obciążeń eksploatacyjnych, opracowano algorytm obliczeń podstawowych parametrów charakteryzujących pracę bloku. W tym celu przeprowadzono wymagane obliczenia, wykorzystując zasady entalpowej i entropowej metody bilansowej. Następnie dokonano analizy porównawczej przydatności obu metod do sformułowania poprawności oceny efektywności pracy badanego bloku.

Słowa kluczowe: *elektrociepłownia, blok upustowo-przeciwprężny, sprawność termiczna, bilans entropii*

1. INTRODUCTION

Performance parameters of a power unit in a combined heat and power plant are typically obtained experimentally. The evaluation of a power unit's performance in a thermal power plant shall be treated as a specific case on account of cogeneration of electricity and heat. The energy analysis of the extraction condensing type power unit is provided in (Tokarz 2006). This study focuses on the evaluation of effectiveness of an extraction back-pressure unit. In the first stage thermal measurements were taken of the power unit BC-100 in the combined heat and power plant "Elektrociepłownia Kraków S. A.", under a wide range of working loads and for the for the heating and pseudo-condensing modes of operation. Two evaluation methods were applied:

1) enthalpy method,
2) entropy method.

The enthalpy method, which is a traditional approach, allows for evaluation of performance of a back-pressure unit through finding the conversion factors for chemical energy of fuels and the efficiency profiles of the investigated unit. This study is extended to cover the entropy method, providing a qualitative approach as the control of entropy generation in particular nodes of the power unit enables us to identify, locate and evaluate the installations in which to investigate the back-pressure unit performance.

Measurement data and investigation results are presented in more detail elsewhere (Karpiel 2003).

2. IDENTIFICATION OF CONTROL POWER UNIT

The control object is the back-pressure power unit BC-100 type, one of the four power units in the combined heat and power plant and one of the two heating units operated in the conditions of the peak demand for district heating. Installations of the power unit no 4 BC-100 type in the heat and power plant "Elektrociepłownia Kraków S.A." include (see Fig. 1):

– Radiant, two-pass, single-drum, natural circulation steam boiler, fired with coal fines.
  Nominal parameters of the boiler:
  • rate of fresh steam production: 430 t/h,
  • fresh steam pressure: 13.53 MPa,
  • fresh steam temperature: 813 K (540°C).
– Turbine 13UP-110 type – axial, tandem-compound back-pressure type impulse steam turbine; the turbine has a high-pressure section HP and low-pressure section LP, with two district heating bleeds, bleed control on the outlet of turbine cylinder and five bleeds for regenerative heating of the feed water.

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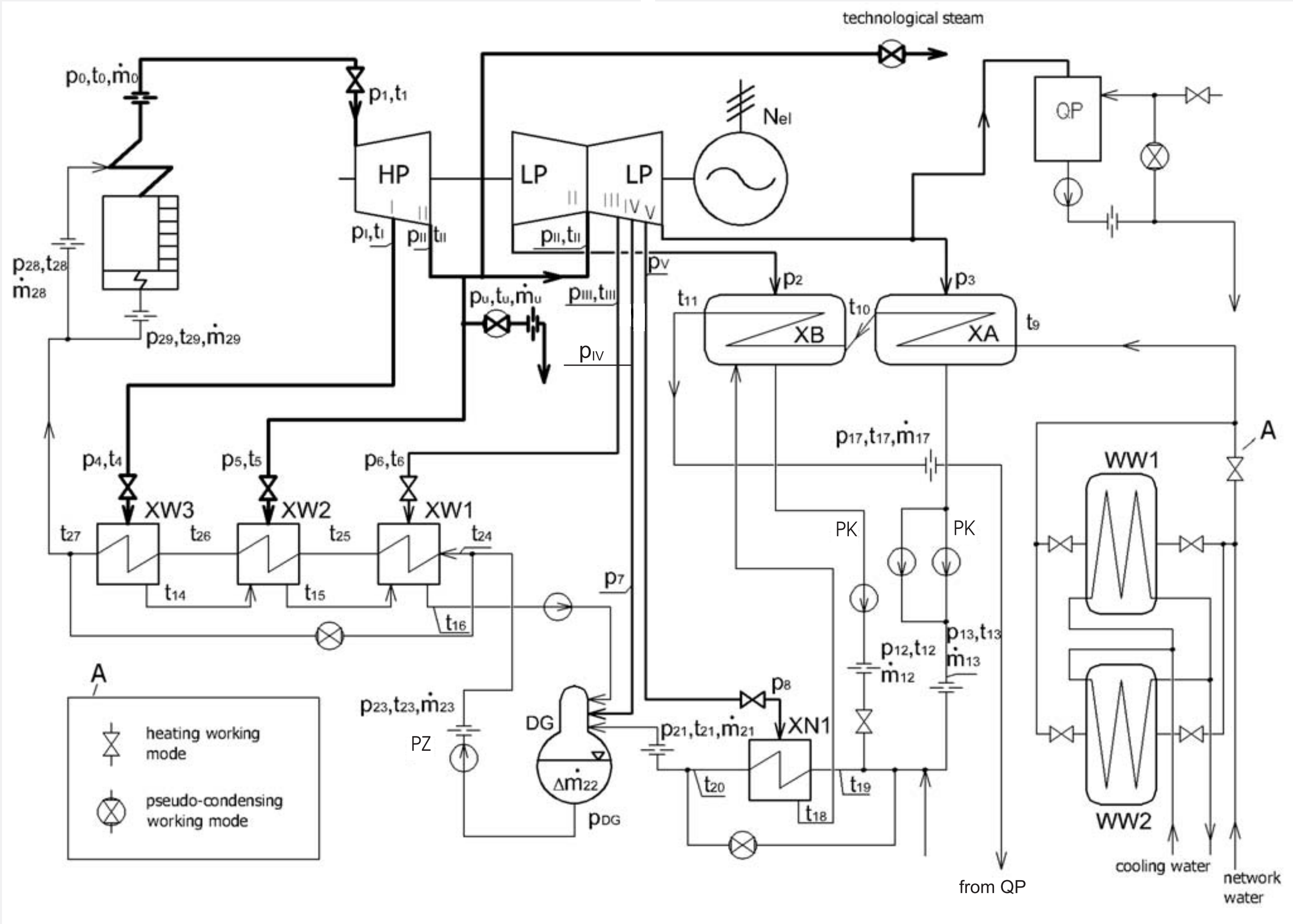

Fig. 1. Power unit BC-100 – schematic flow diagram with measurement points distribution

Nominal parameters of the turbine:
- turbine speed: 3000 rpm,
- fresh steam pressure, nominal rate: 12.75 MPa,
- fresh steam temperature, nominal rate: 808 K (535°C),
- potential thermal power: 191 MW,
- flow rate of heating network water: 1305 kg/s.

– Synchronous generator TGH-120 with hydrogen cooling:
- electric power, nominal rate: 120 MW,
- voltage, nominal rate: 13.8 kV.

– Regeneration system with preheaters: low-pressure exchanger XN1 and high-pressure exchangers: XW1, XW2, XW3.
– Degasifier DG with a supply tank.
– Feed pump PZ and condensate pumps PK.
– Heating exchangersXA and XB.

BC-100 unit can be used for cogeneration of electricity and heat. Two operating modes are available:
1) heating working mode,
2) pseudo-condensing working mode.

In the **heating mode** a stream of superheated steam leaving the boiler {I} is fed to the blades of the high-pressure turbine section – HP-turbine, where it is expanded to the pressure and temperature {II}. Steam passes then to the low--pressure turbine section – LP-turbine, where it is further expanded to reach the parameters given in {2} and {3}. In the turbine energy contained in steam is converted into mechanical energy and then converted into electric energy in the generator.

There are several bleeds in the turbine: three bleeds to high-pressure regeneration heat exchangers XW1, XW2, XW3, a bleed to the degasifier, a bleed to low-pressure regeneration heat exchanger XN1, a bleed for municipal customers and a bleed for other processes involved in electricity and heat generation. The main stream of steam {2} and {3} will condense in heat exchangers, transferring heat to the stream of district heating water. Condensate with the stream of water replacing the extracted water is pumped to the boiler via regenerative preheaters.

In the **pseudo-condensing mode** the flow of water for district heating is arranged such that thermal power contained in network water leaving the heat exchangers XA, XB is not absorbed by municipal customers but transferred to cooling water in coolers WW1-WW2 – water-water exchangers. Water cooling the exchangers WW transfers its thermal power in the cooling tower. Under these conditions the power unit operates as a conventional condensing power plant with lesser vacuum and an extended cooling system.

3. METHODOLOGY

Parameters of water and steam during measurements:
– generator loading:
- in the heating mode of operation: from 79.1 to 100.2 MW,
- in the pseudo-condensing mode: from 76.9 to 96.1 MW,

– inlet steam pressure to the turbine: from 13 339 to 13 767 kPa,

– inlet steam temperature to the turbine: from 783 to 801 K (from 510 to 528°C),
– inlet water temperature to the heater XA:
  • in the heating mode: from 322.1 to 335.0 K (from 49.1 to 62.0°C),
  • in the pseudo-condensing mode: from 337.9 to 344.1 K (from 64.9 to 71.1°C),
– flow rate of network water (nominal value): 1305.6 kg/s
  • in the heating mode: from 1490.2 to 1552.6 kg/s,
  • in the pseudo-condensing mode: 1759.4 to 1792.4 kg/s.

During measurements steam pressure at the inlet to the turbine was higher from the nominal value though the admissible level p_{dop} = 12 750 ± 1275 kPa was not exceeded. Steam temperature was a little lower than the nominal value and exceeded the admissible level t_{dop} = 535°C, the deviation value ranging from – 8°C to +15°C. The fact that the incoming steam was not hot enough might be attributable to the admission of large amounts of injected water as the admissible temperature levels are exceeded in the steam superheater.

The fluxes of steam and condensate were determined by measurements taken at points designated on the schematic diagram, the remaining streams of bleed steam were derived from the energy balance formulas and mass balance for the power unit. It is assumed that the loss of medium during the cycle equals the loss of water in the supply tank (degasifier), assuming that 40% of losses occur in the boiler and 60% – in the turbine and pipelines (35% – HP, 35% – LP, 30% – pipelines).

Basing on separated measurement data compiled in (Karpiel 2003) boiler thermal power ranging from 230–310 MW, the efficiency of the boiler η_k was calculated by the indirect method and it is approximated (at the correlation factor 0.98) by the linear equation:

$$\eta_k = -0.002639 \cdot \dot{Q}_0 + 91.6802,$$

where $\dot{Q}_0$ – thermal power of boiler.

4. THERMODYNAMIC EVALUATION OF POWER UNIT PERFORMANCE

Performance of a power unit in a thermal power plant should be verified by different methods to enable the evaluation of performance parameters characterising the unit operation. The quality factor widely applied to evaluate the performance of a back-pressure unit in a thermal power plant is the real thermal efficiency of the cycle, defined as the ratio of internal power of a turbine to thermal power supplied to the cycle. Two balance methods were employed: the enthalpy and entropy methods which utilise different algorithms for finding the internal power of the turbine.

Assuming the adequacy of the two methods, thermal efficiency values derived by the two methods should display a high degree of correspondence. This is a major aspect for comparison and control when the adequacy of the applied method is to be assessed.

4.1. Enthalpy method

The enthalpy method allows for evaluation of the back-pressure power unit performance. Accordingly, parameters are derived which express how the thermal energy supplied to the block in the cogeneration of heat and electricity is utilised.

A back-pressure power unit typically operates in the heating mode and hence its performance is evaluated using the conversion factor for the chemical energy of fuel, expressed as:

$$\xi_{UP} = \frac{N_{el} + \dot{Q}_c}{(\dot{m}_p \cdot W_d)_{UP}} \tag{1}$$

where:

N_{el} – electric power on the generator terminals,
$\dot{Q}_c$ – thermal power of district heating bleeds,
$(\dot{m}_p \cdot W_d)_{UP}$ – thermal power of fuel combusted in the boiler,
$\dot{m}_p$ – fuel consumption,
W_d – calorific value of the fuel.

The energy balance equation for the back-pressure power unit shown in Figure 2 can be written as:

$$(\dot{m}_p \cdot W_d)_{UP} = \frac{\dot{Q}_c + N_i}{\eta_k} \tag{2}$$

where:

N_{el} – internal power of the turbine,
η_k – boiler efficiency, which is expressed as:

$$\eta_k = \frac{Q_0}{(\dot{m}_p \cdot W_d)_{UP}}.$$

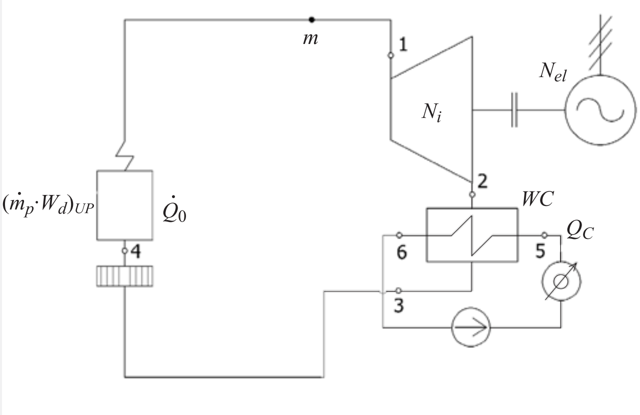

Fig. 2. Simplified diagram of a back-pressure power unit in a combined heat and power plant

The relationship between the generated electric power N_{el} and the thermal power of the district heating bleeds $\dot{Q}_c$ in the back-pressure power unit can be expressed as a cogeneration factor:

$$\sigma_o = \frac{N_{el}}{\dot{Q}_c} \tag{3}$$

Replacing thermal power of fuel in Eq. (1) by inserting Eq. (2) and applying the relationship (3), the conversion factor for chemical energy of fuel for the back-pressure power unit is given as:

$$\xi_{UP} = \eta_k \cdot \frac{\sigma_o + 1}{\dfrac{\sigma_o}{\eta_m \cdot \eta_g} + 1} \tag{4}$$

where:

η_m – mechanical efficiency of the turbine,
η_g – electrical efficiency of the generator.

One has to bear in mind that when the power unit is operated in the heating modes, the values are ξ_{UP}; whilst in the case of the pseudo-condensing mode of it assume that product of $\eta_m \cdot \eta_g$ is nearly unity the conversion factor for chemical energy of fuel is equal to power conversion efficiency of the extraction back-pressure unit i.e. $\xi_{UP} = \eta_{EK}$.

When the enthalpy method is employed, the real thermal efficiency of the cycle is obtained from the formula:

$$\eta_t = \frac{N_i}{\dot{Q}_0} \tag{5}$$

where:

$\dot{Q}_0$ – thermal power supplied to the cycle,
N_i – internal power of turbine.

For the real installation of the back-pressure power unit BC-100 shown in the Figure 1, internal power of the turbine is obtained from the energy balance equation in the form:

$$\begin{aligned} N_i &= \dot{m}_4 (i_1 - i_I) + \dot{m}_5 (i_1 - i_{II}) + \\ &+ 0.7 \cdot 0.6 \cdot \Delta \dot{m}_{22} (i_1 - i_{II}) + \dot{m}_6 (i_1 - i_{III}) + \\ &+ \dot{m}_7 (i_1 - i_{IV}) + \dot{m}_8 (i_1 - i_V) + \dot{m}_2 (i_1 - i_2) + \dot{m}_3 (i_1 - i_3) \end{aligned} \tag{6}$$

Thermal power of the boiler is expressed by the balance equation:

$$\dot{Q}_0 = \dot{m}_0 (i_0 - i_{27}) \tag{7}$$

It is assumed that internal power achieved from the total flux of steam loss in the turbine[1)] equals $0.7 \cdot 0.6 \cdot \Delta \dot{m}_{22} (i_1 - i_{II})$.

Enthalpy flux of steam supplied to the turbine can calculate from:

$$\dot{Q}_1 = \dot{m}_0 (i_1 - i_{27}) \tag{7a}$$

4.2. Entropy method

Finding thermal efficiency by the enthalpy method fails to give us detailed information about the thermodynamic status of particular components of the thermal system.

According to the entropy method, the values of entropy generated in particular devices of unit are found and thermodynamic performance parameters of the back-pressure power unit can be easily evaluated. To illustrate the principles of the entropy method, let us consider a simplified diagram of the back-pressure power unit shown in the Figure 2.

Underlying the power balance equation are the principles set forth in (Chmielniak 1987, 2000, 2004). Accordingly, the balance equation for the back-pressure power unit becomes:

$$\dot{Q}_0 = N_i + \dot{Q}_c + \dot{I}_{ot} \tag{8}$$

and hence:

$$N_i = \dot{Q}_0 - \dot{Q}_c - \dot{I}_{ot} \tag{8a}$$

where $\dot{I}_{ot}$ – flux of enthalpy loss transferred to the surroundings.

The equation of entropy flux balance can be written as:

$$\dot{S}_0 + \dot{S}_{gen} = \dot{S}_c + \dot{S}_{ot} \tag{9}$$

where:

$\dot{S}_{ot}$ – flux of entropy transferred to the surroundings (due to losses),
$\dot{S}_{gen}$ – sum total of generated entropy flux fluctuations in the heating system of the unit: in turbine sections, in degasifier and in the heat regeneration system.

Inserting the relationships expressing the flux of entropy transferred to the system $\dot{S}_0 = \dot{Q}_0 / T_1$ and the flux of entropy transferred from the cycle $\dot{S}_c = \dot{Q}_c / T_2$ into Eq. (9) yields:

$$\frac{\dot{Q}_c}{T_2} - \frac{\dot{Q}_0}{T_1} + \dot{S}_{ot} = \dot{S}_{gen} \tag{9a}$$

and hence:

$$\dot{Q}_c = \left(\dot{S}_{gen} - \dot{S}_{ot} + \frac{\dot{Q}_0}{T_1} \right) \cdot T_2 \tag{9b}$$

Temperatures present in Eqs (9a), (9b) are:

T_1 – mean value of the absolute temperature of heat supplying process to the cycle (in the boiler),
T_2 – mean value of the absolute temperature of heat transferring process from the cycle (in the heat exchangers).

Thermal efficiency of a power unit can be derived from a formula similar to that applied in the enthalpy method, in accordance with Eq. (5):

$$\eta_t = \frac{N_i}{\dot{Q}_0}.$$

Substituting Eq. (8a) into (5) yields:

$$\eta_t = 1 - \frac{\dot{Q}_c + \dot{I}_{ot}}{\dot{Q}_0} \tag{10}$$

[1)] According explanations were presented in part 3, Methodology.

The quantity Q_c in Eq. (10) is then replaced by (9b) and the thermal efficiency equation can be thus rewritten as

$$\eta_t = 1 - \frac{\dot{S}_{gen} \cdot T_2}{\dot{Q}_0} - \frac{T_2}{T_1} - \frac{\dot{I}_{ot} - \dot{S}_{ot} \cdot T_2}{\dot{Q}_0} \tag{10a}$$

The final version of the thermal efficiency equation according to the entropy method is obtained by transforming the first three terms in Eq. (10a) to the product form. Accordingly, we get:

$$\eta_t = \left(1 - \frac{T_2}{T_1}\right) \cdot \left(1 - \frac{\dot{S}_{gen} \cdot T_1 \cdot T_2}{(T_1 - T_2) \cdot \dot{Q}_0}\right) - \frac{\dot{I}_{ot} - \dot{S}_{ot} \cdot T_2}{\dot{Q}_0} \tag{10b}$$

It is readily apparent that the first term on the right hand side of Eq. (10b) expresses thermal efficiency of the Carnot cycle and the second term is less than unity, giving a qualitative representation of thermodynamic processes in the heat generation system of the power unit. Contribution of ambience parameters – the second term on the right hand side of Eq. (10b) usually has only a minor bearing on thermal efficiency and can be therefore neglected in computations on the preliminary stage.

In order to derive thermal efficiency of the back-pressure power unit BC-100 using Eq. (6) it is required that the values of relevant parameters be found on the basis of entropy balance for the heat generation unit – Figure 1.

The sum total of generated entropy flux fluctuations in the heat-generation system can be obtained from the formula:

$$\begin{aligned} \dot{S}_{gen} &= \dot{S}_{HP} + \dot{S}_{LP} + \dot{S}_{XN\,1} + \\ &+ \dot{S}_{XW\,1} + \dot{S}_{XW\,2} + \dot{S}_{XW\,3} + \dot{S}_{ODG} \end{aligned} \tag{11}$$

Particular terms in Eq. (11) are as follows:

– Fluctuations of entropy flux in the HP-turbine is derived from the formula:

$$\dot{S}_{HP} = \dot{m}_1 (s_I - s_1) + (\dot{m}_1 - \dot{m}_4) \cdot (s_{II} - s_I) \tag{12}$$

– Fluctuation of entropy flux in the LP-turbine :

$$\begin{aligned} \dot{S}_{LP} &= \dot{m}_{WS} (s_{III} - s_{II}) + (\dot{m}_{WS} - \dot{m}_6) \cdot (s_{IV} - s_{III}) + \\ &+ (\dot{m}_{WS} - \dot{m}_6 - \dot{m}_7) \cdot (s_V - s_{IV}) + \dot{m}_2 (s_2 - s_V) + \\ &+ (\dot{m}_{WS} - \dot{m}_6 - \dot{m}_7 - \dot{m}_2) \cdot (s_3 - s_V) \end{aligned} \tag{13}$$

where $\dot{m}_{WS}$ – mass flux of steam at the inlet to the LP-turbine.

– Fluctuation of entropy flux in the high-pressure regeneration system: from equations for the heat exchangers: XW1, XW2, XW3

$$\begin{aligned} \dot{S}_{XW\,1} &= \dot{m}_{23} (s_{25} - s_{24}) + \dot{m}_6 \cdot (s_{16} - s_6) + \\ &+ (\dot{m}_4 + \dot{m}_5) \cdot (s_{16} - s_{15}) \end{aligned} \tag{14}$$

$$\begin{aligned} \dot{S}_{XW\,2} &= \dot{m}_{23} (s_{26} - s_{25}) + \dot{m}_5 \cdot (s_{15} - s_5) + \\ &+ \dot{m}_4 \cdot (s_{15} - s_{14}) \end{aligned} \tag{15}$$

$$\dot{S}_{XW\,3} = \dot{m}_{23} (s_{27} - s_{26}) + \dot{m}_4 \cdot (s_{24} - s_4) \tag{16}$$

– Fluctuation of entropy flux in the heat exchanger XN1 of the low-pressure regeneration system.

$$S_{XN\,1} = \dot{m}_{21} (s_{20} - s_{19}) + \dot{m}_8 \cdot (s_{18} - s_8) \tag{17}$$

– Fluctuation of entropy flux in the degasifier

$$\begin{aligned} \dot{S}_{DG} &= (\dot{m}_4 + \dot{m}_5 + \dot{m}_6) \cdot (s_{23} - s_{16}) + \\ &+ (\dot{m}_{21} + \Delta\dot{m}_{22}) \cdot (s_{23} - s_{21}) + \dot{m}_7 \cdot (s_{23} - s_7) \end{aligned} \tag{18}$$

The value of entropy at point 27 in the Figure 1 is governed by the formula:

$$s_{27} = s_{23} - \Delta s_{XW\,1} + \Delta s_{XW\,2} + \Delta s_{XW\,3} \tag{19}$$

whilst entropy at point 20:

$$s_{20} = \frac{\dot{m}_{12} s_{12} + \dot{m}_{13} s_{13}}{\dot{m}_{12} + \dot{m}_{13}} + \Delta s_{XN\,1} \tag{20a}$$

if the values of mass fluxes $\dot{m}_{12}$ and $\dot{m}_{13}$ are nearing that:

$$s_{20} = \frac{s_{12} + s_{13}}{2} + \Delta s_{XN\,1} \tag{20b}$$

The value of thermal power supplied to the boiler cycle can be derived from the formula:

$$\dot{Q}_0 = \frac{\dot{m}_0 \cdot (i_1 - i_{27})}{\eta_r} \tag{21}$$

where η_r – pipeline efficiency.

– Mean value of the absolute temperature of isobaric process of heat supply to the cycle is obtained from the formula:

$$T_1 = \frac{\dot{m}_0 \cdot (i_1 - i_{27})}{\dot{m}_0 \cdot (s_1 - s_{27})} \tag{22}$$

– Mean value of the absolute temperature of isobaric process of heat transferring from the cycle in heat exchangers:

$$T_2 = \frac{\dot{m}_2 \cdot (i_2 - i_{12}) + \dot{m}_3 \cdot (i_3 - i_{13}) + \dot{m}_8 \cdot (i_{18} - i_{12})}{\dot{m}_2 \cdot (s_2 - s_{12}) + \dot{m}_3 \cdot (s_3 - s_{13}) + \dot{m}_8 \cdot (s_{18} - s_{12})} \tag{23}$$

Relationships formulated in the entropy method allow the performance evaluation of the investigated power unit BC-100 in the range of working loads, both in the heating and pseudo-condensing modes of operation.

5. RESULTS

Computed parameters of the power unit BC-100 in two operation modes are complied in form of Tables 1–2 and in graphic form in Figures 3–7, also in Tables 3–4 and Figures 8–11.

Table 1
Compiled results obtained by the enthalpy method for the power unit BC-100 in the heating mode of operation

Description	Symbol	Unit	1	2	3	4	5	6	7	8
Electric power	N_{el}	MW	96.02	95.06	90.31	90.51	79.06	79.12	100.27	100.21
Thermal power transferred to heating water	$\dot{Q}_c$	MW	181.2	178.3	168.0	169.2	149.2	151.0	192.2	192.8
Boiler efficiency	η_k	%	90.94	90.95	90.99	90.98	91.07	91.06	90.90	90.90
Enthalpy flux of steam at the turbine inlet	$\dot{Q}_1$	MW	279.7	274.8	262.0	263.8	232.4	233.4	295.6	296.5
Internal efficiency of the HP section of turbine	η_{iHP}	%	78.6	79.0	77.8	77.6	74.9	74.4	80.6	81.2
Internal efficiency of the LP section of turbine	η_{iLP}	%	88.7	88.8	89.6	87.7	88.2	88.5	89.2	88.2
Internal efficiency of the turbine	η_i	%	83.2	83.5	83.0	82.2	80.7	80.6	84.4	84.4
Cogeneration factor	σ_o	–	0.530	0.533	0.538	0.535	0.530	0.524	0.522	0.520
Conversion factor for chemical energy of the fuel	ξ_{UP}	%	89.69	90.02	89.26	89.12	88.96	89.33	89.50	89.40
Thermal efficiency of the cycle obtained by the enthalpy method	η_t	%	35.57	35.83	35.71	35.54	35.23	35.12	35.14	35.01

Table 2
Compiled results obtained by the entropy method for the power unit BC-100 in the heating mode of operation

Description		Symbol	Unit	1	2	3	4	5	6	7	8
Entropy generation	HP high-pressure section of the turbine	$\dot{S}_{HP}$	kW/K	27.09	25.95	27.35	28.03	27.89	29.62	24.95	24.69
	LP low-pressure section of the turbine	$\dot{S}_{LP}$	kW/K	14.43	12.89	11.15	10.95	8.51	6.96	15.48	14.95
	Heat exchanger XN1	$\dot{S}_{XN1}$	kW/K	1.447	1.435	1.332	2.039	1.508	1.237	1.630	1.421
	Heat exchanger XW1	$\dot{S}_{XW1}$	kW/K	1.219	1.066	1.109	1.101	0.961	0.796	1.602	1.612
	Heat exchanger XW2	$\dot{S}_{XW2}$	kW/K	1.062	1.175	0.965	0.972	1.020	0.910	1.195	1.230
	Heat exchanger XW3	$\dot{S}_{XW3}$	kW/K	1.126	1.091	0.976	1.005	0.960	0.983	1.186	1.218
	Degasifier	$\dot{S}_{DG}$	kW/K	4.52	4.47	1.28	2.88	2.14	1.04	9.75	8.30
	Total	$\dot{S}_{gen}$	kW/K	50.89	48.07	44.16	46.98	42.98	41.54	55.79	53.43
Mean value of the absolute temperature of heat supply		T_1	K	607.3	607.9	603.4	606.0	608.0	604.8	613.3	609.9
Mean value of the absolute temperature of heat transfer		T_2	K	356.4	356.8	355.7	355.5	354.6	353.6	360.6	360.8
Thermal power supplied to the cycle		$\dot{Q}_0$	MW	281.1	276.2	263.3	265.1	233.6	234.6	297.0	298.0
Thermal efficiency of the cycle by the entropy method		η_t	%	35.03	35.26	35.25	35.19	35.29	35.39	34.58	34.52

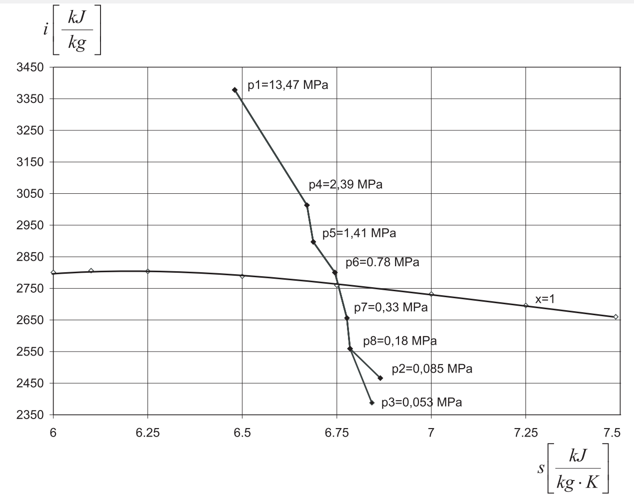

Fig. 3. Steam expansion in the turbine on the *i-s* diagram plane for the working load N_{el} = 100.3 MW – the heating mode of operation

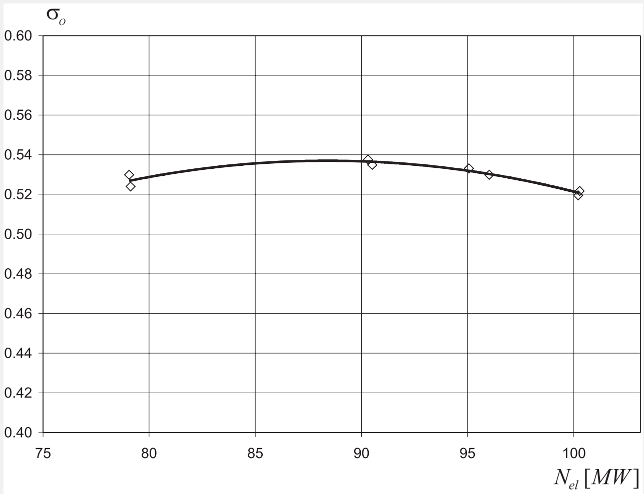

Fig. 4. Cogeneration factor characteristic for the power unit BC-100 – heating mode of operation

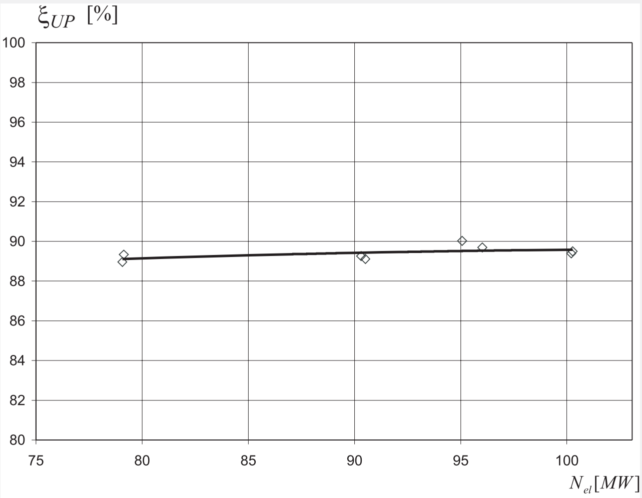

Fig. 5. Characteristic of the conversion factor of the fuel chemical energy for the power unit BC-100 – heating mode of operation

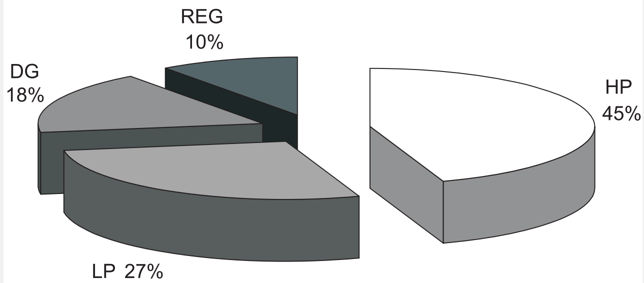

Fig. 6. Distribution of entropy generation contributions in the BC-100 assemblies for the working load N_{el} = 100.3 MW – heating mode of operation, HP – high-pressure section of the turbine, LP – low-pressure section of the turbine, REG – regeneration system, DG – degasifier

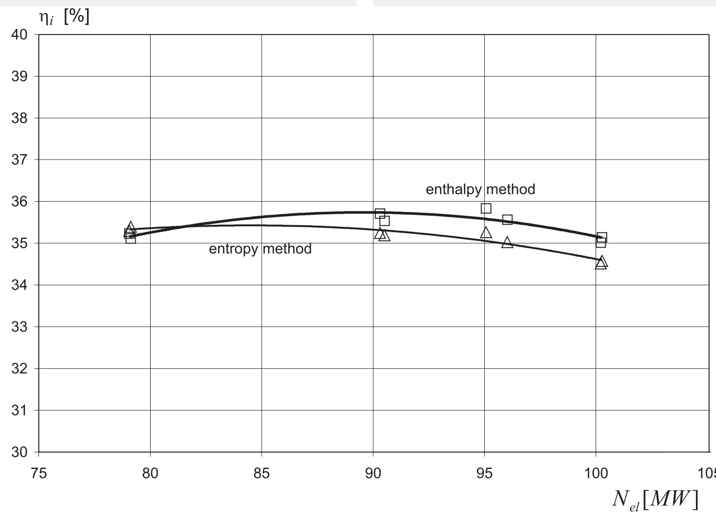

Fig. 7. Graphic display of thermal efficiency profiles obtained by the enthalpy and entropy method for the power unit BC-100 – heating mode of operation

Table 3
Compiled results obtained by the entropy method for the power unit BC-100 in the pseudo-condensing mode of operation

Description	Symbol	Unit	1	2	3	4	5	6	7	8
Electric power	N_{el}	MW	96.13	96.13	89.27	89.54	83.34	83.12	76.93	76.90
Thermal power transferred to heating water	$\dot{Q}_c$	MW	200.7	202.3	184.4	185.3	167.3	167.0	157.1	155.9
Boiler efficiency	η_k	%	90.89	90.89	90.95	90.95	91.01	91.01	91.06	91.06
Enthalpy flux of steam at the turbine inlet	$\dot{Q}_1$	MW	299.1	300.3	275.6	277.4	254.9	255.0	235.9	236.0
Internal efficiency of the HP section of turbine	η_{iHP}	%	80.8	80.0	78.3	78.2	77.5	77.5	75.6	75.5
Internal efficiency of the LP section of turbine	η_{iLP}	%	84.9	85.3	84.3	86.7	86.1	85.4	86.6	88.5
Internal efficiency of the turbine	η_i	%	82.5	82.2	80.7	81.6	80.9	80.7	79.9	80.5
Power conversion efficiency	η_{EK}	%	29.06	28.95	29.31	29.20	29.61	29.51	29.54	29.52
Thermal efficiency of the cycle obtained by the enthalpy method	η_t	%	33.29	33.17	33.55	33.44	33.87	33.76	33.78	33.75

Table 4
Compiled results obtained by the entropy method for the power unit BC-100 in the pseudo-condensing mode of operation

Description		Symbol	Unit	1	2	3	4	5	6	7	8
Entropy generation	HP High-pressure section of the turbine	$\dot{S}_{HP}$	kW/K	24.75	25.99	26.33	27.02	26.64	26.13	27.87	27.29
	LP Low-pressure section of the turbine	$\dot{S}_{LP}$	kW/K	19.06	19.91	19.60	19.57	16.11	16.17	16.14	15.23
	Heat exchanger XN1	$\dot{S}_{XN1}$	kW/K	0.801	0.754	0.507	0.498	0.779	0.798	0.672	0.561
	Heat exchanger XW1	$\dot{S}_{XW1}$	kW/K	1.102	1.216	1.045	0.748	0.896	0.879	0.830	0.798
	Heat exchanger XW2	$\dot{S}_{XW2}$	kW/K	1.244	1.230	1.056	0.926	0.912	0.952	0.887	0.883
	Heat exchanger XW3	$\dot{S}_{XW3}$	kW/K	1.218	1.246	1.134	1.112	0.921	0.877	0.785	0.868
	Degasifier DG	$\dot{S}_{DG}$	kW/K	2.312	5.094	1.049	0.685	1.677	0.866	0.106	3.748
	Total	$\dot{S}_{gen}$	kW/K	49.41	52.94	48.31	47.61	46.51	45.98	44.38	47.57
Mean value of the absolute temperature of heat supply		T_1	K	601.0	601.6	602.0	599.2	604.2	601.3	602.0	600.5
Mean value of the absolute temperature of heat transfer		T_2	K	372.0	372.5	368.4	368.6	364.8	364.5	362.0	361.6
Thermal power supplied to the cycle		$\dot{Q}_0$	MW	300.6	301.8	277.0	278.8	256.2	256.3	237.1	237.2
Thermal efficiency of the cycle obtained by the entropy method		η_t	%	32.67	31.90	32.87	32.71	33.25	33.19	33.28	33.16

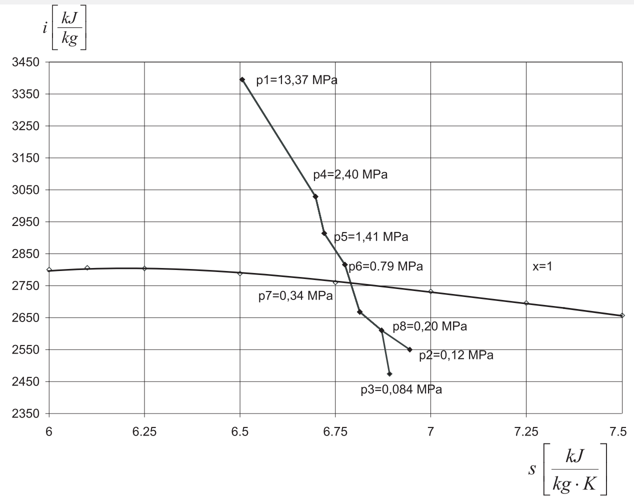

Fig. 8. Steam expansion in the turbine on the *i-s* diagram plane for the working load N_{el} = 96.1 MW – pseudo-condensing mode of operation

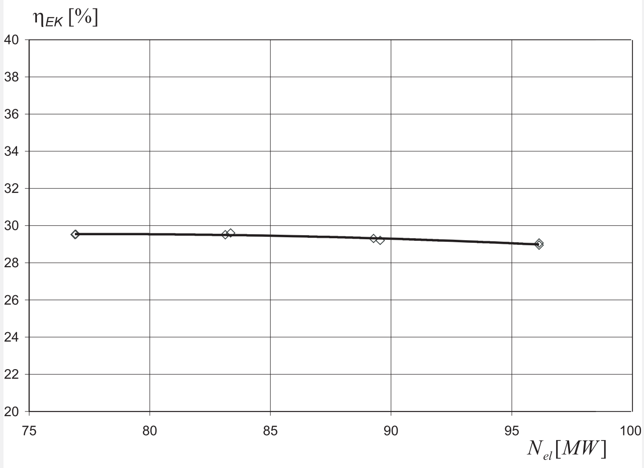

Fig. 9. Thermal efficiency profiles for the power unit BC-100 – pseudo-condensing mode of operation

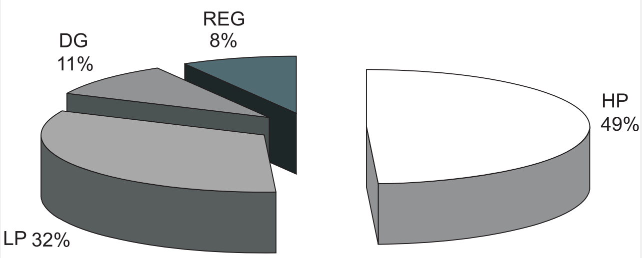

Fig. 10. Distribution of entropy generation contributions in the BC-100 assemblies for the working load N_{el} = 96.1 MW – pseudo-condensing mode of operation, HP – high-pressure section of the turbine, LP – low-pressure section of the turbine, REG – regeneration system, DG – degasifier

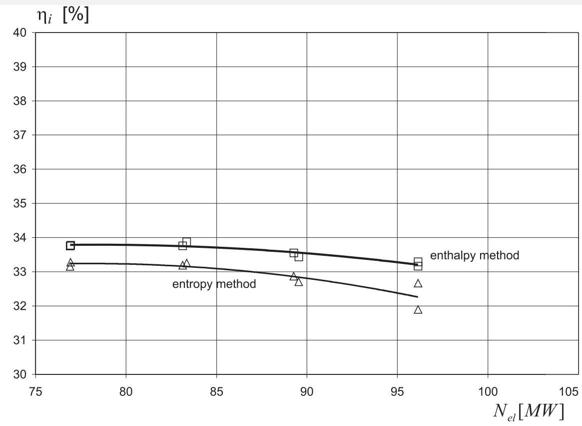

Fig. 11. Graphic display of thermal efficiency profiles obtained by the enthalpy and entropy method for the power unit BC-100 – pseudo-condensing mode of operation

6. CONCLUSIONS

Research data are compiled in Tables 1–4, for the two operating modes of a back-pressure power unit under a range of working loads. Thermodynamic parameters of the back--pressure unit performance were obtained by the enthalpy and entropy balance methods.

Heating mode of operation

– load range N_{el} = 79.0–100.3 MW:

– The conventional process of steam expansion in the turbine[2)] is shown in Figure 3 for the loading N_{el} = 100.3 MW. Turbine's internal efficiency values for the given load range would fall in the interval:
  - for the HP-turbine: η_{iHP} = 74.4–81.2%,
  - for the LP-turbine: η_{iLP} = 84.8–89.6%,
  - mean efficiency of the whole turbine: η_i = 80.7–84.4%.

– Cogeneration factor for the investigated loading range would fall in the interval σ_o = 0.520–0.538, its maximal value of reached for the load N_{el} = 90.3 MW (Fig. 4).

– Conversion factor for chemical energy of fuel for the back-pressure power unit would fall in the range ξ_{UP} = = 89.0–90.0% for the investigated load range (Fig. 5).

– The real thermal efficiency of the cycle in the investigated load range was:
  - η_t = 35.0–35.8%, according to the enthalpy method,
  - η_t = 34.5–35.4%, according to the entropy method.

The efficiency/load profiles are shown in the Figure 7. Discrepancies between the thermal efficiency values obtained by the two method would range from – 0.27% to + 0.57%.

Typically, thermal efficiency obtained by the entropy method would have a lower value. Distribution of entropy generation

[2)] Course of steam expansion line in the turbine in the range of the wet saturated steam as for heating as well pseudo-condensing mode of operation achieved in the back–pressure power unit are shown as percentage fractions on the diagram in the Figure 6, for the working load N_{el} = 100.3 MW. The HP-turbine accounts for 45% of entropy generation, next in line are: the LP-turbine (27%), degasifier DG (18%) and regeneration systems REG (10%).

It is worthwhile to mention that operation of the back--pressure unit is fully autonomous and the amounts of generated electricity and heat to be supplied to the customers must be precisely controlled to make the process cost-effective. The control is achieved through varying the streams of fresh steam fed to the turbine. Hence in the range of nominal loads in the heating mode of operation the value of the conversion factor for chemical energy of fuel approaches the value of boiler efficiency. At that point the power unit has the highest efficiency. When the demand for district heating decreases, the conversion factor for chemical energy of fuel decreases too and outside the heating season the back-pressure power unit is switched to the pseudo-condensing mode. Accordingly, the conversion factor for chemical energy of the fuel equals the power conversion efficiency of the back-pressure power unit.

Pseudo-condensing mode of operation

– load range N_{el} = 76.9–96.1 MW:

– In the pseudo-condensing mode steam expansion in the turbine proceeds in the same manner as in the heating mode, which is borne out by the expansion profiles for the working load N_{el} = 96.1 MW shown in the Figure 8. Internal efficiency of the turbine, being a measure of steam expansion in the turbine in this mode of operation, would fall in the following range:
  – for the HP-turbine: η_{iHP} = 75.5–80.8%,
  – for the LP-turbine: η_{iLP} = 82.0–85.0%,
  – mean efficiency of the whole turbine: η_i = 79.2–82.3%.

– The power conversion efficiency profile for the back-pressure power unit shown in the Figure 9 is flat over the investigated range of working loads and the obtained efficiency values differ only slightly, falling in the range η_{EK} = 28.95–29.61%.

  In the pseudo-condensing mode the distribution of entropy generation contributions is analogous to that in the heating mode. A distribution of entropy generation contributions in the pseudo-condensing mode of operation of the back-pressure power unit is shown in the Figure 10 for the working load N_{el} = 96.1 MW. The HP-turbine accounts for 49% of entropy generation, next in line are: the LP-turbine (32%), degasifier DG (11%) and regeneration systems REG (8%).

– The profiles of the real thermal efficiency of the cycle for the pseudo-condensing mode are shown in the Figure 11 and it is apparent that in the investigated load range the thermal efficiency achieved values:
  • η_t = 33.2–33.9%, according to the enthalpy method,
  • η_t = 31.9–33.3%, according to the entropy method.

Thermal efficiency characteristics clearly reveal that efficiency values obtained by the two balance methods disagree in a minor degree only, the deviation falling in the range of computed values from + 0.50% to + 1.27%.

It is reasonable to suppose, therefore, that the actual discrepancy between the two efficiency values obtained by the enthalpy and entropy methods are minor up to 1% (apart from point 2. for the pseudo-condesnsing mode of operation with derivation achieved value +1.27%) for the investigated load range, both in the heating and pseudo-condensing mode of operation. The results seem sufficiently accurate, particularly when we recall measurement errors and mistakes inevitably made when thermal parameters of steam and water are sought. The results seem also to confirm the adequacy of the applied performance evaluation procedure.

Application of equations governing entropy fluctuations to the model testing of a power plant extends the scope of control of the installation. A chief benefit of the entropy-based model is that the sources of energy dispersion are easily localised and their impacts on the performance of other units are identified. That is why this approach is particularly useful in diagnosing the working condition of nodes and individual system components. Control of entropy generation in machines and installations of a power unit in a power plant or a heat and power plant might be useful in assessing their condition though some sort of baseline data are always required in such case (e.g. technical condition of a new installation or before and after the repairs). Furthermore, the entropy method gives us most valuable information about the effects of modernisation. When constant monitoring procedures are employed, this method allows for quick diagnosis of particular plant installations, which might be useful in prognosticating their service life and the scope of maintenance jobs and repairs.

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